Second Edition

# Troubleshooting Centrifugal Pumps and their Systems

Ron Palgrave

# TROUBLESHOOTING CENTRIFUGAL PUMPS AND THEIR SYSTEMS

Second Edition

**RON PALGRAVE**
Pump Hydraulic Specialist, Gateshead
United Kingdom

ELSEVIER

Butterworth-Heinemann
An imprint of Elsevier

Butterworth-Heinemann is an imprint of Elsevier
The Boulevard, Langford Lane, Kidlington, Oxford OX5 1GB, United Kingdom
50 Hampshire Street, 5th Floor, Cambridge, MA 02139, United States



**Library of Congress Cataloging-in-Publication Data**
A catalog record for this book is available from the Library of Congress

**British Library Cataloguing-in-Publication Data**
A catalogue record for this book is available from the British Library

ISBN: 978-0-08-102503-1

For information on all Butterworth-Heinemann publications visit our website at https://www.elsevier.com/books-and-journals

*Publisher:* Matrhew Deans
*Acquisition Editor:* Carrie Bolger
*Editorial Project Manager:* Carrie Bolger
*Production Project Manager:* Paul Prasad Chandramohan
*Cover Designer:* Christian Bilbow

Typeset by TNQ Technologies

# Preface to second edition

In writing this the second edition, I had a number of points in mind.

The first was to correct any errors or ambiguities that had crept into the core text of the first edition. Any significant updates should also be included.

Secondly, I wanted to expand a little more on the pump hydraulic design aspects that I had briefly touched on in the earlier edition. The first edition dealt chiefly with hydraulic troubleshooting of centrifugal pumps and systems. I hope it was clear that any successful outcome depended on BOTH the pump and system characteristics being considered. But in that volume, any descriptions of pump hydraulic design elements were only introduced in order to give background to the particular troubleshooting issue. In this second edition, much of the hydraulic design elements have now been gathered into separate appendices. A few new components have been introduced to help the structure of this section. This includes a basic description of how pumps do what they do. All this might be useful to those who are only occasionally involved in the use of centrifugal pumps. I hope all this will then leave the troubleshooting sections more coherent.

In the first edition, I deliberately avoided too many references to specific case studies. I have continued this approach in the second edition. My reasons are as follows- Pumping system performance is dependent on both the pump and system hydraulic characteristics. All the different piping system characteristics that exist can be usually grouped into a small number of abstract categories. As it is important, these categories are explained at some length. These categories are independent of industry or application. Hence comment or analyses, including troubleshooting solutions, relating to any particular category, are broadly applicable to a wide range of other applications. For example, pumps that drain mineral mines deep under the earth can face the same hydraulic interaction issues as pumps pressurising an oil field, feeding water to a power generating boiler, or even supplying water to a village that is significantly elevated from its source. At first sight, the underlying interactions behind such diverse applications may not be obvious. If I were to use only industry-specific case studies, it might not then be obvious that its solution might be applicable in several other diverse industries. I hope that maintaining an abstract perspective will allow deeper insights to specific pump –system interactions.

Most of the pumping systems covered by this book would have begun with an assumption that the liquid being handled is almost free from suspended solids. Except where this is unavoidable, most manufacturers will encourage the user to take steps to make sure this condition is met. Where significant levels of suspended solids were foreseen at the outset, then special solids handling pump would most probably have been installed. However, it is known that those conventional clear liquid pumps are occasionally exposed to suspended solids. Mostly this will be unexpected. However, in a small number of cases, suitable solids handling pumps are not widely available and clear liquid pumps have to be used. This would typically occur if the liquid were extremely hot and maybe volatile.

In the case of conventional pumps, re-engineering is often possible to make the machines more tolerant to suspended solids. However, before this can be carried out it is essential to identify the particular damage mechanism causing wear failure. This then avoids the wrong solution being applied to the problem. I devote a new section to the issue of abrasive wear in pumps, and its effect on performance. It is a visual data base of abrasive pumping symptoms. I explain how, in these circumstances, the shape of a pump characteristic curve can influence how fast it appears to wear out.

At the end of many hydraulic troubleshooting exercises, the user is often faced with the dilemma that the pump hydraulics need to be, modified or replaced. I have included a discussion relating to modification.

Chapter 1 explains that the concept of a centrifugal pump has been in existence for a very long time. In that period, different manufacturers have made many different interpretations of the concept. I was fortunate in that by the end of my career, I had worked for 9 pump manufacturers having 16 subsidiaries. In some matters there was general agreement on how things should be done. In others areas there were significant differences, but these often amounted simply to 'opinion engineering'. Despite these differences, each manufacturer still produced pumps that served their purpose. Given such diversity, it is unrealistic to suppose that the outcome of any of the general advice given in this volume could be guaranteed in all circumstances. This advice is given in good faith, but is only intended to illustrate the likely scale of any outcome.

I should close by mentioning that this book deals almost exclusively with matters that are hydraulic in nature. There are only a few references to purely mechanical engineering matters. That is because many good manuscripts already exist covering these facets of pump construction, such as [say] bearings and seals. Similarly, the general subject of rotating machines such as alignment and so on. There seemed little point in simply reiterating this.

# CHAPTER 1

# Introduction

Whom is this book intended for? It is intended for those new to the use [and abuse] of centrifugal pumps. It is also for those whose involvement is so occasional that they often need a reminder of the basics. I hope its content can help avoid just some of problems I have been involved with over the years.

Modern Centrifugal Pumps are not complex machines. Usually they consist of a few castings, mouldings or pressings that are pierced by a few wrought or turned components. They are evidently one of the most ubiquitous machines, being comparable in number to electric motors. Despite their simplicity, centrifugal pumps are still capable of posing complicated puzzles during operation. A medical analogy is in order.

A patient may say to a doctor, 'I have a pain in my arm, what is the cause?' At this level the doctor is almost powerless to make a diagnosis against such a simple remark. So he asks structured questions to try to narrow the problem down.

His experience may suggest that the cause is a likely a broken bone or a badly strained muscle. So he might begin on that premise.

'Has the pain just begun suddenly?'

'Have you any previous experience of this pain?'

'Does it hurt when you turn a certain way?'

Progressively the doctor narrows the problem down to a most likely cause or causes.

In the same way, a pump engineer must follow a structured approach so as to home in on the cause. Sometimes the doctor cannot be absolutely sure because different ailments often have similar symptoms. So it is with pumps. This is particularly true when the pump in question may not be accessible. It may be on another continent and even worse the symptoms are relayed via a third party.

After discussions with many users over the years, I found it useful to draw up a specification for an ideal pump design [1] and relisted here. Real pumps fall someway short of this ideal and so some operating troubles

*Troubleshooting Centrifugal Pumps and Their Systems*
ISBN 978-0-08-102503-1
https://doi.org/10.1016/B978-0-08-102503-1.00001-3

might be inevitable. Evidently the ‘Ideal Centrifugal Pump’ should be very tolerant, flexible and able to:

- Tolerate indefinite dry running
- Handle suspended solids of an undefined size, both granular and fibrous.
- Tolerate entrained gas with little or no performance decay
- Show some inbuilt intelligence and ability to react to unspecified changes to the pumped liquid system
- Have indefinite life, or cost so little as to not be worthy of repair if it fails.
- Demand no routine maintenance, or so little that it can be written on one [small] sheet of paper
- Have Infinite bearing life and, where fitted, infinite shaft seal life.
- Have no negative impact on the environment.
- Have an internationally interchangeable silhouette
- Have no restrictions to its operating window. Ideally Efficiency and NPSH[R] should be constant and not flow-related.
- Self-prime as and when required.
- Be fully transportable and not require any special foundations
- Require no external services such as cooling water, lube oil and so on.
- Be constructed of standard parts and non-exotic materials, both of which should be widely available.
- Able to routinely operate with almost no suction pressure.
- Able to tolerate thermal shocks of temperatures up to 450 °C
- Be able to totally absorb all envisaged pipe loads unassisted by additional pipe support.

Modern machines can fulfil many of these expectations, but not all of them. It might be useful to put the modern machines into context.

# CHAPTER 2

# History

## Introduction

The need to move liquid has long existed. Early machines were simple but worked slowly. The centrifugal pump we know today did not really evolve till the 1600's. Following a long subsequent period of disinterest, its use then accelerated once fast and reliable drivers became available. The theoretical basis for pump design further improved on the strength of potential aerospace applications.

Centrifugal pumps are a comparatively recent invention. Their design and construction require some degree of industrial development. However, the need to move liquid has existed almost from the start of civilisation. As humanity began to farm the land, there was a parallel need, in certain parts of the world, to irrigate crops. It is probable that the Shadoof [Shadouf] became the most widespread method of doing this (Fig. 2.1).

**Fig. 2.1** The Shadoof. This is probably the first widely used pumping machine. Its origins are lost in time, but the device became prolific in the Middle East. The suspended bag is periodically dipped into the river. When full, the counter-weighted swivel arm helps lift the bag over and above an irrigation canal. *(Reproduced from Wikipedia.)*

*Troubleshooting Centrifugal Pumps and Their Systems*
ISBN 978-0-08-102503-1
https://doi.org/10.1016/B978-0-08-102503-1.00002-5

Obviously, this was not a centrifugal pump, but the widespread use of the Shadoof created a route for subsequent invention.

Another need to move liquid emerged just as soon as man began to mine for minerals under the earth. In certain circumstances, groundwater could seep into such mine workings and create a hazard. It could even cause work to cease. Unlike the irrigation problem, there was less agreement on the early solutions. That probably reflected the much wider range of 'lifts' or liquid level changes involved. Many different innovations are recorded.

These two liquid handling problems existed from very early times. But it is not until relatively recently that we find evidence of the centrifugal pump concept. An Italian engineer, Francesco di Giorgio Martini laid down some ideas relating to such a device in 1475, but there is no known evidence that it was ever constructed. It is also recorded that Dr Reti [a Brazilian historian] described a pumping machine relying upon centrifugal force. Again there is no evidence it was ever built.

Evolution of the centrifugal pumping idea depended on the concept of a rotating liquid handling machine. Such a concept was not new. Vitruvius [c 80BC–c 15AD] had mentioned the Aeolipile, which is a form of impulse turbine. The philosopher Hero [c10–70 AD] is alleged to have actually built a model as a novelty (Fig. 2.2).

Centuries later, this would provide the inspiration for a form of centrifugal pump. *[While discussing rotary liquid handling machines it is worth mentioning a contemporary device, the Archimedean screw. Though not a centrifugal machine, it did show that the rotary principle could be successfully applied to the pumping problem.]*

The first practical centrifugal pump was conceived and built by the Frenchman, Denis Papin, in around 1689. *[He also, controversially, laid claim to invention of the steam engine.]* He drew inspiration from the Hero Aeolipile device but considered his pump as a reverse turbine. We now know that good pump hydraulic designs can make good turbines when run in reverse. However, the opposite is not true. Without some compromise good turbine hydraulic designs rarely make good pumps. This was indeed the case here. His reverse turbine was not efficient but it did function. It showed the way ahead. Papin's original pump is shown here (Fig. 2.3).

But by 1705, he appears to have made some substantial improvements (Fig. 2.4).

Unlike his first pump, which mirrored the Hero Aeolipile in having only two vanes or ducts, his later impeller employed a large number of

**Fig. 2.2** The Aeolipile of HERO. Liquid was stored in the hemispherical base. When this was heated, steam rose up the hollow supporting legs and into the spherical section. Steam then issuing from both the jets caused the sphere to rotate. *(Reproduced from Wikipedia.)*

vanes. He also seems to have realised that the flowpath out of the casing in his original pump was too small and not very smooth. Therefore, his later pump incorporates a spiral volute casing. Many modern pumps employ both of these features.

It is likely that Papin developed the proportions of his pumps by trial and error. Little if any relevant hydraulic theory existed at that time. But by the mid 1770's, the French mathematician Euler developed a theory that attempted to describe the flowpath just as it leaves a multiblade impeller rim. This helped to put the discipline of pump hydraulic design onto a firm footing. However, there was certain hydraulic phenomenon that Euler could not include, and so practical pumps based on his theory would only develop approximately half the predicted theoretical pressure head. *[One aspect of his theory assumes that the liquid inside an impeller passage more or less follows the blade shape and angle. Liquid inertia effects conspire to make this untrue. The liquid 'lags behind' the blade shape and leaves the impeller at a lower angle than*

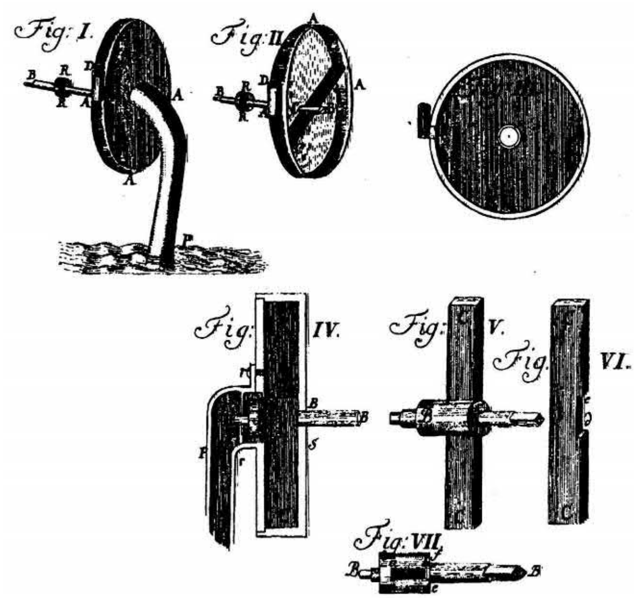



**Fig. 2.3** The early Denis Papin centrifugal pump design [1689] shows some resemblance to the HERO Aeolipile turbine, but in reverse. *(Reproduced from E. Grist, Cavitation and the Centrifugal Pump-A Guide for the Pump Users. Taylor and Francis, 1998.)*

*Euler would predict].* However, by factoring this departure into the design calculations, workable pumps could still be produced. Yet, sadly, development then more or less than stood still. More than 100 years would pass before Papin's original work was rediscovered and interest renewed. Although the Papin machines were impressive enough, their performance had been constrained by the speed at which they could be driven. This was still the era of such rotating machines as windmills and water wheels. Higher speed prime movers were not yet widely available. Rotary speed increasing systems were based on wooden cog-wheels or fabric belts.

By the early 1800's the industrial revolution was under way and several new ways existed of driving pumps at speeds in excess of triple digits. Increasing pump shaft speed was a more cost-effective method of raising pump performance than the alternative of just increasing its size.

At the Great Exhibition in London [1851], prominent engineers produced competing centrifugal pump designs. This list included Appold, Gwynne and Bessemer. There was apparently some acrimony, but the design of Appold won the contest. *[John Gwynne went on later that year to*

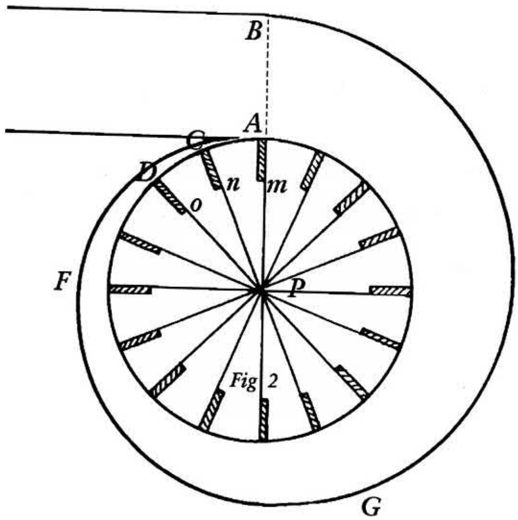



**Fig. 2.4** Later PAPIN impeller design [1705] shows increased vane number, of a straight paddle-like form. A volute spiral was applied to the collector passages. *(Reproduced from E. Grist, Cavitation and the Centrifugal Pump-A Guide for the Pump Users. Taylor and Francis, 1998.)*

*patent the concept of a multistage pump, though Osborne Reynolds had Mather and Platt manufacture his design for a multistage pump]]* (Fig. 2.5).

The Appold pump is very interesting in that it embodied features that are still familiar today. It included non-radial impeller blades. In contradiction however to modern designs, these blades leant forward and against the direction of rotation. Euler's theory would encourage this in order to maximise the pressure head that the impeller develops. However, as previously noted, this simplistic theory is not without its shortcomings and so in most modern designs, the blades lean backwards relative to the direction of rotation. *[Thereby inducing less hydraulic losses within the impeller].*

After leaving the impeller, the liquid enters a duct in which the passage area, for the most part, increases along the flowpath. This approximates the now ubiquitous volute passages of many modern pumps.

The impeller was of the form we would today call double suction or double entry. Modern designers do this chiefly in order to reduce cavitation resistance over an otherwise comparable single suction or single entry design. It is most unlikely that this formed the motive for Appold. More likely, it was the matter of axial impeller forces. In continuously turning liquid from an axial to radial direction, a single entry impeller can subsequently experience hydraulic forces. This momentum change can produce

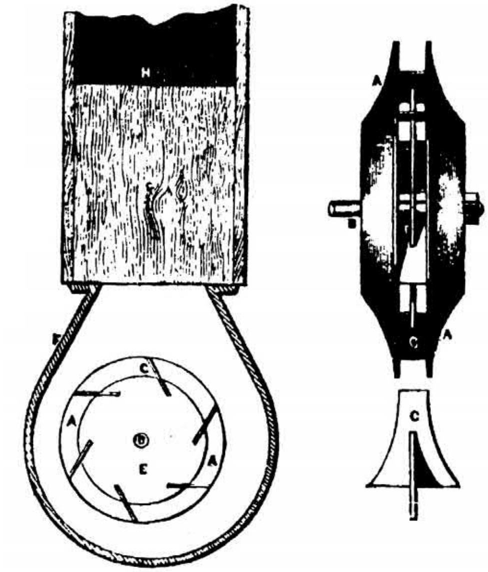


**Fig. 2.5** The APPOLD pump of 1850, begins to resemble some current designs. The advantages of straight non-radial blades seems to have been understood. Doubtless this contributed to the much improved efficiency of the machine. *(Reproduced from E. Grist, Cavitation and the Centrifugal Pump-A Guide for the Pump Users. Taylor and Francis, 1998.)*

significant axial loads on the shaft/rotor system. But by arranging a mirror image impeller back-to-back, these loads can more or less cancel out leaving the rotor in a state of almost complete axial balance.

Developments after Appold followed an evolutionary path. Not a great deal of this was applied to impeller flowpath analysis. In fact, machines of very high efficiency were evolved based chiefly on empirical and statistical hydraulic analysis. This despite a poor understanding of the actual internal flow conditions.

In the 1930's Buseman [2] attempted to predict the inertia corrections that Euler was unable to include. This was termed a 'slip factor'. Others had proposed such a concept, but the Buseman slip factor was probably the most widely accepted among pump designers.

Evolutionary development continued, but received a significant boost when pumps were used in aircraft fuel systems, or more specifically military aerospace systems. Here the incentive was to increase the 'output-to-weight' ratio. This required of course better understanding of material sciences as well as of mechanical design. But it also made stronger demands on our knowledge of the flow physics inside pumps. This initiated some advanced research, much of which trickled down to industrial pumps at all levels.

The pumps of the Saturn rocket, used to launch the Apollo space shuttles, give a useful snapshot of the current pump design frontier (Figs. 2.6 and 2.7).

**Fig. 2.6** General view of the SATURN rocket engine, showing the Liquid Oxygen pump.

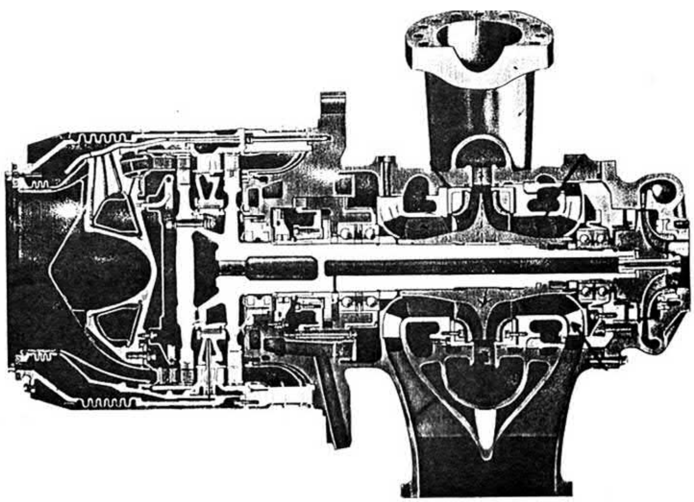


**Fig. 2.7** Section of the Liquid Oxygen [LOX] turbopump, by Rockwell International. This extremely powerful machine is only about 2.5 feet long [0.75 m].

**Table 2.1**

| Machine | Year | Flow (USGPM) | Head (FEET) | Efficiency | Power (BHP) | Speed (RPM) |
|---|---|---|---|---|---|---|
| Appold | 1850 | 1483 | 19.4 | 68% | 10 | 788 |
| LOX pump | 1978 | 6830 | 8590 | 78% | >17,000 | 28,000 |

The Saturn engine creates huge amounts of thrust. In simple terms, this is achieved by combusting kerosene with liquid oxygen in a conical chamber or nozzle. The temperature in the combustion nozzle can be very high indeed. To help combat this, the walls of the chamber are cooled by pumping the liquid oxygen propellant through conduits that line the walls of the combustion nozzle.

The design of, say, the Liquid Oxygen pump forms an interesting contrast with Appold's Great Exhibition machine (Table 2.1).

This is an impressive comparison in showing how far pump technology has advanced. However, the design life of the LOX pump could be measured in hours. Appold's machine would have been expected to last many times longer!

## Common myths associated with centrifugal pumps

Since the centrifugal pump has become so ubiquitous, many myths have grown to surround them. I list some of them here;

- Pumps pull [or suck] liquid towards themselves.
- In any pumping system problem, the pump is always guilty until proven innocent.
- Flexible shaft couplings can tolerate a great deal of misalignment [the makers say so!] so care in alignment is not necessary.
- The final adjustment to soft packing shaft seal [where fitted] can be accomplished within a few moments of start-up.
- A poorly primed and vented pump will always clear itself after a few moments.
- All pumps are intended to act as pipework anchors.
- Soft packed shaft seals can tolerate much more abuse than mechanical seal.
- Any pump can be successfully installed in any system and it will still always deliver its design flow.
- The pre-start checks have already been done, by someone else.

- The suction isolation valve is never NORMALLY left closed.
- The factory has already done the motor—to—pump shaft alignment. and it does not need to be checked.
- Some sense of a pump's criticality can be gained from its power consumption. Small pump are never important and are not worth worrying about. They are incapable of causing significant disruption.
- There is never time to check everything properly, but always time to correct any subsequent damage afterwards.
- Two pumps operating in parallel will deliver twice the flow of one pump.
- Published NPSH [R] curves show the conditions under which a tiny amount of cavitation is present in a pump. By providing a even a small margin over this curve, cavitation damage is completely avoided.
- Impellers can only ever be installed one way
- Minimum Continuous Safe Flow is another word for Minimum Flow. Both refer to the low flow point at which the pumped liquid temperature begins to rise due to churning.
- Pumps with unstable curves can never run properly.

Over the years, I have found that a few empirical rules can be applied to pump troubleshooting. They might help others.

## Palgrave's rules

### First rule

Over time, a pumps performance characteristic will not normally improve. If it appears that this rule is being violated, then there are some unusual circumstances behind this.

### Second rule

When pump performance decays, the slope of the characteristic curve increases [the curve becomes steeper]

### Third rule

The pump is always guilty until proven innocent.

## Fourth rule

Pumps are never purchased for the real duty requirements. There is always a contingency.

## Fifth rule

The degree of owner respect is directly proportional to the absorbed pump power.

# CHAPTER 3

# Centrifugal pump basics; part 1

## Introduction

Firstly, a clarification: This book is entitled Troubleshooting Centrifugal Pumps and their Systems. A large part of the pump population can be described as rotary machines, as opposed to, say, reciprocating. Of the rotary segment, some have a positive displacement volume, such as gear pumps, lobe pumps and the like. Some do not have a positive displacement. These have been termed rotodynamic pumps. As we shall see, this includes pumps where, at one extreme, the flow approaches the rotor axially, but is turned to a radial direction as it has work done on it. These are termed Radial Flow machines. At the other extreme, the flow also approaches axially but leaves more or less axially. These are often called propeller pumps, or more exactly Axial Flow machines. Between these two extremes are machines where again the flow approaches axially, but leaves at some angle between radial and axial. These are known as Mixed Flow or diagonal flow pumps. The term centrifugal pump strictly relates to the first group only, those where the flow leaves radially. In these cases, centrifugal forces are by far the dominant contributor to the way in which pump pressure head is created. But this is not the only contributor. At the other extreme, in cases where the liquid leaves the impeller axially, it does so at the same radial distance from the shaft as when it entered. With no change of radius, there can be no obvious centrifugal force. Instead the pressure head is generated almost entirely by a different mechanism. In the case of mixed flow pumps, the contributions from both mechanisms might be comparable.

In fact, most practical members of this part of the pump family generate head from combinations of these two mechanisms. However, the term Centrifugal Pump has come to describe any member of the family (Fig. 3.1).

The main purpose of almost every centrifugal pump is to create flow in a liquid system. To do so they have to simultaneously generate pressure

*Troubleshooting Centrifugal Pumps and Their Systems*
ISBN 978-0-08-102503-1
https://doi.org/10.1016/B978-0-08-102503-1.00003-7

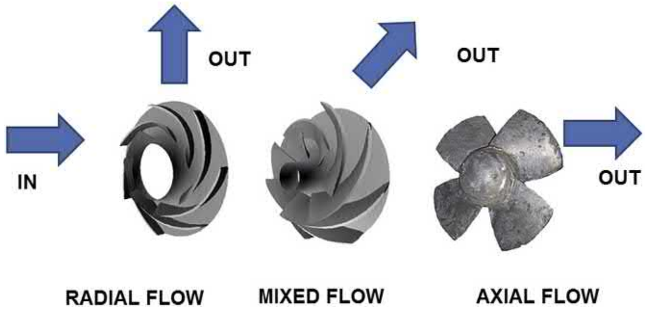



**Fig. 3.1** The general appearance of rotodynamic pump impellers.

head in the liquid. This pressure head is then dissipated by the liquid system resistance while handling the above-mentioned flow. Though the pump still has to generate pressure head as a by-product, its chief purpose is to create a specified system flow. On very rare occasions, centrifugal pumps are installed for the main purpose of creating hydrostatic pressure. But generally, other classes of pumping machine are better suited to this aim.

Compared to many other machines, the inner workings of a centrifugal pump are largely invisible, except in certain laboratory research pumps. This means that the underlying physics are less than obvious. The detail physics of the liquid flow path are well covered elsewhere [3, 33], but at a level inappropriate to this volume. Therefore, a simpler description is given here. I hope this helps to generate a deeper general insight into the inner workings.

Practical centrifugal pumps consist of three distinct and important hydraulic design zones (Fig. 3.2).

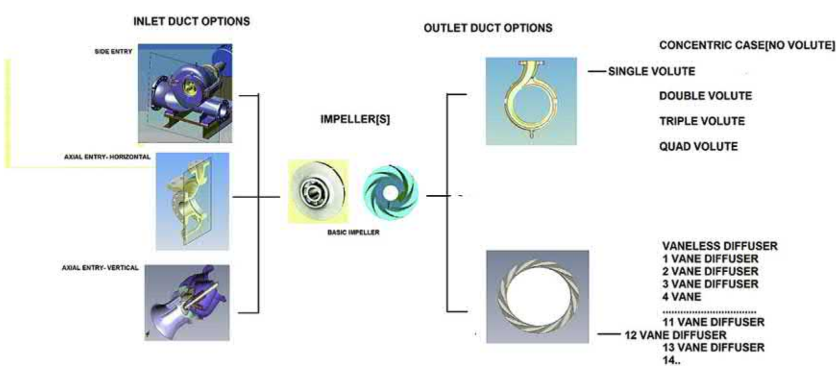



**Fig. 3.2** The three key hydraulic design areas in a centrifugal pump.

- An **INLET DUCT** that leads liquid to the impeller and from the source. Most often this will connect to system pipework. With wet pit/vertical shaft pumps this duct will usually be submerged in the liquid and have little or no associated pipework.
- An **IMPELLER**, which does work on the liquid.
- An **OUTLET DUCT**/collector which brings together liquid leaving the impeller and leads it towards its destination. Usually this is also via system pipework.

## Inlet duct

This leads liquid from the connection with the external pipework system up to and into the impeller entry. Three different configurations are common.

## Axial approach; closed

The most common form of inlet duct leads the liquid along the shaft axis and towards the impeller. There will be associated connecting pipework. This might appear the most elegant arrangement since it creates the possibility of uniform axisymmetric flow, without rotation/swirl being presented to the impeller. However, in most practical cases, there will be elements upstream of the impeller such as valve or bends or strainers that impact on the flow picture. These distort the otherwise uniform flow. If sufficiently distant upstream, say 15 pipe diameters, this effect can be negligible (Fig. 3.3).

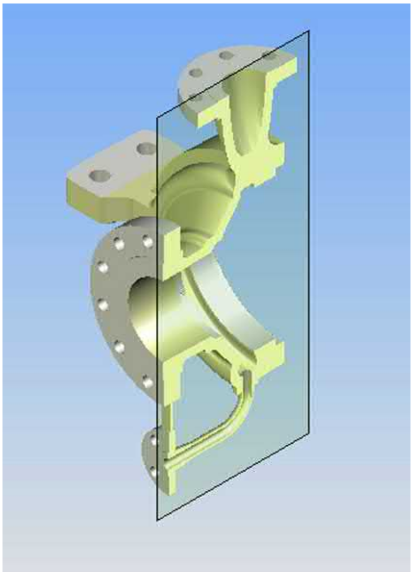


**Fig. 3.3** Axial flow approach to the impeller. This is by far the most common layout.

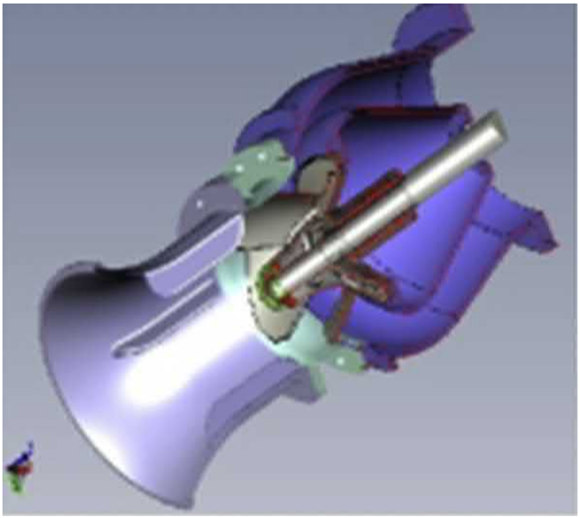


**Fig. 3.4** Axial impeller approach when the pump shaft is vertical. As shown the inlet bell mouth has no flow restrictions. Sometimes a form of strainer is installed.

## Axial approach; open

In some cases, the pump shaft axis is vertical. Here the inlet duct is also axial but open and unpiped, other than a bell mouth and/or strainer. Typically, this class of inlet duct is submerged in an open volume [sump] of liquid. This plan is often attractive because it means that with some thought, the pump can be automatically primed (Fig. 3.4). At first sight, this layout might seem to also offer uniform axisymmetric flow. However, for this to be true, a great deal of thought has to be put into the configuration of the surrounding sump, including its flow entry arrangements. In poorly thought-out sumps, the liquid approaches the inlet duct with some degree of rotation/swirl imprinted onto the flow picture. The pump will rarely have been designed or tested with such conditions in mind and so there may well be a performance departure on site.

## Side/radial approach; closed

Axial approach ducts are most often related to machines in which the impeller is overhung from the bearing system. From a hydraulic design standpoint, this configuration offers the hydraulic designer the best design choices (Fig. 3.5).

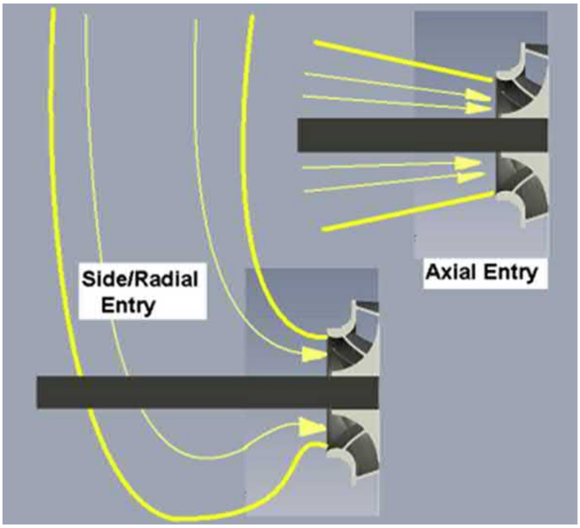



**Fig. 3.5** In the side/radial configuration, the flow approaches at right angles to the shaft, then turn to an axial direction before entering the impeller inlet annulus.

However many machines can only be successfully configured with the impeller[s] 'Between-Bearings'. In this case, axial approach ducts are difficult to arrange. By far the easiest solution to this is to then configure the inlet duct radially and perpendicular to the shaft axis. In fact, this layout often coincidentally produces pipe connections that are particularly attractive to certain types of plant design. However, turning the liquid from a radial to an axial direction inevitably loses some flow symmetry. This dilemma has been traditionally approached in two contrasting ways (Fig. 3.6);

- In the first approach, considerable effort is devoted to flow management within the inlet duct such that something approaching axisymmetry is eventually presented to the impeller, but without any significant rotation/swirl. Great care is demanded for this to be successful and simultaneously avoid stall cells in the flow.
- In the second approach, flow management results in deliberate and significant amounts of rotation/swirl being induced into the flow picture presented to the impeller. This is a less elegant method, but aims to sweep away any static stall cells that might otherwise form in the inlet

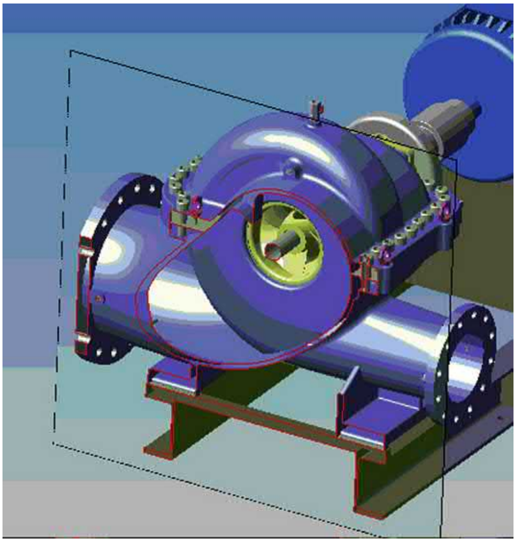


**Fig. 3.6** Side/radial entry arrangements can combine with certain outlet duct designs to give a connection arrangement that is often attractive to pipe layout designers.

duct. Since the impeller will have been designed to accept such swirl, the only penalty incurred is that the impeller diameter, and hence the machine as a whole, will be need to be slightly larger. On the plus side, the NPSH[R] should be a little less.

## Impeller

We will later discuss the impeller, how it works, and its various executions.

## Outlet duct/collector

Collects the liquid emerging from the impeller, gathers it and directs it to the connection with the external pipework system. Again, two common forms exist each with sub-divisions.

## Volute collectors

- The simplest collector consists of a concentric chamber arranged around the impeller rim (Fig. 3.7). This does not comply with the strict definition of a volute, but it has nevertheless become known as a 'concentric

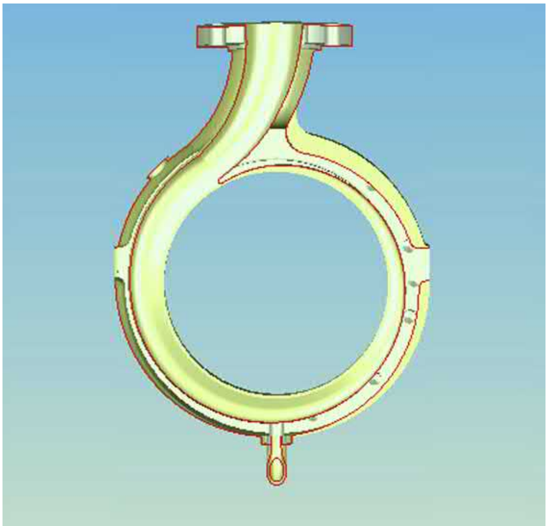


**Fig. 3.7** Collectors with volute spiral passages are most commonly employed.

volute' It is most often applied to very simple pumps and/or to those of very low design flow coupled to high-pressure head per stage. It is known to produce hydraulic radial loads on the shaft system that are lowest at or near to zero throughput and then increase in step with flow.

- By far the most widespread arrangement is where the collection chamber takes the form of a single volute spiral. Favoured for its simplicity, it is known to induce radial loads on the shaft system that are lowest at design flow.
- A more elegant approach opposes two-half volute spirals, which results in an almost complete cancelation of any induced radial loads. These are known as Double Volutes.
- In rare cases, usually vertically shafted cantilever shaft pumps, a triple volute spiral is included. The 'tripod' effect of three radial forces provides somewhat of a self-centring effect. These can be considered Tri-Volutes
- Certain specialised pumps employ what is called a Quad-Volute. Looked at in abstract, these designs are hard to differentiate hydraulically from a four vaned crossover diffuser. They represent a hybrid approach.
- Later we shall see that each impeller carries with it a rotating flow field. Simplistically speaking, this resembles a repeating saw tooth pattern and results from the way in which flow is obliged to segregate inside each vane-to-vane passage space (Fig. 3.8).

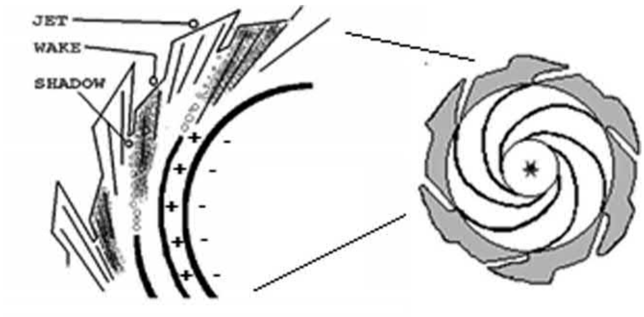



**Fig. 3.8** The flow pattern that leaves a typical impeller can be idealised as a high velocity 'jet' and a lower velocity 'wake'. A turbulent 'shadow' trails from the vane end.

As the impeller rotates, this uneven flow pattern cyclically encounters fixed parts of the collector, and pressure pulsations are created in the liquid. With a single volute, there will be only one encounter per blade per rev. With a double volute, there will be two encounters per blade. With a triple volute three encounters and so on. Most volute collectors are of the single or double volute class. So normally there will be a significant difference between the impeller vane count and that of the volute. This means that the pressure pulsation frequency will be relatively low, though there may be higher frequency harmonics of course. If both the impeller and collector vane count are of an even number, then simultaneous encounters occur, producing an additive effect. If the vane count of one component is even and the other odd, then a rotating pattern of encounters occur-the direction depending on which has the greater number (Fig. 3.8).

- Volutes have a minimum/cross sectional area called the volute 'throat'; A3. The total area of all throats will play a large part in deciding the optimum flow rate of the pump. All other things being equal, increasing this cross sectional area will **tend** to increase the pump flow.
- The throat control effect is greatest in pumps where the impeller Total Impeller Outlet Area between Vanes OABV is large compared to the Total Casing throat area A3. *[Say Area Ratio of 1.5 or more].* The first pumps built by Denis Papin [and described in Chapter 2] fell into this category.
- Where OABV is small, relative to throat area A3, *[say Area ratio of less than 1.5]* then performance flow control rests more with the impeller (Fig. 3.9).

Whatever forms the volute collector takes; its collective passage will usually join and flow seamlessly into the discharge duct.

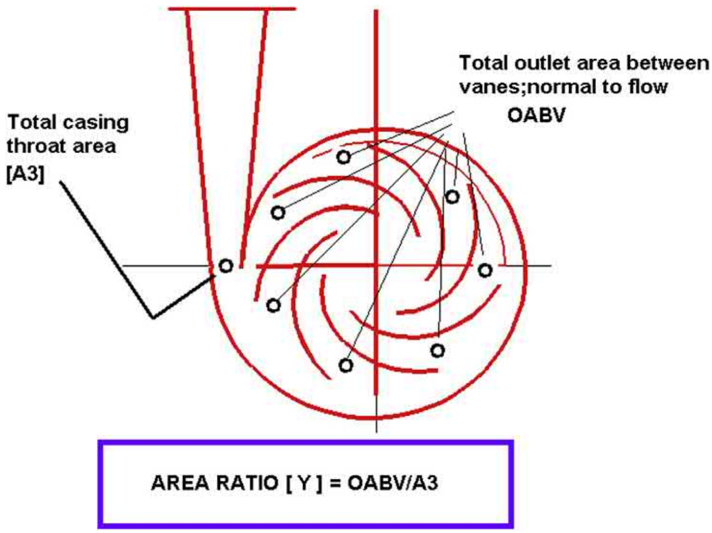



**Fig. 3.9** The ratio of **total impeller passage outlet area** to the **total collector throat area** is an important parameter in some pump design approaches. It is widely known as the **area ratio** of a pump. This definition applies equally to Multiblade collectors as well as volutes.

## Multivane diffuser collectors

- Centrifugal compressors can sometimes incorporate what is called a 'vaneless' diffuser. In such designs retardation of the flow leaving the impeller is accomplished chiefly by the flow spiralling radially outward and slowing in the process. Such designs are rarely if ever deployed in pumps, though the concentric volute principle shares some similarities. Multiblade diffusers have been found to be more cost effective (Fig. 3.10).
- With Multiblade diffuser machines the impeller will be surrounded by an array of blades, which are set at an angle closely corresponding to that of the average flow leaving the impeller rim. As with volutes, there will be a total/sum throat area that plays a large part in deciding the optimum flow rate of the pump. All other things being equal, increasing this total cross sectional area [A3] will increase the pump flow.
- Though diffusers with two or three vanes are known to exist, the vast majority employ vane counts, which exceed four and often significantly exceed the vane count of the impeller. The periodic interaction between the discharging impeller saw tooth flow pattern and diffuser/volute vane inlet tip is known to produce important forces. This effect

**Fig. 3.10** Multiblade diffuser design. Blades are set to match the fluid angle leaving the impeller at the pump's design flow.

has been widely studied and the choice of vane count is no longer an arbitrary design decision. It turns out that having a diffuser/volute vane count equal to that of the impeller is the poorest choice from a pressure pulsation point of view. That is because non-uniformities in the impeller flow field interact with the diffuser/volute vane simultaneously with a constructive/additive effect. The impeller flow field 'cogs' or 'ratchets' as each vane is passed. The next least favourable configuration is to have a difference of one between impeller and diffuser/vane counts. Although the impeller/diffuser encounters are not now simultaneous, there is instead a rotating pattern of encounters. These rotate forward or backward depending on which component has the greater number of vanes. With even greater vane count differences, the encounter intensity tends to dilute.

- In one or two stage machines-generally between bearings configuration, it is common to encircle the diffuser with either a concentric chamber or a passage closely resembling a volute.
- Should the machine have three or more stage, then flow leaving the diffuser will be normally be guided to the next stage inlet via a set of Return Guide Vanes [RGV]. Their purpose is to reduce the liquid

swirl before it enters the next impeller. In some cases, the RGV are designed to almost totally remove the embedded liquid swirl. In others, they are designed so that some remnant swirl exists as the flow enters the next impeller. Zero swirl machines will be slightly smaller and generate slightly more pressure head, whereas those with finite swirl make it easier for the designer to achieve a constantly falling performance curve.

- Diffusers are known to result in lowest radial shaft loads at design flow.
- Generally, Multiblade diffuser are separate components, so it is easiest to subsequently exchange the collector hydraulic characteristics than with fixed volutes.

## Impeller basics

The first two discussed elements are of course important to the functioning of a pump. But of the three, the impeller is at the heart of the matter. Therefore, I want to generate a mental picture of the flow conditions inside a pump in general and the impeller in particular.

To do this I begin with a brief description of vortices. This might seem odd at first. However, a centrifugal pump impeller is essentially a vortex generator. The vortex size and strength governs the pump flow and generated head. However, this particular genre of vortex is not unique in pumps. In fact, many sub-members of the vortex family occur in pumps. They can be grouped into two main types; Free and Forced vortices. In a free vortex, the liquid velocity decreases with increased radial distance from the centre of rotation. In a forced vortex, the liquid velocity increases with radial distance. For the purpose of this simple explanation, the internal impeller flow path can be described as a forced vortex.

Imagine a cylindrical drum with a relatively thin/shallow depth of liquid in it. The drum is arranged in such a way that it can be spun around its axis at various angular velocities (Fig. 3.11).

When at rest, the entrained liquid surface is essentially flat.

Now imagine that the drum is spun around its axis, slowly at first. Viscous friction forces between the liquid and the inner walls of the drum will start to drag along the layer in immediate contact with it. *[For the moment ignore any viscous drag effects from the circular base of the drum].* With time, this viscous drag is transmitted to the next inner layer and so on until the whole contained volume is rotating as a forced vortex. It will be clear

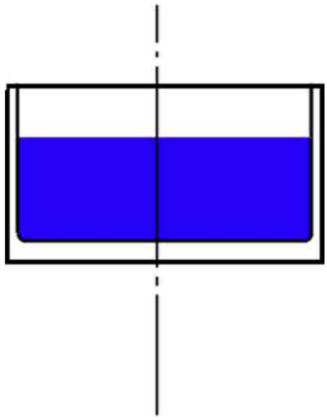


**Fig. 3.11** Stationary drum-liquid filled.

that with this mechanism, the liquid volume at the drum inner wall can never rotate faster than the drum. It has lower angular velocity. This phenomenon is generally called ‘slip’ Due to this ‘slip’ effect, the angular velocity of the inner layers gets progressively lower as the axis of rotation is approached.

Furthermore, the previously flat liquid surface will now adopt a curved profile shape. A similar but slightly different effect takes place when liquid in a cup or beaker is stirred with a spoon. Liquid nearest to the cup wall will be most elevated relative to liquid at the centre (Fig. 3.12).

Imagine that the rotational speed of the drum is increased. The liquid layers in contact with the drum wall will also rotate more quickly, but due to the ‘slip’ mechanism, can still never achieve the same levels of speed. The same will be true for all the other liquid layers.

At the same time, due to the increased speed, the liquid surface will adopt an even more curved profile. The point here is that the liquid surface will become more and more curved as the speed of rotation/angular velocity increases. In fact if we carry on increasing the speed of rotation, then a point will be reached where the liquid level at the drum wall can almost, but not quite equal the height of the drum rim. For any further speed increase, the liquid will spill over the rim. Intuitively a low drum rim will require less speed to reach this spill over condition than will a higher rim. To be more correct, it is actually a combination of drum speed and drum diameter that will define this spill-over condition. A small drum rotating at a higher speed can achieve the same spill-over height as a larger drum

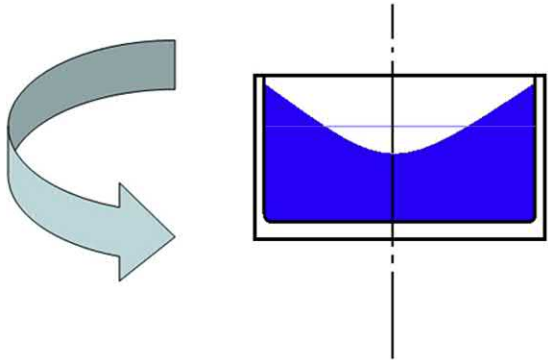


**Fig. 3.12** As drum spins, the entrained liquid eventually follows, but at a lower speed. The initially flat liquid surface now adopts a parabolic shape.

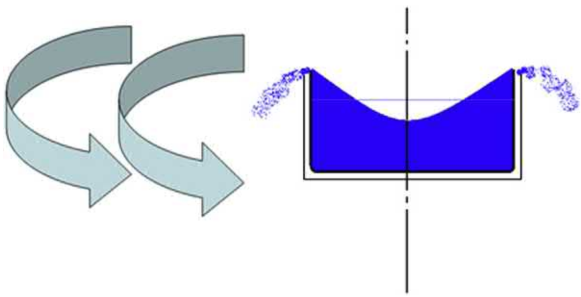


**Fig. 3.13** At a certain speed the entrained liquid may spill over the rim of the drum.

spinning more slowly. Running the mental experiment at a slightly higher speed, then clearly a condition can JUST be reached where sufficient liquid has been spilt to lower the entrained volume and so start to empty the drum (Fig. 3.13).

Simplistically, this description of the flow conditions in the drum also applies to centrifugal pump impellers when no flow passes through them. We shall see later that actually, true zero impeller flow can rarely if ever be actually accomplished in an impeller without special measures. There is always a small flow passing *[through the wear rings for example]*. But for the moment, this is irrelevant. In the next part of our mental experiment, imagine further modifications to the previous arrangement.

Firstly, the rotating drum has a small hole pierced in its base. The diameter of this hole is not critical. But to give some sense of proportion, it will lie in the range of *[say]* D/5 TO D/3, where D is the drum inner diameter.

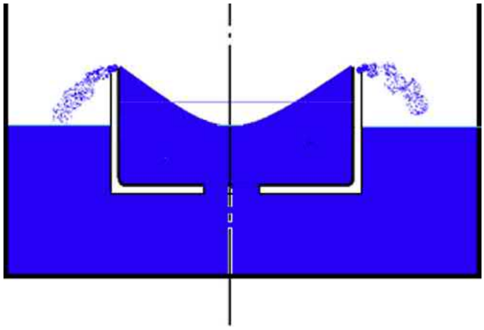


**Fig. 3.14** Second drum may be arranged to catch and circulate the rim flow.

Secondly, the newly pierced rotating drum is positioned inside a larger static vessel, which is co-axial with it. The whole apparatus is filled with liquid to the same level as before (Fig. 3.14).

The inner drum is again spun at just below the spill-over speed. As before, the liquid height just fails to exceed the height of the drum. But once the speed is further increased, something interesting happens. As before, liquid spills over the rim. But unlike before, the entrained volume in the drum is not depleted. As liquid spills over the rim, it is replaced by fresh liquid that pass through the central hole of the rotating drum. The entrained drum volume stays more or less the same.

If speed is increased, further, then more liquid spills over the rim, to be simultaneously replenished through the central hole. We are now starting to build the hydraulic elements of a true centrifugal pump, and how it works.

The last stage of this mental experiment is to imagine further modifications to the apparatus. A source of liquid is connected to the stationary vessel base. Then a gutter or rim is formed around the inside of the stationary drum. This rim fits very closely to the drum outer, but does not quite touch it. The conduit thus formed leads to some destination point that is distinctly separate from the source (Fig. 3.15).

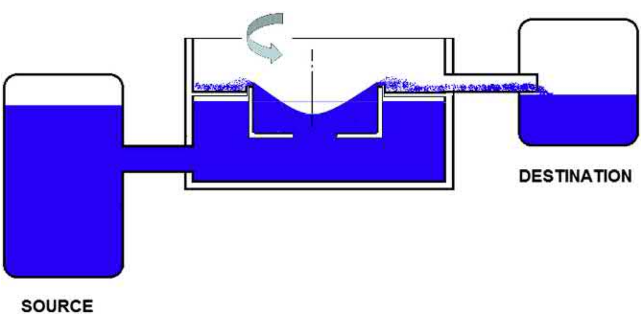



**Fig. 3.15** Drums may be configured to produce a continuous pumping effect. The destination surface is slightly elevated relative the source.

Now the apparatus is run at a speed exceeding the spill over speed. Some liquid continues to leak and recirculate through narrow annulus between stationary and rotating drum. But most will pass away on to the destination vessel. Of all the liquid passing through the rotating drum, most will be arriving from the source and passing to the destination. Only a small amount will be arriving through the annular clearance around the rotating drum. For practical purpose, this latter liquid can be considered as continuously recirculating over the rim and back again. It adds nothing to the drum output and is analogous to the internal leakage flow of real pumps.

This describes the basic hydraulic operation of a true centrifugal pump. The generated head of such a simple pump is closely related to the difference in levels [but less so once flow takes place!]. Pumps operating on this simplified principle actually exist. One such example, from the late sixties, is shown [4] (Fig. 3.16).

Here the rotating drum is in the shape of a tapered cone. Unlike most conventional pumps, this drum [impeller] does not possess any blades or vanes. That is because the chief purpose of this particular pump is to lift water a small distance from one river water canal to another close by. This water may contain live fish, which would otherwise be damaged by the vanes normally employed in impellers.

In this pump, as in our mental example, the energy is transferred to the liquid from the drum and its power source, via viscous friction at the drum surface. In fact there is a whole subset of centrifugal pumps that use this principle. Their invention is often accredited to the scientist Tesla, after

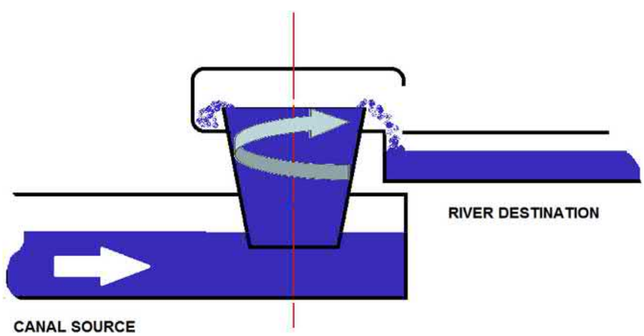



**Fig. 3.16** Elements of a cone flow pump. One aim of this design was to pump water containing live fish and to do so without injury.

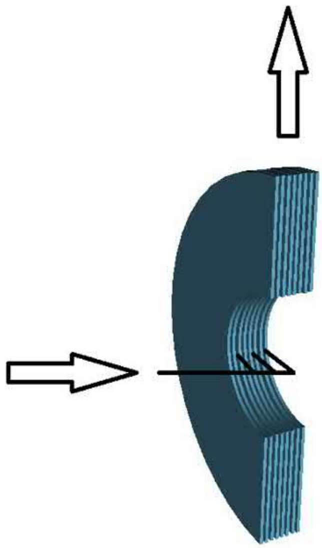


**Fig. 3.17** Conceptual illustration of a Tesla vaneless impeller. Several closely spaced disks are stacked together. The viscous drag forces created at each disk surface generate a radial flow pumping action.

whom the unit of magnetism is named. In this version, a series of closely spaced radial disks provide the viscous driving force (Fig. 3.17).

[*Keen eyed readers will notice that in our subject mental image pump, there is also bound to be a contribution from this radial disk flow made by the base of the drum. But I have for simplicity I ignored this].* The Tesla principle was later employed by Barske [5] in early aircraft engine fuel pumps and later by the Discflo Company in more commercial pumps. Even though the vortex strength is lower than an equivalent vaned impeller, the absence of any vane obstruction can prove helpful when the approaching liquid contains abrasive particles, or is at close to its vapour pressure. This design avoids the 'tearing action' of the blades on the liquid. However, most practical centrifugal pumps employ vanes or blades to improve the energy transfer process (Fig. 3.18).

The addition of vanes to the impeller/drum is a significant step, with some huge advantages. Without them, the maximum liquid angular velocity/rpm [at the drum wall] are much lower than the drum itself. It might achieve perhaps 10% of the drum velocity. Moreover, this velocity will rapidly diminish at smaller radii from the axis of rotation. We already know that the head generated by the impeller is related to the strength of the

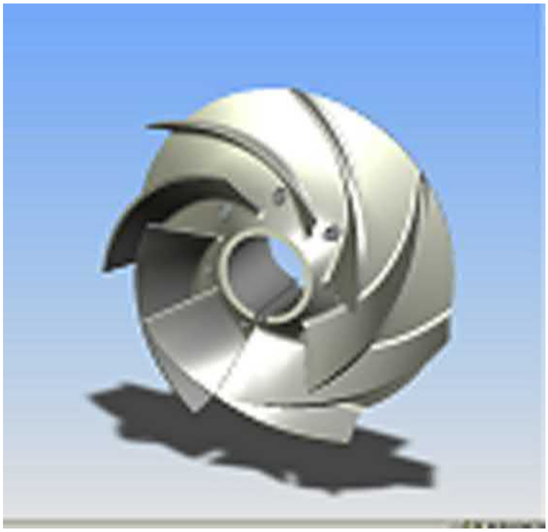


**Fig. 3.18** Most impellers are equipped with pumping vanes to improve pumping action. Some vanes curve in only two dimensions, others [like this one] have a three dimensional surface.

vortex, as described by the [Maximum-to-Minimum] difference in the liquid surface profile. This difference in liquid levels is known as the static lift and is an important concept in pump and system definitions. Anything that improves this vortex strength will improve the effectiveness of the machine.

The action of impeller vanes oblige almost all the entrained liquid to now rotate at a more nearly constant angular velocity/rpm. This establishes a much stronger vortex and thus a capability of generating a much higher pressure head than an unbladed impeller.

As useful as vanes are, early centrifugal pump makers did not yet understand the importance of the vane shape. Radial 'paddle' vanes were initially chosen. This category has some unwelcome properties (Fig. 3.19).

As the flow enters the impeller radial paddle vanes, it will suffer high losses. This is chiefly because the radial setting angle of the vane departs significantly from the highly tangential approaching flow-path angle. Some flow separation is very likely. In addition, the individual flow ducts formed between each vane will probably diverge too much, causing extra losses. High speed turbo compressor impellers usually employ radial vane, but that is because non-radial blades would experience unacceptable bending stresses in operation. Radial blades are a necessary compromise here.

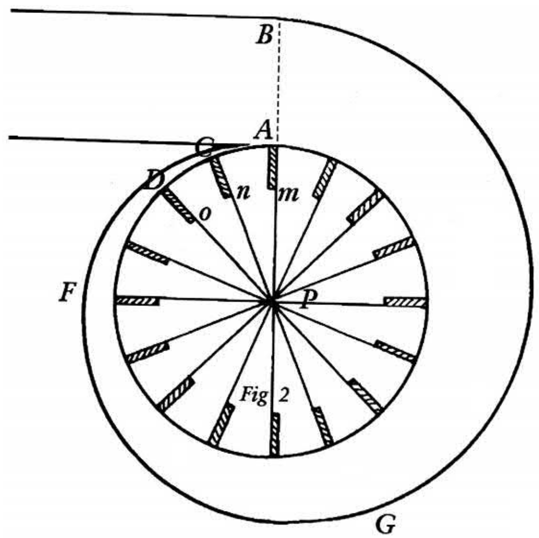



**Fig 3.19** Early PAPIN design of 1705 was equipped with simple radial 'paddle-type' vanes. The concept of a volute collector had already been included.

However in one part of the pump impeller family, such radial paddles are acceptable. This is in pumps where at a given speed; the flow is relatively small, and the stage pressure head relatively high. Such impellers are termed very low Specific Speed [Ns] designs. Because the internal impeller flow velocities are also relatively low, so are any impeller losses. In such configurations, the collector passages contribute the bulk of the hydraulic losses, the impeller contribution being much smaller.

Early pumps employed radial paddles, but by the time of Papin the general importance of more refined vane shape does seem to have been appreciated. We start to see what are called 'backward leaning' vanes, which are vanes shapes that lean away from the direction of rotation (Fig. 3.20).

Modern pumps have vanes that are mostly curved in very subtle ways. They push the liquid about in a more gentle and controlled manner than that of paddle vanes. Small or simple pumps have impeller vanes that only curve in two directions. Large and or more sophisticated machines use impellers which curve in three directions. Impellers with three dimensional curvature are, by and large, more efficient than those with two dimensional curvature. However, this difference diminishes with reducing size, so at some point, the advantage is largely lost or becomes insignificant. However, two dimensional vanes are a lot easier to manufacture so below this cut-off point the advantage lies with them.

**Fig. 3.20** Modern impellers utilise backward curved blades for more efficient liquid guidance. This impeller blade surface has only two dimensional curvature-unlike Fig. 3.18.

Among the important dimensions of any impeller vane systems are (Fig. 3.21);

- The minimum diameter [D1] inscribing the vane
- The maximum diameter [D2] inscribing the vane edge- [$D2_{RMS}$ for mixed flow Impellers]. Often referred to as the 'eye'
- The passage height, [b2] of the vane at outlet. *[Actually, b2 is just a convenient analogue of the total outlet area between [OABV] vanes measured normal to flow]*

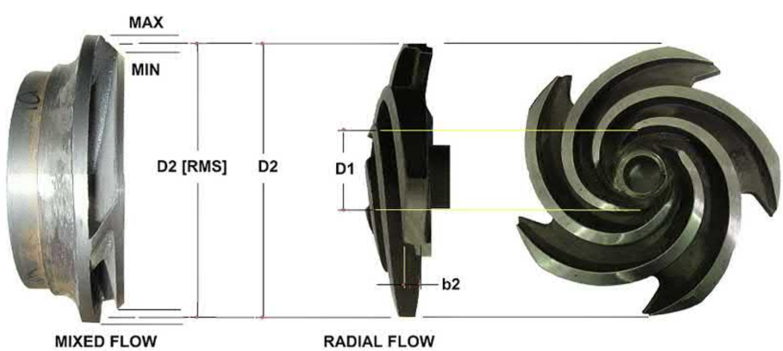



**Fig. 3.21** Key hydraulic dimensions of a centrifugal pump impeller.

It might be guessed that the inner and outer diameter of the vane system is influential to the strength of the impeller vortex. This strength helps define in some way the pressure head generated by the impeller.

Without delving into the physics of pump performance, we can simplistically say that, all other things being equal.

- Pump pressure head is closely related to the maximum diameters, *[Root Mean Square [RMS] diameter for mixed flow pumps]*
- For a given mean of the maximum impeller diameters [D2], pressure head reduces as the minimum inscribed diameter [D1] increases.

But it might not be as obvious as to what part the vane height at outlet plays in the performance picture of the impeller. In truth, the important factor is the outlet area between adjacent vanes OABV. But for a given vane spiral shape this is proportional to b2. Since most impeller vane spiral angles lie in a rather narrow range, the passage outlet width, b2, can be used as a more easily measured or visualised analogue of OABV for discussion purposes.

In practice, b2 [or OABV] is a major factor in determining what volumetric flow the impeller has the potential to create, all other things being equal. To understand this it helps to consider a centrifugal pump as a displacement machine. Not in the sense of a positive displacement pump where a definite trapped volume has work done on it. In a centrifugal pump, that trapped volume is less precise (Fig. 3.22).

In Fig. 3.22, a simple impeller is shown in three progressive positions of rotation. In position 1, a particle of water lies at the smallest radius of the vane system. As the impeller rotates to position 2, the vane displaces the particle outward with a radial component. At position 3, the particle has continued its radial displacement and out into the collector. The actual particle flow-path as it moves from position 1 to 3 is dependent on many minor factors. But clearly, if two otherwise identical impellers are compared, the one with the greatest vane height [and hence greatest

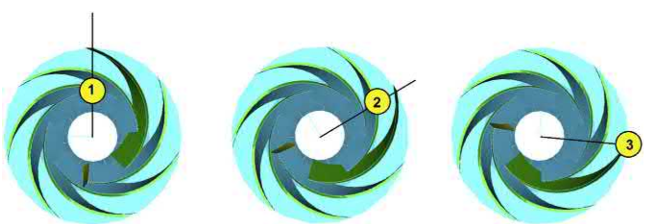



**Fig. 3.22** Through-flow particle position at different instants of time. The particles are displaced radially outward by the rotating vane.

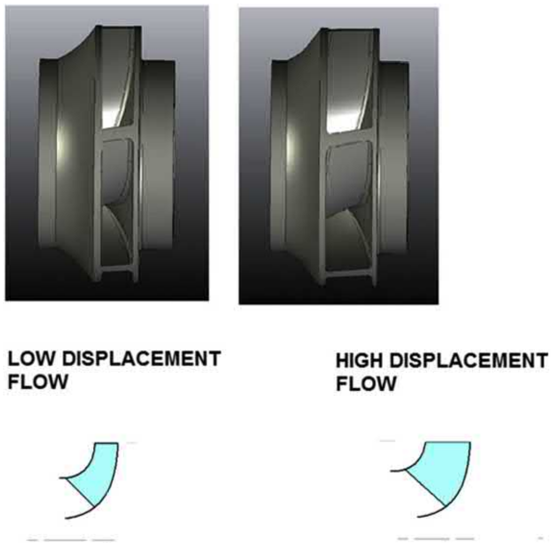



**Fig. 3.23** Impeller outlet widths affects flow potential.

OABV] is potentially capable of displacing more particles than the other displaces. It has the greater flow potential (Fig. 3.23).

We already know that an impeller pressure head is very much influenced by its maximum outlet diameter. So combining these two, leads to very general design point trends that include (Fig. 3.24);

- Impellers of similar Maximum outlet diameter will, when delivering their design flow, tend to generate similar pressure head at a given speed. *[When delivering zero flow, these same impellers will generate an even more similar pressure head]*

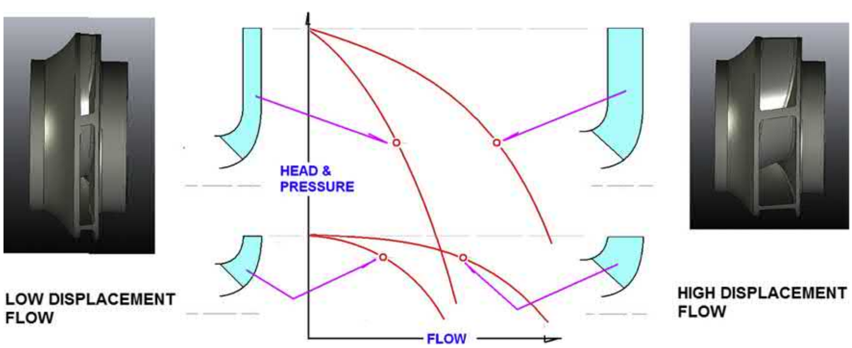



**Fig. 3.24** This attempts to illustrate how different changes to the overall impeller geometry can affect performance.

- Impeller displacement flow increases with increased swept area. So impellers having a similar Maximum diameter will have design flows that are related to their swept area, The passage outlet width, b2, is a close analogue of the vane-to vane area at outlet [OABV]

I have described how as the impeller and its vanes rotate, they displace liquid outwards and into the collector. But an important question is, 'as the particles are displaced outwards, how are they replaced by new ones?'

It has been quite common to say that a pump 'sucks' fresh liquid inwards and indeed the hydraulic ducts that accomplish this replacement are often described as 'pump suction' connections. In fact, most liquids have little if any tensile strength and are incapable of being 'pulled' into the pump. It is actually the pressure acting on the surface of the liquid source that provides the driving force for replenishment. How? (Fig. 3.25).

Well, in radially displacing liquid, the pressure increases on the forward convex face of the rotating vane. There is a corresponding pressure decrease on the concave backward facing face. This pressure difference, acting over the vane surface areas, is responsible for the shaft torque that the driver has to overcome.

It is the difference between the low-pressure zone on the vane concave surface and the pressure acting on the source surface that **pushes** the liquid into the impeller. In successful pump systems, this phenomenon will want to push liquid into the impeller faster than the impeller can displace it

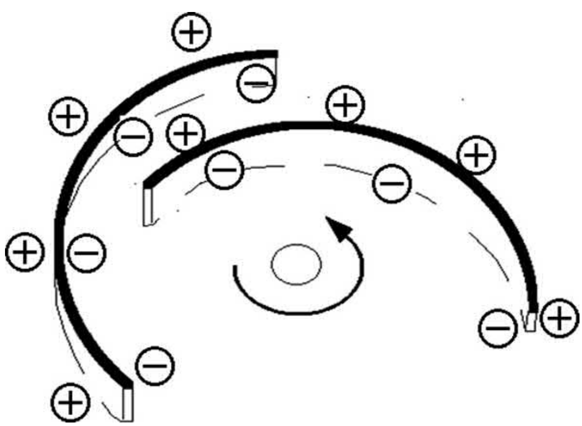


**Fig. 3.25** Simplified pressure distribution on an impeller vane. The differential between concave and convex surfaces causes a torque that the driver must overcome in order to do work on the liquid. The difference tapers-off at inlet to and exit from the vane.

outward. When this is not the case, that is the impeller wants to displace more flow than the source pressure can provide, then a phenomenon called cavitation occurs. Vapour bubbles will form at the lowest pressure point on the flow-path. This phenomenon may be mechanically damaging as well as harmful to pump output

Returning to the different 'shapes' of pump in general and impellers in particular; I have touched on how either generated pressure head, or displacement flow are broadly related to the impeller shape. Let me discuss now these two pump performance properties, head and flow, are measured and related. In this respect, the pump is now simply considered a 'box', where the shape and size of the internal components are irrelevant. We shall see that though the performance characteristic of a pump is important with regard to how a fluid handling system responds, it is not the only factor. The characteristic of the flow resistance put up by the system itself is as important to the outcome. Let's first deal with the pump characteristic.

## 'Black box' performance characteristic

For any pump, one of its most important documents is the chart of flowrate versus generated pressure head. This is unique, almost like a hydraulic fingerprint. Similar pumps can have slightly different charts. In addition, even the same pump can produce different charts depending on its age and state of repair. Knowledge and understanding of this chart is crucial in understanding how the whole liquid system will operate (Fig. 3.26).

Fig. 3.26 shows a typical manufacturer's performance test curve. It shows how much pressure head the pump is capable of putting up over a range of flowrates. Where such a curve is available, this can be an extremely useful tool in troubleshooting hydraulic problems. But they are not always available. In many mature liquid handling systems, the pump characteristic is often an unknown. Records have become misplaced or destroyed. The manufacturer may no longer be trading, or if he is, his records are no longer available. But all is not lost. Faced with this dilemma, the nameplate pump designation might give a **very rough** clue.

## Performance estimate, unknown pump

In the early days of the pump industry it was not uncommon for the manufacturer to assign a name to a particular range or line of pumps; '**The Superior**' or 'The **Superflow**' and so on. As time went on, a pattern

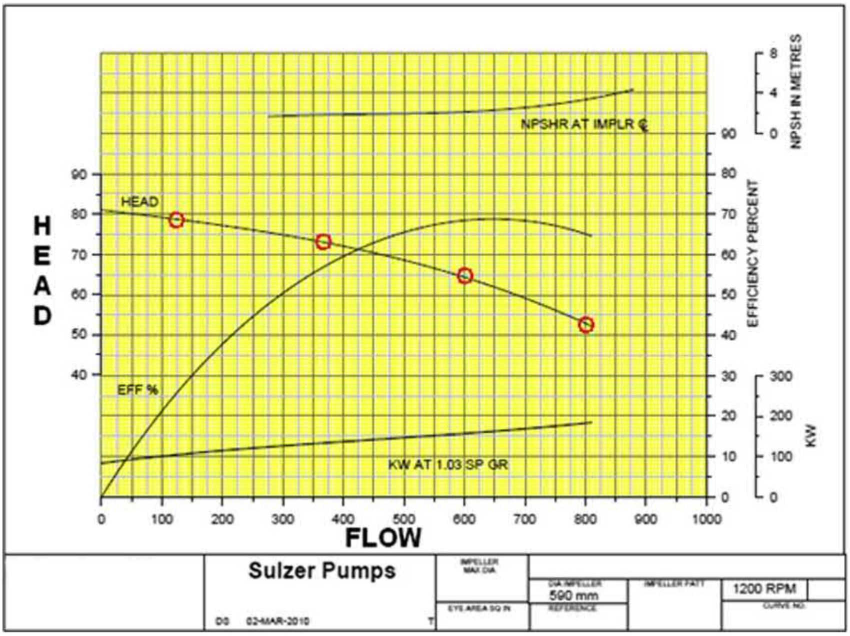



Fig. 3.26 Typical workshop test curve.

emerged of adding a prefix which helped define each specific pump better. These are examples, in inch units, '**4 × 6 × 10 Superior**', or '**8 × 10 × 12 ST-2 Superflow**'. *[There are equivalents in metric dimensions.]* This notation was principally to help the manufacturers' internal processes, but became useful to the purchaser or user. Typically, the first pair of numbers [**4 × 6, & 8 × 10**] represented the nominal inch diameter of the pump connection to the liquid system pipework**.** The smaller of the two numbers would be the nominal outlet pipe connection diameter in inches whereas the larger would represent the inlet connection size. The last numbers [**10 & 12]** would represent the nominal maximum diameter [D2] of the pump impeller. These sets of three numbers can help to yield some useful basic information about the machine. Supplementary integers between 2 and [say] 15 may describe the number of impeller stages within the machine. ST-2 would indicate a pump having two stage or impellers in series.

## Rough estimate of pump design flow

[Refer to Appendix B for a slightly more detailed approach].

**Table 3.1** Typical liquid velocities through pump connection flanges, at pump design flow and speed.

| | Velocity in inlet pipe connection (FPS/[m/s]) | Velocity in outlet pipe connection (FPS/[m/s]) |
|---|---|---|
| Water applications | 12–15 FPS<br>3.6–4.6 m/s | 20–25 FPS<br>6–7.6 m/s |
| Hydrocarbon applications | 10–15 FPS<br>3–4.6 m/s | 25–30 FPS<br>7.6–9 m/s |

It turns out that pump designers have found from experience that the velocity in both the inlet and outlet connections should not exceed certain levels. These depend on both the liquid and the application (Table 3.1).

The table indicates typical criteria. But he will have considered many different conflicting aspects when deciding the pump connection/branch size. *[Note that specialised pumps such as those handling sewage, or those pumping liquids very near to their vapour pressure will have different, generally lower values]*

In general, the inlet connection flow velocity should not be so high that it exceeds the axial velocity approaching the impeller entry annulus, or 'eye'. If that were the case, then the liquid flowpath between the two would be slowing down or diffusing. Diffusion might create stall cells in the flowpath. Experience says this part of the flowpath should be accelerating so as to ensure stable impeller entry flow conditions. This suggests that inlet pipe diameters should be large so as to ensure lowest entry velocities. Low pipe friction losses result. It also helps guarantee liquid acceleration towards the impeller eye.

Similarly, the outlet connection should not be so high as to create high velocity 'jets' in the downstream pipework or components. This is known to have a potential to create noise and vibration issues.

In both cases, hydrocarbon liquids tend to be more forgiving than water based liquids, so higher velocities can be tolerated. These are just general velocities intended as an aid to the unknown performance of a given pump. Special applications such as sewage disposal or the pumping of liquids close to their vapour pressure will dictate lower pipe velocity than indicated in this table (Table 3.2).

The designer would also be aware that the **pump** pipe connections sizes are not directly related to the actual pumping **system** pipe In fact, the selection of Minimum Economic Pipe diameter for any system is a

**Table 3.2** Benefits of high or low pump connection velocities.

| | |
|---|---|
| High inlet velocity-small pipe dia<br>• Avoids gas 'slugs'<br>• Impair NPSH[A]<br>• Hinders solids settlement in pipe<br>• Increase sensitivity to flow distortions from contorted upstream pipework | Low inlet velocity- large pipe dia<br>• Beneficial with organic suspended solids<br>• Essential for vertical double case can pumps so as to not impair NPSH[A] |
| High outlet velocity- small pipe dia<br>• Increase liquid 'white noise' emission.<br>• Increases sensitivity to part closed discharge gate valve<br>• Increased risk of erosion at pip bends | Low outlet velocity- large pipe dia<br>• Raises the cost of ancillary items such as non-return valves, leak off lines and valves |

completely separate matter. It will depend upon the system layout features, the pipe material pipe and the power absorbed in pumping (Fig. 3.27).

If the pipe size is very small, then its capital cost and the cost of installation would be relatively low. However, the liquid resistance put up by small pipe diameters will be relatively large and so pumping will

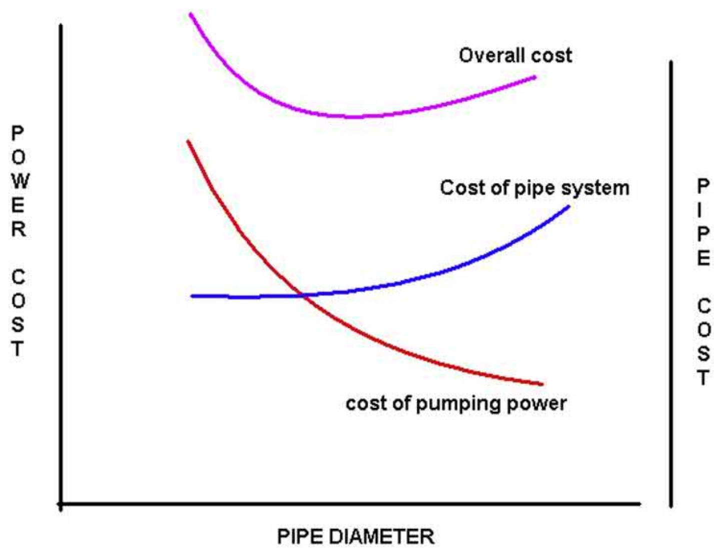



**Fig. 3.27** Chief cost components for a typical liquid pumping scheme. Added together they yield the total scheme costs. Usually a minimum cost point occurs at the minimum economic pipe diameter.

consume a relatively high power. The pump capital cost would also be high.

If the chosen pipe diameter is increased, the capital and installation costs of the system pipework [including ancillaries] will tend to rise. However larger pipe diameter provide less liquid resistance, so the both the consumed power and initially also capital costs of pumps would tend to reduce.

The combination of pipe costs and pumping costs will show a minimum at some pipe diameter and this is known as the **M**inimum **E**conomic Pipe **D**iameter; [MEPD].

Clearly the MEPD will be different for different circumstances. So in addition to acknowledging the trends outlined, the pump designer needs some experience of the MEPD for the expected range of pump applications.

Knowing the inlet or outlet connection size, and knowing the velocity criteria most probably used by the designer, one can gain a rough idea of the pump design flow (Fig. 3.28).

Using the earlier examples.

4 × 6 × 10 SUPERIOR-[WATER APPLICATION].

Based on 6 inch inlet connection and criterion of 15–20 feet/sec flow velocity, the chart yields flows of approximately 1000–1500 Usgpm.

Based on 4 inch outlet connection and criterion of 20–25 feet/sec flow velocity, the chart yields flows of approximately 700–900 Usgpm.

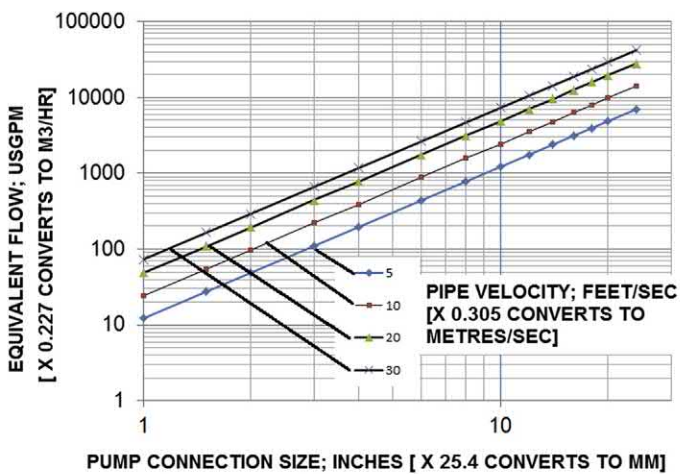



**Fig. 3.28** Approximate relationships between pump design flow and pump pipe connection sizes. Different ranges of pipe velocities are traditional in different industries and applications.

So a rough design flow estimate for this pump would lie between 700 and 1500 Usgpm, with 900–1000 being most probable

8 × 10 × 12 ST-2 SUPERFLOW - [WATER APPLICATION].

Based on 10 inch inlet connection and criterion of 15–20 feet/sec flow velocity, the chart yields flows of approximately 3000 Usgpm to 3900 Usgpm

Based on 8 inch outlet connection and criterion of 20–25 feet/sec flow velocity, the chart yields flows of approximately 3000 Usgpm to 3400 Usgpm. So a rough design flow estimate for this pump would lay between 3000 Usgpm and 3900 Usgpm with 3400 or so being most probable.

Of course this whole exercise is quite inexact, for many reasons. So at such a level it would be valid to just take the average of the inlet and outlet values given by the chart. This will yield a very approximate **design point** flow. Not an exact value, but often better than no information at all. The **real** operating flow will depend on the system characteristics and so of course, may depart from this average.

## Rough estimate of pump head per stage

So far as the corresponding pump pressure head output is concerned we can take advantage of the fact that, to a first approximation, this can be determined from both the impeller diameter and the shaft speed. Also, to a first approximation, the pressure head generated at zero flow of a centrifugal pump increases in the order of 110%–130% over the range of common centrifugal pumps. Levels above this do occur, but are characteristic of mixed and axial flow pumps. At values tending to, 110% the curve shape does not lend itself so well to co-operative parallel operation. Values' tending to 130% produces machines which are heavier. They can have a more useful power curve shape, which can lean towards non-overloading (Fig. 3.29).

Knowing that in the first case of 4 × 6 × 10 Superior, the shaft speed is 3000 rpm, and impeller full design diameter 10 inch, from the chart we can estimate that the pressure head generated at design flow to be about 220 feet per stage [This is a single stage pump].

The second pump, the 8 × 10 × 12 ST-2 Superflow is running at 1450 rpm. The Figure shows we can expect the pump pressure head at design flow to be about 90 feet per stage. But we can guess or know that

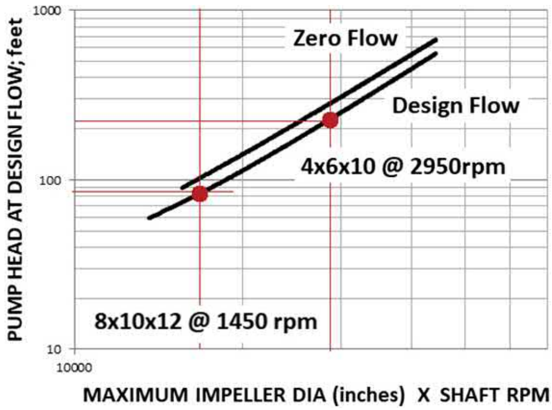



**Fig. 3.29** Approximate relationship between impeller diameter, shaft speed and stage head. This is for pumps with so-called 'stable' curves. Those with 'unstable' curves will have produce slightly more head at a given diameter.

ST-2 means it is a two stage pump hence the total pressure head is 2 × 90, i.e. 180 feet.

Thus far we have estimated the design performance of each pump to be;

4 × 6 × 10 SUPERIOR-

Flow approximately 900−1000 Usgpm

Total Head approximately 220 feet

8 × 10 × 12 ST-2 SUPERFLOW-

Flow approximately 3400 Usgpm

Total Head approximately 180 feet [in two stages] (Fig. 3.29).

## Pump specific speed [Ns] leading to efficiency estimate

With the information deduced so far, we can even make an estimate of the pump efficiency and hence it's absorbed power. It is known that pump efficiency can be broadly related to the hydraulic proportions of the impeller. Within the pump community this general shape is characterised by a term known as the pump 'Specific Speed **[Ns]**' Sooner or later the term Specific Speed **[Ns]** will; arise in any technical discussion on centrifugal pumps. Anyone who is technically involved in pumps will inevitably come across this term at some point. In the distant past, it was a central part of the pump designer's tool kit. Certainly, when one designer spoke to another and defined that pump hydraulics as being of specific speed '1234', then *[provided they had previously agreed to use consistent units of measurement]*

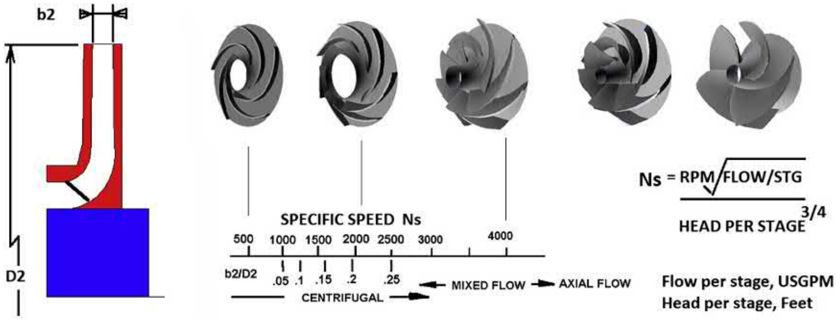



**Fig. 3.30** Approximate relationship between impeller proportions and specific speed.

they would both form very similar, but not identical, mental images as to how that impeller looked.

$$\text{Specific Speed, [Ns]} = \text{[Speed} - \text{RPM]} \times \text{[Flow per eye} - \text{USGPM]}^{0.5} / \text{[[Head perper stage} - \text{FEET]}^{0.75}\text{]} \tag{3.1}$$

Nowadays it is considered far too blunt an instrument for most design purposes, other than that basic description. However, it can often be a convenient way of illustrating design trends. Another problem is that technically it should be a dimensionless number. But different parts of the world have seen fit to use their local units and they have derived a dimensional variant (Fig. 3.30).

Academics will [correctly] argue that the Specific Speed equation should also be divided by a consistent value for g, the gravity constant. This would yield Specific Speed [Ns] as a truly dimensionless number. However, the dimensional term has been widely used for a long time. A great deal of relevant literature survives that is based upon the dimensional value appropriate to different parts of the *world [i.e. Usgpm. Igpm, Litres* per *second or cubic metres* per *hour].* The situation might seem a bit of a mess to academics. However, values calculated in one set of units can easily be converted to another so no real practical problems appear to arise for practising pump engineers. The writer continues to look forward, now with some hope, to the day when designers worldwide will also embrace the non-dimensional Specific Speed [Ns].

It is easy to conclude that Specific Speed [Ns] is an actual speed with some practical value. In fact, consideration of the equation shows that it could be interpreted as the shaft speed that the pump would have to run in order to deliver [say] 1 Usgpm at 1 foot pressure head. This is interesting but not helpful. It implies that a pump of low Specific Speed [Ns], say 800, must have a relatively large impeller because it does not need to turn very quickly in order to generate 1 foot pressure head. Conversely a pump of medium Specific Speed [Ns], say 3000, must have relatively a smaller impeller, since it has to turns that much faster in order to generate the same 1 foot pressure head.

By far the most useful property of the Specific Speed [Ns] term nowadays is to convey a consistent mental image of the impeller between pump designers and engineers. But it is easy to demonstrate the weakness of specific speed as a design tool. Consider a pump designed for a certain flow, head and shaft speed. In practice, there are a wide range of hydraulic designs that can and will satisfy that condition. But there will be significant difference in how the pumps would perform if flow is then increased or decreased. An actual example helps to demonstrate this point (Fig. 3.31).

Both these impellers would generate the same flow and pressure head at their best efficiency point. However, the wider impeller, with the smaller diameter, would have a zero flow pressure head significantly lower than the other. Yet by definition, they have the same specific speed since this parameter only considers the best efficiency point behaviour.

With respect to the entire spectrum of so-called 'rotodynamic pumps' it is convenient to divide them up into discrete groups. This was alluded to earlier (Fig. 3.31).

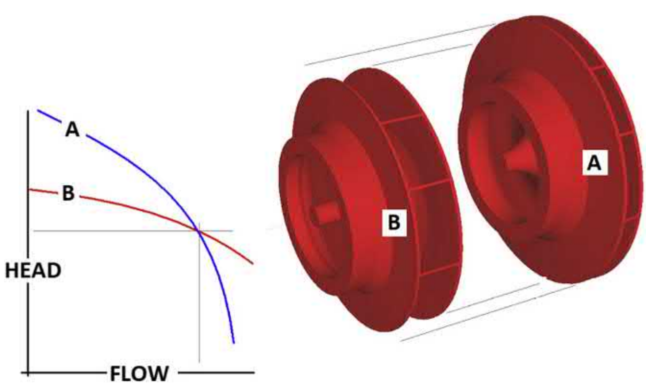



**Fig. 3.31** Impellers with the same specific speed value, but different geometry. This results in the same performance at bep, but very different off-design output.

## Centrifugal [radial flow] pumps

They exist from Specific Speed [Ns] of a few hundreds up to values of around 4000 in Usgpm units. At the low end of this scale, the head is generated almost entirely by centrifugal forces — that is-there is a significant difference between the radiuses at which the liquid enters the impeller compared that at which it leaves. [D2 is much larger than D1.] At the high-end of this scale, the vanes shape itself begins to make a contribution. This is analogous to the 'lift' forces generated by aerofoils (Fig. 3.32).

## Mixed [diagonal] flow pumps

Above Specific Speed [Ns] of 4000, the inlet and outlet diameters are becoming similar. Less head is generated by centrifugal force but the vane shape 'lift' force contribution is becoming more significant. At the upper end of this sector, say around 8000, the lift forces begin to dominate over the centrifugal contribution (Fig. 3.33).

## Propeller [axial flow] pumps

At this upper end of the spectrum, most if not all of the head is generated by vane shape 'lift'. There will be little difference, if any, between D1 and D2 (Fig. 3.34).

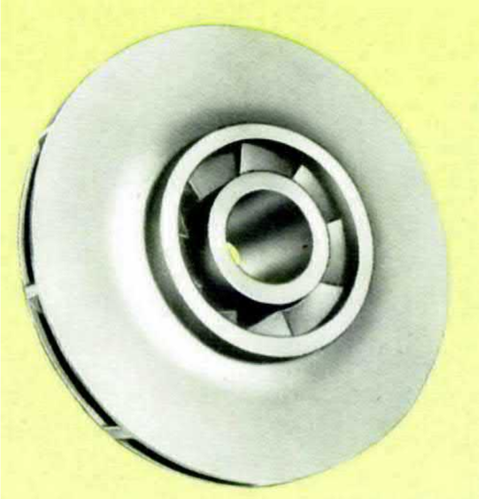


**Fig. 3.32** Typical radial flow impellers exist from specific speeds of a few hundreds up to around 4000 in Usgpm units.

**Fig. 3.33** Between specific speed of 4000 and 8000, the inlet and outlet diameter are becoming similar.

**Fig. 3.34** Above, say, 8000 specific speed are called axial flow, or propeller pumps.

The conceptual relationship between Specific Speed [Ns], general shape, and the mixture of centrifugal/vane shape head is shown here (Fig. 3.35).

## Rough estimate of pump overall efficiency

How does all this help estimate efficiency and consequently power consumption? In fact there is a loose and indirect relationship between impeller stage specific speed and overall pump efficiency. Pump efficiency is actually dependent upon a list of different internal losses. Each of these is affected by factors which in turn relate to the pump hydraulic geometry. So no simple accurate link between Specific Speed [Ns] and pump overall efficiency can

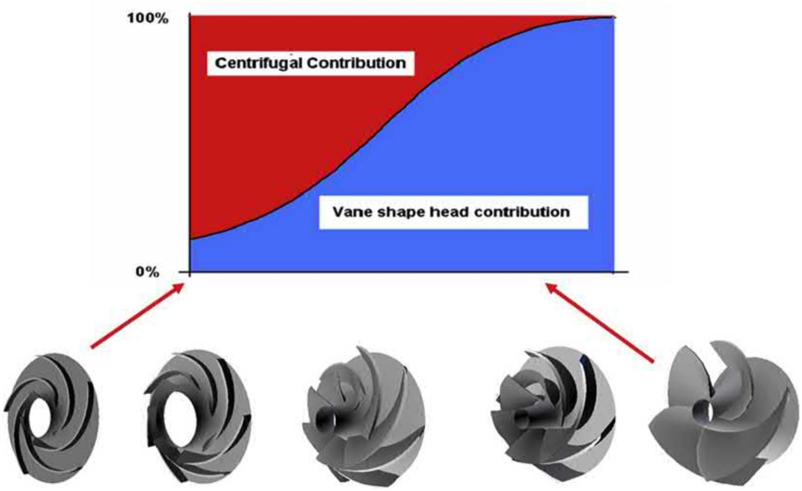



**Fig. 3.35** This shows the general relationship between specific speed, impeller shape, and head contribution components [Sulzer pumps].

exist. But a broad statistical relationship can be shown for pumps of mainstream hydraulic design. Performance data for several hundred pumps of many different types are included (Fig. 3.36).

This chart relates pump overall efficiency to the stage Specific Speed [Ns] and hydraulic size. Hydraulic size is described by the flow discharged per each shaft revolution.

This chart can be useful for a quick rough estimate of a pump at its best efficiency point flow. It also shows the variations that might be expected depending on sophistication and build quality. Its accuracy is consistent with the other performance parameters discussed earlier. Critical cases would need to be examined in a level of detail beyond the scope of this volume.

## Summary of rough perforance estimate

Having estimated the efficiency, the Horsepower absorbed at the nominal pump design flow and pressure can be calculated (Table 3.3);

So knowing only very basic information about the pump we can get some general idea of its output performance. Not precise, but often helpful where no other data exists. It can be useful in the absence of specific test bed information.

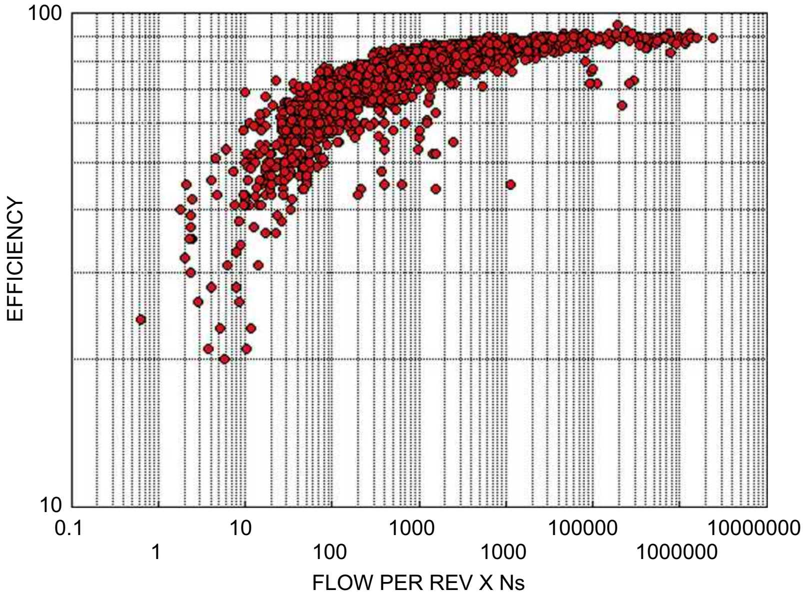



**Fig. 3.36** Pump efficiency as an approximate function of size [Usgpm/rpm] and specific speed [Usgpm, rpm Units]. This chart represents individual pump types known to the author. It extends from simple, dirty liquid pumps, to advanced multistage machines.

**Table 3.3** Summary.

| Pump | Design flow (USGPM) | Design total head (FEET) | Design specific speed | Expected efficiency | Power absorbed (BHP) |
|---|---|---|---|---|---|
| 4 × 6 × 10 SUPERIOR | 950 | 220 | 1600 | 78% | 67 |
| 8 × 10 × 12 ST-2 Superflow | 3400 | 180 | 2890 | 83% | 186 |

## Pump flow-head curve elements

But of course, controlled testing is the preferred method of assessing pump output. And the traditional method of representing the test results is by a series of curves. A typical manufacturer's test curve is shown. The details of pump testing and codes of practice will not be covered here since many excellent National and International standards exist. But the physical meaning of such derived performance charts, while crucial, is often unclear to those not using them regularly. Let me attempt simplify these curves in two different ways.

- Firstly, imagine the pump located in the lower corner of an immense sheet of graph paper. The pump has a flexible hose connected to its outlet. Partway along this pipe is some form of flow measuring device with a digital readout (Fig. 3.37).
- If the hose end is lifted high above the operating point **h3** then very little flow will register on the flowmeter. Indeed a height will exist above which no flow is registered. This special point is called the 'closed valve head' or 'shut-off head', **ho**, and is the maximum for that machine at that speed. Flow through the pump is zero **q0.** *[But words of warning, some pumps achieve their maximum pressure head at a flow that is not quite zero.]*
- If the hose end is lowered, a little flow will start to pass through the pump-represented by head **h1** and flow **q1.** This process can be repeated at several points, noting the corresponding head and flow **hx, qx**
- At some flow the head **h4** will be almost [but not quite] zero. This is also a special point. It is called the run-out flow.
- **q4** and is close to the maximum for that machine at that speed.

The corresponding points can be joined to display the familiar continuous curve. This curve defines the pumps hydraulic characteristic, or 'characteristic curve' (Fig. 3.37).

Another way to imagine how this curve is created is as follows.

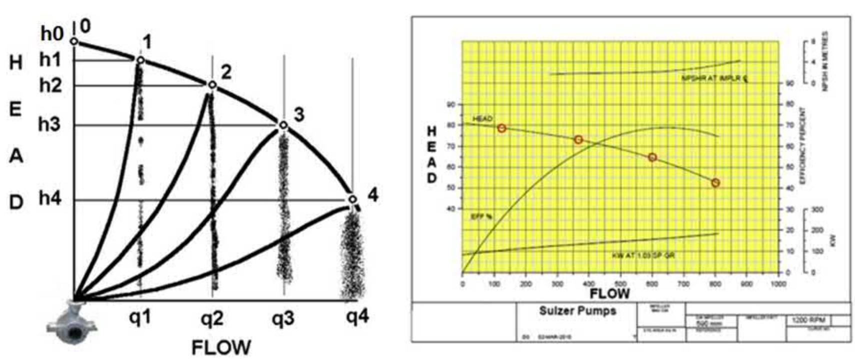



**Fig. 3.37** Analogy as to how a pump performance curve is experimentally derived. A real curve is shown for comparison.

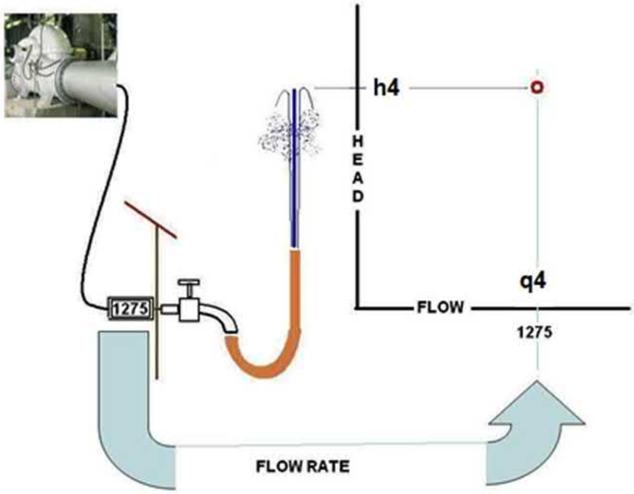



**Fig. 3.38** Alternative analogy-free flows through the hose.

- Consider the pump to have the same flexible outlet hose mentioned above. It lies on the ground at the same level as the pump. An in-line digital flowmeter is also installed (Fig. 3.38).
- The pump is started and discharges near to its run-out flow, as above. The pump flow and pressure head are measured separately and are graphically recorded as q4, h4 (Fig. 3.39).

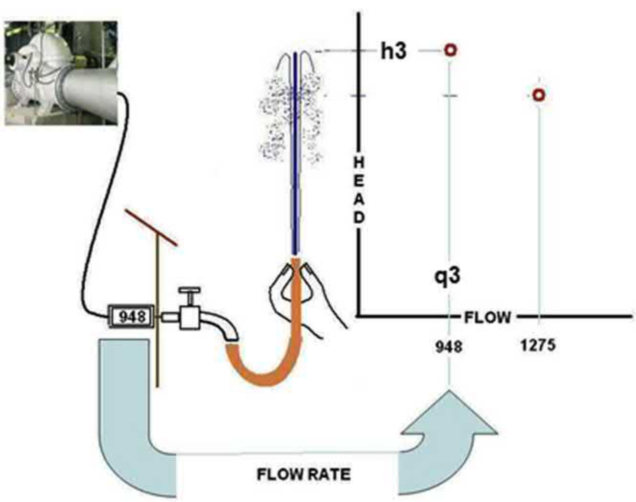



**Fig. 3.39** Alternative analogy-slight restriction in hose.

- Imagine that either the outlet hose is in some way physically 'nipped', by a superhuman hand *[or maybe a truck wheel accidentally gets partly parked on it.]* This nipping will create some pipe resistance or backpressure. Less of the pumps output would then be available to circulate flow, since some of it is dissipated in overcoming this resistance/backpressure. The pump now delivers less flow but correspondingly generates a higher pressure head. This equates to a new point on the graphical record, **h3, q3** (Fig. 3.40).
- With more intense 'nipping' of the hose, the flow continues to reduce while the generated pressure head rises. Eventually, with even more nipping of the hose, the pump output flow falls to zero. In many cases, the generated pressure head also continues to rise to its maximum value. *[But as above, words of warning, some pumps achieve their maximum pressure head at a flow that is not quite zero.* As before, joining these tests points yields the pump characteristic curve. Test pipework is usually metallic and so pump manufacturers replace the' nipping finger' with control valves (Fig. 3.40).

## Flow − pressure head

The curve derived from such pump tests will generally fall steadily to the right. [Stable curve] This is preferred by most pumping system designers, since it gives the most positive interaction between the pump and system

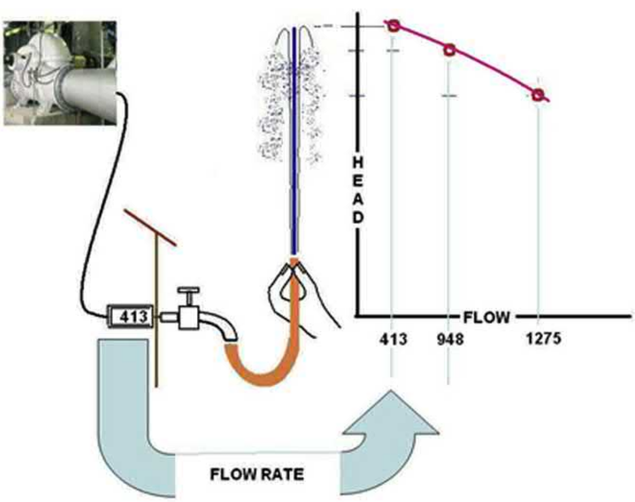



**Fig. 3.40** Alternative analogy-large restriction in hose.

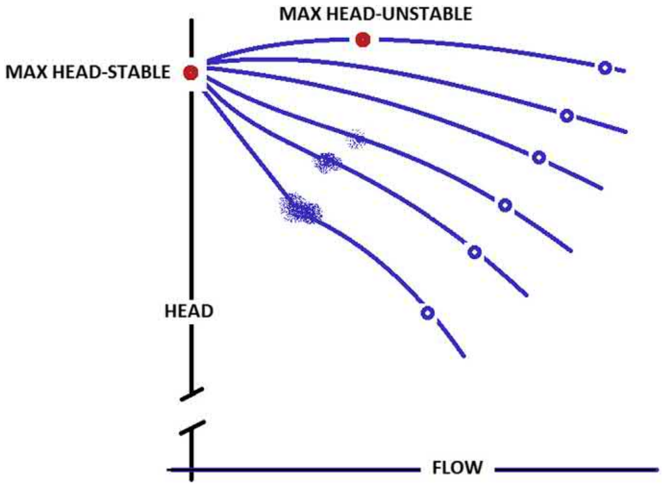



**Fig. 3.41** The range of pump performance curve shapes likely to be encountered. Generally, the maximum head is generated at zero flow. But in some special case, the maximum head occurs at a small but finite flow. Pumps of high specific speed [mixed and axial flow] often exhibit a discontinuity in their curve anywhere between 30% and maybe 60% of design flow. In low specific speed [Radial Flow] impellers this discontinuity will also exist but is rarely obvious.

characteristics. Occasionally, pumps appear which have characteristic curves that do not steadily fall to the right. *[Unstable curve]* Although there is a widely held negative view towards his type of curve, this is only strongly justified where two or more pumps have to operate in parallel, or the static head is an unusually high proportion of the total head (Fig. 3.41).

Pumps with such unstable characteristics are usually physically smaller than those with a steadily falling curve. As a result they may find favour in applications where space or weight is at a premium.

In some cases, the curve may show a discontinuity at between say 20% and 50% of pump design flow. Operation at flow less than this is possible but usually discouraged, since it can be associated with unsatisfactory pump running or even poor interaction with the system characteristic. But in general, the generated pressure head of centrifugal pumps reduce as flow output is increased. Furthermore, impellers with relatively narrow outlets tend to have steeper curves (Fig. 3.42).

This last fact gives a clue as to how a pump performance characteristic curve shape can be manipulated by the designer. This is described later.

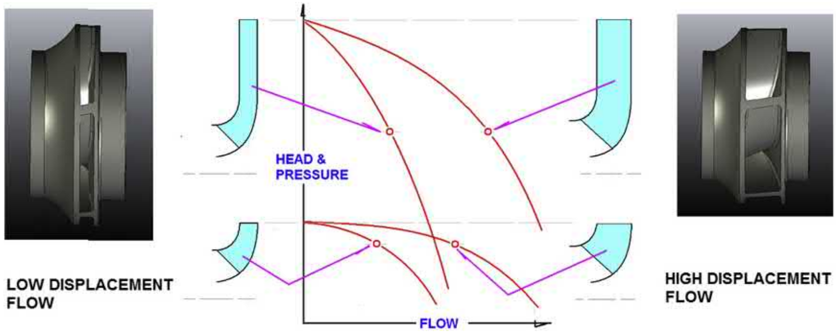



**Fig. 3.42** Impeller width changes the design flow of the impeller. Wider impellers will tend to greater flow. Larger impeller diameters tend to higher heads.

## Pump head and pressure

Fig. 3.43 pump engineers often interchange the terms pressure and head when discussing pump performance. Pump head is a constant property while pump pressure depends upon the liquid specific gravity

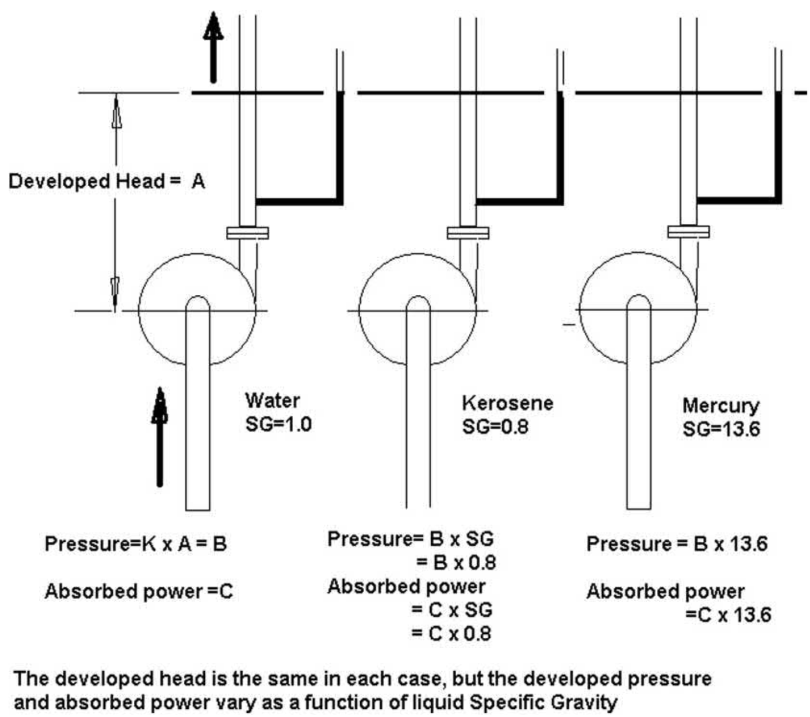



**Fig. 3.43** Explanation of how specific gravity affects pressure, but not head.

## Flow − absorbed power

It is normal for a manufacturer, to routinely measure and deduce the pump flow and its corresponding pressure head. He will also routinely deduce the power that the pump will absorb at each test point. This is of direct interest in sizing the prime mover of course. But he will also want to relate this power consumption to the amount of hydraulic power generated in the output flow. In this way, he can infer the machine's efficiency as an energy converter. The size of the prime mover helps govern the initial capital cost of the pump set, whereas the efficiency helps decide the life cycle costs of the machine.

With the most commonly used pumps, the absorbed power curve will rise as the flow increases. More flow needs more power. However, pumps of high flow and low head may have a power curve that falls as flow increases. This completely different behaviour requires a different approach when it comes to start-up and shut-down. In between these two extremes are pumps with power curves that are more or less flat, or at least show a maximum. Such curves are often referred to as non-overloading. In truth any curve can be considered non-overloading if enough driver power is available. An arbitrary definition considers power curves to be **non-overloading** if the maximum absorbed power **at all or any flow** does not exceed 90% of available driver power (Figs. 3.44 and 3.45).

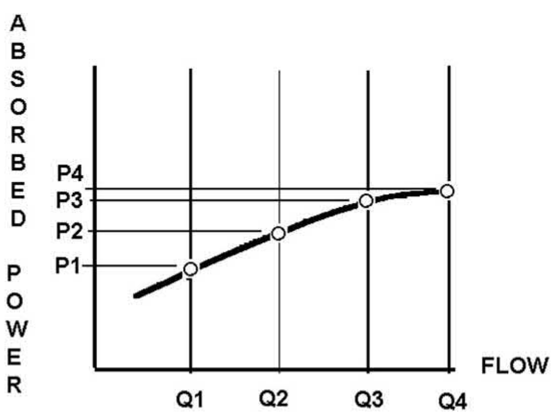



**Fig. 3.44** Typical absorbed power curve shapes for radial flow impellers. Generally the curve rises to a maximum level close to the maximum flow. In low specific speed radial pumps, the power curves seem to rise relentlessly as flow increases. At the higher specific speed end of the radial range, the power curve trends to level off. In the extreme, the power curve will show a maximum inflexion point. These are known as non-overloading power curves.

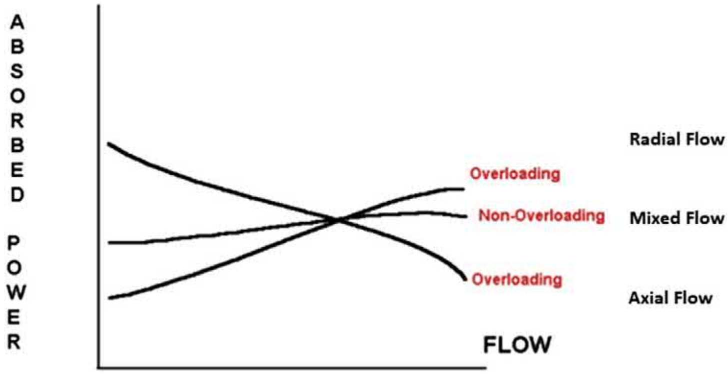



**Fig. 3.45** Typical power curve shapes for mixed and axial flow pumps. Non overloading curves are common in mixed flow pumps but can be engineered into some radial flow pumps. Axial flow pumps trend to the opposite power characteristic than radial flow. Their maximum power is at or near to zero flow. This can pose difficulties in start-up and needs to be acknowledged in matching the speed-torque curve of pump and driver.

## Flow-efficiency

Designers and owners want to know if the pump output is giving good value for the amount of power input to the machine. Manufacturers do this by drawing a flow/efficiency curve. The pump efficiency is defined by.

Efficiency = Output power [Hydraulic]/input power [Mechanical]

Or

= fluid power / power absorbed

Pumps give their best value for the power used at only one flow–pressure head point. Fig. 3.46 this point is, predictably, called the **best efficiency point**, or **bep** for short. Owners will usually want to operate their pumps close to this point so as to demand the least power for the job in hand.

Efficiency cannot normally be measured directly. It is a derived function inferred from the above equation.

However, in certain circumstances, the **THERMOMETRIC** method can more directly yield efficiency. Pump 'in-efficiency' is almost completely dissipated as heat. So the small temperature rise across the pump, as it operates, is a measure of this in-efficiency. From this, efficiency can be calculated.

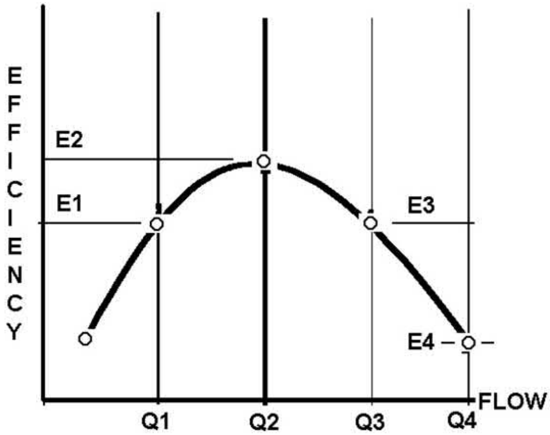



**Fig. 3.46** Curve of overall pump efficiency. With very low specific speed [radial] pumps Q4 may equal twice Q2. At the other extreme, in axial pumps, Q4 might only be 1.3 times Q2. Flow-Efficiency curves cannot normally be measured directly. It is deduced from other measured test data.

All pump efficiency curves follow the same general shape and form, though the proportions may differ. Generally pump of higher Specific Speed [Ns] will exhibit efficiency curves that are more 'peaked' than pumps of lower Specific Speed [Ns]. They will have a narrower flow band of good efficiency compared to their low Specific Speed [Ns] relatives. In some cases, the pump efficiency will display a slight discontinuity at between 20% and 50% of bep flow. Though not always visible, this will coincide with the establishment of flow breakdown/stall inside the impeller. In itself this condition is not always serious, unless the impeller inlet tip speed is relatively high (Appendix D).

## Flow- cavitation

The final information displayed on the performance test curve is the so-called 'NPSH required' or NPSH[R]. This term describes a threshold set of physical conditions at the pump suction branch, below which, a pump tends to develop internal cavitation to an unacceptable level (Fig. 3.47).

The determined NPSH[R] curve shape is unique to each and every pump and is flow-related. Cavitation is the formation of vapour bubbles within the pump flow. It occurs along the flowpath at its point of lowest static pressure. In the short term these bubbles may attain sufficient volume within the flowpath to choke flow and impair pump performance. But

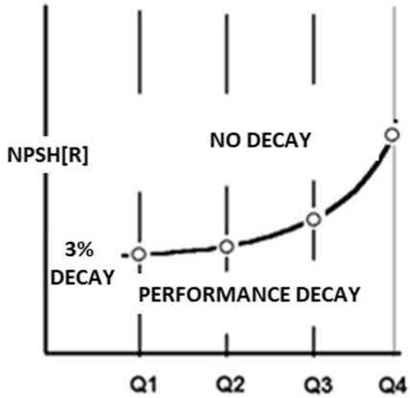



**Fig. 3.47** The widely used NPSH [R] curves are based on cavitation presence sufficient to cause the pump head to decay by 3% (Appendix E).

even if this condition is not achieved, and no performance decay is detected, the collapsing bubbles may still have the potential for slow, progressive long term impeller damage (Appendix E) (Fig. 3.48).

Equally, the pump location conditions, either on site or on test will have a value of 'NPSH available' or NPSH [A]. This value is similarly influenced by both the physical installation details as well as the pumped liquid conditions. It is also flow-related.

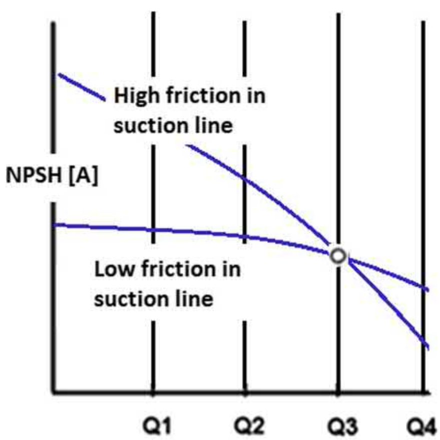



**Fig. 3.48** The shape of the NPSH [A] curve can affect pump behaviour at off-design flow. For instance, a pump operating with a steep NPSH [A] curve is less likely to suffer cavitating surge than with a flat curve. On the other hand it will possess less run-out flow capability (Appendices D and E).

If at any particular flow, NPSH[R] **exceeds** NPSH [A], then cavitating conditions will develop inside the pump. Reiterating the previous point, in the short term, cavitation MAY reduce pump performance below expectations and in the long term it may or may not cause severe physical damage to its impeller. The severity of the outcome depends among other things, on how much NPSH [A] exceeds NPSH[R] and the pumped liquid properties. The most widely used definition of cavitation *[the 3% head decay criterion]* relates chiefly to preventing performance decay on site. Without qualification, it tells us little about the long term life and durability of the pump. However, a considerable body of knowledge and experience has developed that attempts to make impeller life predictions based on the 3% head decay curve.

In most cases, operation in conditions where NPSH [A] = or < NPSH [R] [3%], will ensure cavitation damage in a relatively short time (Fig. 3.49).

However, to prevent long term impeller damage, the NPSH [A] usually has to be higher than that just to prevent head decaying by 3%. This extra margin is largely influenced by impeller inlet tip speed and the difference between operating flow and design flow, but can exceed a factor of 2, or 3. At present, few international standards exist to give guidance on this margin.

### *Suction specific speed [NSS]*

At this point it is worth mentioning that the NPSH[R] performance of any pump can also be ranked in a similar way to the impeller shape. On its own, the value of NPSH[R] associated with a pump gives no sense of how

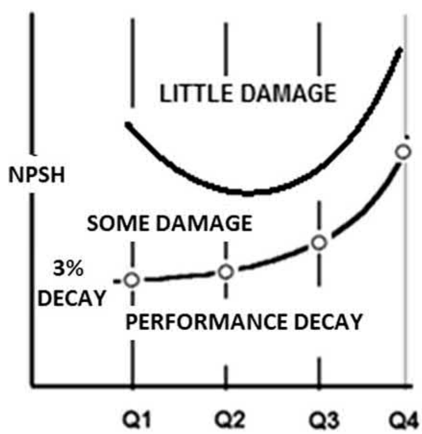



**Fig. 3.49** Continued operation at the 3% head decay level will result in the potential for mechanical damage to the impeller. Operation conditions need to be substantially above this line if damage is to be suppressed. This required safety margin is often highest at part flows (Appendix E).

elegant the impeller inlet design is. And it gives no clue as to the likely behaviour of the pump under off-design conditions. It would be useful to reduce impeller suction performance to a common reference framework.

Earlier I touched on the term Specific Speed [Ns] as a useful aid to creating consistent mental images of pump hydraulics. A comparable concept, of Suction Specific Speed [Nss], serves a similar purpose. It conveys to other designers and engineers how highly 'stressed' the impeller inlet hydraulic design is. As a means of ranking the suction performance of pumps, the forerunner to Suction Specific Speed was the so-called Thoma cavitation coefficient. These days, the Thoma coefficient is rarely if ever invoked in pumps, but it is mentioned just for historical interest.

Suction Specific Speed [Nss] can be defined as.

$$\begin{gathered}\text{Suction Specific Speed, [Nss]} = [\text{Speed} - \text{RPM}] \times \\ [\text{Flow per eye} - \text{USGPM}]^{0.5} \,/\, [[\text{NPSH[R] } 3\% - \text{FEET}]^{0.75}]\end{gathered} \tag{3.3}$$

Like the Specific Speed [Ns] term, this equation can be interpreted as the shaft speed at which the pump needs 1 foot of NPSH[R] at a flow of 1 Usgpm. At first sight, higher Suction Specific Speed [Nss] would appear to be more useful. This would imply that the pump shaft could turn more quickly under set suction conditions. Or put another way, at any given speed, a lower NPSH[R] is implied. Such a choice offers the result of smaller often more efficient pumps. However, and this is a very large however, there are significant constraints to this line of thought.

For 'normal average' pump designs of the 1960's of, Suction Specific Speed[Nss] of Nss 7500 or so was a typical value [Usgpm, rpm UNITS] Experience subsequently showed that certain hydraulic 'bad habits often became more obvious whenever pumps were designed with higher Nss. These bad habits included rough operation and flow surges. The reasons for this were well understood, but were not very well articulated outside of the pump community. Fraser had laid the foundations for this in [6]. Arguably, this relationship was simply defined and verified by Hallam in [7]. He made a thorough study of chiefly, but not exclusively end suction overhung impeller pumps, installed in a certain oil refinery. He showed that pumps with Nss in excess of roughly 11,000 could be expected to suffer from poorer reliability, if, and only if there was a risk of sustained operation at flows less than originally specified. At the time of his study, about the same time as the 1970's Oil Crisis, many Refineries had been forced to operate far below their design output. This tended to highlight the problem.

Reiterating, this study was conducted on a selected population of chiefly similar machine types *[overhung impeller – single stage]* in a specific set of applications *[oil and gas refining]*. However, in the absence of any other such simple criterion, this rapidly gained acceptance as an evaluation benchmark. Nowadays the benchmark is applied to machines well outside of the type, application and circumstances surrounding Hallam's original study. Consequently, 11,000 has become almost an immovable criterion.

In fact the impeller geometry necessary to achieve Nss levels in excess of 11,000 at design flow will simultaneously result in a design in which the onset of inlet backflow also tends to occur at higher fractions of design flow. *[Inlet backflow can become damaging if the inlet tip speed exceeds some value, depending on machine construction and liquid properties]. If* the Minimum Safe Continuous Flow is not consequently increased to reflect this mismatch, then poor reliability clearly becomes a risk. This was the root cause behind Hallam's experiences.

In truth levels of 16,000 are routinely delivered, provided certain operational constraints are rigidly imposed. The chief constraint would limit the amount that operating flow could be reduced. On the other hand, it should be stressed that pumps having a low value of Suction Specific Speed [Nss] will not necessarily bring a guarantee of trouble free operation either. Their low value may simply mask mediocre hydraulic design and so result in other problems (Fig. 3.50).

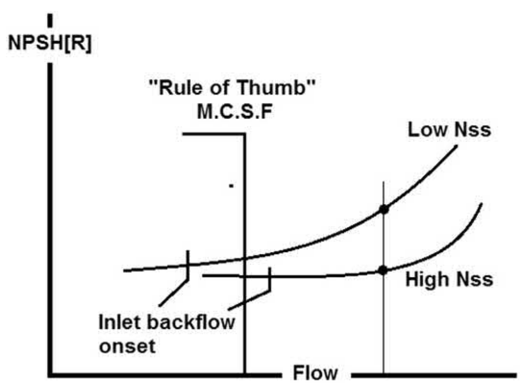



**Fig. 3.50** Problematic pump site NPSH requirements can be reduced by installing impellers of higher Suction Specific Speed [Nss]. Without very special measures, such design will inevitably exhibit a higher inlet backflow point. Consequently the operating comfort zone will be reduced. Defining a higher minimum continuous safe flow will help side-step any resulting problems (Appendix D).

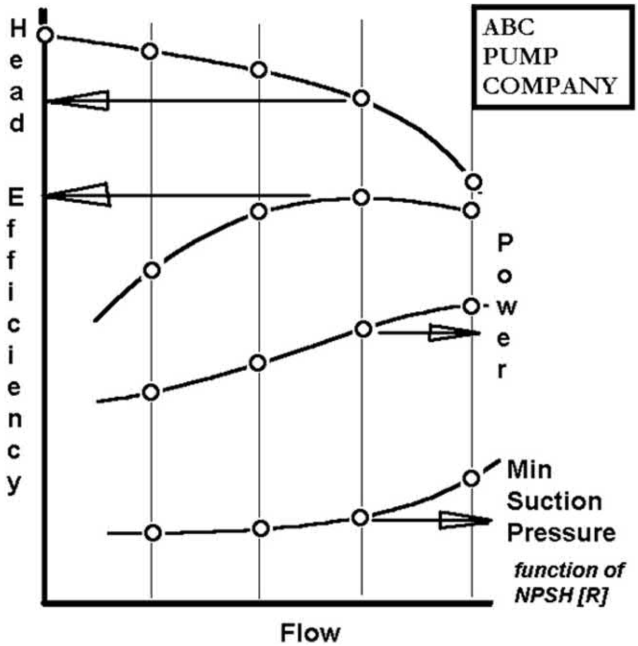



**Fig. 3.51** Components of a typical workshop test curve.

### *Pump characterisitc curve set family*

Traditionally, all these Test Data curves are plotted on one composite called the 'characteristic curve set'.

At a glance, all the salient performance information can be seen. We shall see later that this set of curves is one of the two key factors central in determining the overall performance of any pumping scheme. The format will vary from one company to another but they will look something like this (Fig. 3.51).

## CHAPTER 4

# Pump basics; part 2

## Introduction

Before I begin; an explanation. This Chapter will discuss how impellers can be configured into different combinations as well as continuing the description of different shapes of individual impellers. Within the pump community it is frequently necessary to discuss both matters and so designers adopt a form of shorthand description. This text will also adopt it for the most part (Fig. 4.1).

So when those not routinely involved in pumps see an impeller as in the left hand image, those in the pump technical community will tend to visualise it, or sketch it as in the right hand image.

Hence, in this widely accepted shorthand, the left hand image is condensed as shown on the right. This simplification will help clarify some of the different configuration in use (Fig. 4.2).

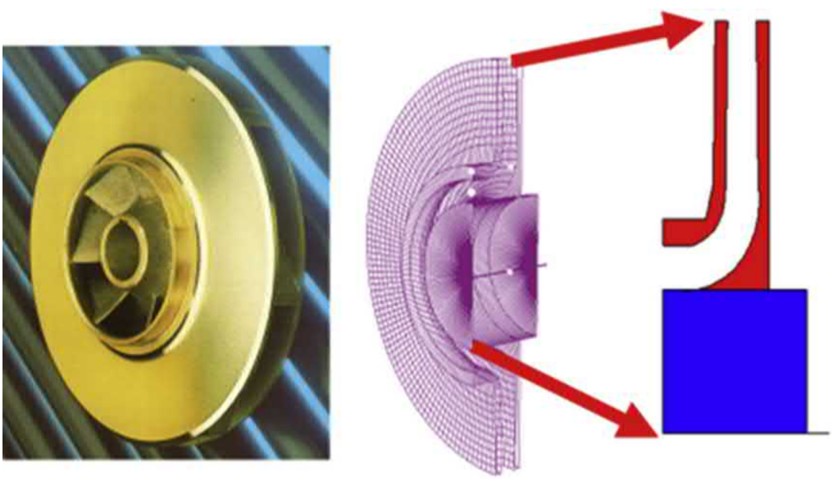


**Fig. 4.1** Transposing real impeller shapes to hydraulic designer's 'shorthand'.

*Troubleshooting Centrifugal Pumps and Their Systems*
ISBN 978-0-08-102503-1
https://doi.org/10.1016/B978-0-08-102503-1.00004-9

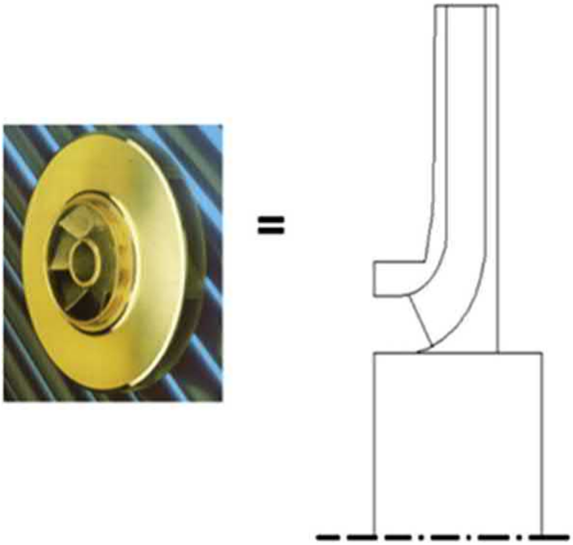


**Fig. 4.2** Transposition used in this volume.

In this chapter, I will use the simple single entry–single stage impeller concept, just described, as the building block for the various other hydraulic configurations (Fig. 4.3).

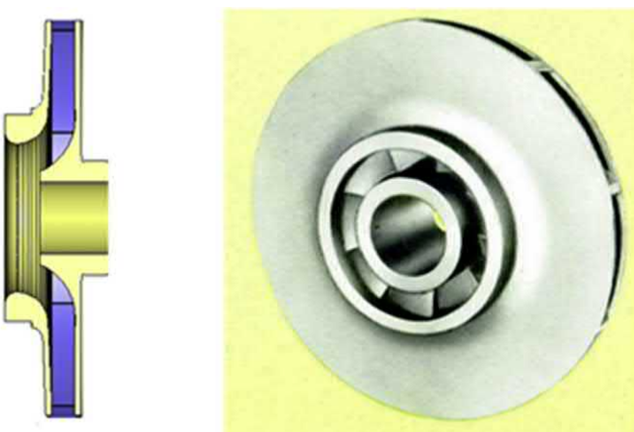


**Fig. 4.3** Single Entry-Single stage impeller. This is by far the most common impeller configuration. It is furnished with only one inlet annulus-commonly known as the impeller 'eye'. It forms the basic building block for other configurations. Vane counts will normally range from 4 to 8. Impellers with 1, 2 or 3 vanes are normally associated with special applications, often handling large unscreened solids. Impellers with more than 8 vanes are often handling liquids with entrained free gas. *[Mixed and Axial flow impellers tend to lower vane counts and 2 or 3 vanes is not unusual.]* The example shown here is known as 'fully shrouded' Later we shall see examples with partial shrouds.

## Single entry-single stage impellers

Centrifugal pumps clearly do not all look the same. They are available in several different hydraulic genres with a choice of mechanical layouts. So far, we have only referred to a simple basic impeller. Here, flow axially approaches the impeller via an inlet duct and enters through only one orifice or annulus. Having done work on the liquid, the impeller discharges radially outward into a collector duct. This is generally referred to as a 'single entry-single stage' layout (Fig. 4.4).

There are some significant design parameter for this and all other centrifugal pumps.

## Eye diameter D1

In conjunction with the pump speed, the so called 'eye diameter' D1 will largely determine how much flow the pump can accept, as well as the minimum suction pressure conditions to avoid cavitation. Increasing D1 allows the impeller to accept more flow. It will also help to reduce the minimum suction pressure requirements, but only up to a point. Furthermore, when D1 approaches D2, there is too little radial space within the impeller silhouette to install efficient blades. Such extreme designs might be associated with operating 'bad habits'. For example, their band of useful operating flow might be rather limited and they become a source of hydraulically excited vibration.

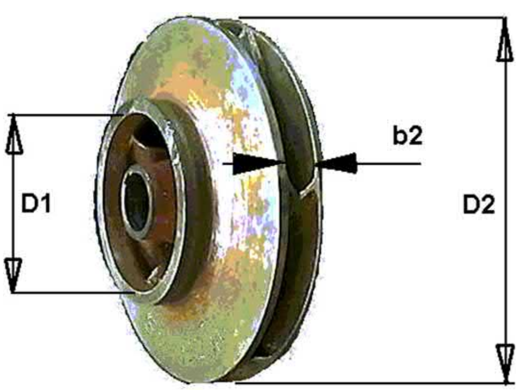



**Fig. 4.4** Illustrating the three key impeller geometry dimensions. As shown here, the impeller rim is machined cylindrically. In other versions the rim is machined to a conic profile.

## Impeller outer diameter; D2

In conjunction with pump speed, this will dictate how much pressure head the pump generates. Increasing D2 will increase the pressure head and vice versa. As shown, the radial outlet area forms part of a cylindrical surface. The diameters of front and rear shrouds are the same. This is typical of so-called Low Specific Speed impellers. In impellers of the Mixed Flow variety [higher specific speed] the radial outlet area forms part of a conical surface. Here the diameter of the rear shroud is smaller than that of the front. In practical terms, the effective impeller diameter is the Root Mean Square of these two diameters. Occasionally, impellers of the Radial Flow variety have their cylindrical outlet surface re-machined to the typical Mixed Flow conical profile. This is a design nuance aimed at locally increasing the pump performance curve slope in the low flow region-say 0—30% of design flow. Appendix K describes this.

## Impeller outlet width; b2

In conjunction with pump speed, this *[or more exactly OABV]* will help decide how much flow the impeller can discharge. In combination with D2, this will also help decide the slope of the head flow curve. *[I.e. how quickly pressure head reduces as flow increases.]*.

Narrow impellers *[that is where b2 or OABV is relatively small]* will slope away from the zero flow head more steeply than a wider one.

## Collector throat total area; A3

This refers to the total passage minimum cross sectional are-, normal to flow, of the collector (Fig. 4.5).

A single volute collector is shown but for multiple volutes [i.e. doubles triple or quad]; it is the total of all throats. With Multiblade diffuser pumps, it is again the total of all throats.

## Impeller total outlet area between vanes; OABV

This refers to the total impeller passage outlet area, normal to flow. It is known that the ratio of this to the total collector throat area *[known as the Area Ratio, after Anderson* [Ref. 8]*]* also influences the shape/slope of the performance head flow curve. In instances where the OABV significantly

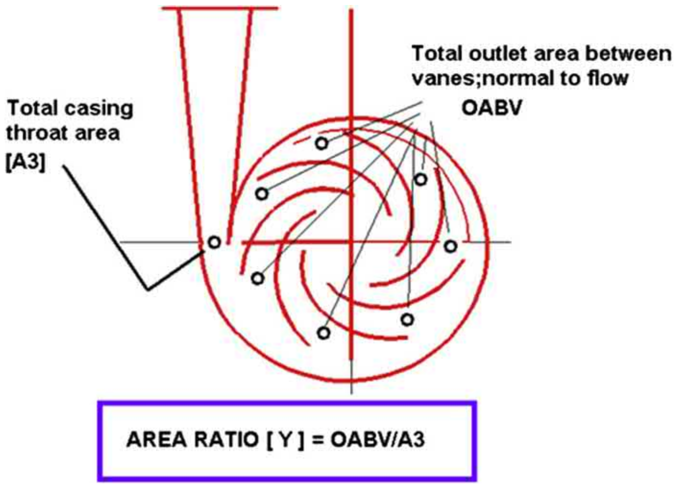



**Fig. 4.5** Illustrating two key cross sectional area on the liquid flowpath through the pump. This so-called Area Ratio is most relevant to the design of Radial Flow machines. It is much less applicable to Mixed Flow and particularly Axial Flow machines.

exceeds the throat area A3 in the ratio of 3 or more, then the head flow curve shape will be relatively flat, that is the pressure head at zero flow might only be 10–15% higher than at the design flow. If on the other hand, the ratios are reversed and the OABV is down to 1/3 of A3, then the head flow curve will be relatively steep. The pressure head at zero flow might be more than 50% higher than at design flow.

It is worth mentioning that the Area Ratio concept, whilst extremely powerful, is best suited to the design and discussion of so-called Radial Flow machines with Specific Speed [Ns] less than, say, 4000. This approach views an impeller as a collection of *[chiefly radial]* ducts or passages. However, Fig. 3.35 indicates that at the higher specific speeds of Mixed and Axial Flow machines, the head contribution from the vane shape starts to dominate. So their design leans more on aerofoil test data. Both design methods seems to work at or near to the design flow-where the vane setting angle closely matches the approaching flow angle. But at off-design flow, the Area Ratio takes no account of the incidence losses [see Appendix D and Figs. D4–D6]. As Specific Speed increases, these losses become more pronounced.

Whenever a pump is dismantled for repair or service, it is worth recording as much of this data as possible-including the number of impeller vanes. Manufacturers often have a wide range of impellers that fit a particular casing and occasionally spare part supply errors occur.

The single entry-single stage is the most prolific configuration and the one with which most people are familiar. Even so, there are some design aspects, which may need explanation. Liquid leaving the impeller rim will have a higher static pressure than when it entered at the eye. The natural tendency for the leaving high pressure liquid would be to flow back to the nearest low pressure zone. In a basic single entry impeller that will be the entry duct zone. So without some other arrangement, much of the liquid discharged from the impeller would naturally recirculate back to the entry zone instead of out and onward to the outlet connection. The most obvious method of preventing this recirculation is to provide an annular restriction on the front shroud *[front wear ring]*. The front shroud is chosen because this is the path of least resistance for the recirculating liquid. *[We shall see later that there are good valid reasons for providing a path on the rear shroud, then paradoxically fashioning a similar restriction.]* (Fig. 4.6).

Experience says that this annular restriction will be subject of routine wear by various mechanisms; hence, on most pumps they are made so they

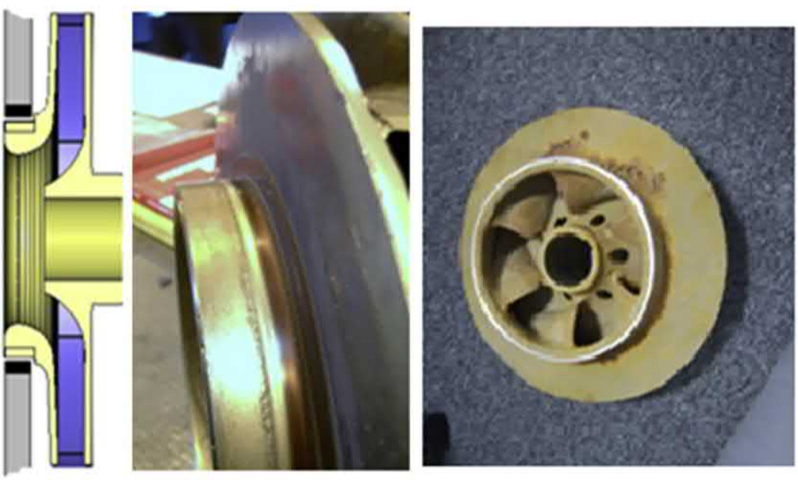


**Fig. 4.6** Shrouded impeller are usually equipped with rings on the front shroud at least. These are a close fit to the bore of a mating stationary rim on the casing. The annulus thus formed helps restrict high pressure liquid at the rim from flowing back to the low pressure region of the inlet eye. Generally, both these rings are replaceable when they become worn. An exception would be in the case of high speed machines, [where the loose ring mass may unsettle dynamic balance], or very inexpensive pumps [where they would be an over-refinement].

can be easily renewed/replaced as a consumable item. By restricting this wasteful recirculating flow, the pump obviously becomes more effective and efficient. Therefore controlling this recirculating leakage will be advantageous, and a great deal of engineering effort has been centred on this. First thoughts might be to make this restriction clearance very small, but this raises the risk of contact due to any combination of mechanical alignment, suspended solids in the pumpage dynamic shaft deflection and thermal growth. Continuous developments aim to either reduce the leakage flow for a given restriction clearance, **or** reduce the clearance and massively reduce the consequences of contact. The former approach has resulted in the constriction flowpath taking on complex labyrinthine passage shapes. The latter approach utilises advanced non-metallic material in less labyrinthine flowpath designs.

At least one user industry [Ref 9] has long held strong views on how small these constriction clearances should be. Based on historical experience, they have been happy to trade-off the slight negative efficiency effects of relatively high constriction clearances against the reduced risk of contact and associated reduced reliability. This historical experience was based on pumps with relatively flexible shafts. However, modern designs will tend to have much stiffer shafts than those in the past.

Though many impellers follow this simple single stage basic layout, many also utilise a ring on the back shroud as well *[back wear ring]*. This was alluded to earlier, and at first it might seem counterproductive to introduce an additional leakage flowpath. But there are two main reasons for this.

Firstly; the pressure head generated at the impeller rim will be applied all the way down both back **and** front impeller shroud. The pressure at the rim will depend upon the Specific Speed of the machine but typically will be in the region of 80–90% of the pump discharge pressure head (Fig. 4.7).

The product of differential pressure distribution and effective shroud area results in a piston force. To a degree, some of these opposing forces cancel out, at any diameter larger than that of the inlet. But significantly, in the most basic configuration shown, the pressure distributions on back and front shrouds are not the same. Considering the back shroud distribution first; at the rim it will be close to the pump discharge pressure head. The liquid trapped between the impeller back shroud and the stationary casing wall have little if any radial flow component applied, at least on a macro scale. But viscous drag forces will oblige this trapped liquid to whirl around the shaft axis at an angular velocity roughly half that of the impeller. In doing so, it loses some of its static pressure the nearer it approaches the shaft

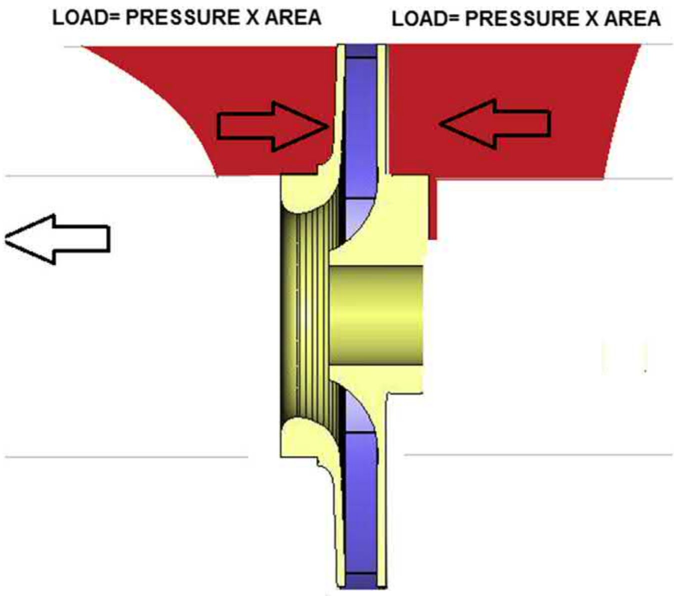



**Fig. 4.7** Typical static pressure distribution that will occur over the back and front shrouds of an impeller. Though they have similar values at the impeller rim, they change at different rates at similar radii from the shaft.

axis. Hence in the vicinity of the shaft seal [where fitted] the static pressure might reduce to 50%–70% of pump discharge pressure (Fig. 4.8).

The situation at the front shroud is quite different. By definition, the pressure head at the wear ring diameter will be close to, but somewhat above suction pressure. That is in order to provide enough differential pressure to drive liquid against the flowpath resistance in the clearances. Furthermore, this inward leakage flow introduces a radial flow component to the swirling flow field that is not present on the rear shroud flow field. The nett result is that the pressure distribution on the front shroud will be less intense than on the back. Furthermore, the effective piston area upon which this acts will, in most cases, be less than on the back shroud. So in the absence of any other arrangements, there will be inherent axial unbalanced load acting in the direction towards the suction inlet duct. This resultant axial unbalanced force has to be absorbed by some external bearing system.

A second consequence of this back shroud pressure distribution is that any shaft seal may be subject to a high percentage of the pump discharge pressure head. This might not be helpful to seal life.

Both of these last two issues can be effectively countered by simultaneously providing both a constriction ring on the rear impeller shroud and a connection to the low-pressure entry duct zone (Fig. 4.9).

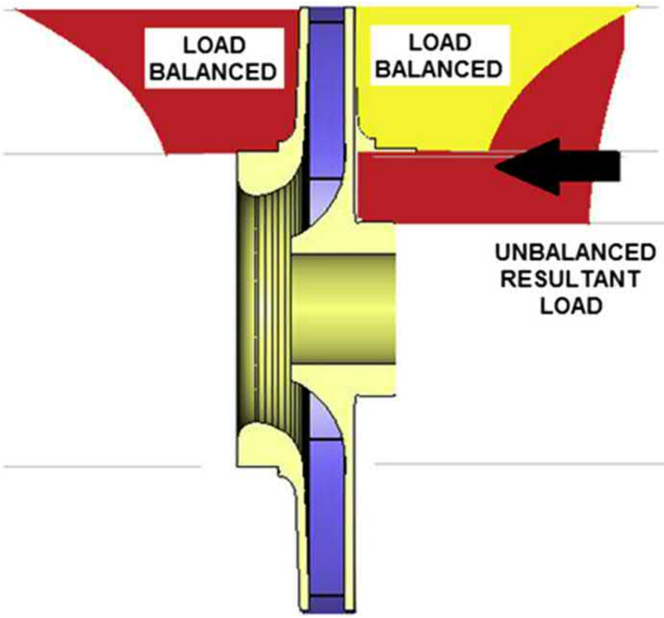



**Fig. 4.8** The difference between these two pressure profiles, acting over the shroud area, results in nett unbalance axial load being applied to the shaft bearing system.

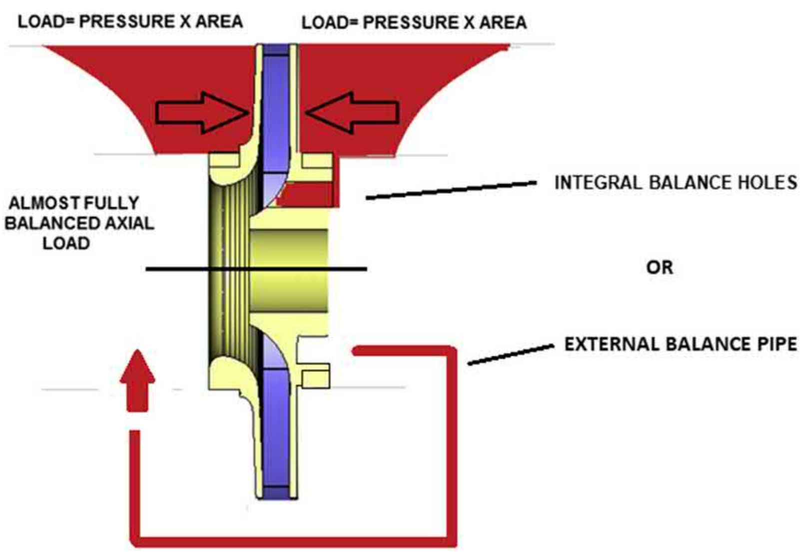



**Fig. 4.9** The nett unbalance can be largely eliminated by providing a similar flow constricting rings on the back shroud. This leakage flow is channelled back to the suction region, generally through so-called 'balance holes' but sometimes through an external 'balance pipe'.

Generally, this is achieved by holes in the impeller back shroud called 'balance holes'. *[In rare occasions, an external 'balance pipe' replaces these holes. This flowpath can have some detail benefits where it is known that the pumped liquid will contain a small amount of abrasive solids See Fig. J.106].*

In this way, the level of liquid whirl and radial inflow can be made comparable to that at the front shroud, ensuring a very similar pressure distribution. Obviously some small pressure head will needed to drive the flow through the balance holes/balance pipe. But this can be comparable to that needed to drive liquid through the front wear ring. There will almost always be a little residual pressure acting on the small annular area between wear ring and shaft. This creates a small remnant axial load. Hence the resultant axial load becomes nearly zero with the added benefit that the seal environment pressure is now substantially lowered. Both back and front wear rings are usually identical. But occasionally they are made different in order to stabilise the resultant axial load magnitude and direction *[This can also help avoid tendency for rotor to 'shuttle'].*

## Fully shrouded/closed impellers

Impellers thus configured as in the previous section, are called 'shrouded impeller', for obvious reasons. An alternative description is 'closed impeller' because the impeller liquid passages, *[mostly rectangular in section perpendicular to the flowpath]*, are closed on all four sides. Some closed impellers are produced from an assembly of welded metallic pressings. Others consist of a two-piece fabrication. This latter development has been common practice in the manufacture of turbocompressors

However; most closed pump impellers are still cast or moulded in one piece. This means the manufacture of the raw casting or moulding is slightly complex in that the internal passageways must be formed by additional tooling components that are called 'cores'. This is a routine process and possesses little or no problems during production, unless the passageways are unusually narrow.

We have mentioned that low flow impellers will have a small outlet width/area. We also know that relatively high pressure head designs will possess large outer diameters. There comes a point where the ratio of outlet width to maximum diameter ***[b2/D2]*** becomes a practical constraint (Fig. 4.10).

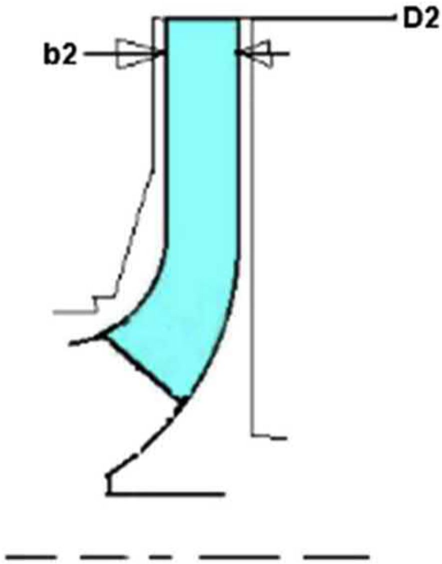



**Fig. 4.10** Impeller 'castability' is influenced by the impeller width to diameter ratio, b2/D2. Values below 0.03 may require special attention in the foundry.

Experience says that once the outlet passage width b2 becomes less than say 2–3% of the maximum outlet diameter, D2, then castings may become troublesome. That is not to say they are impossible, just that they are prone to difficulties in the foundry process. This difficulty centres on the fragility of the 'cores'. Blade–to–blade passageways on the inside of shrouded impeller are formed by what is known as 'cores', mentioned above. These are often fashioned in sand-like materials to which various binders have been added. Molten metal flows around these cores in order to form the passage shape. Once the metal has cooled and frozen these sand cores are 'knocked out' or disintegrated-leaving the expected passage. Narrow outlet widths mean thin sand cores and these are susceptible to distortion from the molten metal, as well as handling during their manufacture. Furthermore, it is difficult to 'knock out' the sand core of narrow impellers. And it is also difficult to clean out the casting because narrow width can hinder tool access. Core materials that are superior to sand can help to extend this limitation, but may not to avoid it completely.

However, the production of cores adds to the cost and timescale of casting or moulding manufacture.

## Semi open impellers

Fortunately, the cost of 'cores' can be largely sidestepped by the manufacture of so-called 'semi-open' impellers. In such designs, only three of the four water passages walls are formed in the impeller casting/moulding. A stationary face on the pump casing forms the fourth side. In practice, there will be a small setting gap between the rotating impeller and the stationary casing face. In principle the impeller manufacture can be accomplished with a simple coreless two part mould. This design brings with it some advantages as well as disadvantages, relative to the closed design described previously (Fig. 4.11).

Its chief advantage is its lower cost and weight. A further advantage is more direct access to the water passages for cleaning during either manufacture or service. In addition, it is known that semi open impellers are better able to handle fibrous material entrained in the pumped liquid. That is because, up to a certain fibre size, they become mechanically fragmented in the small setting gap. In 'closed' designs, there is often a risk that fibrous material will just 'choke' the liquid passages and degrade performance. To an extent, the semi-open impeller helps avoids that risk (Fig. 4.12).

**Fig. 4.11** In semi open impellers, the front shroud does not exist. Nor does the front wear ring.

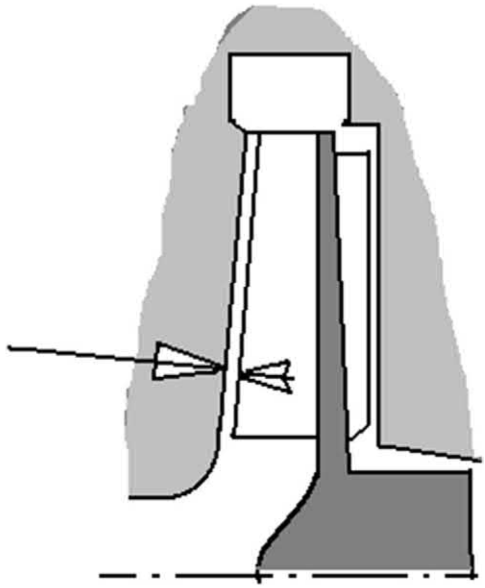


**Fig. 4.12** In semi open impellers, the stationary casing wall acts as the fourth impeller passage wall. The fine clearance between impeller vane tips and the casing functions as a leakage restriction. In such designs leakage occurs over the vane tips, rather than radially-as with shrouded impeller.

The small axial setting gap in a semi open impeller is hydraulically comparable to the radial gap in the wear rings of conventional closed impellers. In both cases, there will be a finite amount of liquid flow 'leakage' that would otherwise be discharged down the outlet connection. In other words, this leakage represents a small loss in pumping capacity. Clearly this also reduces pump overall efficiency. When the wear ring gaps of 'closed impeller' increase due to, say, corrosion or erosion from solids suspended in the liquid, then this efficiency reduction is a one-way process. The performance decay can only be halted by renewal and replacement. With semi-open designs, provisions can be made in the mechanical design to readjust its setting gap, sometimes even when the pump is running! Hence any efficiency decay can, up to a point, be reversed.

But semi open impellers also have some negative points relative to closed impellers. The chief concern is of unbalanced hydraulic axial load.

This Fig. 4.13 portrays the internal pressure distribution acting on a semi open impeller. A closed impeller is shown for comparison. Straight away it is clear that a closed impeller can be in nominal axial balance, as articulated

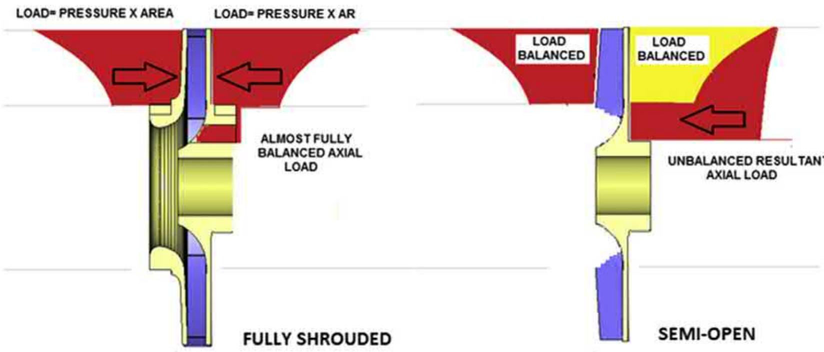



**Fig. 4.13** Comparative pressure distributions for closed and semi-open impeller configurations. Bring eye up to pressure.

earlier. As a refinement, it could be deliberately unbalanced by some specific degree to [say] compensate for downward dead load in a vertical shaft configuration. This could be achieved by making the back and front wear rings different diameters. But in most cases, the rotor system will be in near axial balance. However, the natural pressure distributions in the most basic Semi-Open impeller will result in a finite axial load in the direction of impeller inlet duct. The chief contribution to this will come from the overpowering high-pressure acting down the outside of the back shroud. As with those shrouded impellers without back rings, the average pressure will be in the region of 80%—90% of the pressure head at the pump outlet connection. To some extent, this back shroud pressure distribution can be modified to more closely approach that of a shrouded impeller. This will obviously help reduce the nett axial load. The most obvious approach would mirror that for the shrouded impeller and directly connecting the [region near to the shaft and seal] to the pump suction, via a cylindrical wear ring. In this way, the back shroud pressure distribution can be made to closely approach of the front. As was the case with shrouded impellers, near axial balance might be achieved. But there is another way to improve the back shroud pressure distribution and that is to arrange vestigial radial vanes on the back shroud. Like the front/main vanes, there is a small controlled running clearance between them and the stationary casing. In the simplest case, these are straight radial paddle vanes.

Their aim is to generate a shroud pressure distribution that closely duplicates that of the front main vanes. While this is a good first step with pumps that are to handle liquids that are chiefly clear, they are not ideal if the

**Fig. 4.14** Back pump-out vanes are more efficient if they somewhat follow the spiral shape of the main vanes. Simple radial paddle vanes are sometimes used instead, but they may promote accelerated erosion effect if dirty liquids are being pumped.

liquid contains fine hard abrasive solids in suspension. A later Chapter describes how local vortices create erosion hot-spots. [Appendix J] (Fig. 4.14).

In a refinement, these straight paddle vanes are instead curved to more or less follow the spiral shape of the front vanes. It is known that such designs consume a little less power and so more efficient than simple radial vanes.

*[While such back pump-out vanes are ubiquitous among open and semi open impeller designs, they are occasionally seen on shrouded impellers. In this instance the pump will most probably be handling abrasive solids in suspension. The purpose of the pump-out vanes here is simply to try and centrifuge the heavier particles away from the annular wear ring clearances.]*

On the downside, the effectiveness of any back vane configuration rests on their running clearance against the casing. Like the front vanes, they can be susceptible to wear-out and this will affect their pressure distribution profile. Furthermore, significant liquid temperature changes can, through differential expansion, cause the clearances to change. The nett axial thrust will change accordingly. Often this issue can be controlled by choosing a suitable thrust bearing capacity margin. Small or less critical pumps are designed this way. However, a similar margin cannot be always be attained in larger, more powerful machines.


**Fig. 4.15** Reducing the projected area of the back shroud by 'scalloping' can help to reduce axial thrust load levels.

Some Industry—Specific user codes do not permit axial thrust balance to be achieved by the use of such vanes, due to their less predictable long term thrust performance, when compared to the combination of back and front wear rings [9].

Another way to help accomplish better axial balance is to reduce the 'piston area' upon which the pressure distribution acts. In the example just described, the impeller shroud or disk is circular and both main [front] and back vanes are attached to it. By removing 'bites' from this disk, between the vanes, the projected shroud area is reduced and so therefore is the nett axial a load (Fig. 4.15).

As a matter of course, the back and front pressure distribution profiles then approach similarity, which helps further. The main disadvantage of this design approach is that the outer sections of the main vanes are less well supported. Thus, they are less able to resist the bending forces induced by both the centrifugal forces acting on a spiral vane, as well as pressure distribution existing across each vane.

The concept of reducing or 'scalloping' the shroud disk area can be extended very considerably to the point that very little remains. In this way, axial load can be very substantially reduced. Such configurations are termed 'Open Impellers' as opposed to the 'Semi open' and 'Closed' designs previously discussed. But the transition between open and semi open is ill defined. Because of bending force limitations just described, this configuration is mostly limited to small or low-pressure head pumps.

So, even the hydraulically simple impeller arrangement of Single Entry-Single Stage has a number of nuances-Closed, Semi Open and Open. In addition, as discussed previously, they can exist in a wide range of hydraulic shapes, from the low flow-high pressure head designs to the high flow —low-pressure head types (Figs. 4.16 and 4.17).

## Single entry-single stage design limitations

Are there any hydraulic design limits to this basic single entry-single stage design? Well at any given speed, the design flow will dictate the diameter of the inlet aperture, the so-called impeller eye. [D1]. The impeller pressure head output is chiefly dictated by its maximum outer diameter [D2] (Fig. 4.18).

Between these two diameters, the pump designer must lay out a blade shape that efficiently does work on the liquid passing through the impeller. A low flow/high head pump will tend to have a small eye diameter and a large outer diameter. Consequently, there will be ample radial space to layout a satisfactory blade design (Fig. 4.19).

**Fig. 4.16** Closed or Shrouded impeller designs are available in a wide range of Specific Speeds, ranging from Low [narrow relative to diameter] to High [wide relative to diameter].

**Fig. 4.17** Semi-open impeller designs are also available in a wide range of Specific Speeds, ranging from Low *[narrow relative to diameter]* to High *[wide relative to diameter]*. Because very Low Specific Speed closed impeller are very narrow relative to diameter, they can be difficult to cast, so semi open designs are the only practical option.

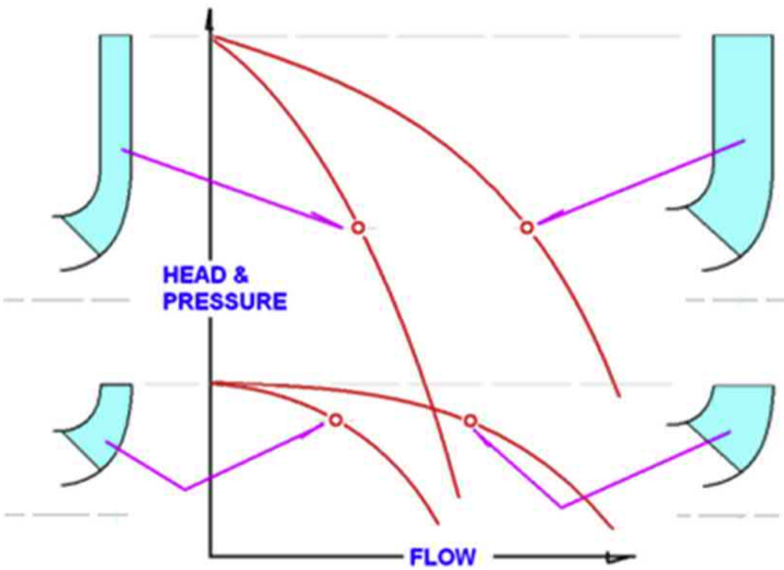



**Fig. 4.18** Reminder of how impeller outlet width and diameter can influence pump performance.

As design flow increases and simultaneously design pressure head reduces, these two critical diameters converge. A point is reached where there is insufficient radial space in which to layout an efficient blade designs.

One solution to this lies in the strategy of double-entry design configurations. A double-entry design will normally have a smaller eye

**Fig. 4.19** This Single Entry impeller is near the limit for High Specific Speed radial flow designs. The radial space available *[between inlet diameter D1 and outlet diameter D2]* for efficient vane design is restricted.

diameter than its equivalent single entry *[Each half impeller entry now only handles half the total flow]*. So the maximum design flow limitation can be increased before the radial space constraint again applies. This concept is discussed next.

## Summary – single entry-single stage impellers

This basic single entry – single stage configuration can cover many practical applications and is the most commonly encountered (Fig. 4.20).

The vast majority of centrifugal pumps follow this format in its various nuances.

- It is available as a Shrouded, Semi open and Fully open design.
- At any shaft speed;
  - o The lowest design flow for single entry – single stage shrouded designs is constrained by the outlet width ratio, b2/d2. [*The constraint is overcome by the multistage concept]*. No such constraint exists for open or semi open designs.
  - o The highest design flow for Shrouded, Semi open and open designs is constrained by the Inlet to Outlet ratio, D1/D2. This constraint is overcome by the double entry single stage concept.
- This configuration can have high axial shaft load if not compensated for in some way.

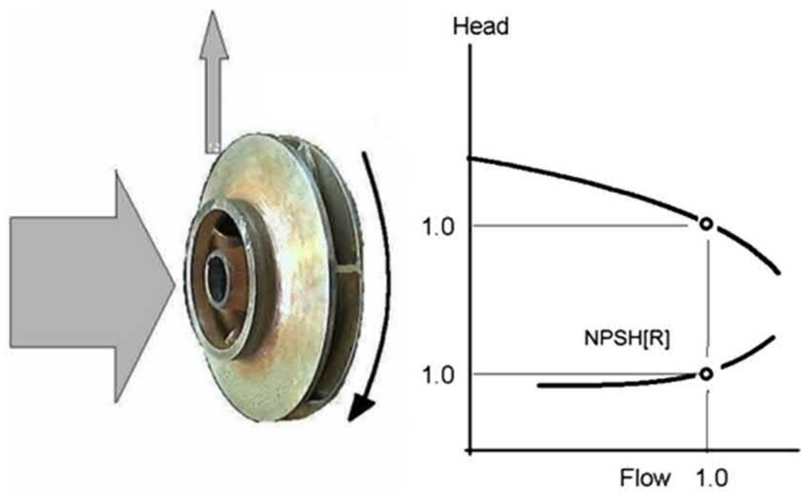



**Fig. 4.20** Baseline performance of a single entry impeller.

## Double entry-single stage – parallel flow impellers

This is the second most common configuration. Its most obvious feature is that it has two entrance apertures or 'impeller eyes'. Conceptually they can be imagined as two Single Entry-Single Stage impellers arranged back-to-back and operating in parallel (Fig. 4.21).

Usually the two impellers are produced as an integral casting. But in some rare cases, they are manufactured as two separate items. It is most common for both impeller halves to be mirror imaged relative to each other. But sometimes, in pumps of high energy, the two halves are angularly rotated [or staggered] from each other so that the respective emerging passage flow mixes-out to a more uniform pattern. This has the benefit of reducing any associated hydraulic excitation, and so can reduce vibration levels.

This layout has some advantages over a single entry design, the chief one being that, at any total flow they can operate with a lower suction pressure than a comparable single entry design. Earlier, I discussed how in a performance test, the limiting suction pressure can be measured in terms of the property known as NPSHR. Operation close to or below this limiting value will be associated with 'cavitation' within the machine. As a reminder cavitation, if well-established, can have an immediate effect of reducing machine performance and is often accompanied by noise and vibration. It can also have the longer-term effect of progressively damaging the impeller.

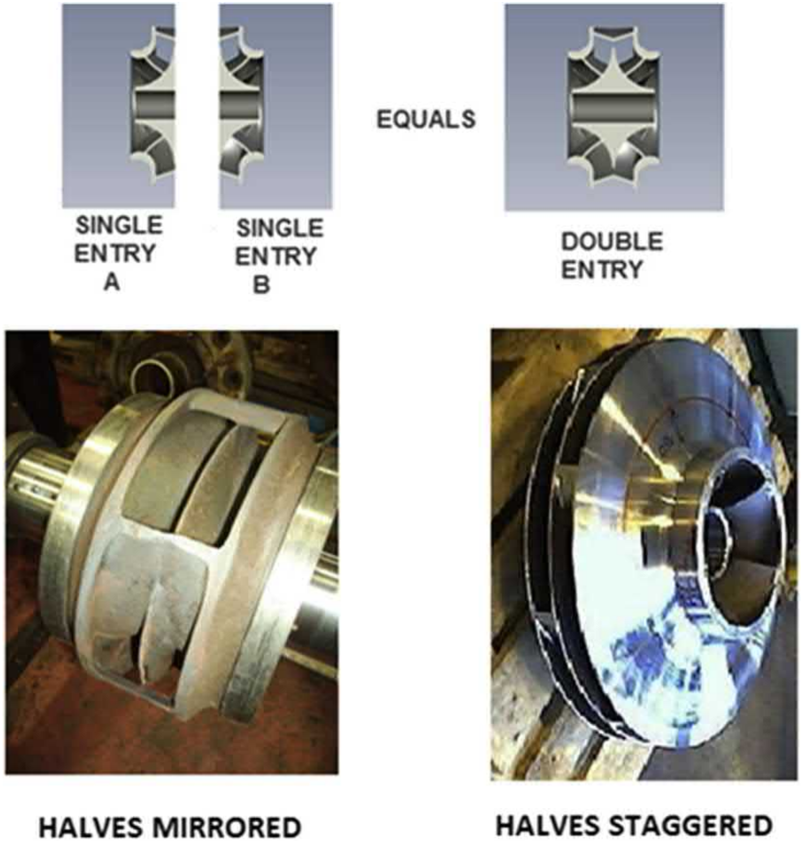



**Fig. 4.21** Conceptually, a double entry impeller can be imagined as two single entry designs arranged back to back, and pumping in parallel. Sometimes the two halves are actually produced separately, but more often they are integrated into one casing or component. Staggering the two halves can help to reduce overall pump vibration levels. Sometimes the stagger is equal-as shown here. In other cases, the stagger is unequal and further vibration reduction is claimed.

But this sinister damaging effect can occur even when cavitation is less well established and there are little if any outwards signs in terms of performance decay, noise or vibration.

It is known that, all other things being equal, one of the most influential design factors controlling this NPSHR is the maximum diameter of the entrance aperture or 'eye'. Up to a point, increasing the eye diameter brings about a reduction in NPSHR [limiting suction pressure]. But as in so many other design decisions, this NPSHR improvement comes at a price. *[Higher* *backflow onset point, narrower operating COMFORT ZONE, Lower efficiency].* Other negative factors simultaneously increase and so a compromise often has to be struck. Beyond this 'optimum' inlet diameter, NPSHR actually begins to increase again.

Splitting the flowpath into two entry apertures or 'eyes' allows their maximum diameter to be reduced along with the limiting NPSHR. Consequently, a 'double entry' impeller can have the same flow as a single entry impeller, but a lower NPSHR (Fig. 4.22).

Conversely, a double entry impeller may deliver a higher flow, under the same suction pressure conditions as a single stage.

Apart from one small design detail, double entry designs are normally symmetric. This means that the internal pressure distribution acting on the shrouds is in balance, at least around the design flow point (Fig. 4.23).

While at first sight, this may seem attractive, it is not always a welcome condition if the pump has a horizontal shaft, and is furnished with rolling element rotor location bearings. Such bearings are unhappy running in a very lightly loaded state and can experience destructive 'skidding' within the elements. For that reason, in horizontal shaft pumps, it is not unusual for the wear rings of one 'eye' to be of a slightly different diameter than the other. This design detail produces an inequality in the opposing pressure distributions and hence a SLIGHT axial loading on the location bearing.

So, one might assume that double entry impellers are either in complete axial balance or at least consistent unbalance right across the flow range of

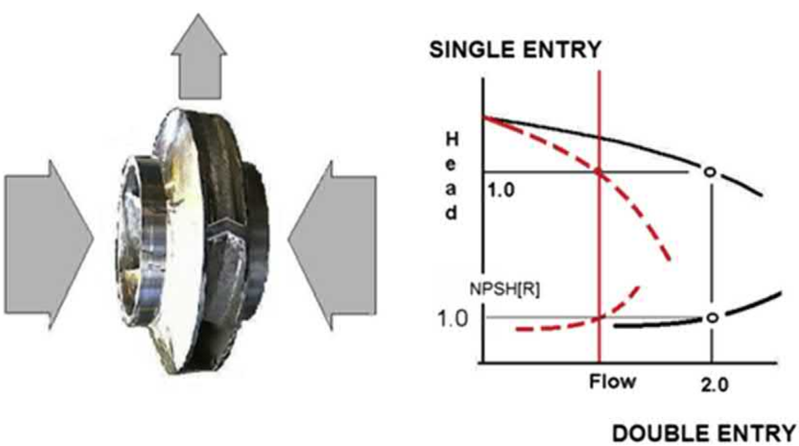



**Fig. 4.22** Two impellers in parallel deliver twice the flow but only need the same NPSH [R] as one single impeller.

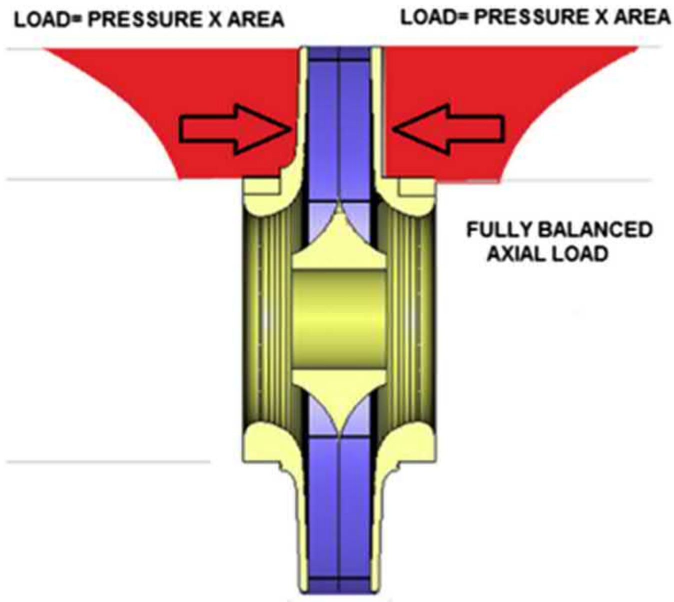



**Fig. 4.23** Internal pressure distribution acting on impeller shrouds counter can balance each other and nominally negate any axial load. Complete balance is unsustainable at every flow, due to manufacturing discrepancies. So the wear ring diameters are often made slightly different so as induce a slight positive load in one direction.

the machine. But it is known that at low flows, [say less than 20%–30% in 'commercial' pumps], this is not the case. Some pumps will experience random axial movement of the shaft, known as 'shuttling'. The root cause of this lies either with manufacturing variations of the hydraulic castings, or with inherent hydraulic behaviour of the inlet apertures. It is discussed in more detail in Appendix D. Offsetting the wear ring diameters, as just described above, can help to reduce this tendency.

## Summary – double entry-single stage impellers

- Can help to extend the high flow constraint of single entry designs
- Can help reduce the NPSH [R] constraint of single entry designs.
- Can have very low axial shaft load over most of its operating range
- This type of impeller is almost always produced in the Closed/Shrouded configurations. But Semi open versions are occasionally found in applications, such as Pump and Paper production, where the pumpage might contain fibrous material. Semi open designs can help to macerate the fibres to an extent.

Double entry impeller extends the design **flow** constraint of a single entry design. But how can the **pressure head** constraint of a single or double entry impeller be extended?.

## Single entry-multistage-series flow – stacked rotor

At any given speed, the stage pressure head is largely dependent on the impeller diameter; [D2]. To gain more output head, the impeller diameter can be increased. However there is a limit to how far this is practical. As explained earlier, the width to diameter ratio b2/D2 becomes a limitation.

Manufacture of narrow impeller design by the metal casting process can be quite challenging. This issue can be overcome to a large extent by fabricating the impeller as a 'semi open' design, to which the front shroud is later attached. This fabricated impeller approach is popular in non-metallic designs, particularly where volume production is concerned. Another solution involves fabricating the impeller from pressed metal sub-components. Here the initial tooling investment can be high compared to that for sand castings. Both approaches can be seen in compressor design.

But even when the manufacturing difficulties are overcome, large narrow impellers have another drawback. And that relates to their efficiency, which will be inherently low. This is because the viscous energy wasted in spinning a large impeller disk can become significant, even though some of that energy may reappear as a small contribution to the pump's pressure head.

A strategy that can reduce both these concerns is that of multistaging. In this approach, liquid is fed successively through two or more impeller stages set out in series along the flow path. *[Two stage pumps are the most popular of the multistage variety, and are dealt with separately later]* As it passes through each stage it has incremental work done on it to achieve the higher desired pressure head (Fig. 4.24).

Since the pressure head per stage is now reduced, so is the diameter. Consequently, the width to diameter ratio is also improved of course. So, up to a point, a multistage version should always be more efficient than a comparable single stage design, all other things being equal. Even so, once the geometry of each individual stage begins to approach the previously mentioned width to diameter ratio, then the limit of this strategy is reached and other approaches must be explored. This might involve increasing the running speed, or even alternative designs, such as reciprocating pumps.

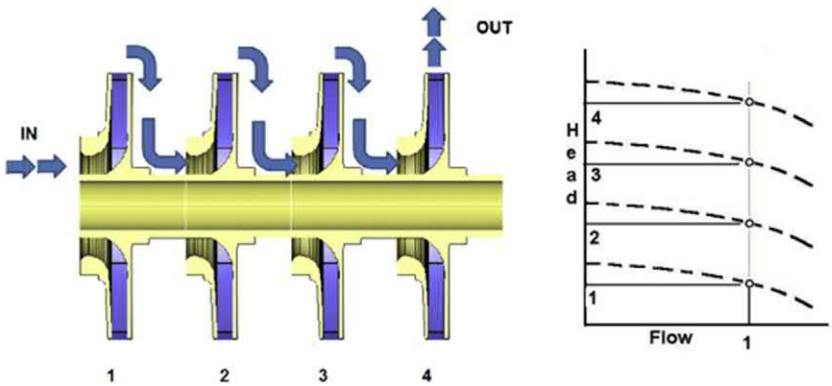



**Fig. 4.24** The Multistage effect: Liquid passes from stage 1 to stage 4, receiving incremental pressurisation in the process. Liquid leaves the having gained 4 times the pressure head of a single stage. The arrangement shown here is of a 'stacked rotor'.

Rotors in which all the impellers face the same way are called 'stacked rotors'. This reflects the way they in which they are built, with the impellers stacked onto the shaft one after the other.

Horizontal Multistage rotors are almost always built around shrouded [fully closed] impeller designs. Very few cases of semi open or open impeller multistage pumps exist. Those that do were built for very special purposes. Multistage pumps with semi-open impellers are more often arranged with in a vertical shaft configuration.

In the stacked rotor configuration, the flowpath starts at one end of the machine and progresses to the other. Each stage is more or less identical, though sometimes the first stage is different. Typically it may have an enlarged 'eye' [D1] since it has sole responsibility for operating under difficult suction pressure conditions. If and when it does this successfully, the second and subsequent stage will not face this issue of course. Their NPSH [a] has been boosted by the special first stage, so can be designed free of any compromise in this respect. This tactic has some limitations, but these can be overcome by a double entry first stage alternative. *[Although below suction branch sizes of 3 or 4 inches, the real practical NPSHR benefit of this alternative is questionable.]*

Since in this execution, the impellers are "stacked" one behind the other, it is inevitable that any associate axial loads are also stacked and accumulated (Fig. 4.25).

If the pumps are relatively small, *[less than, say, 3 inch discharge size]* then the impeller can use the concept of back and front wear rings, just as with

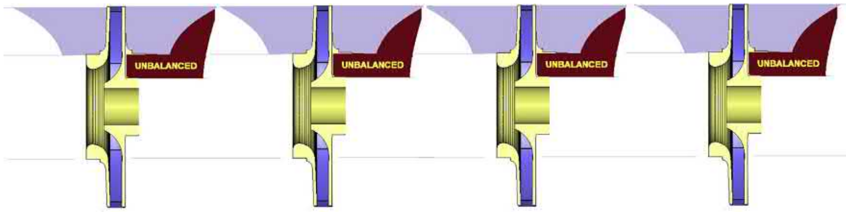



**Fig. 4.25** In stacked rotor multistage pumps, the unbalanced axial load is also multiplied.

shrouded basic designs. As before, this helps equalise the back and front pressure distributions. This will tend to suppress unbalance in the resultant thrust bearing loads (Fig. 4.26).

But of course this second ring also has some leakage associated with it. One way of changing this leakage rate is to change the back wear ring diameter. This will simultaneously increase the remnant axial rotor load. The designers thus has to trade-off the reduction in remnant axial load against the reduction in efficiency.

Usually the remnant axial loads can be handled by simple rolling element bearing, or similar. But in medium to large machines, the loads exceed the commercial limits. The constraint dictated by rolling element thrust bearings can be extended by more complex and costly bearing systems, such as pressure fed hydrostatic systems. But ultimately even these will also pose a finite limit.

In these cases of higher energy machines, the unbalanced axial thrust is usually counteracted by some form of internal hydraulic load balance or compensation device. Only the [*often still significant*] remnant axial load is now absorbed by an external thrust bearing (Fig. 4.27).

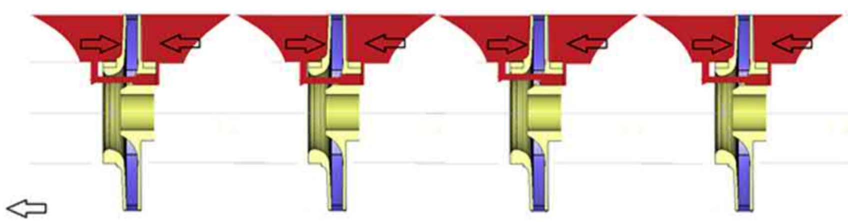


**Fig. 4.26** Here a stacked multistage rotor consisting of impellers with back AND front wear rings is shown. Again the residual thrusts are multiplied. But the total is much less than with the unbalanced impeller shown above.

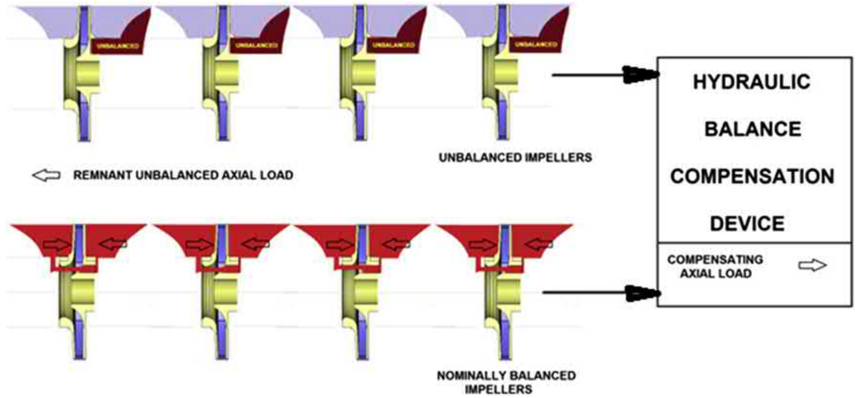



**Fig. 4.27** With either nominally balanced or unbalanced impellers, the residual axial load can be largely compensated by an hydraulic device. These devices are discussed later.

Hydraulic balance compensation devices are described later in the Chapter 10; 'Has Pump Started Satisfactorily?' But to pre-empt; the pressure head of the pump is arranged to act over an area that is coaxial and rotating with the shaft. The resultant force is arranged in such a way as to oppose the impeller axial unbalanced load and so significantly reduce the remnant loads that the external bearing system has to absorb. Since such devices use the active pump pressure head, they are largely self-adjusting.

## Summary – multistage series flow – stacked rotor

- Will significantly increase the threshold pressure head limits of single stage machines.
- Will inherently generate high axial shaft/bearing loads unless otherwise addressed by;
    - o Balanced impeller design
    - o Specialised thrust bearings
    - o Inclusion of 'Hydraulic Balance Compensation' devices
- In the past, stacked rotor pumps were available with surrounding pressure casings that were either radially or axially split. Nowadays, the vast majority of stacked rotors are only surrounded by symmetric radially split casings.

## Single entry-multistage-series flow – back to back rotor

In the previous discussion, the impeller flow was arranged in series and the impellers were essential identical. They all faced the same way when

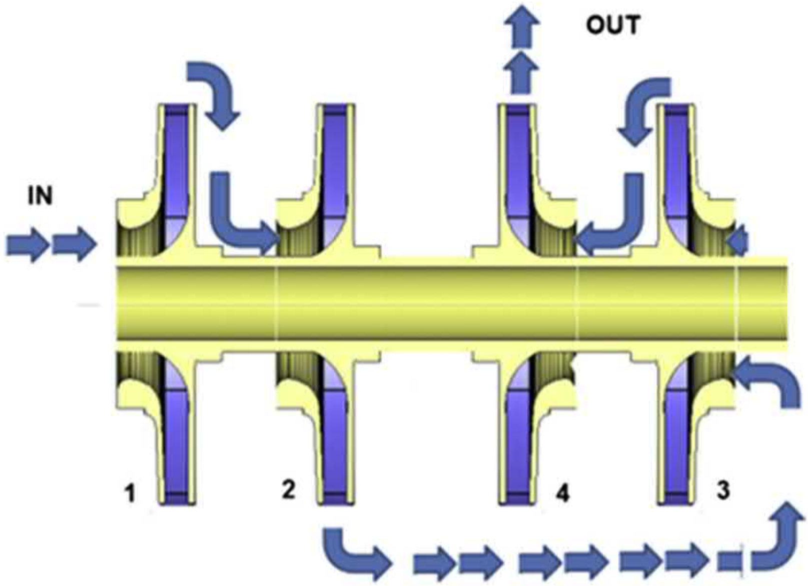



**Fig. 4.28** In contrast to stacked rotors, the Back-to-Back arrangement can [nominally] result in very little axial unbalance [At least, when the stage count is also even].

positioned on the shaft. Flow passed consecutively from one impeller to the next (Fig. 4.28).

An alternative exists. Here the number of impellers is divided into two sets, where one set is the mirror image of the other. It might appear that this option is only available with an even stage count. In fact odd stage counts are also possible. At first sight this might imply that there will be a remnant axial load caused by the odd stage. In fact that is not necessarily the case. By proportioning other components on the shaft, they can be made to behave in much the same manner as Hydraulic Compensation devices.

Each group is then arranged in opposition on the shaft. Flow that has progressed through the first part of the machine is ducted to the opposing end *[through 'cross-over passages']* and progresses back in the reverse direction.

Immediately it is clear that in such an arrangement, the opposing hydraulic axial loads in each half of the machine almost completely cancel out and leave the rotor with very little remnant axial loading. This is probably the most attractive feature of this configuration. It's often assumed that because the mechanical layout is symmetrical left and right, that the axial thrusts will exactly cancel out and leave zero remnant thrust. But that is not the case. Although the mechanics are symmetrical, the hydraulic picture is not.

*[These last remarks of course refer directly to pumps with an even stage count. They still apply to pumps of an odd stage count, with the proviso that the pseudo-hydraulic balance device helps overwrite the asymmetry].*

Looking at stages 3 and 4 in the Figure; the static pressure at the rim of impeller 2 will be 2/4 stage pressure head. At the rim of impeller 4 it will be 4/4 stage pressure (Fig. 4.29).

So the liquid at the rim of the final impeller will want to naturally flow back to the rim of impeller 2. Hence there will be leakage flow radially INWARD on the back shroud of 4 and radially OUTWARD on the shroud of impeller 2. This causes the pressure distribution on the back shroud of stage 2 to be different from all the other stages. Although relatively small, this pressure difference may be acting over a large back shroud 'piston area' and will result in substantial departure from the axial balance first assumed. This 'hidden' thrust results from the inter-stage leakage flow effect.

Pumps with back-to-back rotors are available with either axially split or radially split surrounding casings. The majority have axially split casings, with radially split casing tending to be the preserve of advanced specialised machines.

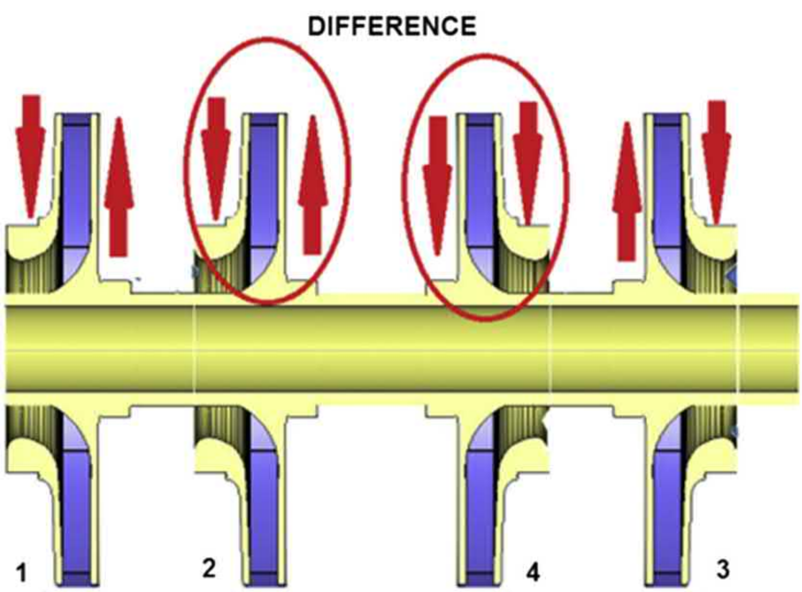



**Fig. 4.29** Though Back-to-Back rotors are nominally in axial balance, there will be some unbalance created by the non-symmetric flow conditions that exist on either side of the central bush.

## Summary – multistage series flow; back-to-back rotor

- Will significantly increase the threshold pressure head limits of single stage machines.
- Can achieve very low axial thrust loading
- Are available chiefly in axially split casing, but a number of radially split versions are also available.

## Single entry-multistage-series flow – special case – two stage back-to-back

In terms of quantities manufactured, the Single Stage-Single entry configuration is definitely the most popular. The double entry layout described earlier then extends the **flow** capability of this configuration. Similarly, two stage machines then extend the upper **pressure head** capability (Fig. 4.30).

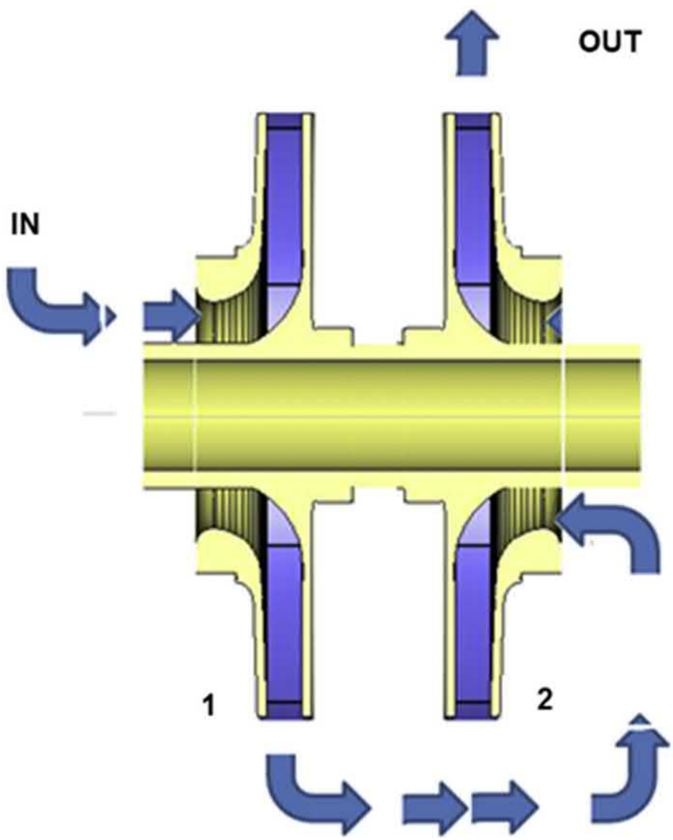



**Fig. 4.30** Two stage Back-to-Back impeller rotor.

Two stage impeller pumps of series flow -stacked rotor construction are quite normally available in radially split casing layout. The chief constraint will probably be that in certain orientations, the inlet and outlet branch flanges may clash. The same 2 stage stacked rotor configuration with an axially split casing would be much less common nowadays.

However two stage axially split case pumps with Back-to-Back rotors are also quite common. In fact two special sub categories exist. The impellers can be positioned such that each eye faces away from the other. This is the Back-to-Back version just described. The chief characteristic of this layout is that only the second stage shaft seal environment is naturally pressurised. The centre throttle bush can provide some rotordynamic support in certain cases-due to the inter-stage leakage effect.

## Single entry-multistage-series flow – special case – two stage front to front

In the alternative configuration, the impeller eyes face each other. Logically this would then be called a Front-to-Front version, but is rarely described this way. One feature of this front to front arrangement is that each mechanical shaft seal will be at a higher static pressure than if in the back-to-back layout. When pumps handle liquid that is close to its' vapour pressure, this can be an advantage (Fig. 4.31).

## Double entry-multistage-series flow

Nowadays multistage series flow designs are almost always built around the concept of singe entry impellers. In the past, however, it was common to also base such machines on double entry impeller. Today it would be most unusual to see new multistage pumps designed along such lines. The associated flow ducts are indeed complex, but not impossible to produce. One of the chief advantages of this layout is that each impeller is more or less [but not quite] in axial balance. So the need for special purpose thrust bearings can often be avoided. This helps avoid what was once the Achilles heel of multistage-series flow, stacked rotor machines.

## Double entry-multistage-parallel flow

This configuration is more or less restricted to engineered and special purpose machines. It does have attractions in certain cases. In such a layout, the pump can, by change out of the interconnecting stage pipe be set up as a two stage parallel flow multistage or two-stage series flow machine. Such flexibility might be an attractive way of approaching initial and final flow conditions on a complex pipeline configuration.

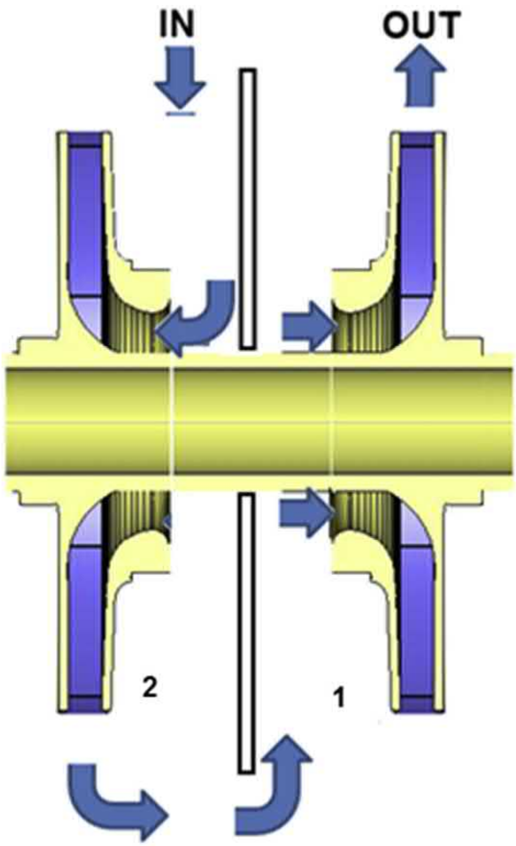



**Fig. 4.31** Two stage Front-to-Front impeller rotor.

## Multistage-series flow; hybrids

This is a very common configuration. In the most basic multistage machine, all impeller are identical hydraulically. They may have different directions of rotation of course if arranged Back-to-Back. In particular, the size of the inlet annulus, D1, will most probably have been selected to help the pump attain its highest efficiency. However, it is quite usual for this first stage to have a larger diameter, D1. As explained earlier, this will help to lower the operating NPSHR threshold constraint. This improvement is not dramatic, but it is often useful on the occasion where the available NPSHA is marginal. Since this is executed on only one stage, the negative effects on efficiency are vanishingly small. However, there are distinct limits to this tactic. If overdone, then the first stage will be prone to a number of operational bad habits such as surge and pressure pulsations if the NPSH margin becomes too small.

In that case, a double entry first stage impeller is a viable alternative. A parallel flow [double entry] first stage is best able to handle conditions of low NPSH itself, while able to pass flow to the subsequent single entry stage at a pressure high enough to avoid cavitation. This combines the low NPSH [R] benefits of the double entry impeller with the high generated pressure head benefits of single entry multistage series flow designs. This very popular hybrid is found on both horizontal and vertical shaft machines. It turns out that the benefit of a double entry impeller over a single entry, in the size range of popular multistage machines is about 3–4 inch suction branch diameter. Below this branch size, the benefit can be questionable, except in special circumstance (Figs. 4.32 and 4.33).

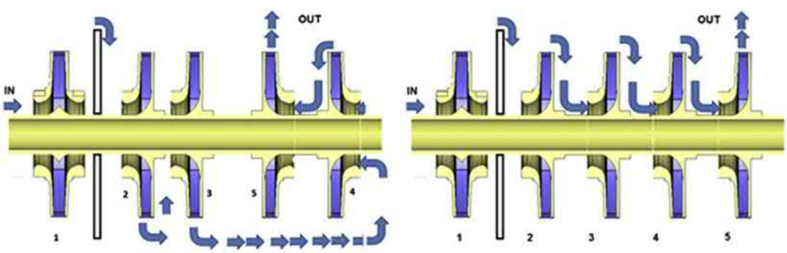


**Fig. 4.32** Hybrid Multistage rotors, with double entry firsts stage impellers.

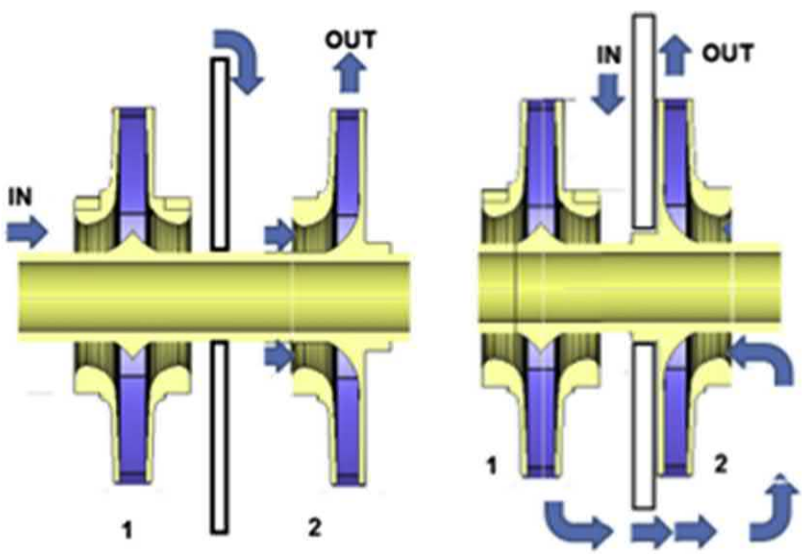


**Fig. 4.33** Two stage rotors with double entry first stage impellers.


**Fig. 4.34** Family trees for the most common rotor configurations. Tables 4.1 and 4.2 attempts to lists their popularity.

**Table 4.1** Common configurations when arranged with a Horizontal drive-shaft (see Fig. 4.34).

| Designation | Occurence | Comment |
|---|---|---|
| SS1 | Very common | Ubiquitous |
| SS2 | Very common | Ubiquitous |
| SS3 | Less common | |
| SS4 | Very common | Ubiquitous |
| SS5 | Uncommon | Special applications |
| SS6 | Very uncommon | Little if any commercial use |
| MS1 | Very common | Ubiquitous |
| MS2 | Uncommon | Special applications |
| MS3 | Very uncommon | Little if any commercial use |
| MS4 | Uncommon | Usually less than 4 stages. One MS4 stage may be hybrid with several subsequent MS1 stage. |
| MS5 | Very uncommon | No commercial use |
| MS6 | Very uncommon | No commercial use |

## Impeller configuration navigator (Table 4.2)

**Table 4.2** Common configurations when arranged with a vertical drive-shaft [see Fig. 4.34].

| Designation | Occurence | Comment |
|---|---|---|
| VSS1 | Very common | Ubiquitous |
| VSS2 | Very common | Ubiquitous |
| VSS3 | Less common | Mixers? |
| VSS4 | Very common | Ubiquitous |
| VSS5 | Uncommon | Special applications |
| VSS6 | Very uncommon | Little if any commercial use |
| VMS1 | Very common | Ubiquitous |
| VMS2 | Very common | Ubiquitous |
| VMS3 | Very uncommon | Little if any commercial use |
| VMS4 | Uncommon | One VMS4 stage may be hybrid with several subsequent VMS1 stage. |
| VMS5 | Very uncommon | No commercial use |
| VMS6 | Very uncommon | No commercial use |

# CHAPTER 5

# Pipe system basics

## Introduction

Liquid system designers *[quite rightly]* express interest in many aspects of a pump's performance and construction. Yet they often seem unwilling or unable to correspondingly disclose very much about the hydraulic characteristics of the pumping system that the pump faces. The information is mostly restricted to a forecast of the system resistance at a given flowrate, followed by information on conditions at the pump inlet connection. The role played by the complete pumping system resistance curve is often dismissed as secondary.

This Chapter aims to highlight the overlooked importance of the **whole** system curve. It shows that the system curve shape can govern how a pump will respond to transitional flow conditions, to multiple pump operation and will help determine how fast a pump appears to wear out. It encourages purchasers to routinely provide an indication of system head at zero flow and also to disclose how much 'contingency head' is included at rated flow.

Pumps are normally selected to best 'fit' the hydraulic characteristics of the system they operate in. To get a good fit is the objective of most pump application engineers. To be given a good fit is the expectation of every pump purchaser, since he infers this to mean the risks of poor selection have all been minimised. But this is not just a 'nice to have' property. It can be shown that a bad fit will also have hidden effects on cavitation performance and impeller life.

In discussing pump troubleshooting one might be surprised that the pumping scheme pipe system is even mentioned. Many users seem astonished to find that the hydraulic characteristics of the scheme can really have any effect at all on how the overall package responds. This feeling is generally most acute in aftermarket troubleshooting.

For example, it often happens that a pump user wants to rerate his pump for a significantly higher or lower flow. He may have a clear idea of how

*Troubleshooting Centrifugal Pumps and Their Systems*
ISBN 978-0-08-102503-1
https://doi.org/10.1016/B978-0-08-102503-1.00005-0

much more [or less] flow he requires from his pump. But to prepare a solution it will be necessary to know how much more [or less] resistance his system will put up at this new flow. In many cases the facility that includes the pumps was designed by or purchased from a third party, often some time ago. This means that the calculations leading to a hydraulic resistance curve may not be readily accessible. On the other hand, the pump output curve will generally be much more accessible, either in the instruction manual or from the manufacturer records. If the scheme now requires more or less flow, then engineers involved in re-rating the pump will expect that the pressure head needs of the pump may also have changed, if only slightly. When queried, the user very often says that 'the resistance head is the same as before'. Sometimes this is nearly true, sometimes it is definitely not. Where it is not, the consequent solution is bound to disappoint. To understand why, and to help underline the influence of the system resistance curve, it will be useful to describe some of the basic factors (Fig. 5.1).

It's well understood that centrifugal pumps generate both flow and pressure head independently. However, they are rarely bought primarily for their ability to generate pressure. Centrifugal pumps are almost always bought with the intent of creating flow in some liquid system. *[Other types of pumps are better suited to static pressurisation tasks.]* Their capacity to simultaneously generate pressure is a necessary by-product. We know that, in general, a centrifugal pump's ability to generate pressure head **reduces** as its throughput flow increases.

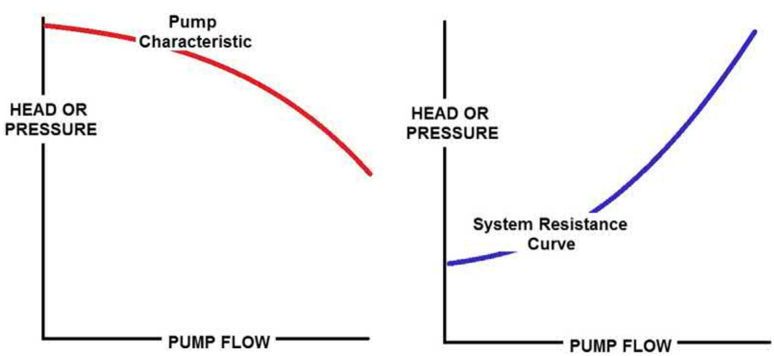



**Fig. 5.1** This contrasts the characteristic curve of a pump with that of a typical system resistance curve. The pump curve generally slopes downward as flow is increased. The system curve does the opposite; it generally slopes upward as flow is increased.

We will also see that, conversely, the overall resistance put up by the pumping system **increases** as throughput flow increases. This increase actually comprises of two components;

- A fixed element relating to both;
  - How and where the pipework is placed, as well as,
  - Local conditions acting on the liquid surfaces
- A flow-related component that depends on the choice of pipework and its trimmings

The pump and system will be in balance and equilibrium at the unique flow where the falling pump pressure head curves crosses the rising system resistance head curve (Fig. 5.2).

The point where the pump and system characteristic intersect dictates the above mentioned 'fit'. Ideally this should be in the region where the pump efficiency is high, the so-called best efficiency point *[bep*]. Such an **exact** fit is often not possible [*or even always necessary]* from standard commercial/catalogue pumps. A close fit is good enough. Each pump has its own 'COMFORT ZONE' [Appendix D] and a 'fit' that occurs somewhere near the centre of this zone is enough to help side-step most common problems (Fig. 5.3).

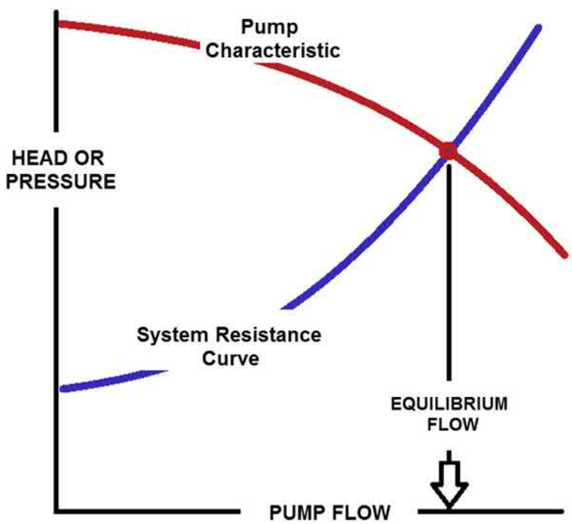



**Fig. 5.2** When **[or if]** these two curves intersect, it defines the equilibrium flow for the pump and system combination. Here a single pump and system is shown. The behaviour of multiple pumps and systems is discussed later in this chapter.

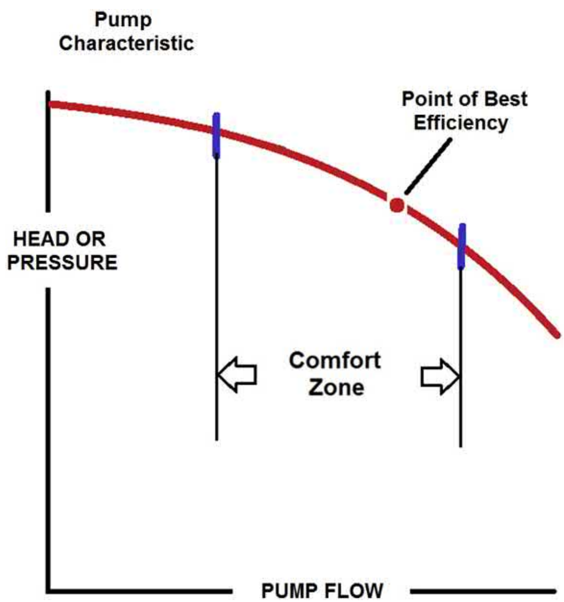



**Fig. 5.3** All centrifugal pumps exhibit an operating comfort zone. Ideally the conditions of service should be contained in this zone [see Appendix D].

However, there is more to selection than just determining a 'fit'. We shall see how the relationship between the shapes of the pump characteristic curve and the system characteristic can influence the routine behaviour of the complete system. For example it can determine how quickly the pump might appear to wear out. In this sense, the system resistance characteristics are at least as important as the pump output characteristic. This very important point often gets lost during any troubleshooting analysis. Failure to at least establish a rough opinion as to the system features might hamper any troubleshooting analysis.

The pump performance characteristic is experimentally determined by the manufacturer and is unique to each model in that range. Its accuracy may be known to within a few percent. Centrifugal pumps characteristics share many common features. As mentioned above for example, their output pressure generally reduces as flow increases. Also, they will all have a flow where their efficiency reaches a maximum.

Liquid system characteristics, on the other hand, have a wide range of differing appearances. This reflects the different contributions made by their three main components. It is very rare for any system resistance curve shape to be known to anywhere near the same level of accuracy as the pump curve. To acknowledge this uncertainty, a contingency margin is almost always added to the system curve at the outset, so as to ensure the selected pump is not undersized for the service. This step is well intentioned. But it can have unexpected consequences, and is discussed late in the Chapter.

## System characteristic; main components

Despite the large differences in system characteristic shapes, it may seem less than obvious that there are common features that underpin them all.

How does a system, intended to raise large quantities of water from a river and onto a crop field, have anything in common with system feeding more modest quantities water into a power station boiler? Or on the other hand, how do systems feeding water to high buildings share any elements with systems draining underground mineral mines or flooding oil wells.

Well, in all liquid systems curves, there are some common features;

- There is a liquid source, and a liquid destination. In all but a few cases, the pump will connect these two by a scheme that includes pipes, valves, bends, and filters and so on. These items are responsible for the so-called 'friction head' component.
- The source and destination liquid surfaces may or may not be at the same geographic altitude. This creates the so-called 'static head' difference.
- The pressure acting on free liquid surface of source and destination *[the local atmosphere]* may or may not be the same. This has an effect equivalent to a supplementary static head.

Liquid system resistance curves can very broadly be put into three categories, reflecting the relative dominance of their three basic components. In what follows I describe three discrete basic systems that attempt to highlight these fundamental components. But be aware that distinctions between the categories are always blurred in practice.

### *Category 1 – mainly pipe system hydraulic friction*

In this, the simplest case, both the liquid source and destination point are at practically the same elevation above ground (Fig. 5.4).

Normally it is good practice to positon the pump close to the start of the pipeline. This aspect is covered later in the Chapter.

- The **SOURCE** or feed is typically a vessel, a river or a pond.
- Liquid flows from the source to the pump via some duct or pipe system.
- From the pump, liquid is passed to the **DESTINATION** via another duct or pipe system.

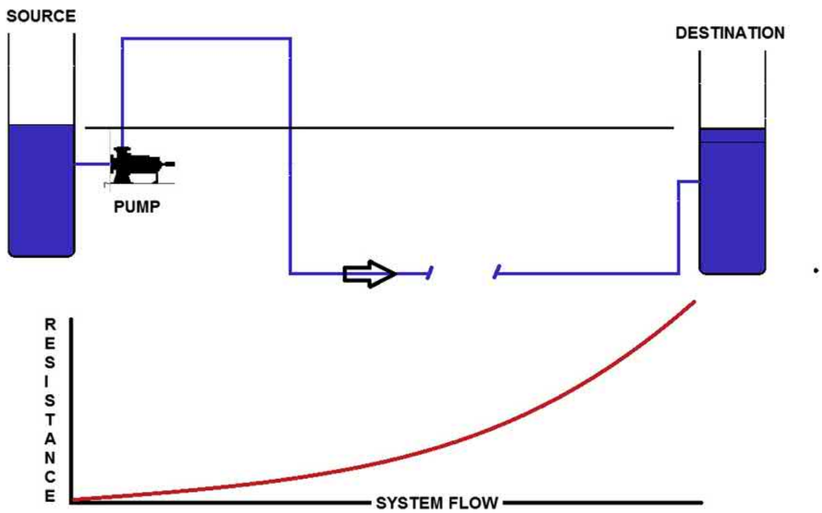



**Fig. 5.4** The elements of a Category 1 system. The pump is employed chiefly in overcoming viscous liquid resistance in the pipe. The pressure acting on Source and Destination surfaces are very similar. There is very little, if any, significant liquid elevation differences between the two surfaces.

- At the destination point, liquid may discharge freely onto the vessel surface, or below the surface. This difference has a small impact on the final resistance curve, but at this discussion level is not important.

In this category, the pump has to mainly overcome the liquid friction resistance put up by the viscous drag in the system pipework and its accessories. In most practical cases, a square law can approximate this resistance; *[i.e. flow resistance is a function of flow squared*], so if system flow increases by a factor of two, then the liquid resistance put up by the system increases by a factor of two squared, that is a factor of 4 (Fig. 5.5).

Similarly, if the pipe diameter is reduced, or its length increased, this will bring about an increase in the resistance curve slope, but it will still approximate a square law. The only time this approximation breaks down significantly is when the liquid pipe velocities become so low that conditions become laminar. Laminar flow is said to exist when the Reynolds Number Re is less than some critical value. Above this value, the flow is said to be turbulent. As a rough guide the transition between the two regimes occurs at a Re value of about 2000.

$$\text{Reynolds Number,}\quad \mathrm{Re} = \rho \times U \times D \div \mu \tag{5.1}$$

where $\rho = \textit{Density}\ U = \textit{Velocity}\ D = \text{Pipe Bore},\ \mu = \textit{Absolute viscosity}$

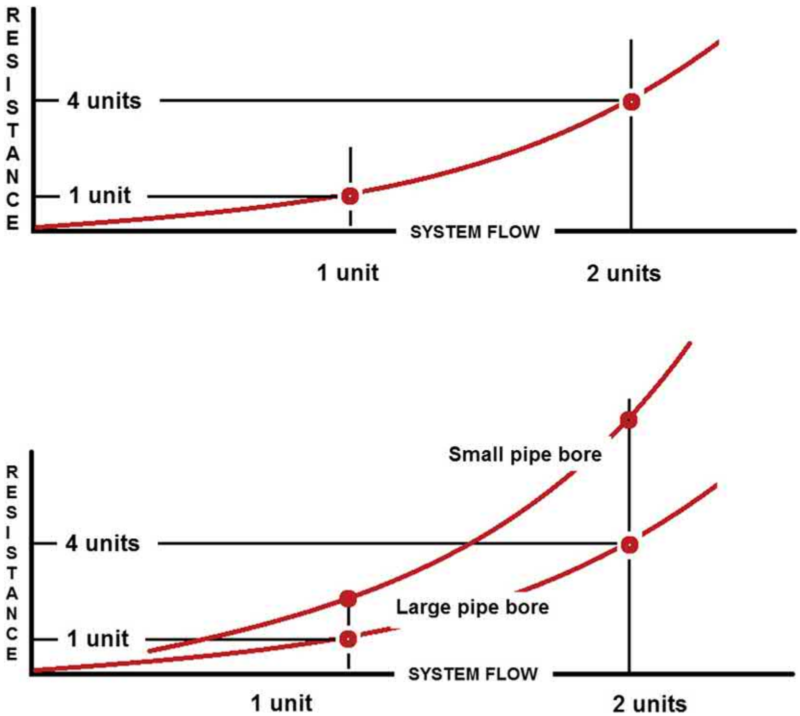



**Fig. 5.5** The typical 'Square Law' resistance relationship for viscous pipe friction. The lower chart depicts the general effect of changing the pipe bore size.

In laminar conditions the liquid particles behave like well drilled soldiers on parade; they maintain constant spacing between each other and all proceed with exactly the same speed and direction.

In turbulent conditions, the particles behave more like the runners in an inner city race. The spacing between runners is constantly changing and although they are all heading the same way, their instantaneous speed and direction change all the time.

It is very rare to experience laminar flow in commercial systems where the Minimum Economic Pipe Size concept has been employed [Fig. 5.19]. In such case, the flow will almost certainly be turbulent in nature. However, if the flow is progressively reduced there generally will come a point where the flow flips to a laminar condition. The losses in laminar flow do not follow the Square Law approximation. This transition will occur at flows too low to be of general interest and in fact the square law can still be used for simplicity. But if the calculation outcome is critically important, then appropriate calculations should be carried out.

Beware, however, if the pump operates in a system where the pipes are significantly larger in diameter than the pump connection flanges. Pipe systems designed along the principles of Minimum Economic bores yield design liquid velocities in the pipe of 5–10 m/sec. Sometimes, pipes are deliberately oversized *[To resist abrasive wear for example]* and this may result in velocities as low as 1 m per second. With any subsequent reduction, laminar flow could be encountered earlier than expected.

So far, I have simplified the description by assigning all the friction to viscous drag on the pipe walls. In fact there are other pipe elements that make the same square law contribution (Table 5.1).

This foregoing picture is typical of many supply pipelines [water, oil, solids transport].

It can also characterise many irrigation or land drainage schemes, because in both cases, the static head component is small or negligible, relative to the friction head.

In some installations, the source and destination are actually the same, with the liquid passing through some process device *[a cooler or a filter say]* and then returning. This is called a closed/circulation system (Fig. 5.6).

**Table 5.1** System friction inducers.

- Partly closed valve
- Dislodged or poorly fitting gasket
- Orifice plate
- Fouled pipe clogged strainer

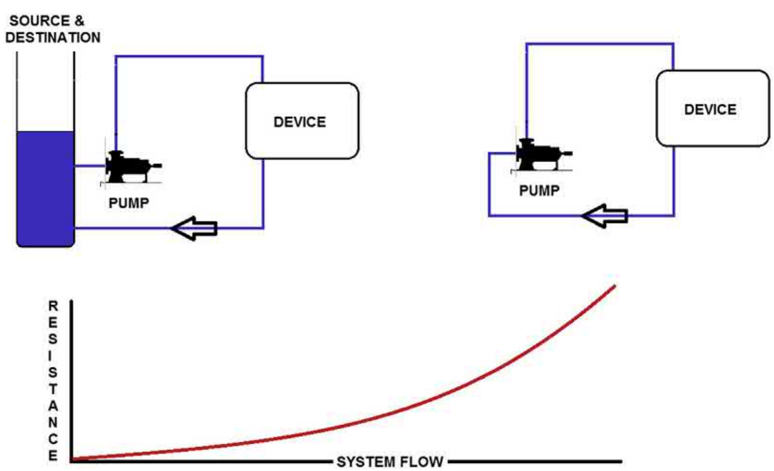



**Fig. 5.6** Liquid circulation systems fall into Category 1. Again the Square Law will generally apply.

In reality, the connecting system may be fully enclosed, or may be a combination of closed pipe and open conduits, such as a river, canal or lake.

In principle, it does not matter, in an open system, how much the connecting pipework rises and falls *en-route*, so long as the end points are more or less at the same level. In practice there are some slight constraints to how much it can be allowed to rise and fall. It must still respect the hydraulic gradient line. Conceptually, the hydraulic gradient line relates to the pressure at any point relative to that at the destination point. It represents the liquid static pressure at different points on the flowpath. For the purpose of this simple explanation, we can imagine this to be a straight line (Fig. 5.7).

If or when the pipe run rises significantly above this imaginary line, a potential problem appears. Depending on temperature, the static pressure *[difference between the two lines]* could go below the vapour pressure. In this case there is a risk that vapour voids will appear in the liquid column, and these can disrupt or even interrupt flow.

Realistically, this is an unusual set of circumstances, but it can occur. Picture a pump that has its source on one side of a hill or mountain and that it discharges to a destination on the other side. The picture is shown below (Fig. 5.8).

As the pipe passes over the hill or mountain peak, its centreline height might exceed that of the hydraulic gradient. Vapour voiding might then occur. A simple solution here is to install a backpressure/breakdown orifice in the downhill section of the pipeline. This artificially raises the hydraulic gradient above the physical centre line of the pipe and avoids vapour formation. Of course, this is a wasteful solution, since the pump pressure head has also to be artificially lifted, only to be dissipated and lost downstream. A more useful solution would be to install a power recovery turbine

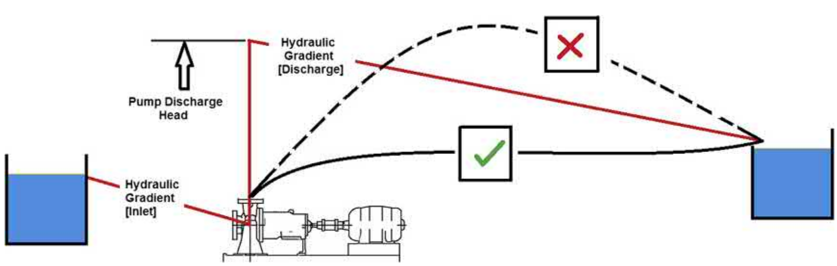



**Fig. 5.7** Concept of hydraulic gradient. Running a pipeline above the gradient line is not advisable.

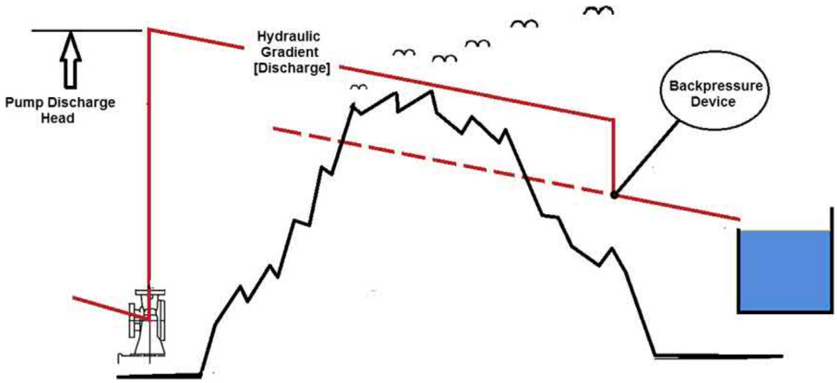



**Fig. 5.8** Geographic profiles may oblige a pipeline [say] to pass above the hydraulic gradient line. Additional backpressure should be strategically located in order to avoid this.

[say] that would achieve the same backpressure objective with less power wastage.

In **summary of Category 1 systems**, the salient features are that;

- The source and destination liquid levels are similar and so have little influence
- The pressures acting on the respective surfaces are also very similar and have little influence.
- In such cases, the resistance curve is chiefly flow-related and almost entirely dictated by the system pipework flowpath design.

This last, flow-related component is a common feature of all liquid systems. In this first example it is the only or at least key component. In the remaining two broad categories, other components start to come into play.

### *Category 2 − hydraulic friction + geographic surface level differences*

While the Category 1 layout described above is quite common, it is much more common to find that the source and destination points are at significantly different elevation levels (Fig. 5.9).

These differences are termed Static Lifts. The Total Static Lift comprises of the lift on the discharge side plus the lift on the suction side. For the moment, the local atmospheric pressure acting on the source and destination surface is assumed to be the same.

On the **Suction** side, the static lift is the physical distance of the liquid surface below the pump centreline. This is called a suction lift. In the first

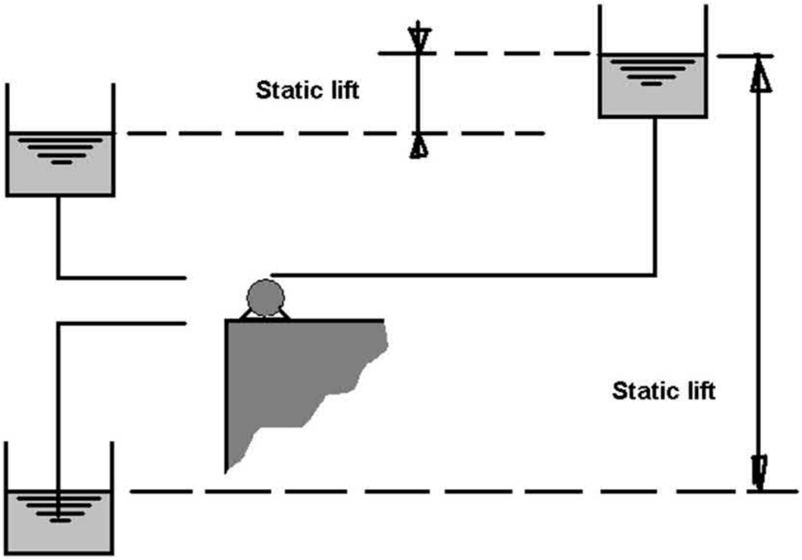



**Fig. 5.9** The total static lift is the algebraic difference between the levels of source and destination. Here discharge at the destination is submerged. If the discharge is above the surface, then that becomes the relevant point. Such extra elevation will usually incur a corresponding power increase so can be wasteful.

example, the suction lift was zero. If the source surface level is already above the pump centreline, then, in order to reach the destination level, it that means that the pump has to lift less. This is termed a flooded suction and has benefits from the standpoint of priming and venting (Fig. 5.10).

On the **Discharge** side, the static lift is the physical height of the destination surface above the pump centreline. There are many possible combinations of suction and discharge static heads (Fig. 5.11).

In most cases, the destination surface level will be above the pump centreline. If the pipe terminates below the destination liquid, the static head is measured to the surface. If the termination point is free and above the liquid, the static head is measured to the termination point.

In some rare cases, the destination surface may actually be below that of the source. The same rule applies. Any difference in elevation levels [Total Static Lift] has the effect of simply shifting the entire hydraulic friction component curve up or down (Fig. 5.12).

Dealing firstly with the more common case of the destination level being higher than the source level; the effect is simply to just slide the friction component curve **UP** the chart by an amount equal to the total static lift.

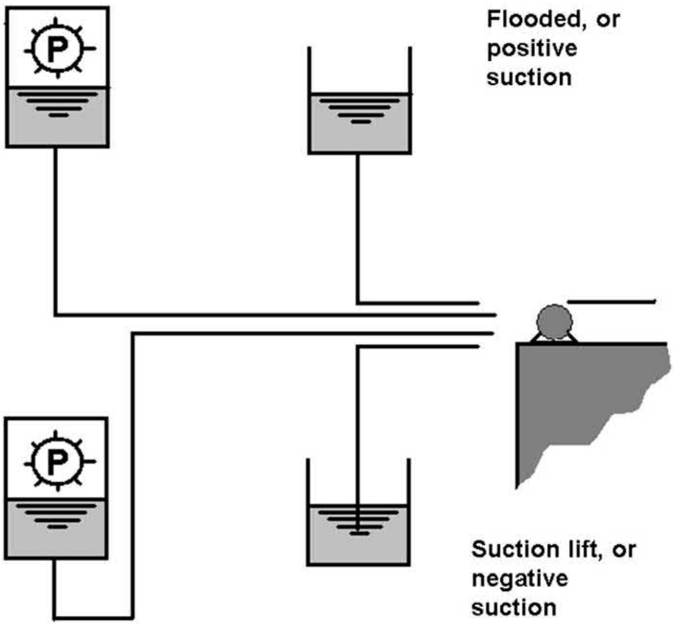



**Fig. 5.10** This indicates the range of possible suction feed conditions.

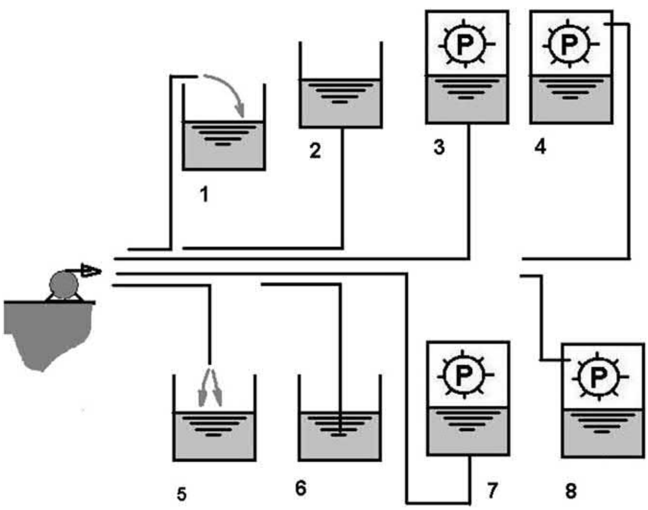



**Fig. 5.11** This shows the range of possible discharge arrangements. Note the near equivalence of [1] and [2]. Option [1] can be wasteful if the total static lift is small.

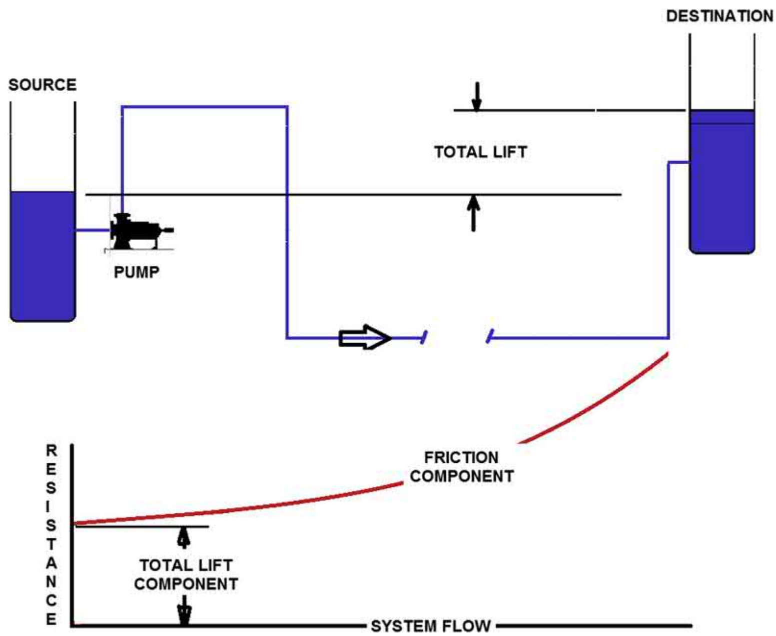



**Fig. 5.12** Introduction of a total static lift between source and destination has the effect of sliding the friction component curve up [or down] the Y axis. This is a Category 2 system.

Practical examples of such schemes would include;

- Pumping water from rivers to elevated communities/villages
- Elevating warm liquid from the base of cooling towers for re-circulation. It differs from a closed loop system in that the source and destination points are not at the same level
- Flood drainage. This only differs from Irrigation examples in that the static lift is now too large to be ignored.
- Dewatering deep mineral mines. In such instances, the static head component may be more than 95% of the total head.

Secondly, in rare cases, the destination level is below that of the source (Fig. 5.13).

Here the friction curve is now slid **DOWN** the chart by an amount equal to the total negative lift. The point where the friction curve crosses the X axis now defines the natural downhill flow under gravity. If more flow than this is required, then installing a pump is necessary. The head required from the pump is the positive difference between the x axis and the friction curve at the specified flow. Such cases of 'downhill' pumping mostly arise where the natural gravity flowrate is insufficient.

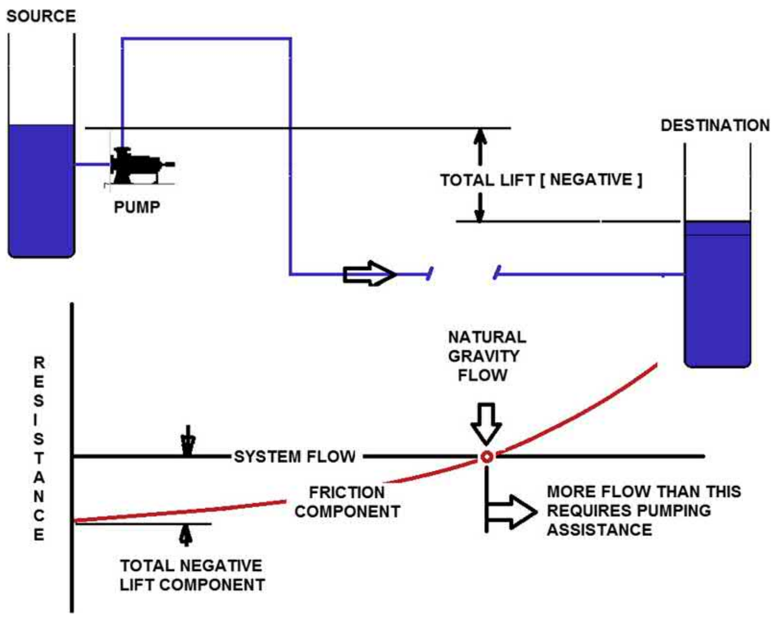



**Fig. 5.13** Where the Destination level is below the Source, the friction curve will cross the X axis at the natural flow point. This flow would be achieved without a pump. If an even greater flow is needed, then pumps will be required. If the natural flow is excess to needs, some form of throttling will be called for. In essence, the throttling loss is added to the friction component and results in a steeper total resistance curve. This results in a lower natural flow.

In **summary of Category 2 systems**, the salient features are that;

- The source and destination surface levels are considerably different. This can have a significant influence on the system resistance curve
- The pressures acting on the respective surfaces are still very similar, so there is little influence
- The overall resistance curve will still contain an influential flow-related component

The overall resistance curve now has two key components; one dictated by the system pipework flowpath design and one governed by the geographic layout of the pumping scheme.

### *Category 3; − hydraulic friction, surface + level difference + surface pressure differences*

In both the previous categories, the pressure acting on the surface of the liquid at the source was more or less identical to that at the destination. This is quite often the case, the most common being atmospheric pressure.

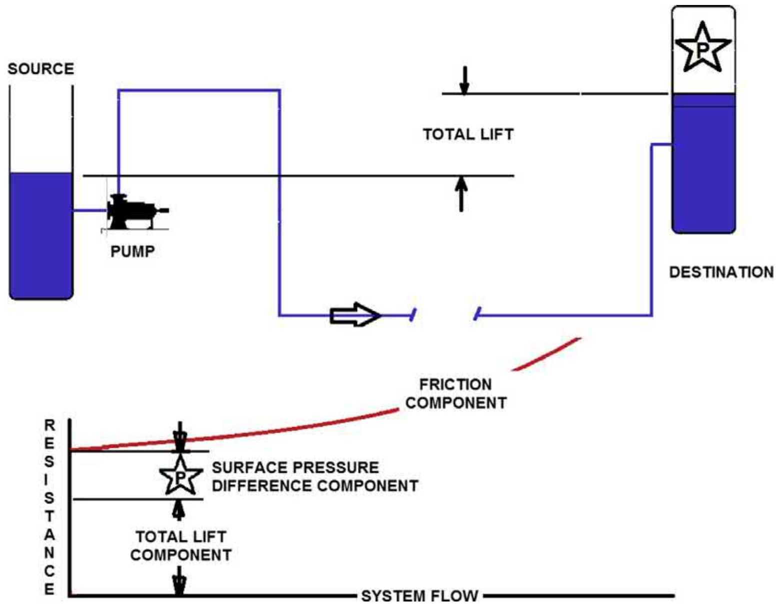



**Fig. 5.14** Any difference between the pressure acting on the Source and Destination surface results in the Category 2 curve being slide up [or down] the Y axis. This is a Category 3 system.

Sometimes however, these two pressures are significantly different *[when the pump is part of some process chain for example].* Yet the effect of this is quite simple; the previous 'friction + static' curve is just further shifted by the algebraic surface pressure difference (Fig. 5.14).

As before, the total resistance curve still has a 'flow-dependent' element and a fixed 'flow independent' element. This last component now has two subcomponents; that due to the 'total static lift' and that due to the 'surface pressure differences'.

This description also applies to many practical systems. For example,

- Pumping water, at high pressure, into a power station boiler so as to be turned into steam.
- Injecting water into naturally pressurised rock strata so as to displace the absorbed oil.
- Most chemical and hydrocarbon processing application.

In **summary of Category 3 systems**, the salient features are;

- The source and destination surface levels are considerably different. This can have a significant influence on the system resistance curve

- The pressures acting on the respective surfaces are significantly different. This can also be very influential
- The overall resistance curve will still contain an influential flow-related component

The overall resistance curve now has three key components; one dictated by the system pipework flowpath design, one governed by the geographic layout of the pumping scheme, and one related to how the liquid is processed in that the source and destination surface pressures become different.

## Summary

Every pipe system – from a garden hose sprinkler to a Power station boiler feed pump scheme or an Oil refinery process-has an associated system resistance curve, which will be unique. I have described three discrete system categories here but I only do this for purpose of illustration. I attempted to show how each feature influences the final system curve makeup. In the real world, there is no such clear distinction. Practical system curves include all three features to a greater or lesser extent.

- A flow-dependent element – **FRICTION HEAD.** This will most often increase with flow in accordance with the 'square law'
- A fixed flow-independent element – **STATIC HEAD.** That depends on both;
  - Any physical difference in height between the source and destination liquid levels. This comprises of the algebraic difference between the source level, relative to the pump, and the destination level relative to the pump.
  - Any difference in the pressure being exerted on the surface of the source and destination liquids. This may result from the liquid being processed in some way *en route*

Returning to the re-rate question posed at the start of this section: If the system is a 'flat' Category 2 or 3, then the head change is probably small and the statement that the head requirement is unchanged is correct, in practical terms (Fig. 5.15).

However if the system is a 'steep' Category 1or 2, then the presumption of negligible head change is INCORRECT. The Figure shows two system curves passing through the same design point. Conceivably, the same modified pump would be selected in both cases.

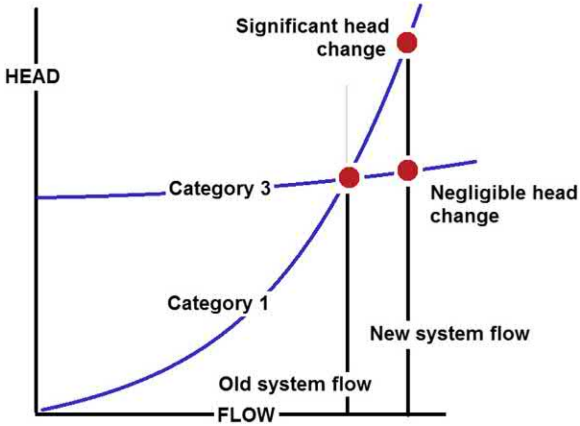



**Fig. 5.15** When more flow is needed, the extra head demands on the pump depend upon the nature of the system resistance curve.

If the pump has been installed in a Category 3 system, where the resistance curve consists largely of static and backpressure head, then an increase in process flow requires only a small increase in pump output head.

However, if that same pump were installed in a category 1 system, where the resistance curve is comprised almost entirely of hydraulic friction, then a substantial increase in pump output head would be required for the same increase in flow. In this case, ordering a pump rerate for increased flow at 'the same resistance head as before' would result in disappointment.

## Other related system matters

### Contingency head/safety margins

I have mentioned that the system resistance curve is at least as important as the pump output curve. While the pump performance characteristic is known, and often test measured to within a few percent, the system resistance curve is not. In practice, and for many valid reasons, the system resistance curve is very rarely known to the same accuracy. In predicting the system resistance curve the plant designer is often faced with some uncertainties while making his calculations. And even if these uncertainties can be made small, there is still often some doubt as to how quickly and severely the condition of the wetted surface will deteriorate over time. This uncertainty or ignorance is usually overcome by applying a contingency factor to the calculated system resistance. This means that in practice; most pumps are actually shipped giving a performance that is initially oversize for the service. A popular strategy is to specify a pump pressure head requirement that is 10% larger than calculated. This plan is predicated on the fact that an oversize pump can subsequently be throttled or constrained in some way. On the other hand, an undersized pump is just that-undersize! At first sight, this oversizing might seem an innocent measure, which has been applied with the best of intent. However, this can have unexpected results. Let me go back to the system examples discussed earlier, and look at the two extreme examples.

Consider first the case where the system resistance is made up chiefly of pipe friction (Fig. 5.16).

Recall that the pump will logically operate at the flow where its natural output curve intersects with the natural system curve. Therefore, the oversize pump curve will naturally intersect at a higher flow than desired and over-pumping is the inevitable outcome. This may be acceptable in some cases. But in other cases, this over pumping may move the operating flow outside of the pump's 'COMFORT ZONE' [Appendix D]**.** It is a fact that, in many cases, the pump owner is not the original procurer. So he may be unaware that his pump is most probably oversized from the outset and likely to over-pump, absorbing more power in the process.

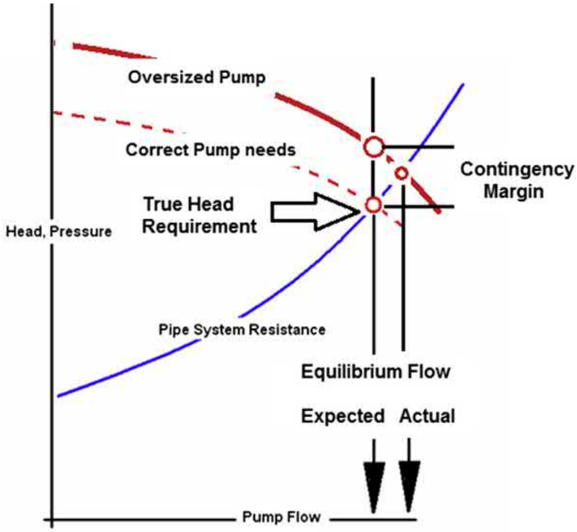



**Fig. 5.16** The unexpected consequences of applying a contingency margin to the pump specified head will result in overpumping. The smallest overpumping effect occurs if the system has Category 1 characteristics.

Now let's examine the other extreme that is a system, where the friction head component, is only a small part of the overall.

In almost every case, the static head component can be assessed directly and very accurately. Similarly, the local atmospheric pressure differences can be judged with some precision. The chief unknown is, again, the friction losses, since their calculation relies on generalised empirical loss coefficients. While application of an uncertainty margin is often justified, it is only valid to apply this to the friction head component. The other components will normally be much better defined. But I am aware of cases where, in this type of system, the 10% margin has been innocently applied to the overall head, not just the lesser friction component.

Based on Fig. 5.17, this would clearly result in very substantial over pumping-possibly taking the pump way outside its 'COMFORT ZONE' and taking it nearer to prime mover overload.

Therefore, safety margins or factors of ignorance, while justifiable, should not be applied indiscriminately without reference to the component make-up of the total head.

Of course I should mention that sometimes pumps are deliberately oversized in order to allow flow regulation via a control valve (Fig. 5.18).

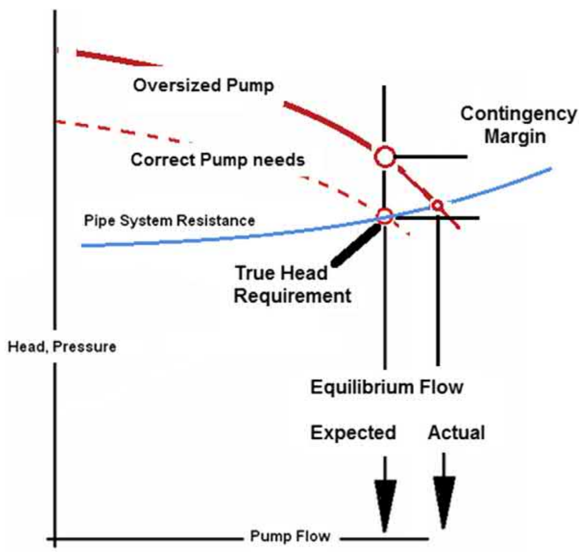



**Fig. 5.17** The unexpected consequences of applying a contingency margin to the pump specified head will result in the largest overpumping effect if the system has the Category 3 characteristics. There is even a risk of the pump getting outside of its comfort zone.

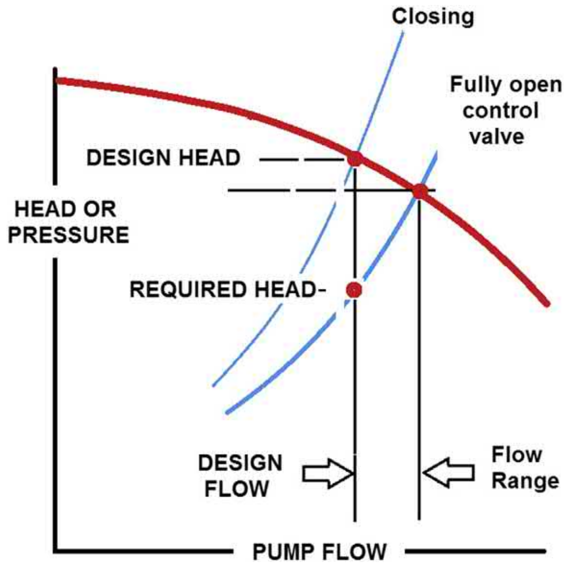



**Fig. 5.18** Flow control is most efficiently accomplished by speed control. But often throttling control valves are used instead. The same care is needed to ensure that with a fully open control valve, the pump does not go outside of its comfort zone.

What happens is that the natural system resistance curve head is artificially increased by a control valve loss, in order to produce a new higher flow intersection point. This might allow an emergency over pumping capability, for example. Or reverse logic could be applied and result in controlled flow reduction. Or a midpoint might be chosen. Where flow control is envisaged, the pump head surplus might be increased to about 15%.

## Concept of system efficiency

Once the friction head component becomes significant in comparison to the total head, some people introduce the term '**System Efficiency'**. Essentially this is the ratio of the friction head component to the overall head, i.e.

$$\text{System Efficiency} = [\text{Total Head} - \text{Friction Head}]/\text{Total Head}. \quad (5.2)$$

This has sometimes been called the '**Friction Ratio'.** The concept predicates that the chief function of the liquid system, is to change the liquid static level – *[acknowledging any surface pressure differences*].

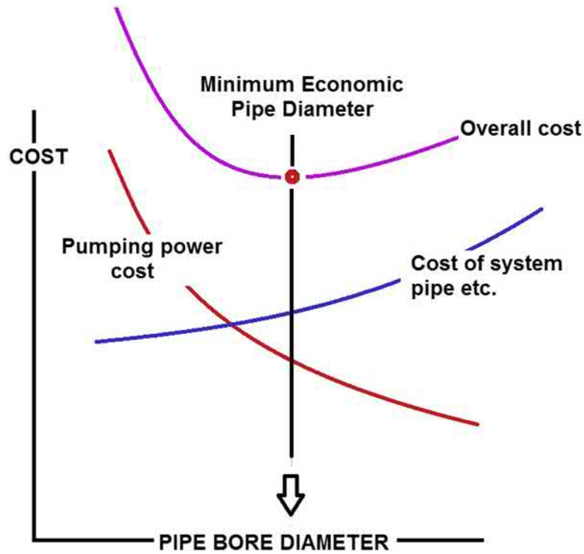



**Fig. 5.19** Each pumping scheme will have an associated minimum economic pipe diameter which balances capital costs against running costs.

Therefore, the most 'efficient' system would be ones that have little or no friction component. Almost all of the pump work output would be directed only to that part of the system problem that is largely fixed. Hence, it is likely to have the lowest pump power costs. But this would rarely if ever also be the lowest cost system to install. This most efficient low power system implies a low pipe friction component, which would normally mean large pipe bores, in order to reduce pipe velocity. But experience says that by reducing the pipe diameter, the capital cost of the pipe reduces faster than the pumping power costs increases.

This leads on to the previously concept of 'Minimum Economic Pipe Diameter', mentioned earlier in Chapter 3. This strikes a cost balance between the two opposing elements.

This Figure conceptually shows how component costs can vary with diameter. Obviously, choosing larger pipe bore will incur more capital cost than a small pipe. However, large bore pipe will incur lower pumping power costs because the system's flow-related friction resistance component will be lower. This is also shown as a curve. If these two costs are summed over a discrete range of pipe bore sizes, then an overall cost can be deduced that is a function of specific pipe size. This third curve will show a minimum at some point, known as the Minimum Economic Pipe Diameter.

This is usually the starting point for most significant pumping schemes since its influence on the lifetime cost of the scheme can be very substantial.

## Pump location in the system

Pumps are almost always located near the start of a pipe system flow path. *[The chief exception would be in cases where the pipe system is essentially an endless loop, such, as is the case in circulation system all or nearly all friction]* The reason is that pumps generally have a very much higher capacity to discharge liquid 'out' than to ingest it 'in '. This limited ingestion capability is relates to an internal flow condition called cavitation. *[Cavitation is discussed in* Appendix E*]*. Cavitation occurs when, all other things being equal, the ingested flow possesses insufficient pressure head to avoid local vapourisation. By locating the pump close to the beginning of the flowpath, the least pressure losses will be incurred and so the incoming flow will have the highest pressure head.

I might seem to be putting too much emphasis on the conditions of the flow as it approaches the pump. But this is deliberate. No amount of advanced hydraulic pump design/manufacturing technology is effective unless the liquid successfully enters the pump impeller in the first place. Until and unless this condition is fulfilled, the pump cannot function as an effective machine and cannot do efficient work on the liquid. So I make no excuses for highlighting this aspect *ad nauseum*.

Pump and system designers evaluate the liquid pressure conditions immediately upstream of the pump by a term called NPSH [A], Nett Positive Suction Head Available. This is a term that is unique to each installation and involves the liquid temperature, pressure and velocity.

We have already come across the term NPSH [R] [Nett Positive Suction Head Required] in Chapter 3. This term describes the point, established during shop testing, where cavitation becomes self-evident, according to international standard. But as a rule, NPSH [A] must exceed NPSH [R] by some margin, depending on pump owner/user life expectations. The pump designer often has to balance pump speed and impeller design in order to operate within the constraints set by NPSH [A]. He assures that the pump NPSH [R] is less than NPSH [A] by that margin.

In Chapter 3, cavitation was described as a boiling process. It will occur when the local liquid pressure/head drops below the vapour pressure. Generally, vapour pressure increases with temperature (Fig. 5.20).

At room temperature and pressure, water boils at 100degC. But it boils at a lower temperature at high altitude/reduced local atmospheric pressure,

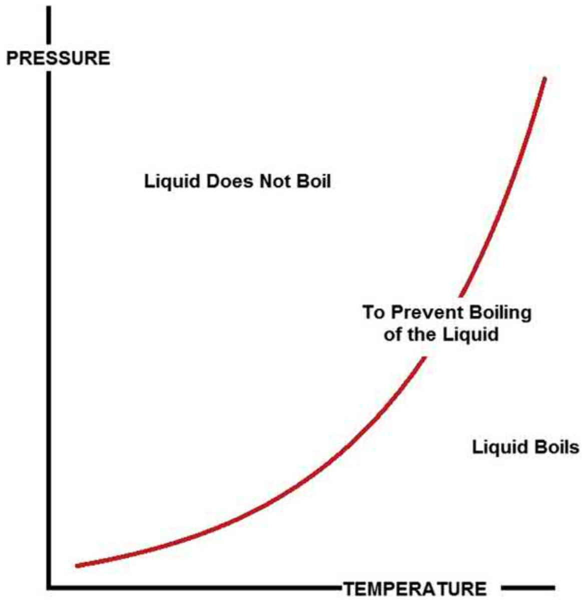



**Fig. 5.20** Typical vapour pressure curve.

say atop a mountain. At significantly reduced local pressure, water can even boil at room temperature. Most other pure liquids demonstrate similar patterns of behaviour. Crude oil however, consists of various constituents or 'fractions'-each with their own value of vapour pressure. So each constituent boils at a different temperature-pressure point. Their vapour pressure curve is more complex.

Once the flowpath passes across the suction flange, it experiences further pressure drop. This drop is caused by liquid friction on the pump casing walls as well as the 'tearing' effect that the impeller blades have on the incoming liquid. We shall also see that as far as pumps are concerned, the extent and intensity of cavitation can dictate the long-term life. For example, In order to suppress cavitation, pumps require that the suction head/pressure exceed the liquid vapour pressure by some amount. This margin is called the NPSH [R] [Nett Positive Suction Head Required]. As mentioned elsewhere, the pump community has adopted the definition of cavitation onset as '…. that NPSH [R] that induces a 3% reduction in pump output head/pressure.' We shall see that in many cases, operating at this level is insufficient to attain satisfactory life (Fig. 5.21).

- A **small** margin [A few percent] over NPSH [R] produces significant cavitation and a still-detectable decay in performance.

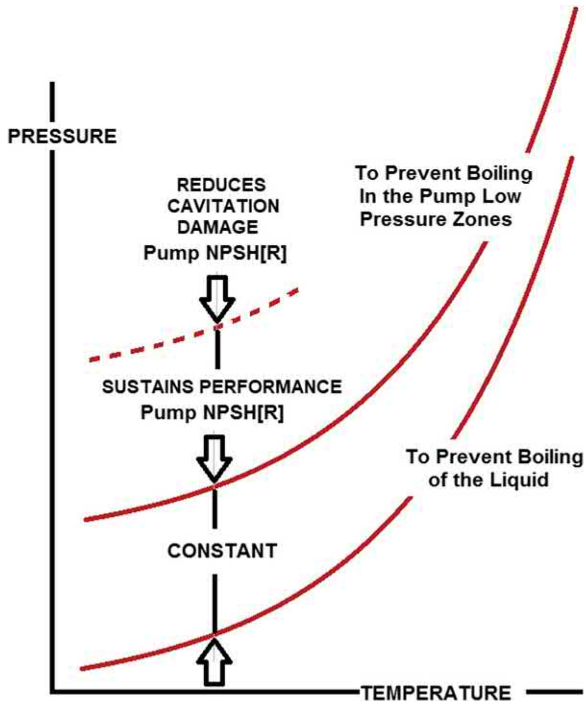



**Fig. 5.21** NPSH [R] is that margin over the liquid vapour pressure that would otherwise cause pump head output to decay by 3%. An even greater margin is required if long term cavitation damage is to be avoided. Manufacturers generally standardise on water as being the subject liquid.

- A larger margin over NPSH [R] [say 10%] will not produce any significant performance decay, but the impeller may still suffer slow long-term damage
- A **very large** margin over NPSH [R] [say 100% or more] is required to eliminate all noise and damage. These values are seldom economic to attain.

Let us now examine how the precise location of the pump relative to the source and its free level. We are interested in how the location might influence the incoming pressure head and consequently the risk of cavitation developing (Fig. 5.22).

In this first example, the pump is alongside an open-topped vessel at the same level as the liquid. On entering the pump suction line the pressure head will be at or near to atmospheric. By the time the liquid arrives at the pump inlet it will have lost some pressure head due to internal hydraulic

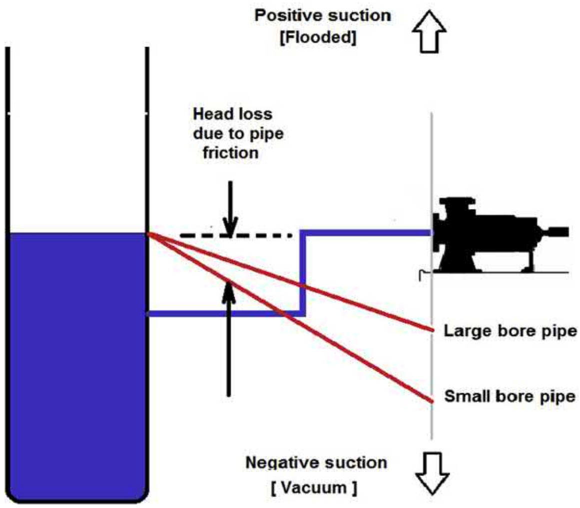



**Fig. 5.22** Simple case with pump centre-line approximately at Source liquid level. In travelling towards the pump, the liquid looses static pressure. The liquid will experience more loss in small bore pipes than with large.

friction in the entry system. It will lose more if the pipe bore is small than if it is large. *[ Already there is a lesson here; to maximise pressure at the pump inet the pipe bore should not be too small.].* A pressure gauge located on the pump inlet would register a sub-atmospheric value, that is, a vacuum.

In the next example, the liquid level is above the pump centreline. As noted earlier, such conditions are called 'Flooded Suction'. Again, the liquid enters the suction line at atmospheric pressure. Similarly, the liquid will experience a pressure head drop en route to the pump (Fig. 5.23).

But by flooding the suction, the pipe losses are offset to an extent. The pressure profile curve is slid upwards. This may or may not reduce the pressure to below atmosphere by the time it reaches the pump. In this regard, large pipe bores are again beneficial. Flooded suction conditions are generally preferred because they mean the pump can be more definitely primed and hence catastrophic 'running dry' conditions can be avoided.

Often the most convenient or only pump location is above the liquid level. These are the previously described 'Suction Lift' conditions. Again, the liquid enters the suction line at atmospheric pressure (Fig. 5.24).

As before, the liquid will experience a pressure head drop *en route* to the pump. But by positioning the pump above the liquid, the pipe losses are now compounded. The pressure profile curve is slid downwards and this

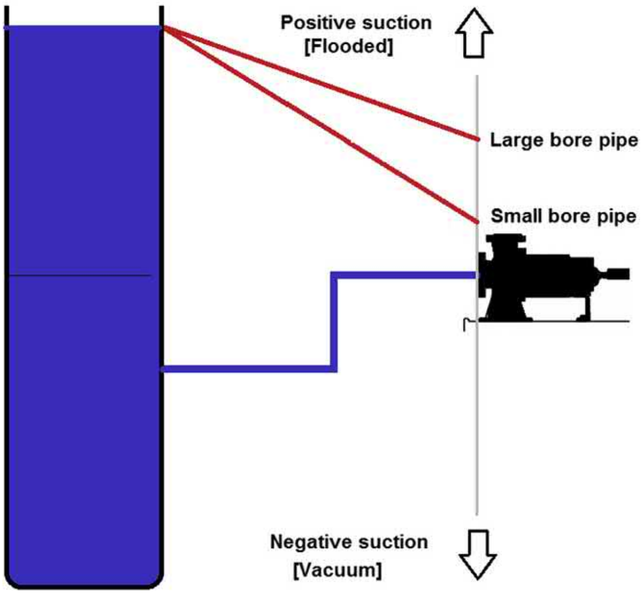



**Fig. 5.23** Allowing the Source liquid level to heighten above pump centerline can help compensate for the losses experienced en route to the pump inlet. Small bore pipes would detract from this benefit.

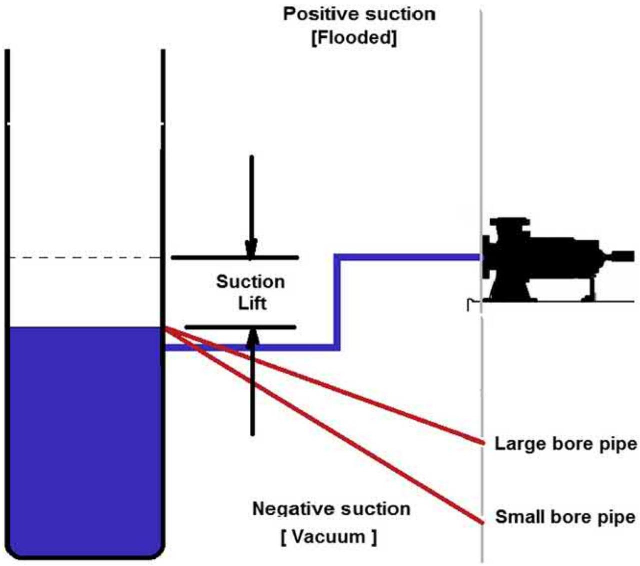



**Fig. 5.24** Allowing the Source liquid level to fall below the pump centreline will reduce static pressure at pump inlet. Agai, small bore pipes have the most negative impact.

will further reduce the pressure to below atmosphere by the time it reaches the pump. In this regard, large pipe bores are again beneficial. Suction lifts are best avoided if possible, because additional measures will be required to prime the machine prior to start-up and to avoid 'dry running'. Pumps operating at flows significantly below their 'COMFORT ZONE' boundary are particularly prone to upsets in the suction line that may lead to de-priming. In any event, the Laws of Nature dictate that Static Suctions Lifts in excess of 25–30 feet or so can rarely be reliably achieved.

For a time, the writer was heavily involved with powerful pumps used to dewater deep underground mines. The pumps would be mounted many hundreds of feet below ground level, often in expensive dedicated pump houses. A question that arose with monotonous regularity was 'Why not locate the pumps on the surface, and simply suck the liquid up those many hundreds of feet. Would that not be a lot easier?' To be fair, the question was usually [but not always] posed by non-engineers. It would of course be a lot easier, but regrettably, Mother Nature is only prepared to help us for the first 25–30 feet.

So far, the discussion has centred on a source vessel in which the free surface is at atmospheric pressure. In some circumstances, this is not the case. If the free surface pressure is positive *[greater than atmospheric]* then clearly this can be an advantage. The negative impact of hydraulic friction can be offset. It can increase the pressure at the pump suction flange and hence help increase the margin over cavitation conditions setting in. The friction curve is slid upwards in the diagram.

But if the surface pressure is negative *[below atmospheric]* then this can make the situation worse (Fig. 5.25).

This composite Figure compares the cases of where the surface pressure is first at atmospheric pressure, then above atmospheric. This has the effect of sliding the friction curve upward. Clearly the effect of negative pressure [not shown on the diagram for clarity] would then be to slide the whole friction curve **downward**. This reduces the pressure at the pump suction flange, and can decrease the margin over cavitation onset conditions.

The foregoing section attempts to show how different location points can affect the liquid pressure conditions as it enters the pump. This, of course, directly influences the risk of cavitation occurring. As mentioned, in the liquid handling industry, the standard method of assessing the risk of cavitation is via the term NPSH.

The NPSH [A] value at rated flow is of importance, because,

- it will dictate whether the pump will operate or not

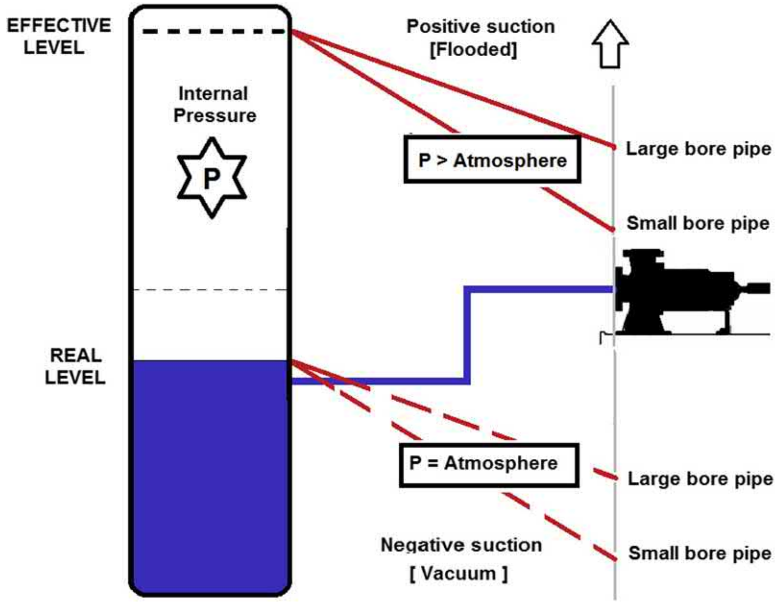



**Fig. 5.25** Pressurising the Source vessel will help increase the static pressure at pump inlet.

- It will affect how long the pump will last before the Impeller becomes too damaged by cavitation, if any

We will also see that the shape of the NPSH [A] curve will dictate whether a pump will be susceptible to a damaging condition known as cavitating surge.

## NPSH [A], NPSH [R]

At this point in the text, a simplification might be useful. This will create a strong mental image of the NPSH concept for those not routinely using the term.

Imagine that the pump is handling water at its boiling point. In that case, if the pump is positioned very close to the source vessel, the pipe friction term can be ignored. Then the liquid vapour pressure and the atmospheric pressure [See Appendix E] cancel out. NPSH [R] then reduces to the static liquid level above pump centreline. This is shown in below (Fig. 5.26).

If there is a significant distance between the vessel and pump, then, the NPSH [A] has to be increased to allow for the further pressure drop caused

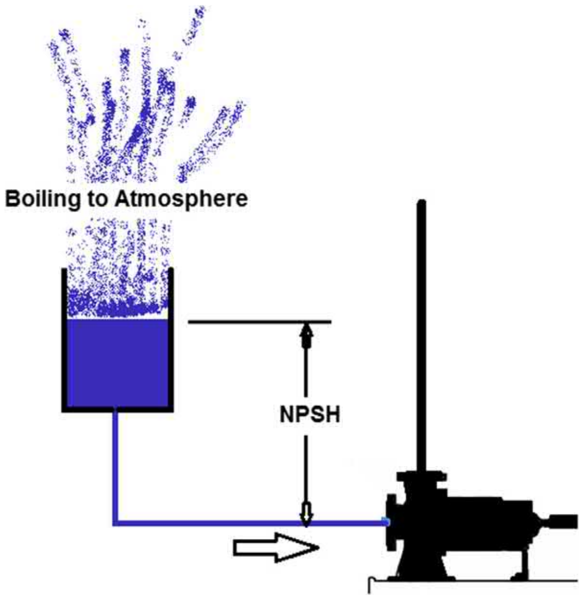



**Fig. 5.26** If pumping boiling liquid from an open Source, is placed very close to the pump, then NPSH available [NPSH [A]] is the Static level of that Source above the pump centerline.

by viscous friction in the suction line. But NPSH [R] remains constant (Fig. 5.27).

As simple as this appears, there are some qualifications. Firstly, the value of NPSH [R] declared by pump makers is, by international definition, that point where the pump output is sufficiently affected by cavitation to reduce output by 3%. While the 3% head decay criterion is important, because of its international acceptance, there are other definitions that are arguably just as important. These have previously been discussed, but are restated here (Fig. 5.28).

- A: Just avoid complete collapse of pump performance output
- B: Just avoid detectable performance output decay [3%]
- C: Avoid long-term cavitation damage with relatively hard impeller *materials [including stainless steel-which work hardens],* i.e. 17-4 PH, Duplex stainless steel.
- D; Avoid long-term cavitation damage with relatively soft impeller [cast iron, bronze, steel etc.].

The numeric values for these four conditions are different, as this Figure shows (Fig. 5.29).

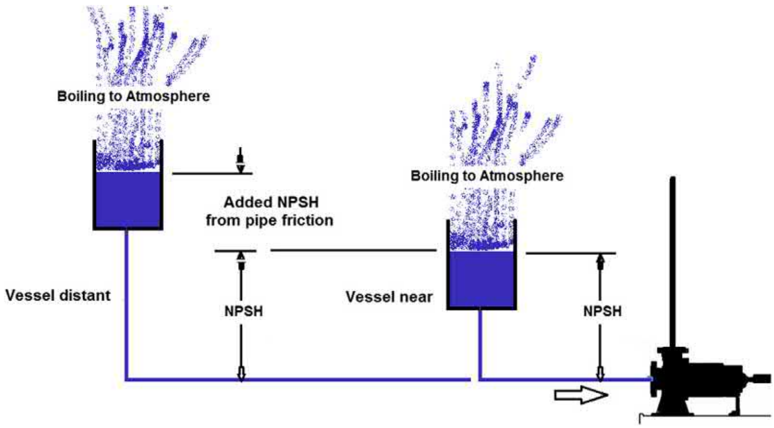



**Fig. 5.27** If the Source is located at some distance from the pump, then the surface needs to be elevated further in order to compensate for viscous friction loses en route to the inlet. In the preceding example these losses were negligibly small.

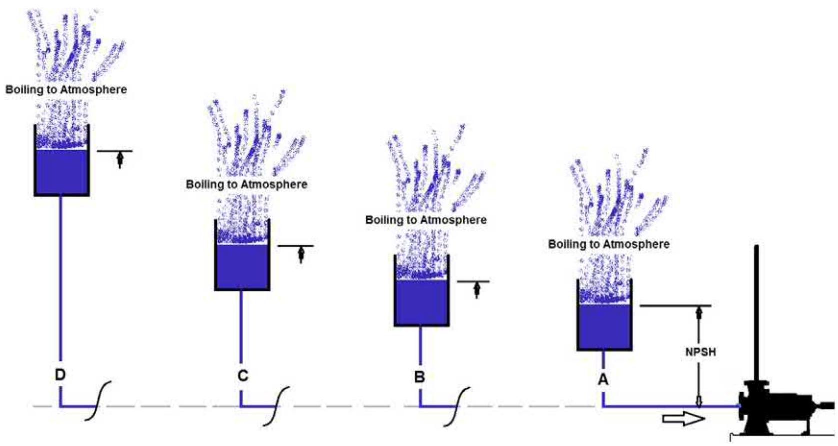



**Fig. 5.28** The Source liquid levels need to be further elevated in order to comply with the four criteria listed below. In all cases it is assumed that the pump is again located so close to the Source that friction losses can be ignored.

This highlights that; all other things being equal, in order to totally avoid cavitation and associated very long term damage, the source vessel surface will need to higher than that just to avoid long term damage. This, in turn, will need to be higher than that to avoid detectable performance decay. Furthermore, the difference between values for B and for either C or D increases very significantly at flows greater or less than design flow.

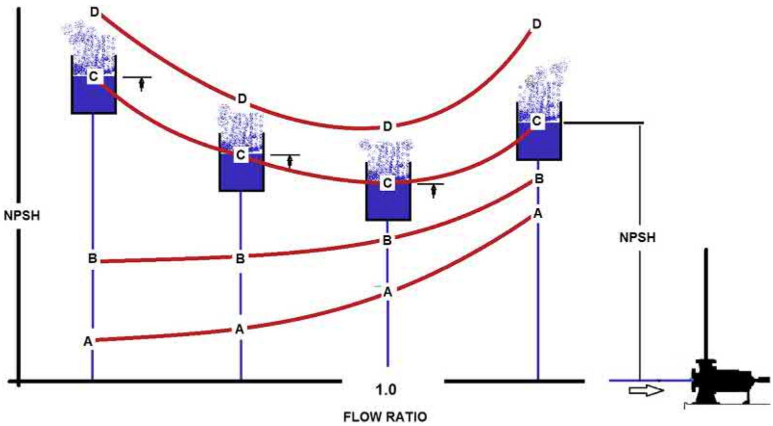



**Fig. 5.29** Conceptual illustration of the flow related the variation in critical Source elevation as a function of the four criteria previously listed.

In an attempt to simplify such a complex picture, I have so far retained the condition that the water is boiling to atmosphere. What happens if I now change this? (Fig. 5.30).

Well the liquid vapour pressure will change but NPSH [R] will not. NPSH [R] is a flow dependent margin/property that is unique to each

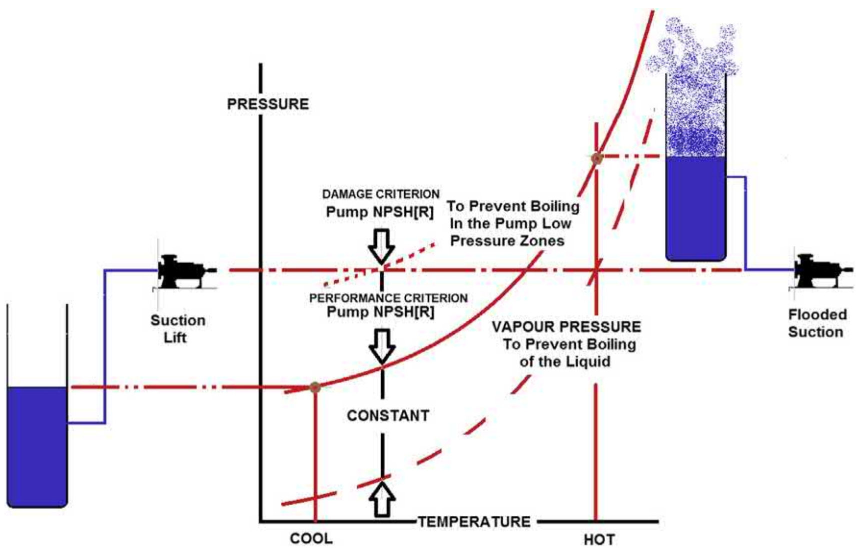



**Fig. 5.30** Showing how, as temperature is reduced below boiling, less level difference between source and pump is required. In some cases, it is permissible to operate with source levels lower than the pump.

pump. By definition, this is the margin over vapour pressure. *[In fact NPSH [R] is known to change with different liquids. But at this level it is sufficient to parenthetically note that].*

If we cool the liquid, then the vapour pressure *[pressure to suppress boiling]* reduces, but the NPSH [R] does not, or at least not significantly. Referring to the previous Figures, this simply implies that in order to contain internal cavitation intensity to the desired level, the pump needs less suction pressure head at above its centreline.

Carrying on this mental experiment, and cooling the liquid further will get to a point where the static liquid does not even need to be above the centre line-it can be below. This is termed a 'Suction Lift', and was discussed earlier. In both these cases, the extent of cavitation will be essentially the same.

Here I have used water as the experimental liquid but the effects would have been the same for other liquids having a definite vapour pressure curve. Some other liquids, such as crude oil, have a vapour pressure that is complex and comprises different elements that relate to the different oil 'fractions'. But the general effect is still the same.

Whenever the suction is flooded, then the pump can be easily and automatically primed. But in the case of a suction lift, special measures are required for priming. In flooded suction installations, there is little if any tendency for the suction line to 'deprime'. But with suction lifts, there is always a small risk that a non-flowing suction line could deprime. This might be due to small air leaks or even deaeration of dissolved air in the inflow liquid.

## Interaction between pump and system: equilibrium flow

Having spent some time outlining the basic types of piping system curves, what relevance do they have to the pump and its curve? Fig. 5.2 illustrates the equilibrium flow concept and how the two curves relate. Obviously, both curves play a part in this determination.[1–3] If the pump curve changes, then so does the intersect. Hence the equilibrium flow changes, (Fig. 5.31).

Pump curves can change due to wear, for example. Similarly if the systems curve changes, then the equilibrium flow will also change. System curves may change due to, say, fouling of the pipe walls.

Paradoxically system curves seldom rate a mention, whereas the pump curve is always in demand. Yet, both are equally important, as this example shows.

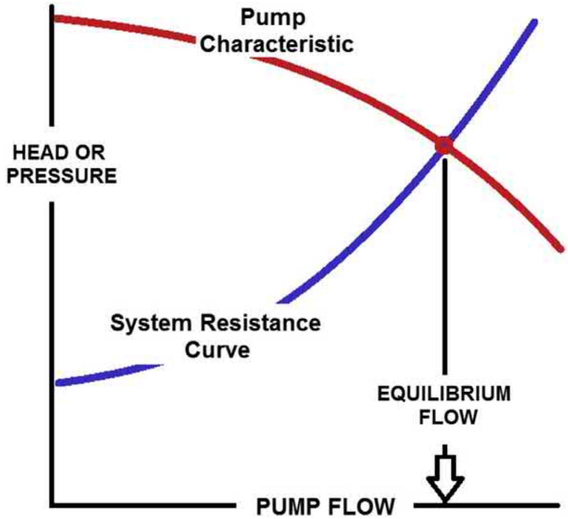



**Fig. 5.31** Equilibrium flows for pump and system.

Purchasers are often eager to know precisely what the pump characteristics are: yet they are often quite unclear or even seem disinterested in the details of their own system characteristics.

It has also been my observation that system calculations are most often restricted to the design flow condition. This is defensible to an extent, in that it does calculate the head required by the pump. Calculations to yield the system curve shape are infrequently conducted or disclosed. This is a shame because its system curve shape, coupled with the shape of the pump curve can give glimpses as to how happily the two will interact. We have seen in the foregoing how a certain pump head can be comprised of mostly friction head, mostly static head, or combinations of the two. The required pump head is the same, but the system responses to changes in the pump characteristic will be different.

Changes in the pump characteristic can occur by design, or accident.

- Examples of accidental pump changes would be
  - Deterioration due to increased and worn wear ring clearances, or
  - Corrosion of the pumps internal wetted components.
- Deliberate changes would include
  - Speed variation,
  - Impeller diameter changes,
  - Impeller design changes,
  - Casing hydraulic modifications.

System curves changes have already been mentioned.

- Accidently, they may change due to
  - Fouling,
  - Blockages.
- Deliberate changes would be
  - The use of control valves to modify the curve and hence manipulate the flow. [Pump manufacturers do this every day on their test bed!]

As discussed, pumps are often tested against acceptance standards that only permit performance deviations of a few percent. Yet, the associated system curve is rarely if ever known with anything like the same precision. This amplifies the point mentioned earlier. In fact, system curves are frequently in error by more than 10%, as mentioned earlier in CONTINGENCY HEAD/SAFETY MARGINS and reiterated in **Palgrave's fourth Rule.** This is ironic, when the success of the combination depends as much on the system as the pump. Many pump problems can be related to this approach, or misunderstanding of the system curve significance.

We shall now see how the **intersection angle** between the pump and system curve:

- Plays a great part in how quickly a pump appears to wear out for example
- Governs how sensitive the pump is to changes in the system characteristic, or elements within it, such as fouling
- Plays a part in the pump behavior during start-up

Let's consider how three different system curve shapes interact with two pump curve shapes. All three systems have the same design flow and the same design head. However, their shapes are different.

- 'All Friction' system
- 'Largely Static' system
- A combination of static and friction

The pump curve shapes are:

- 'Flat' [small head changes as flow increases]
- 'Steep' [large head changes as flow increases]

Strictly speaking, the pumps curve shape or slope should be defined mathematically by its first derivative. But for almost all practical purposes, the slope can simply be defined by the ratio between the head at zero flow to the head at the best efficiency point. This is called the **H**ead-**R**ise-**t**o-**S**hut**O**ff, or **HRTSO**.

$$\text{HRTSO} = \text{Head at zero flow/Head at bep flow} \qquad (5.3)$$

This definition is very widely used in the pump industry and has been proven to be an adequate indicator of curve shape or slope. Normal centrifugal pumps have **HRTSO** in the range of 1.05−1.3. Mixed flow pumps have values in the region of 1.5 while axial flow pumps might reach 2.0 or more. However, pumps of the centrifugal configuration are most prolific, and a **HRTSO** of 1.2 is often assumed in initial estimates.

## Significance of pump curve shape

### Flat pump curve shape

Let's focus first on three different systems with an identical pump in each one. The pump has what is called in the industry a flat pump curve.[4] Notionally, this means an **HRTSO** in the region of 1.1, (Fig. 5.32).

Initially, all three systems pump at the same rate, because they have the same intersection, despite their curve shape differences. Now imagine that the pump characteristic curve decays by a specific amount. This is shown by the short parallel line, [2].

In the case of the 'All Friction' system [5], the new intersection point results in a much smaller flow shift than either of the other two. In this example, it decays to 94% of the rated flow.

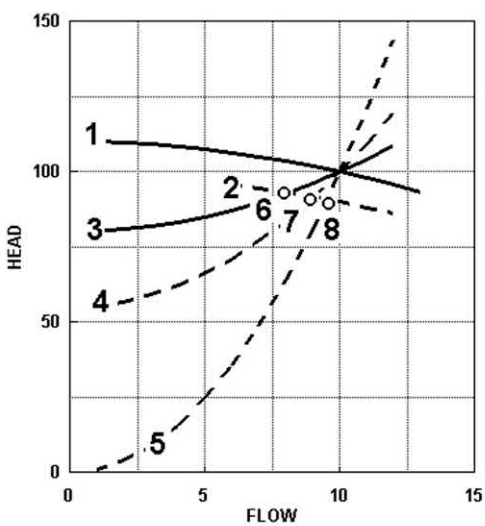



**Fig. 5.32** Pump with 'flat' curve operating and three different system curves. Note the significant difference in consequential equilibrium flowrates.

The biggest shift comes in the 'Largely Static' system [3], where the pumping rate falls to only 80% of the rated flow.

Note that in all three cases the pump curve has decayed by the same amount. But the pump in a 'flat' largely static head system *appears* to 'wear-out' very much more quickly than when in a steep 'all friction' system.

This sort of system influence is almost never considered, even at the system design stage. In reality, there is often little that can be done to improve on the inherent weakness of a configuration, once all the hardware has been committed.

In practical examples, it is rarely possible to change the system static head component by very much, if at all. Either the piping system has been designed to convey liquid from one level to another, or it is confronted with an internal vessel pressure difference dictated by the process. These parameters cannot be easily changed. However, an appreciation that this sort of pump-system combination is most susceptible to wear may justify a re-appraisal of the basic system concept.

From the pump standpoint, it seems at first that there is also little room for manoeuvre. The natural reaction of the pump designer is to create a machine that has a 'flat' performance curve [a low **HRTSO**]. Why? Because this configuration is smaller, has lower capital cost[5] and may have a slight efficiency advantage. Pump designers compete with each other for market share in most cases. Therefore, it should come as no surprise that commercial machines all tend to have a flat curve [low **HRTSO**] as a default. But steep curve options are often available.

If a commercial machine is only available with an unusually steep curve [high **HRTSO**] this is because the hydraulics were originally custom designed, or the designer was inept.

## Steep pump curve shape

Turning now to the case where the pump curve is rather steep, (Fig. 5.33).

The **HRTSO** may be in the region of 1.4,[6] compared to the value of 1.1, which typifies flat curves. Again the flow in all three systems is initially the same, as might be expected. Again, as the pump curve decays due to wear [2], there is a different response depending upon the system shape.

Once more, the pump operating in an all friction system shows the smallest flow shift. But in this situation, the flow shift is much smaller than in the case of a flat pump curve. In fact, the flow shift is 96% in this example, compared to 94% with the flat pump curve. However, even on

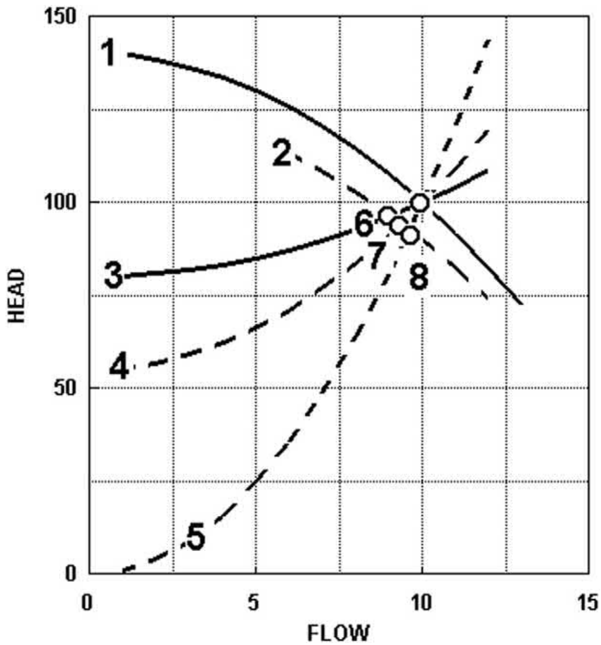



**Fig. 5.33** Pump having 'steep' curve operating with three different system curves. The difference in consequential equilibrium flowrate is much smaller than in the case of 'flat' pump curve.

the largely static system, where the flow shift is the most extreme, the flow in the system still exceeds 85%, compared to 75% with the flat pump curve.

The lesson from this exercise is that a given amount of pump wear and performance loss has:

- The most profound effect on pumping rate if the pump curve is flat and the system characteristic consists largely of static head.
- The least effect on pumping rate if the pump curve is steep, and the system characteristic is largely frictional.

## Parallel pumping

The concept of operating two impellers in **Parallel** was discussed earlier. The outcome of parallel impeller pumping is to increase the total machine flow. The concept can be extended to operate two or more machines in parallel. The system characteristic has an influence on their behaviour.

Parallel pumping is best reserved for systems where the friction ratio, Eq. (5.2), is low. This is because if a single pump has to operate, there may be an

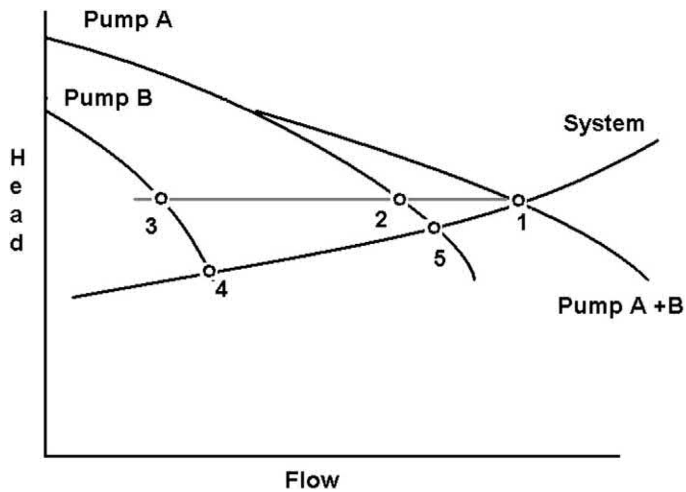



**Fig. 5.34** Parallel operation of 2 dissimilar pumps. Pump A contributes most to overall output.

issue of insufficient NPSH at the run-out flow. This is touched upon below (Fig. 5.34).

## When pumps operate in parallel, the flow of each pump, at the same head, is added

This gives a new curve for the combined efforts of all pumps. The interaction of the systems curve and the combined pump curve are at point [1]. This describes the total flow in the system, comprised of flow [3] plus flow [2] If only pump A operates, and then the system flow is point [5]. If only pump B operates, then the flow is point [4]. In this example, the two pumps have different characteristics.

Often identical pumps are used, (Fig. 5.35).

In this case, the combined flow from 3 pumps is [1] each pump delivering [3]. If one pump operates on its own, it would want to operate at point [5]. This might be way beyond the bep of the pump, and at a point where there is not enough NPSHA to prevent pump cavitation. Cavitation would curtail the pump flow to point [6].

A common misconception is that the flow from 2 pumps is twice that from one pump. From Fig. 5.36 it should be clear that this would only be true if the system curve was absolutely flat *[little friction, if any.]*

Real systems have friction, so the flow increase becomes a little less than double. The higher the friction ratio, the more acute this becomes. With

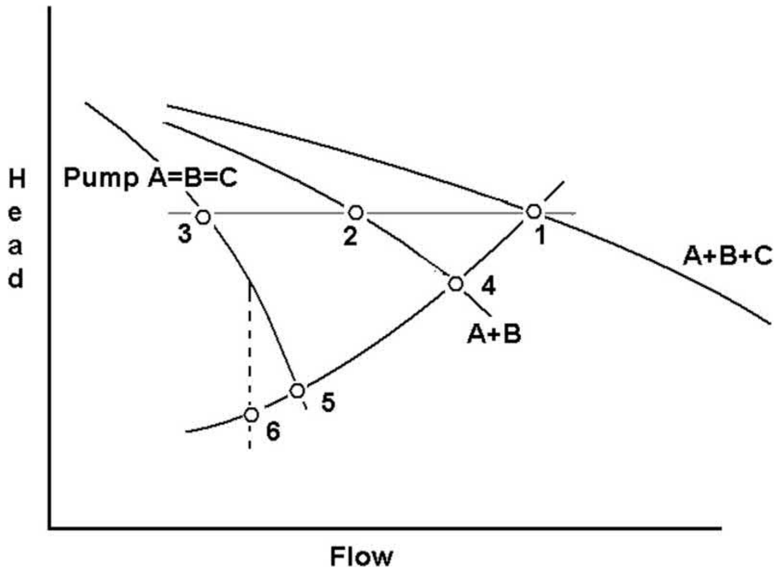



**Fig. 5.35** Parallel operation of 3 identical pumps. Each pump provides an equal share to the overall output.

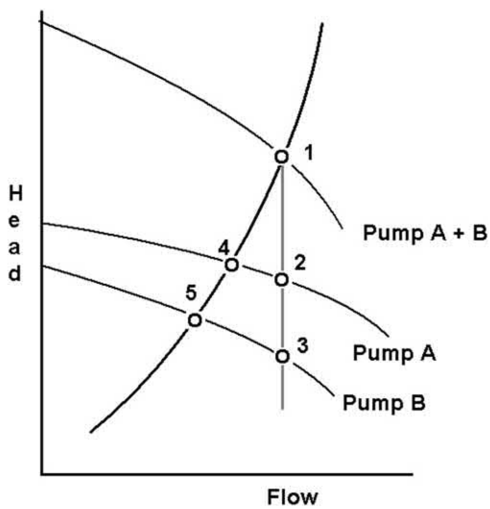



**Fig. 5.36** Series operation of 2 dissimilar pumps. Each pump delivers the same flow, but Pump A contributes most pressure.

very high friction ratios, the concept of parallel operation should be replaced by series pumping.

## Series pumping

The concept of operating two or more impellers in series was also discussed in earlier. The outcome is to increase the total head at any flow. The concept can be extended to operate two or more machines in series. The system characteristic has an influence on their behaviour.

Series pumping is best reserved for systems where the friction ratio [Eq. 5.2], is high. This is because if a single pump has to operate, there may be an issue of insufficient pump head to meet the minimum static head. Pump churning would result.

### When pumps operate in series the head of each pump, at the same flow, is added

This gives a new curve for the combined efforts of all pumps. The interaction of the systems curve and the combined pump curve is at point [1]. This describes the total head in the system, comprised of head [3] plus head [2]. If only pump B operates, then the system head-flow is point [5]. If only pump A operates, then the system head-flow is point [4]. In this example, the two pumps have different characteristics.

Often identical pumps are used, (Fig. 5.37).

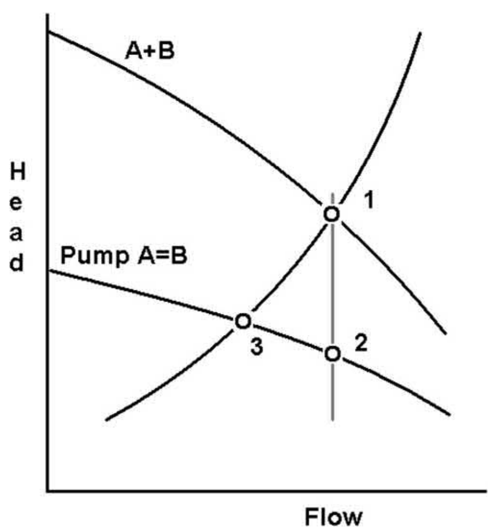



**Fig. 5.37** Series operation of 2 identical pumps. Each pump provides an equal share of flow and pressure.

In this case, the combined head is [1] each pump delivering [2]. If one pump operates on its own, it would want to operate at point [3]. As shown, this is not a problem, but the reader could visualise pump and system combinations where the maximum head of the pump was still less than the static head of the system. The pump would develop no flow and churn away, only pumping down the re-cycle line [where fitted].

## Multiple pipe systems

So far, the discussion has only considered single systems, but often a pump delivers in to a complex network. To give some sense of the logic behind this, consider a pump delivering to two different sized nozzles located at different heights above the pump (Fig. 5.38).

This will help give some insight as to how complex systems interact with the pump. Very complex networks are beyond the scope of the current volume.

The system characteristic of the first nozzle is shown as N-A and for the second nozzle N-B. The resistance of the combined systems is as shown. The flows of each nozzle, at the same head, are added to give the total system flow for both nozzles. The total system curve intersects the pump curve at Point [1]. The flow at point [1] comprised of flow [5] from nozzle B and flow [4] from nozzle A. If nozzle A is shut, the flow through B

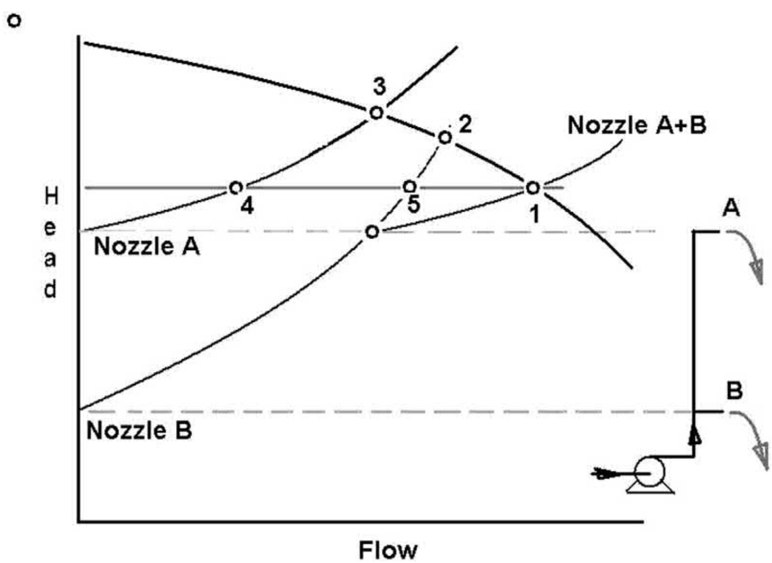



**Fig. 5.38** Multiple systems-parallel nozzles.

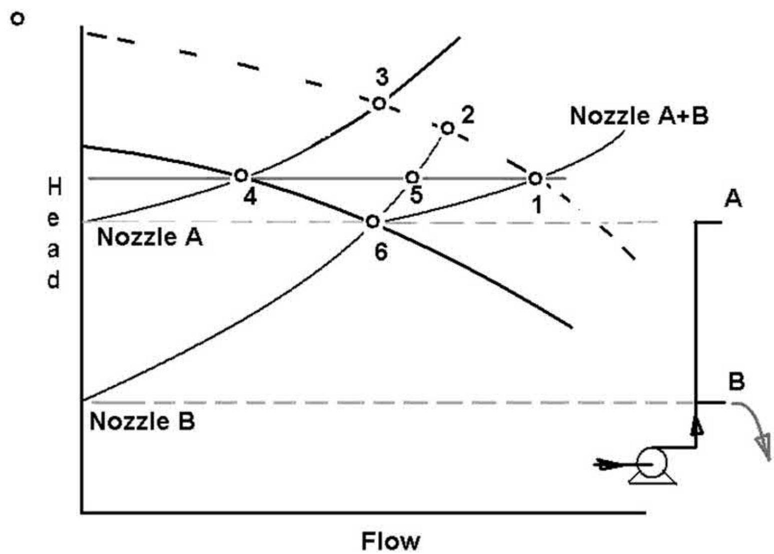



**Fig. 5.39** Multiple systems-parallel nozzles with pump head not sufficient to meet static head of nozzle A.

increases from [5] to [2]. If nozzle B is shut, then the flow through A increases from [4] to [3].

If the pump output reduces–say due to speed change, the intersect with the total system reduces and so does the output at each nozzle. At a certain speed reduction, the total flow reduces from point [1] to point [6] and the flow from nozzle A ceases completely, and only nozzle B discharges (Fig. 5.39).

## Multiple pump and system

The most difficult analyses occur when multiple pumps discharge into multiple systems, (Fig. 5.40).

A good example is when a number of pumps feed a number of different nozzles. This will serve to introduce the principles involved.

Let's look at the behaviour as different pump and nozzle combinations are invoked:

- If all pumps and nozzles are in operation, then the total system flow is at point 1, consisting of an equal contribution from each pump. If the pumps were not identical, then the contributions would be different. If one pump is stopped, then the system flow reduces to point 2.
- If only one pump operates, then the system flow reduces to point 3.

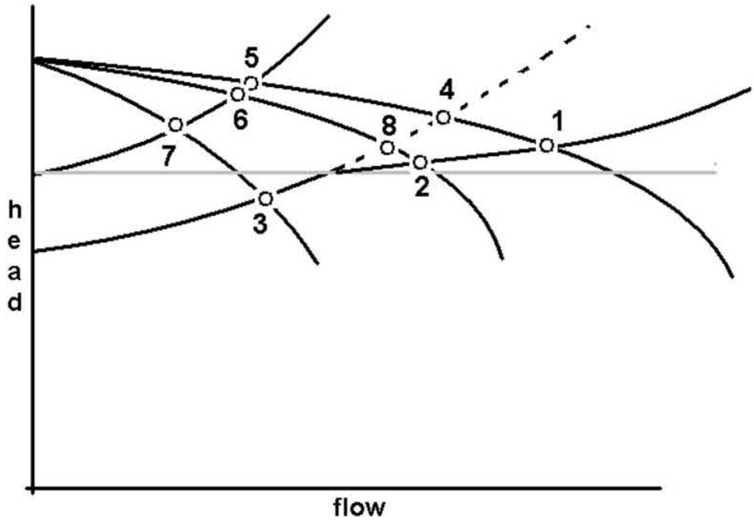



**Fig. 5.40** Scheme consisting of multiple pumps, and multiple systems.

- If one nozzle is shut, then the flow would be at point 7, or 3-depending upon which nozzle was closed.
- If a second pump was started up, but still only one nozzle opened, the system flow would be point 6, or 8 depending upon which nozzle was open.
- If the third pump was then brought in, then the system flow would be either point 5, or point 4, depending upon which nozzle was open.

This simple example helps give some sense of the complexity in system response when multiple pump and systems are operated.

## Pipe work design details

Some attention to the detailed design of the pipe work can help avoid problems at a later date. The following notes may be useful.

## Suction piping

### *Avoid air traps*

The pipe may be fully connected, and the entry fully submerged, but the pipe run may permit collection of an air or gas pocket. This will occur if instead of continuously rising towards the pumps, there is a 'hump' or high spot where air can collect in the suction line. This could arise by bad design, casual installation, or by accident. The air/gas could be dissolved in the

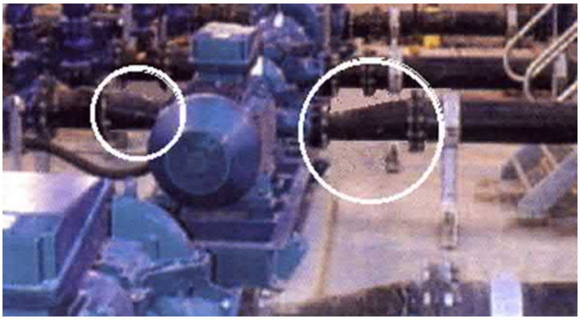


**Fig. 5.41** Incorrect installation of pipe reducers on suction and discharge of axially split case pump. This arrangement encourages air to be trapped in the pipe high spots.

liquid and come out of solution for a number of reasons, see later section (Fig. 5.41).

The Figure shows a typical case where a pipe reducer has been fitted upside down. The consequence of this is that air can collect in the high spots. After a while the accumulated air achieves a critical mass and gets swept into the pump. The effects of this can be serious. When pumps are faced with a mixture of liquid and gas[7] it can be presented as either a well homogenised mixture, or as discrete slugs of gas in the liquid. Generally slugs are more serious. There is rarely enough time for them to disperse inside the impeller, so they create mass unbalance effects. This probably has more effect on the mechanical seal, than on the bearings, though both are affected. The most serious consequences will be on pumps that operate on a suction lift, because there is a significant risk of de-priming if air or gas is ingested–even briefly (Fig. 5.43).

### *Bends in suction piping*

Most pump manufacturers would strongly discourage fitting a pipe bend straight onto the suction flange of the pump. Instead some minimum length of straight lengths between the pump and last bend is defined. The reason for this is that the flow field emerging from a pipe bend is very uneven compared to the flow into it, (Fig. 5.44).

The 'tighter' the bend, the more uneven is the exit flow.

Pump manufacturers would typically define a minimum bend radius as 4 times the bend bore, even with the recommended straight pipe length between it and the pump of 5—10 times the bore. The logic here is that the uneven flow field has a chance to 'mix out' to a more uniform picture by the time it arrives at the pump inlet.

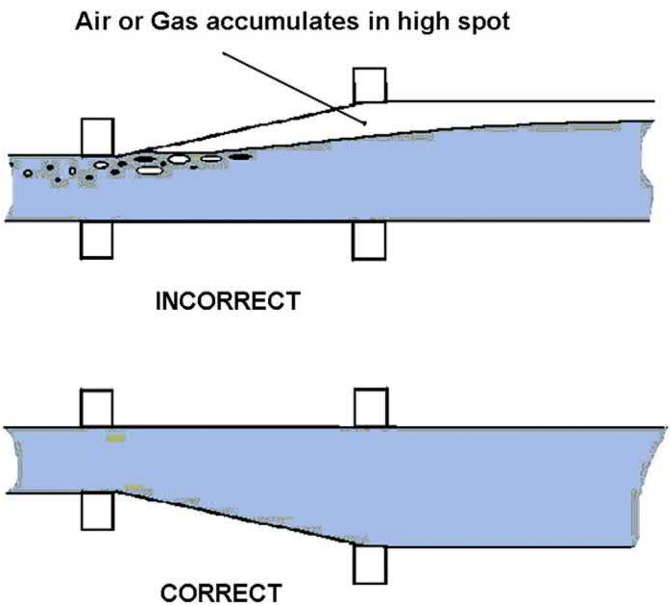



**Fig. 5.42** Pipe reducers on the suction side should be arranged to discourage the accumulation of air or gas in the high spot.

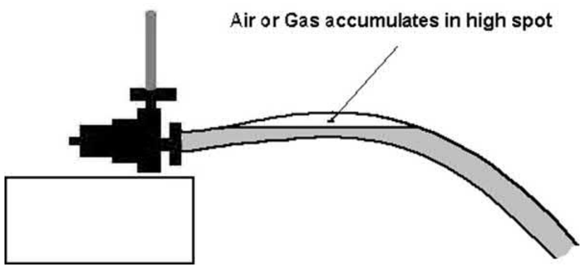



**Fig. 5.43** Suction line should constantly rise towards the pump and high spots should be avoided.

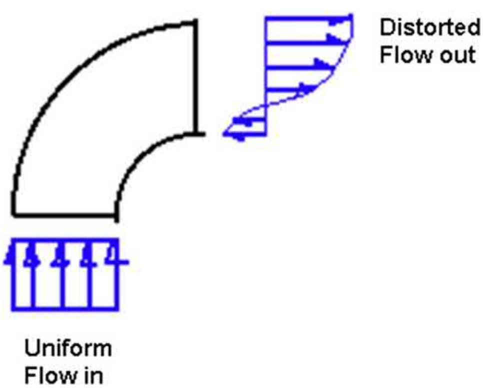



**Fig. 5.44** Flow distortion in a normal bend.

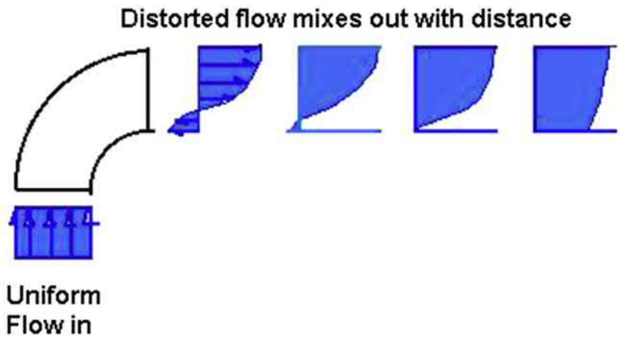



**Fig. 5.45** Straight lengths are necessary to allow flow to stabilize into a more uniform pattern.

If the area ratio of the bend relative to the pump nozzle is high, say 2 or *more [implying a reducing section after it]* then the subsequent acceleration of the liquid can also overcome many of the unsatisfactory aspects mentioned above, (Fig. 5.45).

This requirement of straight length on to the pump nozzle is often a nuisance to pump plant designers. This apparent extravagance can increase the size of the installation and of course the cost. Often there are temptations to compromise on this in order to shrink the installation cost. The manufacturer will rarely, if ever, sanction such compromise, so other solutions get tabled. One common suggestion is to put splitter vanes into the pipe, in the hope that this will create order from disorder. In practice, all that happens is that the uneven flow picture gets divided into to two separate but slightly less disordered flow fields. So the approach has little to recommend it.[8] (Fig. 5.46).

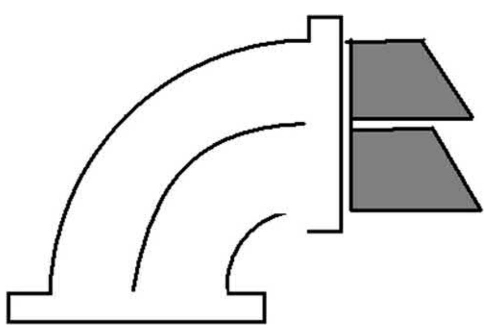


**Fig. 5.46** Central splitters do divide the flow distortion into smaller patterns, but do little to even out the flow.

However, this does point the way to a more useful approach. A large number of splitters would break down the flow into a large number of disordered cells but the combined effect would still be to reduce the disorder level. Practical problems prevent a large number of splitters being fitted to a circular elbow. However, fitting many splitter/turning vanes into a 90-degree mitre bend is very easy to manufacture. *This is precisely the same as the turning vanes in the bend of an aerodynamic wind tunnel.* Furthermore, this so-called **'cascade'** type bend adds very little disorder to the flow across it so it requires no straight length between it and the pump.

In fact it can be bolted straight onto the pump flange. A **cascade bend** mounted straight onto the pump is superior to a bend plus straight length, yet demands much less space.

The author has had excellent experience with these devices in difficult pump installations, within the size range 6″ to 36″ (Fig. 5.48).

But why is an uneven inlet flow field to the pump undesirable? Let's simplify and imagine what happens in an impeller passage throughout one shaft revolution. As the passage inlet traverses the low velocity area of the flow field, the angles of flow into the impeller are lower than would be the case if it had a long straight pipe. As it moves into the area of average velocity, the flow angles are the same as with a straight pipe. As it then

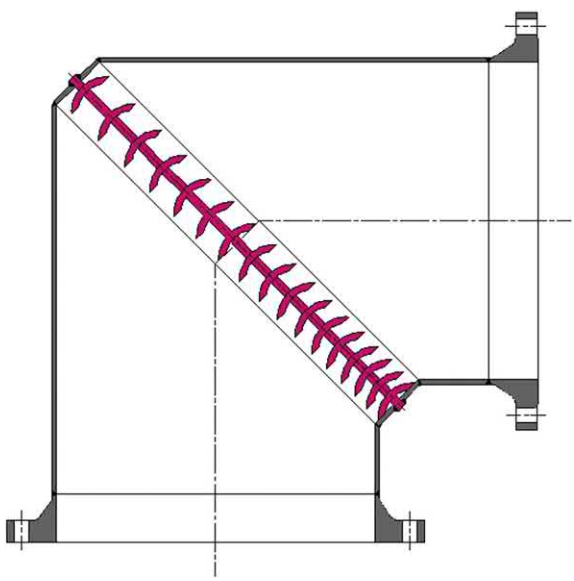


**Fig. 5.47** Cascade turning vanes largely eliminate flow distortion in the outlet flow. The unequal spacing of the vanes is a critical feature. [Weir Engineering Services].

**Fig. 5.48** This 36 inch cascade bend was retrofitted **directly onto** the inlet flange of a large double suction water pump supply in a public utility. It made a significant improvement in flow quality over the original short radius bend. By suppressing cavitation, that helped to reduce the overall pump noise level by more than 5 dB [A].

moves in to the high velocity part of the flow field, the flow angles exceed that in a straight pipe situation. So in a single shaft revolution, each impeller passage experiences a change in fluid approach angle. As the angle changes, so does the performance of each passage. The collector passages are designed on the assumption of constant flow being spewed from the impeller rim. If the flow is not constant at every point on its rim, then the radial forces exerted by the collector on the rotor has a cyclic component added to it. This appears as an additional vibration component (Figs 5.49–5.52).

### *Asymmetric flow to double entry impeller*

Double entry pumps require a special mention. It is bad practice to lead the suction pipe parallel to the shaft and then turn into the pump (Figs 5.53 and 5.54).

If the suction pipe bend of a double suction pump is in the same plane as the shaft, then the uneven flow discharging from it is presented to the impeller. The change in flow angle caused by the flow disorder, plus the velocity changes, means that one impeller eye can receive more flow than the other.

This can cause cavitation in marginal situations. Since double suction impellers are often employed *specifically* to address NPSHR/Cavitation possibilities, anything, which might undermine this benefit, is undesirable.

**Fig. 5.49** Combined bends, with little or no straight pipe lengths between them and the pump, feed badly distorted flow to the suction nozzle. Only small pumps, such as shown, can tolerate this. Larger pumps would suffer seal and bearing decay.

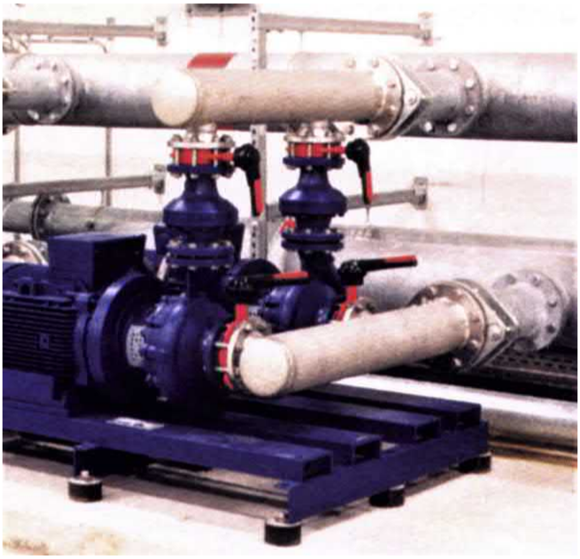


**Fig. 5.50** Abrupt flow change on suction of small pump, with little or no straight pipe lengths before the suction nozzle. This arrangement is not recommended unless the NPSHA exceeds NPSHR by a factor of 2, and at least 4 straight lengths can be included.

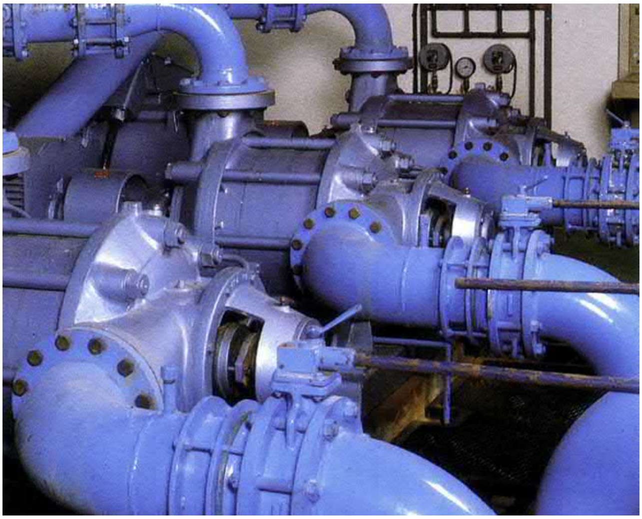


**Fig. 5.51** An example of a non-preferred arrangement — two suction bends very close together.

**Fig. 5.52** Another example of a non-preferred arrangement — two suction bends very close together.

**Fig. 5.53** Poor suction pipe arrangement with bend in the same plane as the shaft.

**Fig. 5.54** Good suction pipe arrangement at right angles to plane of shaft. Photograph Ingersoll Rand.

Such unsteady flow inequality can also raise the vibration level of the pump, as previously explained. It can also help create axial unbalance in an otherwise nominally balanced rotor. Disappointing bearing life may result.

All these risks are heightened; if the impeller has a rather large inlet eye [say Nss greater than 12,000 per eye.[9]]. Such exaggerated hydraulic designs are particularly sensitive to any sort of inlet flow disturbances *see* Appendices D and E.

Good practice dictates that the suction pipe approaches the machine at right angles to the shaft, Any flow disorder carried over from a preceding bend is fed equally to both impeller eyes, and symmetry is maintained.

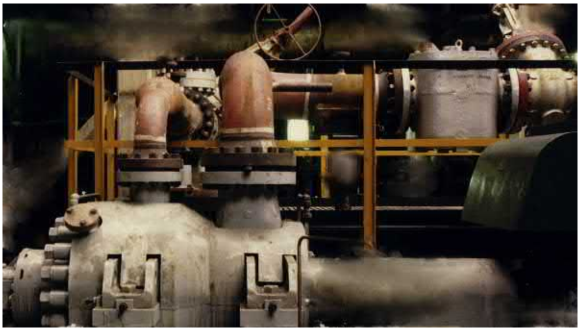


**Fig. 5.55** Non preferred suction pipe arrangement with first bend in the same plane as the shaft, turning to become preferred at the second bend.

On a note of reality, the negative effects of poor suction pipe design are somewhat size-dependent, as far as small pumps are relatively more robust and more cavitation resistant. Pumps of 1″ to 2″ suction branch are largely immune to these effects, whereas pumps of over 6″ become increasingly more susceptible.

Fig. 5.55 shows a high temperature double casing pump equipped with a double suction impeller. The double suction impeller has an unusually high suction specific speed. At first sight, this installation does not appear to violate the rules set out for double suction impellers, because the flow approaches the pump at right angles to the shaft. However, the preceding bend is parallel to the shaft and would feed distorted flow the bend on the pump. In this particular situation, the risks were very well debated before the installation went ahead. The pumps are fitted deep inside an existing plant and the only available route for the suction pipe that would have been parallel to the shaft was to bring it over the motor [seen to the right of the picture]. The pump owner was very reluctant to do that – for obvious reasons. This non-preferred arrangement was sanctioned by the manufacturer on the basis that:

- The suction pipe diameter greatly exceeds that required to match the impeller eye diameter. This ensures steady flow acceleration towards the first stage impeller.
- The suction pipe diameter itself is oversize, ensuring lower than normal pipe velocities.
- The distance between bends exceeds 4 diameters.
- The pump thrust bearing capability very exceeded the normal steady state requirements substantially

- Both the manufacturer and the user had entered into open discussion on this arrangement before construction began. The user very clearly understood the risks and threats of this arrangement.

### Inlet submergence

In the situation where suction draws from a submerged bell mouth, there must be adequate depth of liquid above in order to prevent air entrainment and vortices. Fig. 5.56 gives some guide as to the minimum acceptable submergence. Upper and lower limits show current industry experience, though it should be said that there are no negative hydraulic effects from having too much submergence. Note this guide assumes that there are no other major faults in the wet pit sump design, particularly those that would induce high levels of swirl into the liquid (Fig. 5.57).

### Suction pipe design

- Suction pipe should be equal to pump nozzle diameter or preferably one size larger, NEVER smaller
- If pipe reducers are installed, then see Fig. 5.42
- At least 5 pipe diameters should be provided after bend. If not possible, consider cascade bends, *see* Fig. 5.47

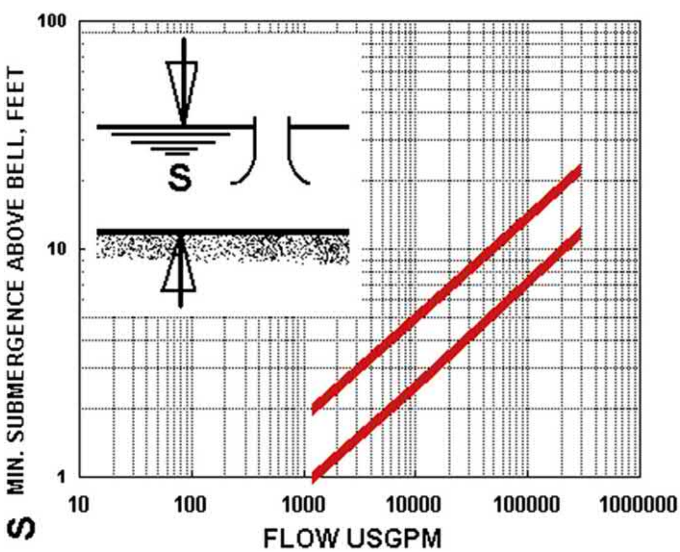



**Fig. 5.56** Submergence guidelines for prevention of air entraining vortices.

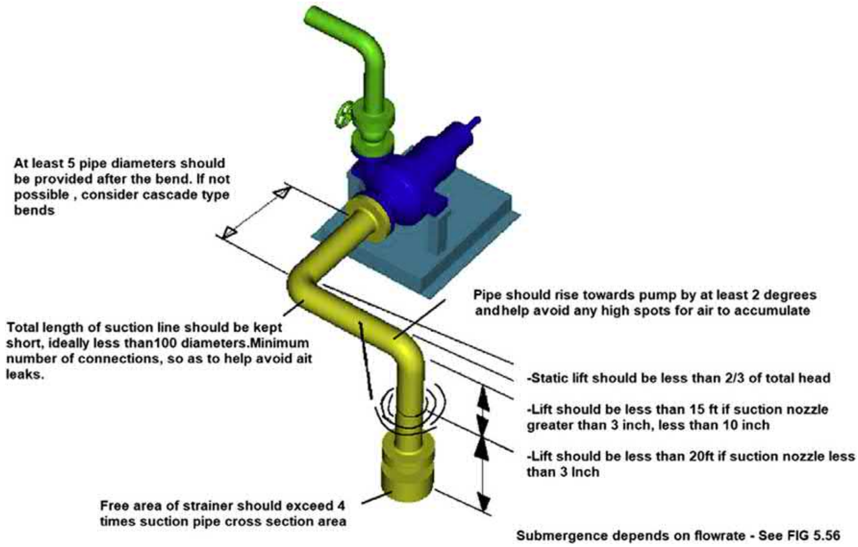



**Fig. 5.57** Guidelines for good suction pipe design. Majority of pump hydraulic problems originate from poor suction design.

- Total length of suction line should be kept short [less than 100 diameters] with the minimum number of connections. *[To avoid air leaks]*
- Suction pipe velocity should not exceed levels given in Table 3.1
- Pipe should steadily rise towards pump by at least 2°. There should be no high spots for gas to accumulate
- Static lift should be less than $^2/_3$ of total head.
- Lift should be less than 15 ft if suction nozzle greater than 3″ and less than 10″
- Lift should be less than 20 ft if suction nozzle less than 3″
- Submergence depends upon flowrate, *see* Fig. 5.56
- Prior to connection, pump and pipe flanges should be parallel to 0.030″ and bolts should be loose in flange holes.

## Discharge piping

### *Bends in discharge piping*

A characteristic of low specific speed pumps[10] is a very high liquid velocity in the casing throat. Typically this is 50% of the impeller peripheral velocity, or more. The role of a volute casing is to diffuse this flow down to a much lower velocity in the discharge branch. However, this is not always accomplished fully and there is risk that a high velocity jet is sent out into

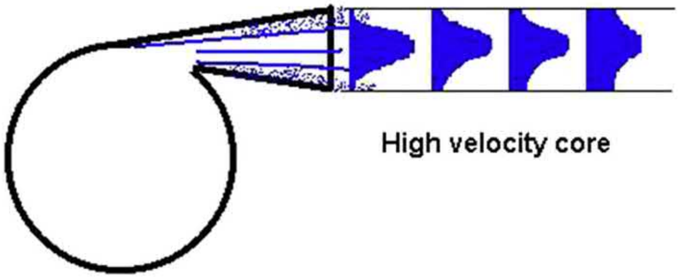



**Fig. 5.58** High velocity core produced in pumps of low specific speed. Pipe bends should not be close to the pump, else vibration may result.

the discharge pipe. If this pipe is straight, then the jet entrains surrounding liquid and mixes out to a uniform harmless flow (Fig. 5.58).

However, if a bend is applied straight onto the discharge of low specific speed pumps, *see* Fig. 5.58, there is a risk of this high velocity fluid interacting with the bend and creating low frequency broad-band vibration. Suggest a straight length of about 5–10 pipe diameters as a minimum (Fig. 5.59).

This is an appropriate time to mention the importance of piping clamps near to pump. The unsteady momentum change resulting from turning the liquid in a bend can generate low frequency vibration in the suction and discharge piping. It is therefore good practice to firmly clamp the pipe close to the pump (Fig. 5.60).

## Discharge pipe design

- Pumps of low specific speed [less than 750 – Fig. 3.30] should have at least 10 diameters before the first bend
- Non return valve, [where fitted] should be downstream of valve, in order to protect it against water hammer
- Gate-valve stem should be parallel with shaft axis, particularly with double volute pumps
- Consider connection upstream of valves for permanent leak off connection to protect against operation at too low a flow.
- Prior to connection, pump and pipe flanges should be parallel to 0.030″ and bolts should be loose in flange holes.

Discharge pipe velocities should not exceed the levels given in Table 5.1.

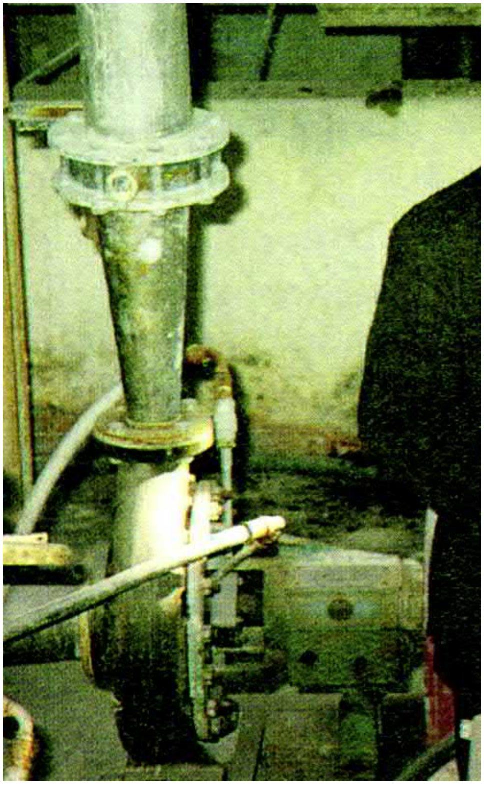


**Fig. 5.59** This illustrates good discharge pipe practice. A slow diffuser/expander section is followed by substantial length of straight pipe, clamped before the first bend is encountered.

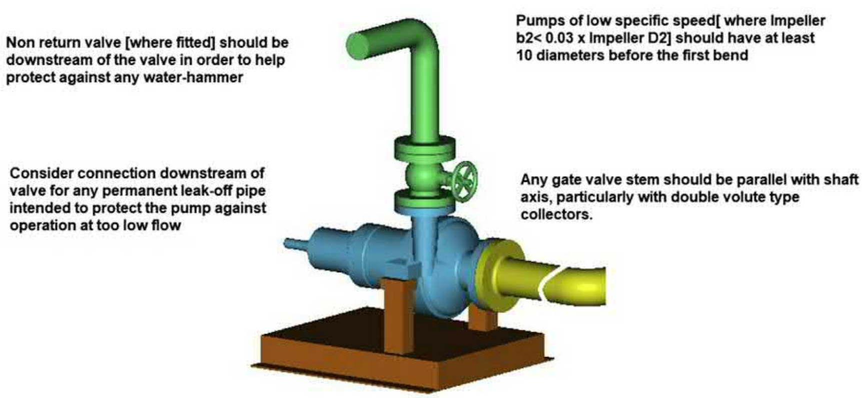


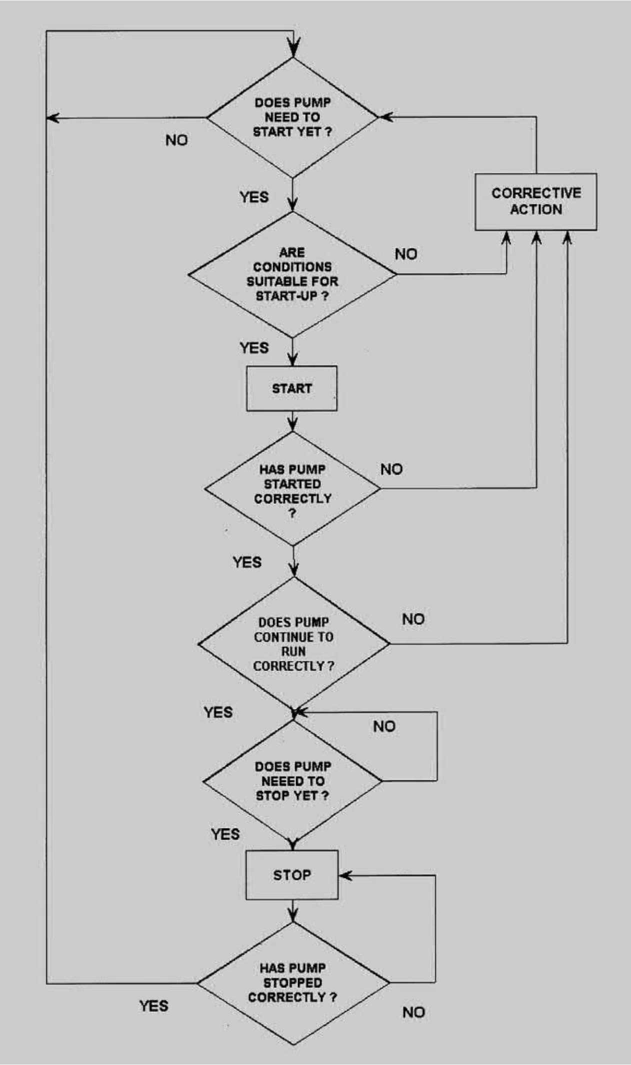

**Fig. 5.60** Guide to discharge pipe design.

## Endnotes

1. Departures from atmospheric pressure acting on the feed surface slightly complicate the analogy and will be discussed later. For simplicity they are ignored at this stage.
2. Strictly speaking, the level of liquid feeding the pump should be at the same level as the pipe exit.
3. Determination of pump characteristic curves is discussed in Appendix I.
4. It is easy to get more system flow from pumps with a flat curve shape. Conversely, the pump flow is very sensitive to system changes.
5. At the time of writing most pumps are bought on the basis of first cost, rather than their life cycle cost. Evaluation based on life cycle costs may favour pumps with steeper curves.
6. It is not easy to get more system flow from pumps with a steep curve shape. Conversely, the pump flow is insensitive to system changes.
7. This is called a 2 phase fluid.
8. Bends right on the suction make the pump particularly susceptible to the effects of inlet backflow if the pump should operate at low flow, Appendix D.
9. In Usgpm-feet-rpm units.
10. Pumps where the impeller outlet width is less than about 4% of its diameter. Passage width ratio is a very crude guide to pump specific speed.

CHAPTER 6

# Troubleshooting prelude





*Troubleshooting Centrifugal Pumps and Their Systems*
ISBN 978-0-08-102503-1
https://doi.org/10.1016/B978-0-08-102503-1.00006-2

The working life of a pump can be well described in the form of a flowchart. This flowchart is used as the backbone of the remaining Chapters. Using this format helps keep in mind that decisions or behaviour in one 'box' can also impact others. For ease of reference, the flowchart is also used as Chapter headers throughout the book. A few words of explanation will be useful at this stage, though the individual Chapters give more detail.

The 'start pump' decision may be an arbitrary one, based on personal opinions. More often than not, it will be an automated decision. Either way, there are some potential pitfalls to be wary of. Sometimes the pump operators were not responsible for, or involved in, the equipment purchase. They may not share the same mental image of how things should be run. Furthermore, the people responsible for running the equipment may have different views from the plant designers as to how and when the pump should start. As a rule, operations staffs are most nervous of the start-up part of the loop. Sometimes they make what the plant designers consider to be illogical decisions, simply in order to minimize exposure to start-up risk, Once the decision to start has been made there are a number of pre-checks that must be satisfied before the start process is put into action. These are detailed in Chapter 8. The scope of these checks varies enormously of course. The intensity of checking is very much higher the first time the pump is ever being started, or is restarted after a substantial overhaul. Once the integrity of the unit has been proven, some pre-checks can be excluded.

- At one extreme, [Central Heating pumps [Domestic Circulators]] require almost no pre-checks and start-up is almost immediate.
- On the other hand, the most complex pumping machines are found in the Oil Industry, or in Power Generation. Here the first pre-check can take several hours. Even when the integrity has been proven, subsequent checks may take more time than imagined.

Once pre-start checks are complete, the start process begins. This is almost immediately followed by a heightened concern that all is as it should be. This may not always be obvious initially. The pump control system may require some adjustment to balance the pump-system interaction. This arises because very often pumps are not started at full load. Instead they are started at low load, and then gradually fed in to the system. *[It is a characteristic of centrifugal flow machines that the absorbed power tends to be lowest at low flow. But, as Chapter 9 shows, this is not the case with mixed or axial flow machines.]* Equally things like bearing temperatures will take some little time to stabilize for example. So some time may elapse before this state of heightened concern can be relaxed.

Concern may then shift away from the start-up risks and now focus on the long-term pump condition. Does it continue to pump satisfactorily? Are there any worrying trends in [say] vibration level, bearing temperature, noise emissions?

With time, a reason to stop the pumps will arise.

- This may be planned; for example a complete plant shut down. In some cases this may be specifically to carry out preventative maintenance on the pumps. In other cases, more major pieces of plant warrant maintenance during shutdown. Here, all that happens is that the spare pump is now designated the main pump. Usually, the spare had previously been overhauled while the plant was on stream. Or perhaps the pump has fulfilled its role for the moment

On other occasions, the stoppage may be unplanned.

- A fault may have developed with the pump and it needs immediate attention. Often there are signals of distress ahead of the event, which, if monitored, may allow damage limitation. This advance warning is usually measured in hours rather than days or weeks.
- A fault may have developed somewhere else in the pumping system.

Having de-energised the pump, it does not infallibly follow that it is stationary – *see Chapter 14.* Further steps sometimes need to be taken to be sure of this.

Some pumps travel round this loop quite often. Pumps installed in a batch process may do so several times a day. On the other hand pumps on a continuous service may only go round very few times. A boiler feed pump in a major utility may only circulate the chart a handful of times in 5 years.

Each block of the flowchart represents an opportunity for troubles to develop. In many ways, the type of risk relates to how quickly the loop is executed.

- Pumps that circulate the loop frequently are subjected to all the risks associated with start-up and shut down.
- Pumps that circulate the loop slowly are more likely to suffer long term wear and tear.

This book will examine each flowchart block in some detail looking at the typical pitfalls, as well as measures for prevention, or cure.

Diagnostic charts are also included to try and narrow down such statements as ‘The pump is not delivering enough flow!’ into a few more helpful statements.

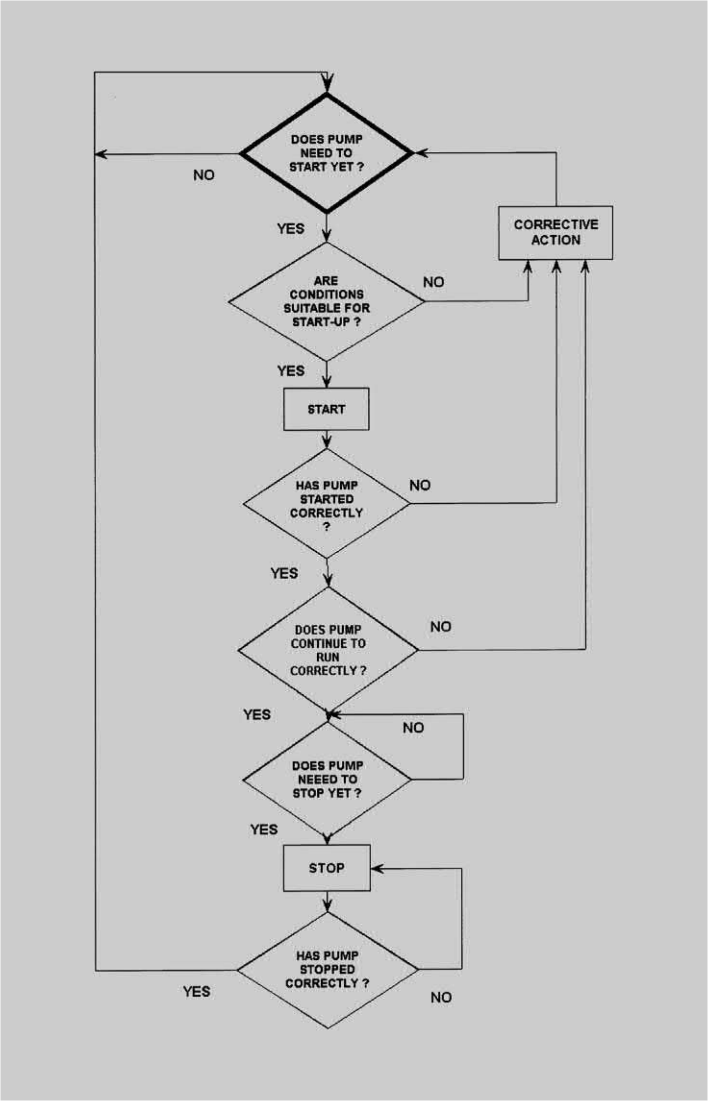
But a warning! A popular saying reminds us that:

'ASSUME' made an ASS out of U and ME.

Keep this in mind when faced with a real problem. Sadly, the path of least resistance is often to make assumptions about some of the issues raised here, rather than going to 'see and touch' the item in question.

CHAPTER 7

# Does the pump need to start yet?




*Troubleshooting Centrifugal Pumps and Their Systems*
ISBN 978-0-08-102503-1
https://doi.org/10.1016/B978-0-08-102503-1.00007-4

## Introduction

Starting pumps can often seem to be that part of the cycle with the highest risk. This is justified by experience, so there is some sense in discussing the need to start, or at least the frequency of starts. Pump operators are almost always comfortable to start or stop small pumps without too much formality. In many cases this can take place automatically and unattended. It is a fact though that when large or complex pumps are involved, operators often feel obliged to be present at start-up. This reflects their historical concern.

Pumps that operate more or less continuously travel round the cycle less often. They avoid many of the start-up and shut-down perils. On the other hand, they most quickly amass operating hours and so are more exposed to long term running risks. Such pumps are typically part of a bigger process. They operate continuously in concert with many other items of plant all closely matched.

Pumps that operate on a batch process cycle round the diagram most often. They are more exposed to start and stop perils, but they more slowly amass running hours. They 'deteriorate' in different ways. Furthermore, if the pumps are driven by electric motors, there may sometimes be constraints put upon the maximum permissible number of starts per hour.

## Manual or automatic?

On start-up, the choice will be between a manual or automatic start processes.

Manual starts take place on the basis of opinion or need. Manual start-up takes place when:

- Automatic start equipment cannot be afforded, or is not appropriate
- The parameters for start are very
  - Opinion-based, or
  - Complex, or
  - Not amenable to algorithmic logic.

Where starts take place automatically, the opinions or needs have been basic enough to have been captured and defined as a set of rules. The benchmark rules are usually based on either level control or pressure control. Pump speed is the usual adjustment in this process, either 'on-off', or step-less variation.

With level control, incoming liquid inflow fills some element of the process. Above a certain point opinion says this is not permissible. This is often termed 'high level' or 'upper level'. The pump or pumps are started and the process element evacuated until the liquid level is within preferred limits. Usually the pumps operate until the liquid level is as far below the upper level as possible. Simplistically, this strategy allows the maximum time interval between pump restarts. There may be a manual override for unforeseen circumstances.

With pressure control, the pump is being used to sustain a minimum pressure in a process element. This is a precondition to possible system flow. That element may be a receiver vessel, or it may be ring main system acting like a receiver.

The relationship between pump flow rate and the pumped volume of the process element dictates the pump behaviour of course-in particular the frequency of stop-starts.

## Starting single pumps

### Does the pump need to stop or start so often?

One example of a batch process is a pump emptying a sump that is filled from one or more independent sources. An effluent drain pump is a typical example.

If the pump is sized for too large a flow[1] it will quickly empty the sump[2]. If on the other hand it is sized for too small a flow, it may run almost continuously as it struggles to keep pace with the inflow. Since they travel round the loop at different speeds, these two operating extremes are therefore associated with different risks and problems (Fig. 7.1).

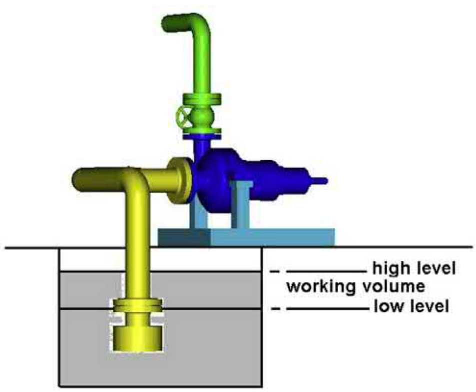



**Fig. 7.1** Sump volume definitions [A English].

**Table 7.1** Maximum starts per hour-related to electric motor size.

| Motor nameplate power kW | Maximum starts per hour |
|---|---|
| 4 | Less than 15 |
| 11 | Less than 10 |
| 30 | Less than 6 |
| 55 | Less than 4 |

It is possible to estimate the optimum sump size for a given flowrate, [*Reference* [*11*]]. Note this equation relates to a single fixed speed pump.

$$T = 4V/QP \qquad 7.1$$

*Where:* T = Cycle time-minutes, V = Sump volume between high and low level and $Q_P$ = Pump discharge volume.

So if 10 starts per hour are desired [cycle time of 6 min] the sump volume should be 1.5 times the pump discharge volume per minute. Table 7.1 gives an indication of the number of starts motors will normally tolerate. More frequent starting may damage the winding insulation by overheating.

This mundane example shows how the need to start can sometimes be managed. We will see other examples later – when we discuss multiple pump operation.

If the pump is over-pumping, there are some simple solutions:

- Reduce impeller diameter. This will help better match the pump output to the system. Other solutions [ permanent or temporary] exist and are discussed in Appendix K
- Consider variable speed drive. This approach helps if there is some doubt over the inflow rates, or if the inflow rates vary over a very wide range
- Install smaller pump. This is rarely necessary
- Increase sump volume. This is the last resort

## Starting multiple pumps

As will be mentioned elsewhere, when operating multiple pumps in parallel, conventional wisdom says that some of the machines should be switched off when pump supply exceeds demand.

However, in practice the plant operators often prefer to keep all the pumps operating, but temporarily throttle the total output back to meet demand. This is defended on the basis that the response time to match a subsequent rapid increase in demand cannot be met by switching pumps back in to the system. Similarly, the risks associated with starting and stopping pumps are unacceptable [particularly if the pumps operate at elevated temperature and require warm-up or warm-through]. These are valid arguments that need to be recognized by the plant designer.

## Variable speed drives

The nuances of Variable Speed drive will not be discussed here. By the time troubleshooting arises, it is too late – the plant has been specified and installed. However, they should always be considered as an alternative to off-on, or throttle control. In the small to medium power levels, the energy savings can usually justify the expense. Enthusiasm for Variable Speed Drives [VSD] has never been higher. However, there often appears a failure to differentiate between the behaviour and savings, associated with on all-friction system [typical of fans and blowers] compared to systems with a significant amount of static head-as typified by many pump installations.

Nevertheless, VSD are often a viable option. The author recommends *References* [12,13] for further study.

In any event, pumps should be shut-down quickly and decisively. With VSD, or steam turbine drive it is unwise to prolong speed reduction below that at which the *Lomakin* effect, *see* Appendix F, vanishes. This would increase the risk of internal seizure in the fine clearances. So turbines should be stopped by manually activating the over-speed trip.

The mechanics of good shut-down are as follows:

## Flooded suction

- Quickly close the discharge valve
- De-energise the driver

## Suction lift

- Quickly close the suction valve
- De-energise the driver

Both approaches ensure that the pump is primed ready for the next start-up. However, it may slowly de-prime on a suction lift so a primed state should not be automatically assumed.

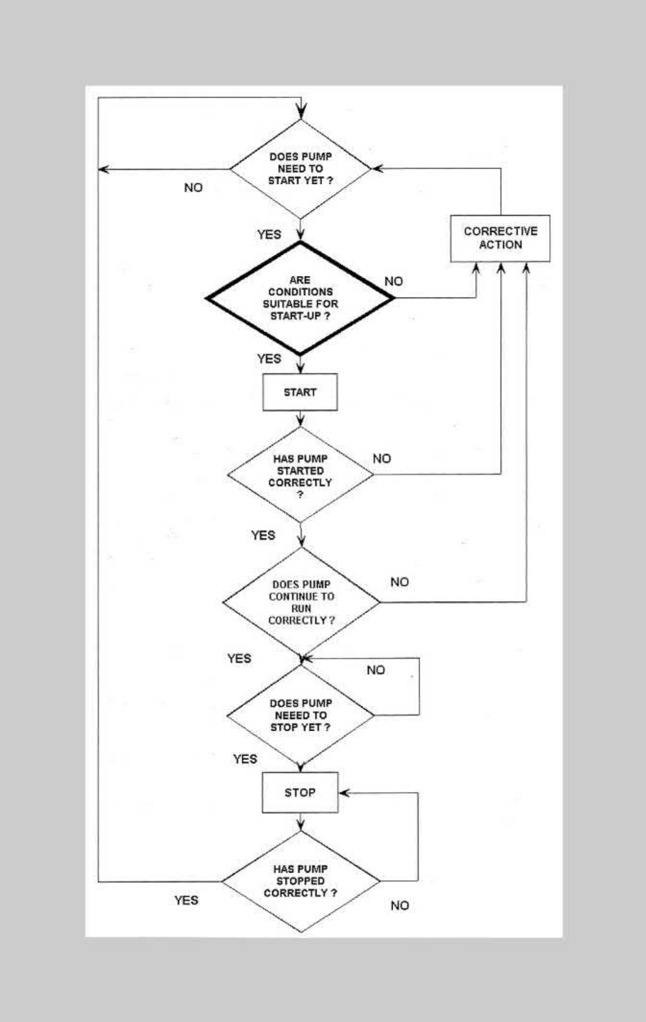
## Notes

1 Almost all pumps leave the factory generating more heads than is actually required by the application. This is discussed later as Palgrave's Fourth rule of pumping.

2 Down to low level

CHAPTER 8

# Are conditions suitable for start-up?




*Troubleshooting Centrifugal Pumps and Their Systems*
ISBN 978-0-08-102503-1
https://doi.org/10.1016/B978-0-08-102503-1.00008-6

## Introduction

There are a number of pre-conditions that need to be satisfied before a pump should be started. Start-up brings risks of both a mechanical and hydraulic nature and, as discussed earlier, there are two different situations where this question has to be asked:

- When the event represents the initial entry into the operating cycle. This may be the first start-up after Installation, repair or maintenance and often happens under supervision. This is particularly true of large, complex, or critical machines.
- When the event is part of a regular start-stop cycle. Such an event may take place under automatic control for example. This reflects operator confidence in the stability of start-up conditions.

It is in this phase that is very easy to make incorrect or hurried assumptions, with dire results. Centrifugal pumps are not very tolerant to certain types of abuse and more than half pump faults can be avoided at this stage. Hindsight most often puts in an appearance after this point.

This is a large section and it is valid to split it into a hydraulic section and a mechanical section. In general, the hydraulic section relates to system issues while the mechanical section mainly relates to the pump itself (Table 8.1).

## Pre start checks − hydraulic

The hydraulic checks can be divided into three major areas:

- Liquid issues
- Piping problems
- Priming and venting

## Liquid issues

It is quite natural to assume that once started, flow to the pump will proceed unrestricted. A previous review of the pipework design and layout may show it to be in accordance with the principles of Chapter 5. However, the real condition of the liquid may not actually promote flow. This will usually be less than obvious, particularly in a closed piping system, since lack of fluid motion would never be self-evident.

It's quite possible that the liquid conditions may change between the source vessel and the pump inlet flange. In summer solar gain may actually warm up a long exposed suction line. There are cases where this unexpected type of change to the liquid has adversely affected the NPSH [A].

**Table 8.1** Hydraulic checklist.

| **Issue** |
|---|
| Is the suction valve fully open? And are all blanking plates removed? |
| Is there liquid available to pump? |
| Is the liquid in a condition to flow freely? |
| Is the suction flow path complete? |
| Is the suction line connection to the supply vessel submerged enough to prevent the formation of air-entraining vortices? |
| Are the suction strainers correctly sized? |
| Are there provisions to measure the pressure drop across the strainer as an indicator of 'clogging'? |
| Is the suction pressure adequate to provide enough NPSH? |
| If this pump has a suction booster pump, do the suction pressure readings of this pump look correct? |
| Will the discharge line stay full once the pump shuts down? |
| Is the pump primed? |
| Is the pump vented and are the vent valves open? Has pump been slowly turned over by hand to clear impeller passages? |
| Pumps handling hot liquid must be warmed-up before starting |
| Are there facilities available for measurement of pump flow and pump power consumption [amps] as an aid to future diagnostics? |
| Are there tapings available for measurement of suction and discharge pressure as an aid to future diagnostics? |
| Is the minimum flow bypass open? [Where provided] |
| Is the automatic re-cycle valve open? [Where provided] |
| Is the discharge line resistance sufficient for start up [*As a default aim for a valve opening of less than 20%*] |
| Is the discharge line empty? |
| Is the discharge line likely to get damaged during operation? |
| Could reverse flow take place in the system? |
| If the pump is of the type that uses an external hydraulic balancing line, are you sure that no obstructions can occur? |

Similarly, exposed lines may experience overcooling or even part freezing in winter.

Problems often occur when the liquid contains something unexpected. This may be a gas dissolved/evolved from the liquid, or it may be suspended solids. Gases have a quite immediate effect on performance whereas suspended solids take a little longer for their effects to be felt.

While it is tempting to dismiss this issue of flow assurance, the consequences are familiar to most pump service engineers.

## Is there actually some liquid to pump?

This might sound an absurd question. Surely no one would ever attempt to start-up a pump which has no liquid in it?

But ask any manufacturer how many times pumps are returned back to their service department, totally seized, due to lack of liquid. They will tell you that this is by no means uncommon. Why did the industry find it necessary to invent the term 'run dry' to describe pumps with internal seizure?

Paradoxically, some pumps are actually designed to have a 'dry run' capability (Table 8.2).

The matter of running dry is so critical it is always worth taking whatever steps are possible in order to positively prove that there is liquid available to pump. In terms of a reality check, there are a number of pre-start checks that can be made to help confirm that liquid is available.

## Flooded suction

If the pump is operating on a flooded suction, a quick check of liquid availability is possible during venting. Liquid eventually issuing from the

**Table 8.2** Dry run capability.

| Pump category | Normal dry run capability |
|---|---|
| Wet recirculating self priming | No restriction |
| Vacuum assisted self priming | No restriction |
| Overhung shrouded impeller: end suction | General purpose pumps [water] — Poor<br>Heavy duty pumps [chemical and Process] — Limited |
| Between bearings single stage | Rarely capable |
| Between bearings multistage | Very rarely capable |

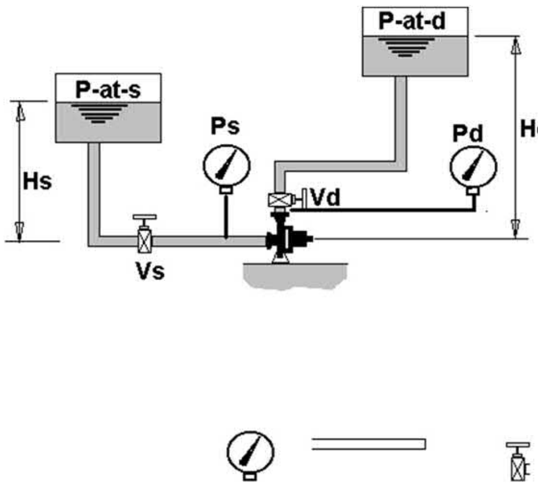



**Fig. 8.1** Schematic of pump with flooded suction showing typical gauge readings prior to start-up.

vent line at least proves that there is liquid there. This is *not enough* to prove that the liquid will adequately flow, once the pump is started. But that can be checked in other ways, as we shall see. The natural tendency in a flooded suction system is obviously to flow towards the pump, so this circumvents one potential start-up problem. It greatly helps priming of course. Fig. 8.1 depicts a flooded suction system. Where:

**HS** = *Static level between liquid level in suction source and suction pipe centre line at suction gauge [Ps] connection point.*

**HD** = *Static level between discharge level [submerged discharge] or discharge pipe exit [free discharge].*

**P-at-s, P-at-d** = *local gauge pressures acting on liquid surfaces. In many cases, the surface is at atmospheric pressure, hence this value is zero.*

**Ps, Pd** = *Suction and pressure gauge reading. NOTE this simplification assumes that the gauge is located at or very close to the pipe centre-line. If the gauge is located significantly above or below the pipe connection, then this distance must be factored in.*

**Vs, Vd** = *Suction and discharge valves.*

With the pump fully vented, the gauge readings should be in accordance with the following table (Table 8.3).

If there is no liquid in the pump the gauge will register zero. But be sure that the gauge is not just broken, or blocked off.

**Table 8.3** Flooded suction checklist.

| | Vd closed | | Vd open | |
|---|---|---|---|---|
| | **Vs open** | **Vs closed** | **Vs open** | **Vs closed** |
| Ps reading = | Hs + [P-at-s] | No | Not advised unless running | Hd + [P-at-d] |
| Pd reading = | Hs + [P-at-s] | No | Not advised unless running | Hd + [P-at-d] |

## Suction lift

If the pump operates on a suction 'lift' or some other negative suction condition, then it is much more difficult to be assured of liquid presence in the pump. In this case, one must make indirect checks (Fig. 8.2).

The natural tendency is for liquid to drain away from the pump, unless there is a foot valve/non-return valve in the suction line.

Study the suction pressure gauge [Ps]: if there is no liquid in the pump the gauge will register zero. But be sure that the gauge is not just broken, or blocked off (Table 8.4).

Even if a foot valve is fitted but is in poor condition, then the suction line may slowly de-prime due to 'passing' across the valve. If there is liquid in the discharge line, then it may be possible to backfill and prime the suction line by opening both valves carefully until the correct reading is shown on gauge Ps. However, the suction foot valve will continue to 'pass' so there must be no delay in starting the pump.

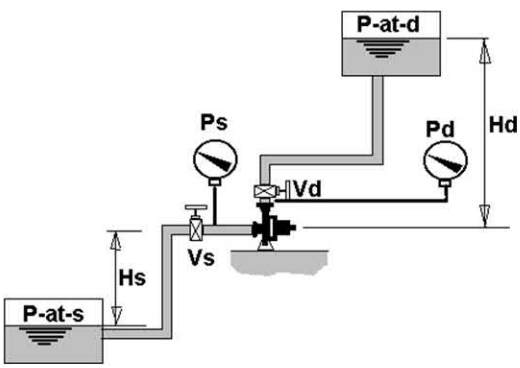



**Fig. 8.2** Schematic of pump with static suction lift and typical gauge readings prior to start up.

**Table 8.4** Suction lift checklist.

| | Vd closed | | Vd open | |
|---|---|---|---|---|
| | **Vs open** | **Vs closed** | **Vs open** | **Vs closed** |
| Ps reading = | -Hs + [P-at-s] | No | Not advised unless running | Hd + [P-at-d] |
| Pd reading = | -Hs + [P-at-s] | No | Not advised unless running | Hd + [P-at-d] |

## Is the liquid in a condition to flow freely?

The plant/pipe system designer will have assumed a certain liquid state in his calculations. But it may be naturally too viscous to flow without [say] some pre-warming, for example. It may also contain 'lumps' or waxy solids. Unusually, some liquids even have to be heated before they will flow sufficiently to be pumped. Perhaps the pumps handle a range of products, one of, which is exceptionally viscous. On the other hand the suction pipe may cool in winter sufficiently to inhibit flow, as mentioned earlier. Two important liquid categories exist:

- Those based on water [Aqueous]
- Those that are oil based products [Hydrocarbons]

Both these liquids lie at the extremes in term of behaviour and properties. Some significant differences are described below.

## Aqueous liquids

Clear, or mildly dirty aqueous solutions, generally flow freely. They rarely display severe viscosity effects. Nonetheless, a pseudo viscous effect does arise when such solutions are very near to freezing. The liquid can become 'slushy'. This has been known to cause isolated problems:

The author is aware of one installation where a sealless pump in a chiller circuit routinely [but inexplicably] burnt out its motor. After much diagnosis, it turned out that the chillers slowly overcooled the liquid, which turned to dense 'slush.' The pump would either vapour lock because of insufficient flow, or it would trip on high motor temperature. In either

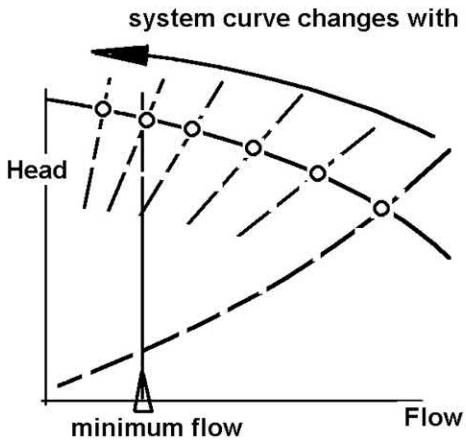



**Fig. 8.3** System curve for chiller gradually became steeper with time as 'slush' content increased.

case, the result was same — the pump was switched off prior to examination. By the time inspection could be authorised, the slushy liquid had melted — along with all the evidence (Fig. 8.3).

The problem was eventually diagnosed by concluding that low flow was the most likely way that the motor could overheat in such a system. Examination of the system characteristics showed it to be mainly frictional. Brainstorming ways in which the friction head could increase without intervention, correctly suggested the 'slushy' liquid cause.

Similarly, aqueous solutions heavily laden with solids can demonstrate pseudo viscous problems. Pumping paper stock or pulp is a case in point.

Aqueous solutions tend to exhibit freezing or boiling within a narrow range of temperatures. Their specific volumes [the ratio of liquid volume to associated vapour volume] tends to be high, which is of significance in their potential to cause cavitation damage.

Aqueous solutions appear with a wide range of acidity-alkalinity. This necessitates an equally wide range of materials of construction.

## Hydrocarbons

The properties of hydrocarbon products differ widely. Even the raw feedstock shows significant variation. At one extreme, the liquid might gush from the ground — helped by reservoir gas pressure. At the other, it has to softened by steam soaking before it can flow at all.

Because this original viscosity problem is widely recognised, it is most unlikely to have been overlooked in the system design and pump selection. Difficulties are most likely to arise if or when the feed characteristics change.

The writer is reminded of a problem experienced by a refinery in high Northern latitudes. The pump in question was a transfer pump exporting a hydrocarbon with a tendency to wax at low ambient temperatures. During the Northern winter, ambient temperatures got low enough for wax to deposit on the very long suction pipe walls-reducing its cross section, and also increasing the frictional resistance. The pumps could cavitate and restrict export capacity. In any event, the increased resistance of the discharge line also restricted performance. The pumps required a larger impeller for the winter conditions. This caused excess capacity in summer but was not a problem once acknowledged. The suction line was steam traced to discourage waxing.

Once refined into hydrocarbon products, the range of viscosity increases again. Since the viscosity of hydrocarbon product is often strongly temperature dependent, this can give a clue as to its propensity to flow. In terms of pump operation, an unexpected increase in viscosity may be damaging. Quite apart from the associated performance reduction, there will also be an increase in absorbed power. This power increase may overload the driver!!

Hydrocarbon products do not show the same narrow range of boiling points as aqueous products. This is because crude oil is composed of various 'fractions' — each with its own vapour pressure.

Neither does oil exhibit the same high specific volumes as aqueous based solutions. Hence hydrocarbons have much less potential for causing cavitation damage. This is particularly true as the hydrocarbon temperature is raised. While examples of cavitation damage to impellers operating on cold water are quite prolific, similar examples on hot oil service are comparatively rare.

Hydrocarbons do not exhibit the same levels of acidity-alkalinity as aqueous based solutions, and so materials of construction tend to be steel or steel based. However some hydrocarbons can be quite acidic [sour] and dictate special variants.

## Viscosity effect on pump performance

It might be useful to briefly describe the effects of viscosity on performance. Earlier, *it* was explained why there is a difference between the theoretical

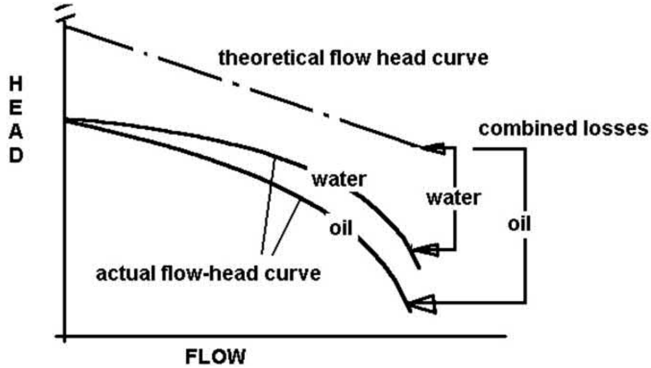



**Fig. 8.4** Viscosity effects on pump performance. Increasing viscosity produces higher internal losses and results in lower pump output.

performance of a pump and its lesser actual performance. This largely results from the effects of liquid viscosity. Viscosity causes frictional losses within the hydraulic passages of the machine (Fig. 8.4).

In all these discussions the tacit assumption is of water being the liquid. As we have discussed, other liquids – hydrocarbons in particular – can exhibit much higher viscosity, and hence much greater departures from theory. This results in the pump efficiency reducing when compared to water tests.

The most widely used method of accounting for hydrocarbon viscosity is described in *Reference* 14.[1]

# Piping issues

## Suction piping

Manufacturers will confirm that most pump operating problems relate to suction conditions. Sadly, many of these problems have trivial root causes which, with hindsight, were preventable. It would appear each new generation of engineer wants to rediscover the basics the hard way. Experience says not to take anything for granted in suction line design. It is always worthwhile taking an inspection walk along the suction line prior to a pump first being started, or re-started. Ideally take a colleague with you. The basic elements of good piping layout are outlined in Chapter 5. This section is concerned with the use and abuse of the line.

## Is the suction flow path complete?

It may not be easy to verify that a continuous column of liquid connects the pump suction to the liquid source. For example:

Although trivial and retrospectively obvious, these are all actual case studies. So why do they [frequently] occur? Let's look at some checks that can be made (Table 8.5).

Before that, note that if the pump operates on a suction lift, then the suction valve must be closed on start-up in order to prevent the pump de-priming – if no other facilities exist [such as foot-valves, non-return valves.] Immediately on start-up the suction valve must then be opened.

### *Are the suction strainers adequately sized?*

Best practices dictate that most pumps should have a means to prevent solid items from being ingested into the pump. Various devices exist to do this, depending upon the relative size of the items. There are three general categories in use:

#### *Screens*

Generally screens are restricted to use on 'Open' system [i.e. liquid systems that are open to the outside world and its associated flotsam and jetsam]. Screens have large spacing and are only intended for macro-filtration, such as large pieces of debris. The flow blockage created by the screen elements should be less than 25% of the free cross sectional area.

#### *Strainers*

Again are generally used on open systems, often in conjunction with screens. Whereas screens are intended to arrest large items that may clog or jam the pump strainers will usually pass items that the pump itself can ingest without too much difficulty (Fig. 8.5).

A commonly used rule is to make the free cross-sectional area of the strainer more than 4 times the area of the suction pipe. Even so it is still possible for strainers to cause significant suction line starvation. For example debris such as fine grass or river weeds will pass through a screen, but can choke a strainer.

Where practical, strainers can be temporarily cleared by back flushing the inactive discharge line back through the pump, though this is clearly not possible if the strainer is working in connection with a suction foot valve [or non-return valve]. Where the need for back flushing can be anticipated, a bypass line needs to be installed around the foot valve.

**Table 8.5** Basic, but common, suction line problems.

| **Common faults** |
|---|
| • The suction entry to source may not be clearly visible. For example the suction line may consist of a long flexible hose that disappears away into the distance. It is not self evident that the suction line is actually connected to the source.<br>• With a complex suction manifold arrangement it is possible that one critical isolation valve is not open when it should be. Best practice locks the suction valve open as part of the start-up pre-check.<br>• The source level may be below suction pipe entry. Just because the suction pipe can be seen entering a vessel, does not mean that the vessel is full, or full enough to permit pumping.<br>• The pump may operate on a suction lift and the foot valve is leaking: thus the liquid column drains away.<br>• The liquid is laden with dissolved gas that slowly comes out of solution, agglomerates, and leads to de-priming.<br>• Incorrectly fitted taper pieces that trap air, agglomerate it, and pass it to subject the pump to large 'slugs'. This may even result in the pump 'air-locking.' [Chapter 5] |

**Fig. 8.5** Typical suction strainer, based on perforated plate.

### *Filters*

Are mostly restricted to ‘closed’ systems with the intent of capturing small particles that are comparable to the wear ring clearances. These are typically hard particles that would otherwise damage the pump clearances. The size of these filters depends upon the application. During commissioning of most plant, filters of 200 μm would be typical in order to capture pipe weld slag, pipe scale and other debris that might damage the pump. This may be retained once commissioning is complete, or removed if confidence in the system hygiene is high. The commissioning of sealless pumps may require a little more care – perhaps 60 mesh.

The dilemma here is that strainers fine enough to protect the pump in every sense, would also be strong candidates for restrictions.

## Discharge piping

Unlike Positive displacement pumps, centrifugal pumps must have a finite hydraulic resistance to work against on start-up. Furthermore this load must continue to be applied, otherwise damage could occur.

Usually, the work done by a pump is mostly expended in the both the discharge line and its elements. Consideration of the pump/system interaction [Chapter 5] shows that with little or no system resistance a centrifugal pump will be immediately obliged to try and operate at its highest flow. This is not desirable. In the case of centrifugal pumps this will also be the condition of near maximum power consumption and will also demand the highest suction pressure in order to operate safely. [Chapter 9 shows this is not true of mixed and axial flow pump]. Review of the discharge line may therefore be useful.

## Is the discharge line resistance sufficient for start-up?

Centrifugal pumps absorb the highest power at the highest flow rates. The high flow situation also needs the highest suction pressure in order to operate. For these reasons, it is never desirable to start a centrifugal pump with little or no line resistance.

In practical terms the best condition for centrifugal pump start-up would bring it up to about 20% of its design flow when running at full speed.

Traditionally the discharge valve/control valve is used to create an artificial start-up load. Ideally this valve should be partly open before start-up — for two reasons:

- It imposes a low power demand on centrifugal pumps — as described above.
- Avoids the churning situation associated with a completely closed valve. While churning, most of the motor power goes into heating the liquid entrapped in the pump body. In some high energy multistage applications, perhaps a megawatt of power is being fed into each stage volume. Since this stage volume may be little bigger than a kettle, it does not take very long at all to completely turn to vapour. This is clearly undesirable. Smaller powers or higher trapped volumes take longer but merely prolong the vapourisation situation. The time to reach this condition can be calculated Appendix H.

But what is partly open? A Figure to start with is 15% opening. With gate type valves the flow is not at all proportional to valve opening. Fig. 8.6 shows data for a typical gate valve, which suggests that the valve should be around 20% open at start-up, certainly not much more.

There is a minor dilemma here that relates to the length of the discharge line.

For safe start-up, the requirement is for the discharge valve to tend towards being closed. On the other hand there may be the conflicting simultaneous requirement to have the pump fill the system. This demands

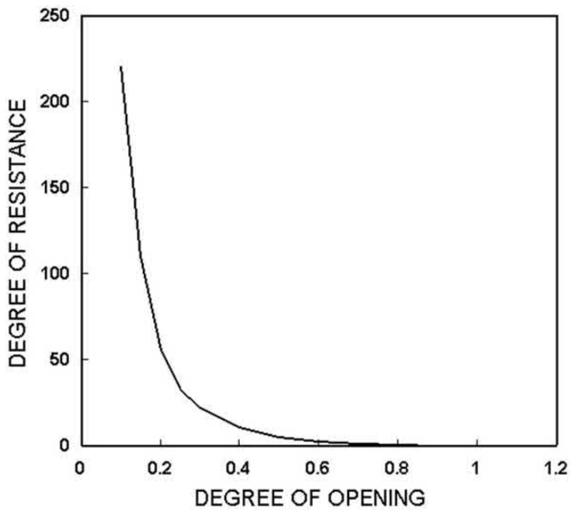



**Fig. 8.6** Gate valve resistance as a function of opening [1.0 = fully open].

that the valve errs towards a larger opening, in order to help avoid the pump operating at low flow for too long.

If the end of the line is in sight, then err towards a small discharge valve pre-opening.

If the end of the line is so far away as to not be easily visible, then one should err towards a larger valve pre-opening.

With some mixed flow and all axial flow pumps the picture is reversed and these machines *must* be started at a somewhat higher flow. Why? Because the power curve decreases with increased flow. [See Chapter 9]. With true axial the power curve falls dramatically and the driver may not be capable of starting the machine at significantly less than design flow.

## Is the discharge line empty?

If the discharge line is empty, then it must be filled before the pump is brought onto full load. Without any discharge resistance, the pump will want to operate at the extreme high flow. The perils of this have already been described (Fig. 8.7).

Traditionally the control valve or discharge valve is used to control the pump and system interaction to within the desired limits on the performance curve. In principle, the valve temporarily increases the frictional component of the total head to ensure interaction at least 20% of best efficiency flow. This should ensure satisfactory start-up. However,

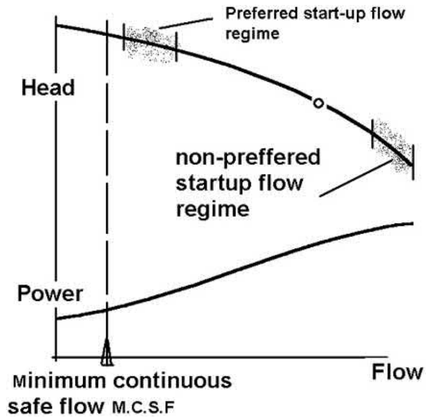



**Fig. 8.7** Preferred start up flow region- particularly for centrifugal pump types. Mixed and axial flow pumps may have different preferences, depending on their power curve shape.

pre-empting the next Chapter, what happens when this valve is opened in an **uncontrolled** manner?

It depends upon the ratio of static head to friction head [Eq. 5.2]. It may depend upon how fast the valve is opened as well. It also depends upon how much liquid was in the pipe to start with. Let's look at the three cases (Fig. 8.8).

### *High static head component*

In the first case, the static component is quite high. In this case opening the valve fully would not be an issue. The interaction point would just move from the start-up flow to the system design flow. Could it run beyond? Well yes, initially. If the line is empty, then the pump has to fill the line.

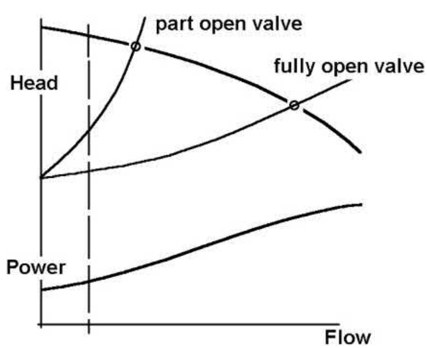



**Fig. 8.8** Discharge valve can control pump scheme maximum flow. This is also how some pump tests are accomplished.

Before it was fully opened, the valve provided all the resistance to ensure that the intersection point was in the desired zone. With the valve fully open and [nominally no resistance], the pump curve will intersect with the static head line (Fig. 8.9).

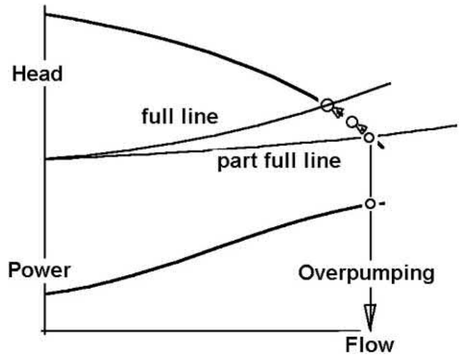



**Fig. 8.9** Run out flow [valve open] on system with high static head.

If the static head is quite high, then this intersection might not be at a too great a flow.

The over-pumping will probably not be too serious.

### *Moderate static head component*

If the static head is moderate or less, then the intersection may lead to gross over-pumping. In either case, the friction component will initially be nearly zero — because there is nearly no wetted area in the unfilled line (Fig. 8.10).

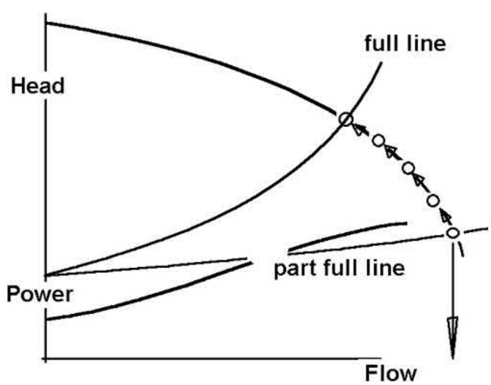



**Fig. 8.10** Starting on low static head systems with fully open valve will lead to over-pumping. The discharge valve-where fitted- can be part closed to help avoid this.

As the line fills the wetted area increase and so the does the frictional component. Not until the line is full will the combination of static and friction head ascend to the system design head. If the discharge line is short enough, it will fill quickly enough. If the line is long, then the [initial] fill process may take some time. This will enforce the pump to operate extensively at its end-of-curve. Judicious use of the discharge control valve can avoid this by creating enough resistance. This will prevent over-pumping, until the line is full.

### *Negligible static head component*

In this case the pump would always be most vulnerable to start-up operation at end-of-curve (Fig. 8.11).

There would be little static head to act as a 'safety net'. In practice, it often turns out that these types of systems are also long physically. Hence if no other steps were taken, such circumstances would oblige the pump to try to function at very high flows indeed. In most cases, the driver would not be sized for this high level of run-out power, so it would simply trip out. As before, judicious use of the discharge valve would help create enough resistance to safely fill the line without driver overload (Fig. 8.12).

An exception would occur if there were not enough NPSH available. In this event the pump curve will just cavitate down to the system curve at the flow where the NPSH available equals the NPSH required by the pump. Flow will stay more or less constant until the system curve ascends to the non-Cavitating part of the pump curve. After this flow decreases as the intersection point converges upon the system design flow.

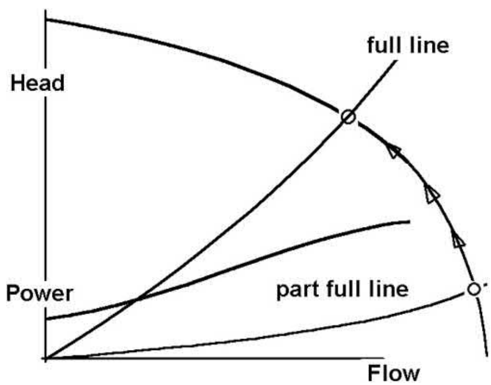



**Fig. 8.11** Starting on systems with negligible static head and fully open valve with lead to so much over-pumping that the driver will probably be incapable.

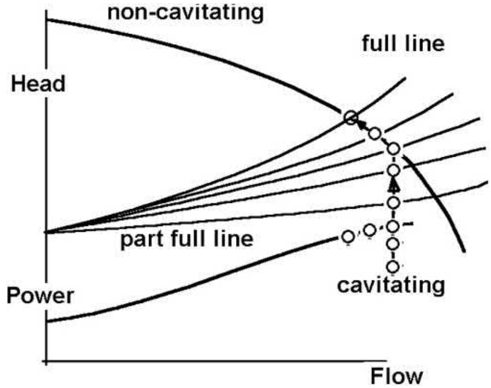



**Fig. 8.12** Pump cavitates down to the system curve, but continues to fill line until flow reduces to point where suction pressure is enough to prevent cavitation.

In all of these instances, the main issue is how fast the frictional component can ascend to the design value.[2] Where this problem can be foreseen; it is worth considering 'packing the line'. In this process, the line is pre-filled which establishes most if not all of the frictional component. One way of doing this is to valve-control the pumps at or near to their Minimum Continuous Safe Flow [MCSF]. The pump can then discharge out into the line in a controlled manner. This process is continued until the line is completely full. In principle, the pump can then be started and the discharge valve opened fully. The operator can be safe in the knowledge that the line is full enough to prevent any of the above over-pumping problems. Pumps driven by variable speed prime movers can use a variation on this theme.

Although it is covered elsewhere, the issue of water hammer should be mentioned in this context. Starting a pump up on a long empty discharge line creates the ideal conditions for surge and even water hammer to take place.

### *Will the line stay full once pump shuts-down?*

It is futile to spend time packing the line if there is a significant risk of it emptying. There may be a known risk of line severance, but one can take preventative measure against this.

Many pump systems are syphonic in the sense that both suction pipe entry and discharge pipe exit are submerged, but at different levels. In the vast majority of cases, the natural flow will drain back from the discharge line, through the pump, and out through the suction line.

Another opportunity for reverse flow arises when the pump discharge line is connected to another high pressure line.

Unless prevented, backflow has the potential to make the pump act as a turbine. If reverse rotation occurs, then more serious risks are presented, *see* Appendix G.

If the flow conditions were insufficient to cause reverse rotation of the pump set then the only effect would be to act as a resistance.

Best practices therefore dictate a non-return valve [NRV] somewhere in the combined system.

Locating the NRV on the suction line can help maintain pump prime. Again best practice locates it close to the pipe entry in order to retain liquid in the line of a pump operating on a suction lift. If in the discharge line, the NRV is located in the best position to maintain a 'packed line' condition (Figs 8.13 and 8.14).

If the natural flow is towards the pump, then the NRV is located as close as possible. It should be located on or near to the discharge flange. This affords some protection to water hammer 'shocks' in the discharge line, should they occur.[3] Locating it on the pump suction flange would just increase the suction pressure requirements, but would expose the pump casing and seal to the full effects of water hammer shocks.

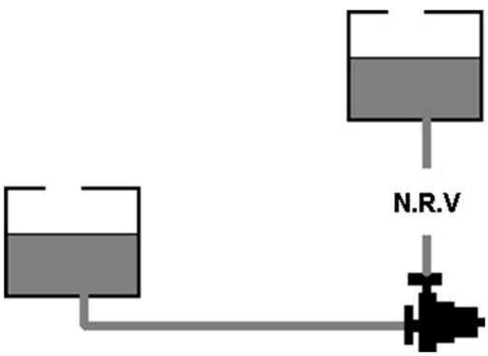



**Fig. 8.13** Positive discharge static head.

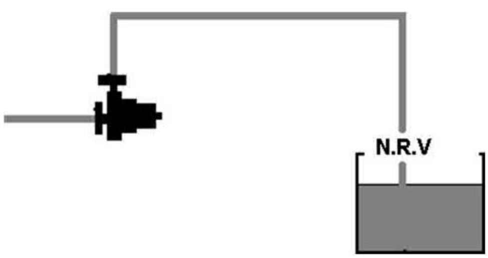



**Fig. 8.14** Negative discharge static head.

The most insidious circumstances are when the liquid in the line just drains away. If the line is constructed of temporarily connected elements, then leakage at the joint connections must be anticipated.

### *Is the discharge line prone to be severed in operation?*

In some applications, this can be a real danger. Working examples of this risk are mobile fire pumps, or portable irrigation pumps. In either case, the discharge line is most often a hose, rather than a pipe. Since the hose may be hastily laid out, it is vulnerable to accidental damage by [say] emergency vehicles. There is thus a finite risk that these pumps will be subject to over pumping if the frictional component of head is suddenly lost. If this is foreseen, the pump should be specified with a non-overloading power curve in combination with a suitable sized driver (Fig. 8.15).

With this provision the only other liability is to cavitation, which is not likely to induce damage over short periods of exposure.

Without this precaution, driver overload and damage may occur.

## Could reverse flow occur in the system?

Earlier attention was focused upon the possibility of flow siphoning or reversing in the system while the pump was not energised. Nevertheless, could reverse flow occur by any other means?

The most common circumstances are when two pumps operate in parallel into a common manifold header. Earlier it was described as good

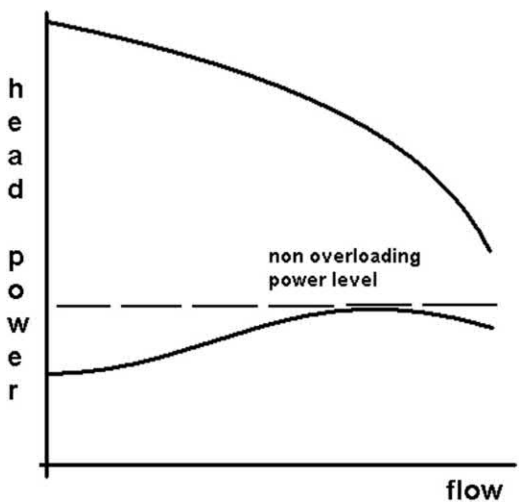



**Fig. 8.15** Non-Overloading power curve, in conjunction with adequate driver, guards against unscheduled loss of system resistance such as discharge line breakage.

practice to install non-return valves on each pump discharge. Apart from all the other good reasons for this, an NRV permits one pump to operate solo.

The NRV prevents flow from the active pump by-passing straight back to the source, instead of out into the discharge line. However, these valves are not infallible, and if they get partially hung up, then bypassing will occur. Some flow will go out to the system while some short circuits back to source. The amount of liquid bypassing will, depend of course, upon how wide open the NRV is when it fails.

Experience says that when NRV fail, they most often fail by not completely closing. They remain slightly open. While this represents a small loss of pumping capacity, and may be less than obvious[4] it is seldom dangerous.

Less often, the NRV fails in the fully open position. This happens when the pump is shut down after long periods of high capacity pumping. The NRV just does not close at all. Usually this is because the pumped product has a tendency to form deposits that inhibit valve motion. This sort of case represents a very large loss of pumping capacity due to the large bypass. It is generally obvious. However, of more significance is the fact that enough flow now reverses from the active pump for it potentially rotate and become a turbine?

As described in Chapter 14, reverse rotation speed at full flow can often be in excess of the design forward speed. This leads to the possibility of the line finally draining back and leaving an empty pump set turning at a substantial speed, thanks to the energy stored in the motor rotor. The results are not too hard to imagine! This situation has even been known to damage electric motor rotors.

One should look very carefully to make sure that, if the above reverse rotation events take place; there is no risk of an attempted start!

## Priming and Venting

Centrifugal impellers need to be fully 'primed' before they will begin to pump. As noted, impellers operate by displacing liquid, and creating a low-pressure region in the inlet [often called the 'eye']. This is then replaced by fresh liquid that is pushed into the low pressure region. If there is no liquid in the impeller, it cannot start to function-hence the need for priming.

Ideally the entire casing should be filled with liquid – including the seal chamber.

As a minimum, the impeller eye should be submerged and the seal chamber vented off prior to start-up. In such a condition the pump will probably start, but not definitely. In most cases, the liquid level needs to be a good deal higher than the impeller eye in order to ensure reliable starting.

Another criterion says that the liquid must at least reach up to the cutwater in the casing, even if the discharge nozzle is vertical/tangential upwards.

Some [but not all] multistage pumps are designed in such a way that every stage can be fully vented. In practice it is often sufficient to fully vent only the first one or two stages, provided that the other stages are part primed. Once the pump has started, the following stages are quickly primed internally. However it is good practice to fully vent all of the stages before start-up if that is possible. This helps establish the *Lomakin* effect that many pumps of this type rely upon. Without entrained liquid the pump may well seize.

It is so important not to overlook the priming issue that a special branch of the centrifugal pump family exists called 'self-priming pumps'.

## Why the need to prime?

Mainstream centrifugal pump designs, can rarely tolerate more than about 5% by volume of air/gas at their inlet, though special designs can far exceed this Figure. Beyond this level, pump performance decays very seriously.

The pumped fluid may already have dissolved gas in it of course. The reduced pressure in the suction column of a pump on a static lift will encourage the gas to come out of solution. Both performance decay and de-priming become a risk. Losing the suction column is seldom more than an irritation with single stage overhung impeller pumps, though the mechanical seals may be unhappy. Between-bearings pumps and multistage pumps can rarely tolerate this situation without seizure threatening. Pumps operating with a suction lift are the most susceptible to gas in the liquid. Not only do they run the risk of separation of the liquid in the suction line, but also the risk of depriming because they cannot establish the necessary vortex in the casing to promote pumping. For these reasons, static suction lifts of more than five metres need to be viewed very carefully.

Ironically, introducing air to the suction pipe is a very crude but effective way of addressing some pump cavitation issues!

## Priming methods

A positive displacement pump, when running can eventually prime itself. However, this is not true of rotodynamic pumps.

To operate on a suction lift, the rotodynamic pumps must be previously 'primed'. They need to be primed in some positive way *before* start-up. Most priming methods try to create the illusion of a flooded suction for the pump. We already know that if this illusion is successful, the pump can start. Such methods fall into two categories-static systems and dynamic systems.

## Does the pump operate on a suction lift?

The concept of suction lift needs a mention. First, visualise an installation where the liquid level is above the pump centreline. Intuition says that even without the inlet pressure gradient created by the rotating impeller, liquid will naturally tend to flow towards the pump. This state is referred to as a 'flooded' suction (Fig. 8.16).

Once the pump is started, the aforementioned gradient further encourages flow. So a flooded suction tends to prime the pump automatically. This is one benefit of vertically shafted pumps-either in dry or wet pit configuration.

Although a flooded suction more or less guarantees priming, it does not also ensure that the pump will then either start or run satisfactorily. There are some, admittedly remote, circumstances where even though the pump is 'flooded', this is still not enough to prevent cavitation within the machine.

At first sight, it is not at all obvious that a pump can and will operate when the liquid level is below the pump centreline. Fluid mechanics would say that in this state, the pressure in the suction line must be below atmospheric pressure. But provided certain conditions are met then the pump may happily operate under these conditions, once it is primed. The pump is said to be operating on a 'suction lift'. The popular perception is that the


**Fig. 8.16** Primed from suction.

pump draws or 'sucks' the liquid into its self. This explains why the pump inlet is often referred to as the pump suction connection.

In fact most liquids have very little tensile strength and the liquid is not so much pulled by the pump as pushed in towards it by the pressure acting on the surrounding liquid surface. The negative pressure created by the rotating impeller vanes increases this differential pressure.

## Prime from discharge

Once a pump scheme has been commissioned and has been in use, it is often possible to prime from the discharge line, if and only if there is a positive static discharge static head. [I.e. not downhill pumping.] (Fig. 8.17).

If the pump operates on a suction lift, then it may be possible to prime from the discharge, provided this does not entail emptying the discharge line. [The perils of starting up on an empty line have already been discussed].

For example, the suction valve must be closed on start-up in order to prevent the pump depriming – if no other facilities exist [such as foot-valves, non-return valves]. Immediately on start-up the suction valve must then be opened, followed by further opening of the discharge valve.

## Static systems-interceptor tanks

In this concept, a tank is placed between the pump and the liquid. For best results, the tank should be very close, the closer the better. The volume of this tank would typically be four times that of the suction pipe between the tank and the liquid surface. The tank must be closed and free from air leaks. Initially this tank must be filled manually (Figs 8.18–8.20).

In start-up mode, the pump operates as if it has a flooded suction. As it draws liquid from the interceptor tank, the level inside the tank falls creating a

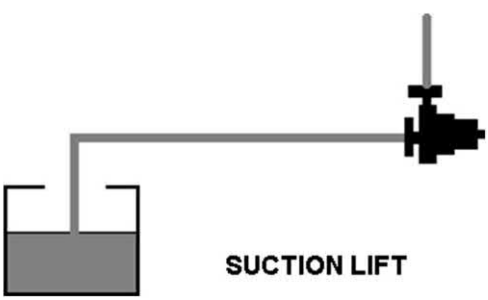



**Fig. 8.17** Not primed from suction.

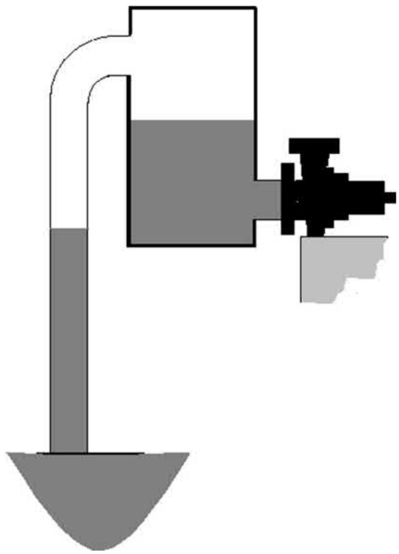


**Fig. 8.18** Start of the priming sequence. Suction leg is empty.

partial vacuum inside itself. This vacuum causes the liquid to be drawn up the suction line and cascade into the tank itself. Some of the air that was in the pipe becomes entrained and mixes with the liquid. For a time the pump is exposed to an air/liquid mixture. With time that mixture dilutes. Once that has cleared and there is a solid unbroken column of liquid between the liquid

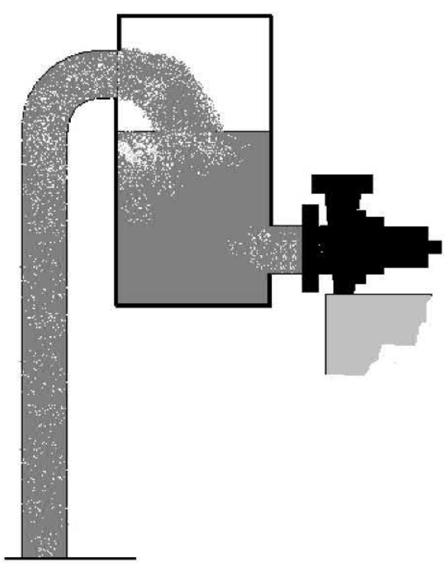


**Fig. 8.19** Part way through the prime cycle. Suction leg is part full.

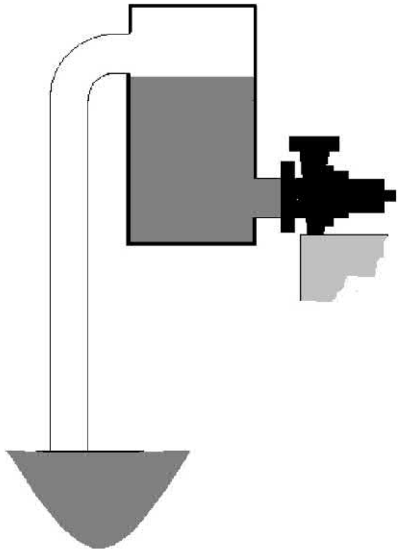


**Fig. 8.20** Approaching end of prime cycle. Suction leg fully primed.

surface and the impeller inlet, the pump should operate normally with the pressure on the liquid surface pushing the liquid into the pump. This assumes that the pump NPSHA needs have been met of course.

On shut down the liquid in the suction leg may drain back to the sump, leaving the interceptor tank largely full and ready for the next start cycle. However, having liquid flowing back through the pump may be undesirable so it is better practice to install a foot valve. This will largely retain the liquid in the suction leg, and hence reduce the priming cycle time. It also prevents backflow problems occurring.

As the pump empties the tank on start-up in the above example, a vacuum is created. This helps draw liquid up the suction leg and eventually into the tank.

In another embodiment of the interceptor tank concept, the tank is integrated with the pump body. Again this tank is manually filled for the first start. On start-up, the pump does not directly discharge the liquid. Instead it re-circulates the trapped liquid within itself, via strategically positioned ports in its casing. As it does this, air in the suction line becomes entrained and is passed out into the discharge leg. In this respect, the trapped liquid becomes a transport medium and air in the suction line gradually becomes evacuated.

As the air is evacuated, the liquid is gradually drawn up and into the casing. Once a solid column of liquid is achieved, the pump picks up and operates normally. This process takes place quite slowly and may take a several minutes to complete. Compare this with the positive vacuum assisted approach mentioned below, where priming might be accomplished in a few seconds.

## Dynamic systems

Another method generates the vacuum mechanically via an auxiliary air pump. Vacuum Assisted Self Primers [VASP] 'fools' the main pump impeller into thinking that is now operating on a flooded suction. This helps promote flows necessary to establish the casing vortex. Once a solid column of liquid is established between surface and impeller, the vacuum pump is no longer needed and can be stopped. However, in some very difficult applications the liquid levels can vary widely and rapidly. In the extreme, the suction pipe extremity may become exposed allowing an air/water mixture to be drawn into the pump. Such pumps can operate on 'snore', a term which adequately describes the noise they make under such conditions. In this case, the vacuum pump runs continuously, being protected from water ingestion by some form of float valve.

These really are specialist machines which are designed such that the operator is isolated from any sort of priming problem. Troubleshooting a priming problem with such machines is beyond the scope of this book.

Another approach is to use jet pump, or eductor pumps connected directly onto the casing (Fig. 8.21).

By definition, vertical shaft pumps are automatically primed, which is one of their benefits. However there is one configuration of vertical where this is not always true. Some are supplied and installed with long suction nozzles: so-called 'tail pipes'.[5]

The logic here is that the pump will be submerged and hence primed at high sump level [2].

As the level drops, the pump moves into a suction lift condition [1]. The pump will stop on low level and, in most cases will then de-prime. In the next part of the cycle, the sump re-fills and reprimes the pump. If attempts are made to re-start the pump *before* the impeller is re-submerged, it may not have yet achieved a primed state. It will probably remain air locked, until re-submerged.

This approach avoids the cost of making the pump long enough to be submerged, even at level [1].

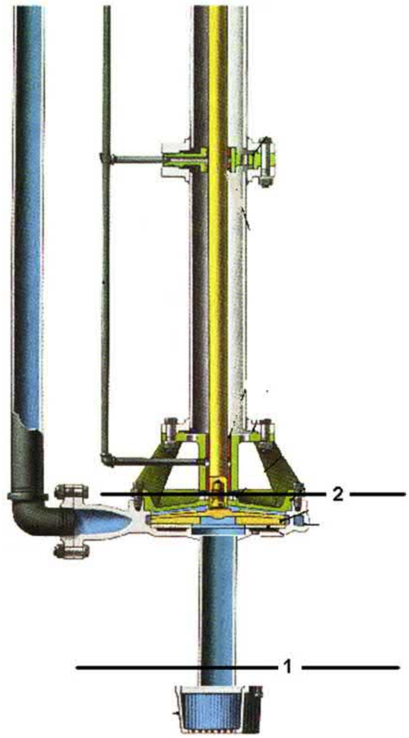



**Fig. 8.21** Vertical pump with tail-pipe may de-prime on low level shut down.

## Is the pump casing vented?

You frequently cannot vent a running pump, because any air gets forced to the centre of the impeller driven vortex, where it stays trapped and does not escape. Even pumps that operate on a flooded suction might not prime if they are not also vented. Conventional wisdom says that an impeller will always prime if the liquid level in the casing is a little above the top of the impeller eye. The logic here is that such a conditions will allow the pump to generate a complete casing vortex without drawing in any air. In practice the liquid level needs to be quite some way above the eye to be sure of priming.

Certain pump layouts are more useful than others in venting respects. Single casing pumps with top-top nozzles hold out the best promise for self-venting. However, the shape of the discharge volute is significant.

Pumps with a tangential discharge, might still trap some air in the volume above the casing cut-water. Automatic venting is by no means certain with this configuration, as tests have shown (Fig. 8.22).

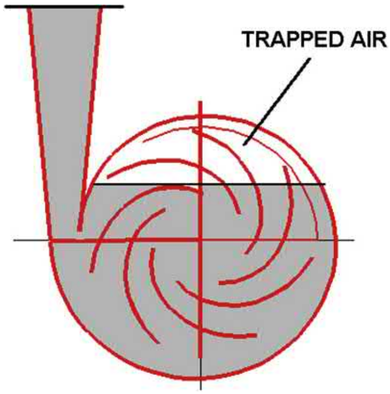



**Fig. 8.22** Tangential discharge may trap air in volute, even though the impeller eye is covered.

Pumps with centre-line discharge nozzle avoid this problem of course, since they allow hardly any amounts of air to concentrate in the top of the casing. The natural flow of entrapped air is up the discharge pipe and out of harm's way (Fig. 8.23).

The risk of air entrapment with a tangential discharge nozzle can be somewhat eliminated by providing either a slot in the cut-water or, more acceptable, a hole drilled across the cut-water (Fig. 8.24).

Either approach permits more of the entrapped air to be vented off and improves the prospects of successful priming.

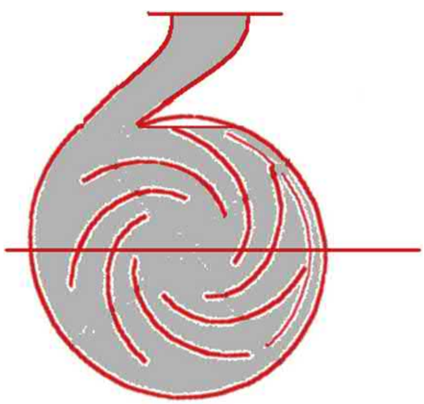


**Fig. 8.23** Centre line discharge eliminates any significant volumes of trapped air in the casing.

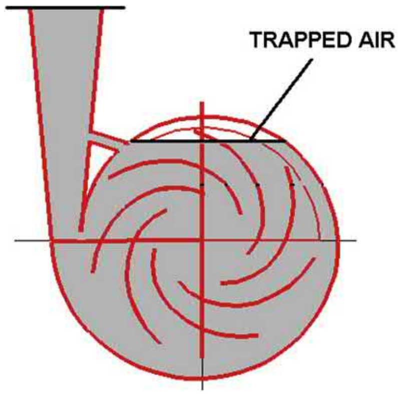



**Fig. 8.24** Drilled hole across the cutwater can significantly reduce the trapped air volume in a tangential discharge casing.

## Pre start checks – mechanical

The necessary mechanical checks are, in the main, more intuitive than the hydraulic checks. They generally reflect the good engineering practise that comes from training. As mentioned earlier, the hydraulic checks tend to relate to the system, whereas the mechanical checks relate to the pump itself (Table 8.6).

## Shaft alignment

Pumps require some other machine to drive them. Most often this is an electric motor, or a reciprocating engine. Other common drivers are steam turbines. Whatever the driver, pumps are classified as close coupled or long coupled.

*Long coupled* means the driven machine [pump] is an element independent from the driver. This permits the mass manufacture of each by specialists. They are then brought together and 'packaged' into a pump set. Both machine elements are mounted on a 'bedplate' that holds them in the correct relative position. Some form of flexible coupling usually connects the rotating elements of the two machines, though in this context the term 'flexible' is relative! (Fig. 8.25).

**Table 8.6** Mechanical checklist.

| Item |
|---|
| Is foundation block sufficiently massive? |
| Is the bedplate pedestal cooling turned on -where fitted. |
| Is the warm-up system [where installed] functioning correctly? |
| Does the alignment change when the pump and pipes are filled with liquid? |
| Does the alignment change significantly after the pump has warmed to its operating temperature. |
| Was the flange alignment witnessed before bolt-up? see |
| When cold, can the rotor be easily turned with [relative] ease, and without metallic sounds? |
| 'Bump' start motor to check rotation is correct |
| If variable speed drive ensure minimum speed is above first critical of multistage pump-typically more than 1700 rpm |
| LOMAKIN effect |
| Check for any mechanical 'looseness' |
| Check oil ring [where fitted] is located centrally and has not become dislodged |
| Check static oil level or constant level oiler, where fitted. |
| If grease lubricated, make sure not over packed. |
| Where fitted, is grease visible from the relief valve? Does the grease look old or new? |
| Can you be sure that there is no chance of the lubricating oil being contaminated with atmospheric moisture? |
| If oil bath lubrication, does the static oil level half cover the rolling elements? |
| If a constant level oiler is fitted, is the pump in a fixed installation |
| Check cooling water to bearings-where provided |
| If pump is soft packed, check that gland bolts are only very slightly tight |
| If soft packed pump is on a suction lift, ensure packing will be flooded with liquid during start-up and operation |
| Is the coupling guard secure |
| Does the coupling guard allow free air movement? but discourage windage |

Can it be verified that the seal chamber is vented?
Pumps handling hot liquids must be warmed-up before starting
In the case of dual seals, can the barrier liquid pressure be monitored? [As an aid to diagnosis of a failed primary seal]
In the case of dual seals, can the barrier liquid temperature [in and out] be monitored? [As an aid to diagnosis of a failed primary seal]
In the case of dual seals, can it be proven that the barrier liquid pressure exceeds the suction pressure BEFORE start up?
If on a hot oil service, can it be proven that there is no water in the seal as a result of [say] a water hydrotest?
Check casing is not frozen
Verify oil mist venting from bearings of oil mist system
Ensure pump is not shimmed, only driver.

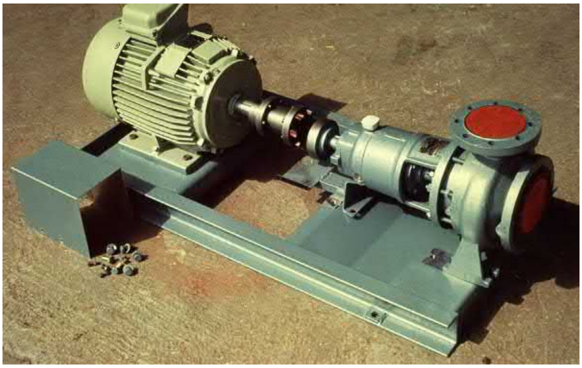


**Fig. 8.25** Long Coupled pumps showing flexible coupling with guard removed.

*Close coupled* means that the pump and driver are closely integrated such as to eliminate the need for any flexible coupling. The impeller is either mounted on the motor shaft, or on an appendage to it (Fig. 8.26).

The pump and driver frames are correctly connected and no bedplate is required. In this latter case, the rotating elements of each machine are automatically aligned by virtue of the machined construction and assembly. No adjustment is required or desirable. In the former case, there is a need to manually align the machines on site. [They will have been aligned by the pump manufacture before shipment, but some slight re-alignment will be necessary on site. This is because the alignment will go out of adjustment during transit due to shocks, or gradual relaxation of locked in welding stresses.

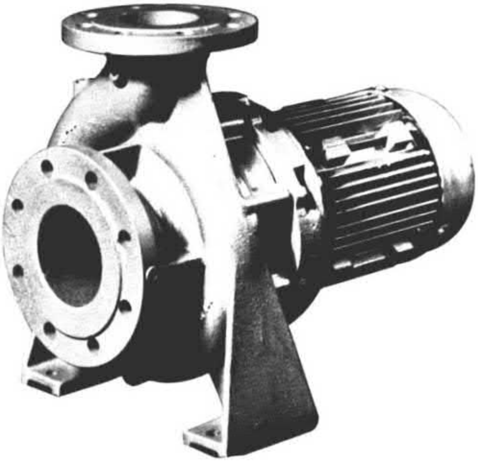


**Fig. 8.26** Close coupled pump and motor.

It may seem an irony that many flexible couplings claim to operate with relatively large misalignments, yet their installation guides insist on near perfect alignment set up!

A poorly aligned machine will often display unusually high vibration in the short term and in the long term, accelerated wear of seals and bearings].

Misalignment appears on FFT analysers as a strong contribution at twice rotating frequency.

With long coupled pumps, the alignment should be checked after the pump has run for several hours, and bearing temperatures have stabilised.

The shaft ends should be true to better than 0.002″ TIR. This should be checked when the pump is empty and then rechecked after the pipes have been bolted up and filled with liquid. If the pipes have been set up in accordance with the advice of Chapter 5, then any substantial departures can be assigned to excessive pipe loads. This needs correction obviously.

## Lubrication

Most centrifugal pumps on the market today employ either rolling element, or journal bearings.

Rolling element bearings [sometimes called anti-friction bearings] are without doubt the most popular choice nowadays. They seem to offer much simpler installation and maintenance when compared with the hitherto popular journal bearings. In addition, they are very readily available as a proprietary item in many parts of the world.

Some bearings rely totally upon presence of a dedicated lubricant in order to function. Others can use little or no dedicated lubricant — while others can actually use the pumped product as a lubricant. For the most part, pumps use bearings that rely upon dedicated lubricant, so we will deal with this in the most detail.

Most rolling element pump bearings are equipped with either grease lubrication or oil bath-oil splash systems. Some specialised pumps are furnished with forced lube oil systems and their troubleshooting is beyond the scope of this current volume.

Looking first at rolling element bearings:

## Rolling element

### Rolling element bearings and grease lubrication

Traditionally, grease has not been associated with heavy-duty pump applications. Issues such as short re-greasing periods and the risk of over-greasing have been cited as a reason. Paradoxically, the electric motor driving the heavy-duty pump was very often grease lubricated. In the

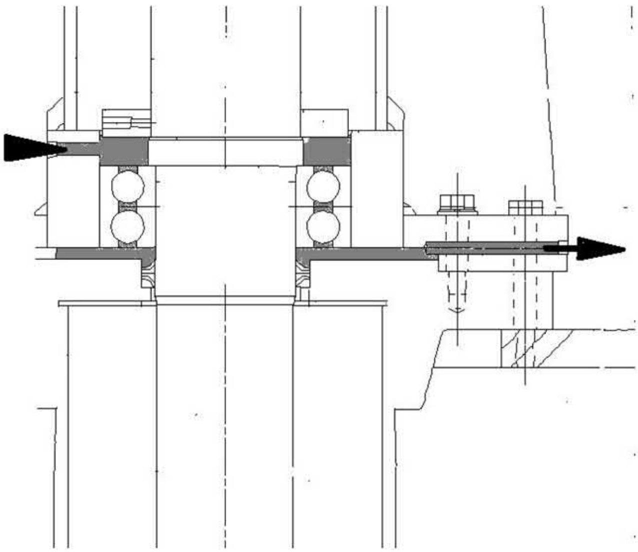


**Fig. 8.27** Simple grease relief arrangements help to prevent over-packing on vertical shaft pumps.

normal sense, grease does not flow at all, so it does not help to keep the bearing cool, as does oil. This places a constraint upon its use. However, it is easily contained in and around rolling element bearings and this makes it very popular (Fig. 8.27).

Nowadays grease is more widely accepted at least in light and medium duty service. The main concern is in over-packing the bearing with grease. Arguably this is a less precise science than oil levels which are very definite and unambiguous for the most part. Grease should occupy about 30% of the housing volume, not more.

Over-packing leads to churning, heating, and grease breakdown. This can largely be avoided by the use of good grease relief facilities in the bearing housing itself (Fig. 8.28).

Such good designs permit the old grease to be easily displaced in one direction, whilst excluding any air entrainment. The bearing runs on a thin film of oil which dries out with time and temperature. If this is replaced in time, the bearings will run as expected.

Clean grease emanating from the relief port is a good sign. Absence of grease or evidence of hard waxy grease is not encouraging and is worth investigating before proceeding.

Good designs include arrangements to fling out the old grease.

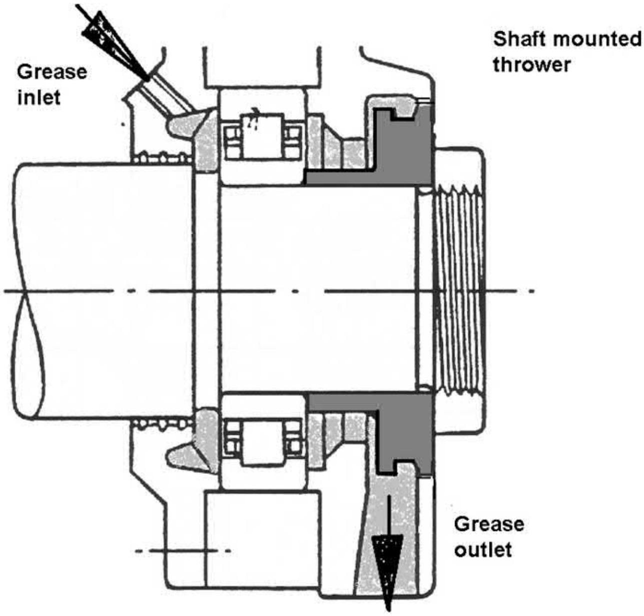



**Fig. 8.28** horizontal grease relief arrangement-equivalent to Fig. 8.27.

The issue of re-greasing period is a valid concern, but a number of innovative solutions exist. Both detract from the apparent simplicity of grease lube.

### *Static feed*

The first method uses internal pressurised grease cans to provide a steady supply of fresh grease to the bearings. Gas evolution or generation inside the can gives a more or less consistent feed pressure with some form of visual indicator when the can is empty. The cans are screwed on to the machine bearing – replacing the manual grease filler cap. The spring loaded, or screw cap type achieve much the same thing manually. Reiterating, the flow of clean grease from the relief hole is the best assurance that the device is working.

### *Mechanical supply*

The second method uses a positively driven displacement pump to discharge very small quantities of grease out into a grease feed line and thence to the bearing housing. An advantage of this arrangement is that the grease pump can be centralised so as to feed several machines simultaneously. This simplifies maintenance enormously as well as improving the chances that a machine will not be missed.

One advantage of grease is that it can form a tenacious good barrier to entry of foreign bodies. If grease is constantly bleeding out from the bearing housing then it is difficult for unwanted debris to get in. Perhaps surprisingly, grease bearings can often be found in very difficult environments, for this reason. Again this strength relies totally upon the grease being regularly replenished. Complex sealing labyrinths prove unnecessary.

A derivative of this concept is the so-called 'sealed for life' bearing. This type is regularly used on smaller or light duty pumps with great success. It has also been used on medium duty and larger machine that operate on intermittent service such as Fire Protection pumps feeding sprinkler systems and nozzles. The thrusts and particularly the speeds, associated with larger bearings, effectively rule them out of contention.

### *Grease choice*

Grease is actually oil with a thickener added. These are commonly Sodium, Lithium, or Calcium soaps (Table 8.7).

Grease should be replenished regularly in order to maintain all its properties. A lot can be learned from the appearance of removed grease.

If it is of uniform appearance, with a tendency to slump slightly, then it is not likely to be over packed.

If it slumps very readily, and appears to have little 'body', it has probably been over packed and or aerated.

If on the other hand it looks dark/cracked and acts like a solid, then the oil base has evaporated or the grease has been overheated and oxidised. Grease that looks like this should be immediately discarded and replaced.

## Rolling element bearings & oil lubrication

Oil lubrication systems will either partly immerse the rotating bearing in oil, will feed a stream of oil, or will attempt to create splashed or atomised oil conditions in and around a non-immersed bearing. This atomisation can be imperfectly accomplished inside the bearing housing itself, or more perfectly in an external, possibly centralised, purpose made system.

## Oil bath

Oil level checks are a formality, along with viscosity. But beware. Experience with one 'end-suction' process pump demonstrated that with no oil

**Table 8.7** Summary of common grease properties.

| Thickener | Water resistance | Corrosion shielding | Upper temperature |
|---|---|---|---|
| Sodium | Bad | Good | 90–100 °C |
| Lithium | Good | Good | 90–130 °C |
| Calcium | Very good | Good | 50–60 °C |

at all in the bearing housing, the pumps was able to run for nearly a day before collapse. The film of preservative grease used when the bearings left the factory was sufficient to lubricate for this period. In other words just because the pump runs for a few hours does not mean that the pump bearings have been correctly lubricated! (Fig. 8.29).

The normal criterion for oil bath bearings is that the static oil level half covers the rolling elements. In the absence of any other indication, this is a good basis. Sadly pumps are often equipped with oil level nameplates that are not very durable, confusing, or missing. Many get over-painted on site, or become illegible, but this rule is still quite general. It is well worth establishing, with some confidence, what the oil level really is, using the half flooded ball criteria as a guide.

Oil bath bearings can generate quite a lot of heat simply due to their churning effect, especially at 2-pole driver speed. For example, a 7313 [double angular contact] bearing running at 3560 rpm can generate nearly as much heat as a small electric kettle!

This can be cured to a large extent by provision of shaft driven fans. The axial flow type is better than the radial flow type. The radial flow fans generate a very strong vortex in their vicinity-so strong that they can actually extract oil vapour from the bearing housing. Not only is this messy,

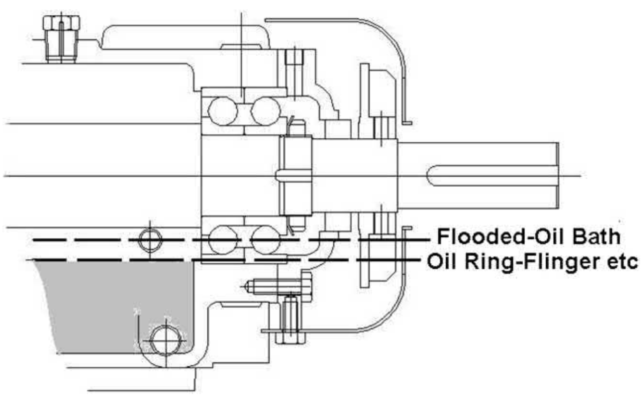



**Fig. 8.29** Default oil levels-oil bath systems.

but it will accelerate oil usage-even when close fitting/running bearing isolators are fitted. Certain couplings can, in conjunction with cylindrical coupling guards, also generate strong forced vortices — with similar extractive effects. Axial flow fans have a weaker vortex and so are less susceptible to fume extraction.

### *Oil splash supply*

Many pumps use a splash system instead of an oil bath. This can take the form of oil rings, oil thrower disks, slingers, flingers, pins, and other splash devices. These approaches may generate less heat. Aspects of oil ring performance are discussed in later. An aim of all these devices is distribute oil onto as much of the bearing housing inner surface as possible. This helps cool the oil and in the process cool the bearing. Direct cooling of the bearing itself, rather than the oil, is strongly discouraged. With this outmoded concept, there is a risk of overcooling the bearing and losing its essential internal clearances. Rapid and destructive runaway temperature rise results.

An enemy of any oil lubricated rolling element bearing is moisture. Moisture in the oil can significantly reduce its life. Volume levels as low as 0.01—0.02% can half the bearing life, a fact that is only now being widely appreciated. Filling from unsealed supply tanks is clearly not preferred! In terms of life cycle costing, the expense of hermetic bearing housing seals is quickly justified.

### *Constant level oilers*

See Fig. 8.32.

### *Oil mist lube systems*

Splash systems basically try to atomise the oil into an oil mist. An improvement is to construct systems that are dedicated to the formation of fine oil mists in a controlled fashion. This is then introduced to the bearings.

## Journal bearings

## Journal bearings and grease lubrication

Grease lubricated journal bearings are largely restricted to vertical shaft pumps (Fig. 8.30).

This is probably because grease is more likely to ‘stay in place’ than oil, when applied to a vertical journal.

**Fig. 8.30** Vertical sump pump with continuously lubricated grease journal line shaft bearings.

Furthermore, there is rarely any significant pressure difference across the journal so the grease has only a slim chance of 'washing out'. Grease lube tends to be used in a secondary role to exclude debris from the journal bearing. This trades on its almost solid appearance of course. Grease is seldom, if ever employed in horizontal shaft pumps.

So long as the grease is exuding from the bearing clearance annulus, the debris is excluded. By supplying the bearings from a constant feed system, this protection seems to be effective. Older pumps relied upon occasional hand greasing and this format must be considered suspect.

However, some sump pump designs encourage product flow across the lower support bearing as sump liquid levels rise and fall. Even though the bearing is grease blanketed, the risk of transporting debris into the clearance is very high. An alternative approach effectively isolated the bearing from this undesirable flushing effect and so is to be preferred. Existing designs can often be corrected quite easily (Fig. 8.31).

## Journal bearings and oil lubrication

Initially, oil lubricated journals were the most prolific bearing arrangement found in horizontal shaft pumps but their popularity has declined as rolling elements have become more reliable.

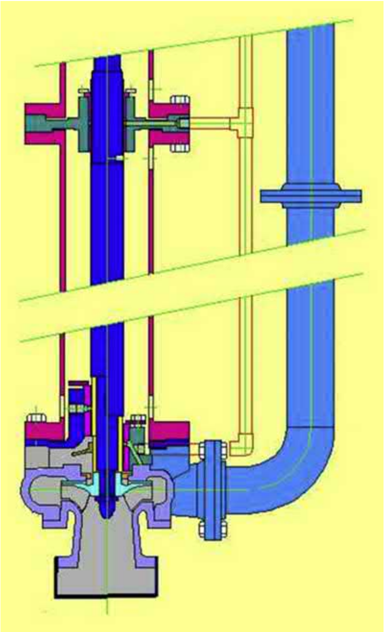


**Fig. 8.31** To the left of the centre-line is shown an improved version of the older sump pump lower bearing. This design avoids any significant pressure differential across the lower journal and helps to reduces the contaminant flow rate across the bearing.

One problem was the requirement to ensure an 80% contact areas between the shaft and bearing surface-as disclosed by the 'engineers blue' test. Often this dictated that the bearing surface had to be 'scraped' by hand-a skill which is ever less available.

This problem was exacerbated by older bearing designs, which, by modern standards, were rather long. Today length-to-diameter ratios of 1:1 are common with non-pressure feed designs, whereas older designs may have exceeded 1.5:1, even 2:1 in some cases. These long bearings inevitable ran rather hot because of the large area of oil being sheared. On the other hand, the loading was so low, that they could even tolerate some invasion of water into the oil.

Journals are rarely if ever immersed in an oil bath, in horizontal shaft applications. Instead, a device is employed to transfer the oil from the sump and up onto the journal itself. This may take the form of an oil ring or an oil disk. The oil ring is currently the most popular device.

### *Oil rings*

The older longer bearing designers usually felt obliged to install two oil ring lubricators whereas in the more modern shorter designs, one ring suffices. The detail design of the ring is important since it can exert a great influence upon the quantity of oil transported to the bearings. The limitations of application should also be understood, and this can be understood best by looking at some test data.

### *Constant level oilers*
### *Principle of operation*

A reservoir bottle is adjusted so that its feed point coincides with the desired bearing housing oil level. If the bearing oil level falls, then the feed becomes

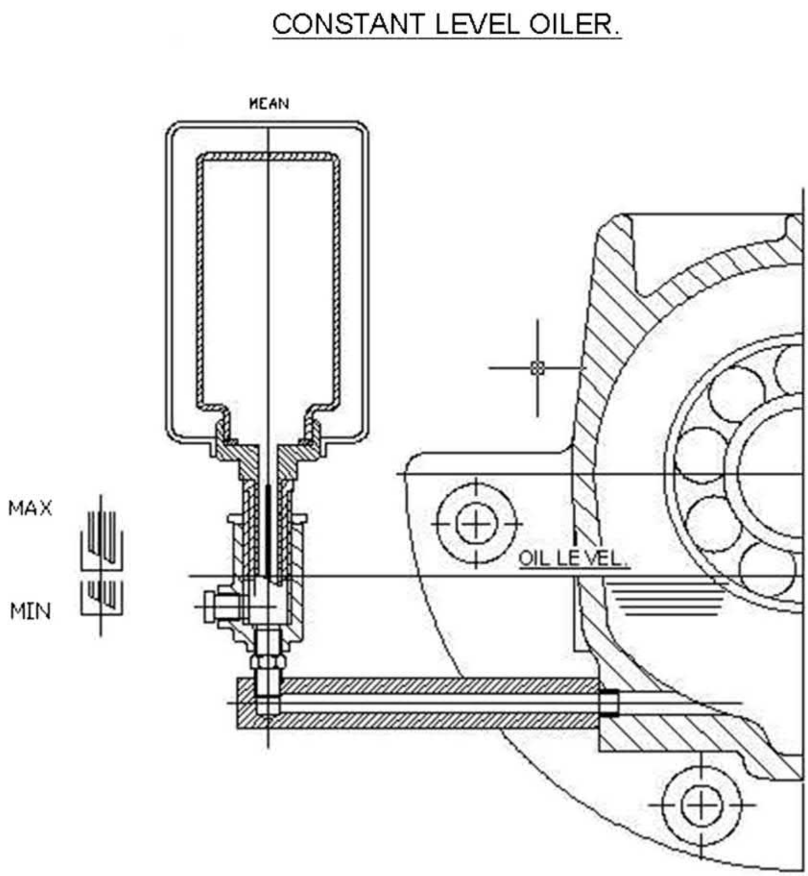



**Fig. 8.32** Elements of a constant level oiler.

exposed, allowing air into the reservoir. This process release oil into the bearing housing until the feed is no longer exposed. Thus the bearing oil level is more or less maintained, until the reservoir is emptied. [The transparent oil reservoir permits ample warning of this] (Fig. 8.32).

Sometimes, these oilers are seen to inexplicably loose oil level. A fundamental assumption of this device is that the air pressure inside the bearing housing is the same as that at the oiler vent hole. Normally this is true. However, if the pump is equipped with a cooling fan, then the air flow impacting on the vent hole can have a Pitot tube effect, upsetting the internal − external pressure differential. This then convinces the oiler to mysteriously function incorrectly. It is best to avoid direct air impact.

Another assumption is that the oil level inside of the bearing housing does not materially alter in operation. In practice, oil flingers on 2 pole pumps can drive the oil against the bearing housing wall. This distorts the oil surface considerably, compared to the non-running level at which the oiler was set. In some cases, this can cause also cause the oiler to mysteriously discharge.

It may seem obvious that these types of oilers are best suited to fixed machines. Nevertheless there are cases where they have [perhaps inadvertently] been used on pumps that operate on floating vessels such as FPSO[6] platforms. They seem to operate provided the swing range is less than $+/-5°$, but even so their consistency of operation is suspect.

## Does pump spin?

If the pump is single stage, check that the shaft spins or at least turns with some quite steady resistance [due to seal and bearing drag]. There should be no mechanical noise or 'grating'/grinding (Fig. 8.33).

Some multistage pumps, particularly those having a large stage number [more than nine, say] should very cautiously be turned by hand because the sagging shaft might allow rubbing contact between the wear rings. If the materials do not have a high galling resistance, this could result in a seized pump. This class of multistage pump relies upon fluid passing across the wear rings to exert a stiffening effect upon the shaft − in effect a product lubricated bearing.[7] This concept is quite commonly used, *see also* Appendix F.

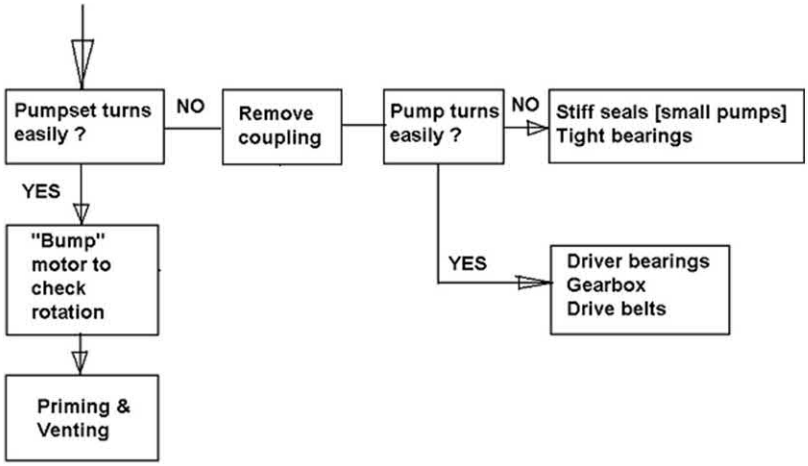



**Fig. 8.33** Pre-start spin check flowchart.

## Direction of rotation

It is essential to check the direction of rotation, because if it is incorrect there is no point in proceeding. A pump that is running in the wrong direction may very well start and pump if installed in systems that have little static head. Tests show a performance from an Axial flow pump running backwards. The renowned Lancastrian pump engineer, the late Harold Anderson, voiced the rough rule that a *centrifugal* pump running backwards would generate half the head at two thirds the flow. As with many of his rules, this proves to be near enough true for most practical purposes.

If the coupling is removed to check for stiffness, then the motor rotation can be very easily checked at that time. If there is no apparent need to remove the coupling, then the contactor can be 'bumped' i.e. briefly started and stopped in which time the motor will just start to turn, disclosing it direction of rotation.

The correct direction of rotation should be indicated by an arrow somewhere on the machine. Often this is cast into a component so is relatively indestructible. In other cases, particularly with older pumps, the direction is shown on a nameplate which may become detached/lost, or over-painted.

In this case the correct direction of rotation can often be deduced from the pumps shape. Focusing on the discharge volute: In which direction do

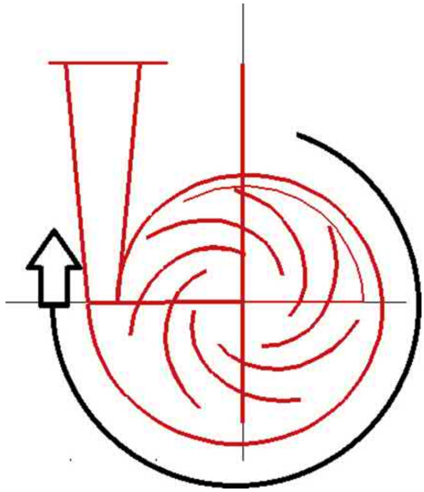


**Fig. 8.34** Schematic to show relation of vane curvature to volute area growth.

the areas increase? Then that is the direction of rotation. In the examples shown below, the impeller rotates clockwise in the viewing direction.

There is a particular hazard with the double suction. Here it is quite possible in some cases to install the impeller the wrong way round. The shaft may rotate in the correct direction, but the impeller does not. As a rule, the impeller vanes should always curve away from the direction of rotation (Figs 8.34–8.37).

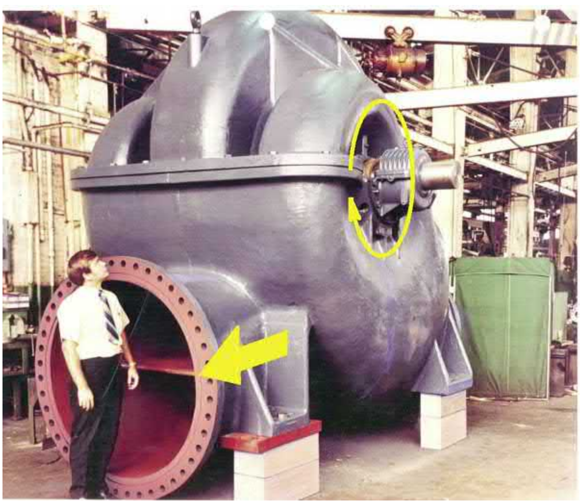


**Fig. 8.35** Double suction between-bearings water pump.

**Fig. 8.36** Single suction overhung impeller pump.

**Fig. 8.37** Double suction between-bearings process pump.

## Pump speed

Even if the rotation direction is correct, the speed may not be. If it is a Variable Speed Drive, the speed setting may not be enough to get the pump head up above the system static head. In this case, the pump will just 'churn' away and heat up the entrained liquid in the casing.

Some multistage pumps rely on the so called *Lomakin*[8] effect to support the rotor. It is not recommended to run this class of machine too slowly, else there is not enough head generated to create a strong support. Contact may occur.

Occasionally, pumps are sometimes energised by a belt drive system. These rely upon correct tensioning to provide the rated power. If too slack, the belt slip and, as above, may also prevent the pump from at least exceeding the system static. Pumping will not then take place.

## Shaft sealing check

Unless the pump is of the sealless design,[9] there will be some sort of shaft seal to check. Increasingly, new pumps are furnished with mechanical face seals. Venting of mechanical seals, prior to start-up is critical since a 'dry' seal can rapidly generate a considerable amount of frictional heat at the faces. This can lead to accelerated face damage. Vertical pumps, particularly those on process service, should have a valved seal vent line, because conventionally, free gas can collect in the seal chamber.

If the pump is 'soft-packed' then there are still a few worthwhile checks to carry out.

## Soft packing

This seal concept works by constricting a gap between the rotating element and a stationary-sealing element called 'packing'. Taking advantage of a material property known as Poisson's Ratio, an applied axial force is transformed into a radial constriction force. A device called a gland follower applies the axial force. In most cases, only the first two or possibly three packing rings exert any significant radial sealing force.

When correctly set, the packing is constricted and deformed in such a way as to closely follow the rotating element shape without touching it to any great extent. The gap created permits a small but finite flow of liquid. There may be local contact and of course local heating. It is the role of this small liquid flow to lubricate and also to remove any of this heat (Fig. 8.38).

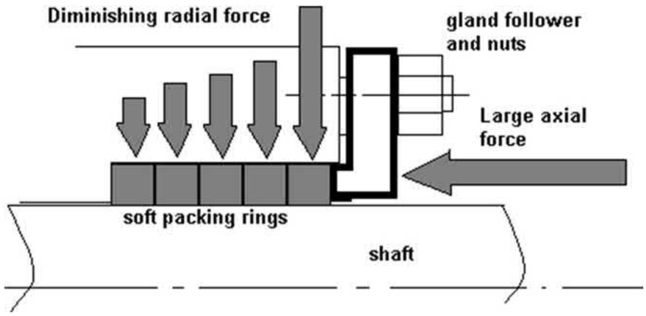



**Fig. 8.38** Basic elements of a soft-packed shaft seal [gland packing].

If over-tightened, the flow gap becomes too small and heat cannot be transferred away. Nor can the interface be lubricated. This causes damage to the surface of the rotating element and decay of the packing properties. This damage **cannot be undone** by releasing the over-tightening. Once damaged, the performance of the device is permanently ruined, so it is absolutely essential that the correct setting be approached slowly and steadily. It may take some hours to achieve such a state. However, when correctly set the leakage from a soft packed device only amounts to a few drops per minute. Once this equilibrium condition is successfully reached, the machine may run some thousands of hours untouched. Achieving this equilibrium state becomes more difficult if the relative rubbing speed is high – say over 12 m/s.[10] Something often overlooked, is that poorly set, over-tight packing consumes a surprising amount of shaft power.

In the early history, packing of some form was the predominant method. A feeling existed that if the correct packing was damaged or unavailable, any old rag or cloth could be 'stuffed' in to the gland area [hence the term stuffing box] and the device would work. While rubbing speeds were low, this feeling was partly true. As the average rubbing speeds tended to increase, only purpose made packing could provide useful service. Modern high performance packing can demonstrate significant advances over the original graphite impregnated materials.

If the packing history is not known prior to a start-up, it is prudent to assume the worst [that it has been inadvertently over-tightened] and back off the gland follower nuts. If this assumption proves true, then you may have saved the situation. If this assumption proves false, then it takes little time to carefully regain the equilibrium state.

Should the machine operate on a flooded suction, there may be enough positive pressure to drive product out to atmosphere and hence lubricate

**Table 8.8** Common faults with soft packed seals.

| **Past faults** |
|---|
| • Lantern ring in wrong place – not in line with feed [particularly when on a suction lift when liquid tends to be drawn into the pump, not into the packing] |
| • Over-tightened and sleeve damaged |
| • Packing scarf joint lined up, not staggered. |
| • Sleeve finish not smooth enough |
| • Leakage under shaft sleeve under high suction pressure |
| • Static suction head too high – more than 900 feet |
| • Makeshift packing used |

the packing. If the pump operates on a suction lift, the reverse is true. Some form of pressurised feed is needed to avoid starving the packing. Often a connection is made back to a high pressure point in the casing. Older machines may have a control valve in this feed line and this should be verified as open. In most, if not all, modern machines, the feed to the soft packing does not contain any form of flow control. So there is nothing obvious to check prior to start-up. Though extremely unlikely, this feed may be blocked internally, preventing flow. The only practical way to check this, is retrospectively – if there is little of no flow from the packing, but lots of heat and smoke! (Table 8.8).

## Mechanical face seals

The use of mechanical face seals has steadily gained acceptance. Face seals rely upon a thin film of liquid to separate the faces. This film really is very thin indeed, and therefore quite susceptible.

Early seals got a poor reputation for reliability, their performance in the field falling far short of that achieved in the development laboratory. Once the important differences between the laboratory and field conditions were identified, pump design attempted to address them and seal life improved. While not ideal, seal life has significantly improved over recent years. Seal life of over five years is now much more frequent than before (Fig. 8.39).

At the pump start-up stage, initially one has to assume such basic parameters as the seal PV value and the balance ratio have all been previously reviewed and are correct. All that can be controlled are the installation factors such as concentricity, shaft deflection, shaft run-out, etc. With single seals, face cooling is effected by the product alone.

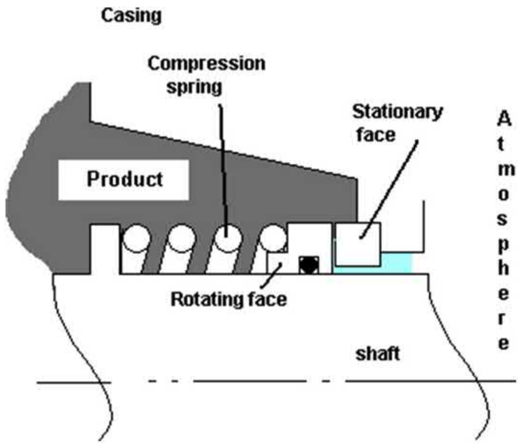



**Fig. 8.39** Conceptual elements of a simple single mechanical face seal.

While simple single seals are the most prolific in use, multiple seals find use in many hazardous services such as hydrocarbon refining and chemical applications. Clearly if a single seal fails, then the product flows straight to the atmosphere. Many products are quite benign and such leakage is a mere nuisance. There are no hazard or safety risks involved.

On the other hand, some products have very bad habits and are best contained. They may be hazardous to health, or perhaps they will auto ignite upon exposure to the atmosphere. Clearly leakage in these situations must be contained. The most popular approaches use two seals operating in concert.

One provides a backup for the other in the case of the primary seal failing. The secondary seal enables any product leakage to be contained from direct exit to the atmosphere.

Another arrangement traps a dedicated liquid between the seals. This liquid is intended to lubricate the seal faces, and prevent any solid particles getting in-between the faces.

The two most common layouts see the seals operating in tandem or back-to-back. Alternative descriptions for these layouts are unpressurised or pressurised dual seals. 'A rose by any other name…'. The main concern at the commissioning and start up stage is to vent and not run the seal dry.

## Dual pressurised seals

The secondary [atmospheric] seal is a mirror image of the primary [product] seal – at least conceptually. The space between the two is filled with a liquid compatible with the product and pressurised to a level in excess of

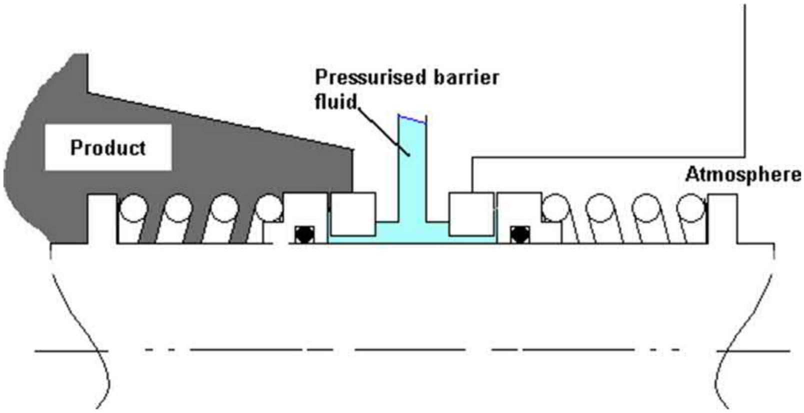



**Fig. 8.40** Conceptual elements of a dual pressurised [back to back] mechanical face seal. The barrier liquid cools and lubricates the seal faces.

that expected to be experienced in the vicinity of the primary seal. This normally means that the barrier liquid very slowly leaks into the product — hence the need for compatibility. This act ensures clean liquid at the seal faces, so the approach is useful even if the product is not dangerous but may be abrasive. Perhaps it also tends to crystallise (Fig. 8.40).

So long as good barrier fluid is passing out across the faces, aggressive product cannot get into the barrier. Even if this seal fails and product does get past, it can be contained in the barrier liquid. This has the added advantage that such behaviour is usually associated with a step change in barrier fluid pressure, and this can be monitored in a number of ways so as to provide seal leakage alerts.

Furthermore, the barrier liquid can act as a circulating transport medium and carry heat generated by the faces, off to a heat exchanger. Sometimes this can be achieved by thermosyphon effects, but usually some assistance is needed. Seal makers employ a variety of crude pumping devices-with the emphasis on the word crude! Modern seal systems and their piping are ever more complex. This spotlights any inadequacy in the crude pumping device. Axial flow or scroll type devices appear less able to confront complex seal system piping than do radial flow devices.[11]

## Dual unpressurised seals

The secondary [outboard] seal is a replica of the primary [inboard] seal, at least conceptually (Table 8.9).

**Table 8.9** Common faults with mechanical seals.

| Common faults |
|---|
| • Seal flush, if any, not turned on *before* start-up |
| • Seal flush impeded and unable to take heat away from the seal faces |
| • Seal chamber not vented before start up |
| • Barrier liquid not pressurised before start up, and so product leaks into barrier system immediately |
| • Alignment clips of cartridge seals not removed prior to start up |

The space between the two is filled with a liquid compatible with the product. This liquid is not pressurised to any great extent–if at all. This means that the product can leak into the barrier liquid, creating a backup capability (Fig. 8.41).

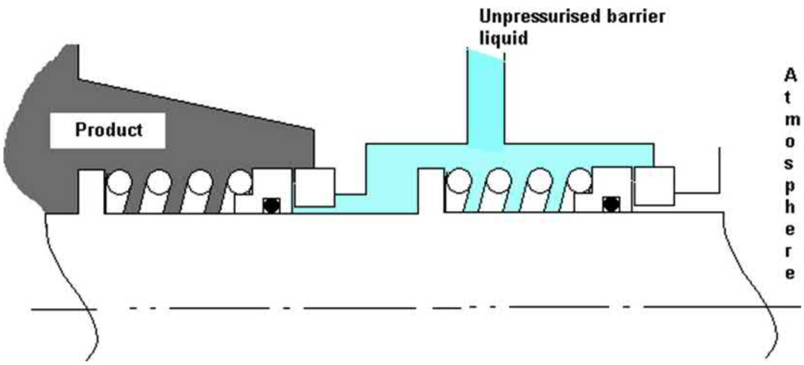



**Fig. 8.41** Conceptual elements of a dual unpressurised [tandem] mechanical face seal.

## Protective guards

The concepts of safety in the workplace have never been higher. In earlier times, the design of guards protecting the rotating parts was based on preventing people accidentally falling onto the equipment. Therefore, it was common to see guards that performed as umbrellas, shrugging off any person or thing that fell onto it, while affording no defence at all to access from below. Nowadays, people accept that complete protection is necessary. Sadly, one motive for this is to protect against deliberate and intentional minor self-injury.

There has been a practice of loosely laying guards in place at first start-up with the genuine intention of fastening them securely at a later time. It is a fact that if a rotating assembly is going to shed parts it is most likely to do so within a short time after start-up. This is the time when secure guards are *most* needed, not later on. There can never be an excuse for starting up a pump without adequate guards being properly fixed.

The guard detail design should be examined closely. Some designs permit the generation of a strong vortex, driven by the windage of the coupling. The vortex has low pressure at its centre-which may be enough to draw oil fumes from the bearing housing, or even impair the functioning of simple bearing labyrinth seals. Oil leaks might result from this.

## Warm-up of pumps on hot service

Pumps that operate on a high temperature service will require an additional consideration. Most pumps will benefit from pre-warming the prior to going in top service. This splits in to two issues:

- How fast should the pump be brought up to the holding temperature?
- How can it be held at the holding temperature

## Warming up

Small pumps can be brought up to temperature very quickly. The reason is that less significant local temperature differentials tend to form in small pumps compared to large machines. With large pumps, quite significant thermal gradients can arise and this can lead to substantial short-term distortion of the machine. Size can be characterized conveniently by the impeller diameter, though other secondary factors come into play. As a guide, warm-up should proceed at about 4 °C per minute. Small pumps [10 inch impellers] can exceed this rate, whereas larger pumps [30 inch impellers] must be warmed more slowly.

The basis for warm-up rates reflects the different thermal inertia of pump components. For example, impeller wear rings have much lower thermal inertia than the impeller on which they are mounted. This means that during warm-up, the rings may 'grow' more rapidly than the impeller and there is a risk that interference fits may be [temporarily] lost. If the pump is started while this picture is still gaining equilibrium, then the rings may be pushed off or at least dislodged.

Similarly, if the pump is cooled down to quickly, then, having a lower thermal inertia, the shaft may contract faster than the impeller, which was shrink-fitted to it. Here the fit may also be lost and the impeller becomes susceptible to dislodgment due to [say] hydraulic shocks.

Symmetrically shaped pumps can obviously warm-up more uniformly than pumps that are not. Problems encountered with non-symmetric pumps are temporary binding of the shaft in the wear rings, or casing rubs. Bracing ribs can exhibit surprising temperature differentials compared to the main casing. This differential can induce more casing distortion.

Such problems are more prevalent in larger pumps where bigger temperature differentials can occur. It should be the aim to limit temperature differentials between top and bottom of the casing to less than 25 °C, while bringing the pump to within 80 °C of the product temperature. Values outside of these would be considered as thermal 'shocks', and while many pumps can tolerate larger levels, they are not without risk, and should be discouraged.

## Holding at temperature

Large temperature differentials are not just a problem on warm-up, they also exist while attempts are made to hold the pump at a given temperature – while on hot standby for example.

The normal method of keeping a pump warmed up on hot standby is to allow product to bleed back through discharge back to suction. Often the pressuriser for this is the working pump and the bleed is arranged via a small hole in the non-return valve (Fig. 8.42).

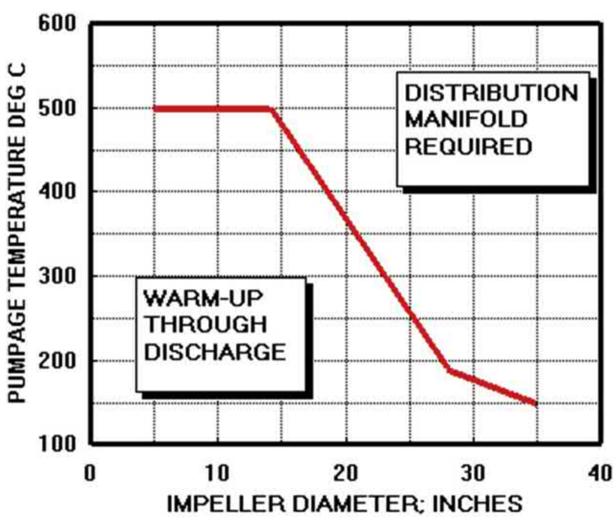



**Fig. 8.42** Warm-up configurations related to pump size and product temperature. This chart relates to between bearings double suction pumps.

With small to medium sized pumps this is adequate unless the temperature is unusually high. With larger pumps this principle is unsatisfactory since the hot bleed has been shown to stratify in the casing and create unacceptable temperature differentials of more than 25°. This may lead to mechanical rubs, making the machine stiff to turn if and when start-up is needed. This whole picture is considerably worsened if there is a [relatively] cool flush to the mechanical seal, since that will enhance the stratification. This can result in significant thermal bending of the static shaft. Large machines are even sensitive to local wind blasts upon the casing.

At a certain point – indicated by the chart, it becomes necessary to distribute the bleed liquid more evenly. One way of achieving this is to arrange a distribution manifold around the pump. The main feature of this is that hot liquid is introduced at the bottom of the casing and entrains the cooler liquid as it rises up. With double entry pumps it is also useful to arrange a similar bottom feed to the suction nozzle – for the same reason. This has proven successful on some of the world's largest and hottest process pumps. [Ref 15].

## Casings for high temperature

The thermal expansion of large hot pumps can impose substantial forces on the bedplate unless provisions are made to absorb it (Fig. 8.43).

Up to a certain size, the expansion can be absorbed by clearance in the pump foot holding down bolts. [They will be doweled to the bedplate pads].

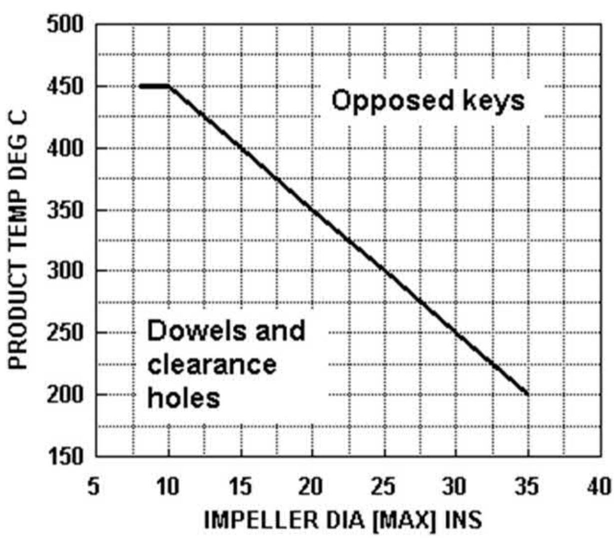



**Fig. 8.43** Criteria for opposed key mounting.

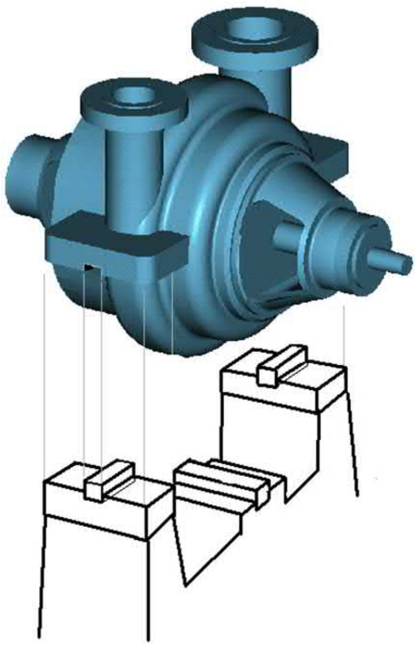


**Fig. 8.44** (right) Opposed keys used in high temperature pumps to maintain alignment while permitting thermal growth of the casing in three mutually perpendicular directions.

Beyond that it is necessary to key the pump feet in some way that allows radial growth of the casing, but constrains the pump from moving in the axial direction. A popular method is shown in Fig. 8.44.

## Applied pipe loads

These days it may seem unnecessary to discuss the topic of applied pipe loads. The consequences of applying excessive pipe loads have been extensively documented, however a few reminders may not go amiss.

Pumps are rarely, if ever, designed as piping anchors, though certain classes of pumps are designed to accept quite well defined levels loads.

For example, those complying with the API 610 standard will have to operate reliably whilst subject to well-documented combinations of loads. The loads are split into two categories:

- Those trying to bend the nozzle off the casing [Forces]
- Those trying to twist the nozzle off the casing [Moments]

These loads occur due to the expansion of the pipes as they come up to temperature. The effects of such loads are to distort the pump casing in a way that might induce internal rubbing. Furthermore, the bedplate may also be deformed in such a way as to take pump-driver alignment outside of limits. In the past, there was a view among pump designers that the actual levels of pipe loads frequently exceeded that set by the standard. However, modern software appears to reduce the uncertainty in the plant designers' calculations.

In general, the larger the pump branch sizes the higher loading it is expected to be able to live with. Unscheduled loading over and above the design levels has the effect of:

- Increasing the casing stresses under the flanges
- Distorting the casing so much that the wear rings may rub
- Deforming the bedplate and putting the pump out of alignment with the motor

API pipe loads are a good starting point, but they do not absolve the purchaser from all responsibility. However, pipe load capability is one reason why process pumps cost more than industrial machines.

As mentioned above, API tables maximum pipe loads based on the machine branch sizes. It even gives the purchaser the option to verify shaft end motion by testing. Unfortunately, no such similar test exists for verifying the actual applied loads on site, so the test is not absolute. API pipe loads are a good starting point, but they do not absolve the purchaser from all responsibility.

In contrast, users of industrial centrifugal pumps are normally discouraged from applying any substantial pipe loads to their machine at all. Why?

Well such machines are often constructed of brittle or low strength material – not suitable for the extra loads applied by piping. Furthermore, the bedplate would rarely have been designed with such loads in mind. This is an important consideration because the loads not only influence fracture; they also govern alignment of pump and driver.

With a typical process pump, the pump shaft end motion, relative to the driver shaft can be split into two causes. Typically half is due to deflection of the bedplate and half due to casing. In contrast, industrial pumps have a bedplate contribution to shaft end displacement that is much higher, due to their lighter construction.

Stories abound, in the pump community, of users deploying forklift trucks, or block and tackle to pull pipes into sufficient alignment. Once aligned, the bolts are inserted thorough the mating flanges of the pump.

Offsets as high as 15% of the pipe diameter have been reported. In these cases, the pump may already be heavily pre-loaded before the bona fide plant loads are applied.

Ideally, the pipes a should be set such that a sheet of paper is *just* nipped by the mating pipe/pump flanges before being bolted up to the pump. In practice a gap of $0.020'' - 0.030''$ is useful. In any event the fasteners should be loose in the bolt holes. Both these measures should ensure that when the fasteners are tightened, undue static pipe forces are not transmitted to the pump.

As a matter of course, the pipe hangers and expansion loops [where fitted] should be examined for binding, since this can induce surprising loads.

## Mechanical looseness

During an installation, or re-installation, mechanical fasteners sometimes get overlooked. The following are all real examples and to err is human.

**Pump-Pipe mating flanges.** A leak here may be dangerous, especially on a high-pressure pump. At worst, the high-pressure jet may cause injury, at best it may travel substantial distances showering equipment not designed to resist.

**Bedplate holding down bolts.** Sometimes they get overlooked in the grouting process.

**Pumps or Motor holding down bolts.** Again they may be overlooked after shaft alignment.

**Coupling assembly bolts.** It has been known for these to be only finger tight. Shortly after start-up while everyone is trying to trace the cause of the vibration, one or more fly off. Off course the coupling guard should be in place, minimising the risk from this sort of thing. But this is not always the case. Familiarity breeds contempt and the writer well remembers as a young engineer being involved in a development test and eagerly awaiting the pump to start. He had not yet learnt the lessons that preached here, and on start-up a coupling nut flew past his ear — so close as to emit a low pitched buzz. After ricocheting between roof and floor for a while it came to rest. Fortunately the only thing that was imprinted was a practical safety lesson.

**Pump assembly.** I cannot ever remember, or have ever heard of a pump being started where the major fasteners have not been tightened. It may have happened to others, though so it should be considered.

**Gland bolts.** One of the few areas where looseness is encouraged is the gland follower bolt on a soft packed pump. The importance of this cannot be over-stressed, but in any event, this issue is covered elsewhere.

## Bedplates

Integral pump-drivers sets have no need of a bedplate. Some designs may actually be hung directly in to the pipe itself.

All other pump sets require a bedplate of some form. The bedplate holds the pump and drive in correct spatial alignment as well as absorbing pipe forces and moments. Sometimes this bedplate may be mobile – on wheels or perhaps sledge shaped and capable of being dragged around.

Large machines on offshore oil platforms may be mounted upon three point bearing supports.[12] This helps absorb the minute flexure of the rig structure, to which it is attached, but without unduly stressing the pump set.

The vast majority of pump set with bedplate are permanently installed. Ideally, the bedplate should rest upon a prepared concrete block with a mass at least five times that of the pump set. Experience says that the block should be reinforced. Both these steps help provide a solid base for the bedplate.

The pump and driver will have been fully aligned in the manufacturer's shop – probably with the bedplate lying loose on the assembly floor plates. Neither machine will generally have been doweled to the bedplate at this point.

For shipment, the coupling flexible will often be removed so as to avoid transportation shock loads being transmitted to the pump set shafts. Larger pump sets will also have their motor removed from the bedplate.

On site, the pump set – complete with coupling shall be roughly aligned prior to grouting.

Then check aligns. Do not rely upon the manufacturer's alignment setting – this is just his internal check that everything is capable of being aligned correctly. But his settings can easily be disrupted during transit – hence an on-site verification is necessary.

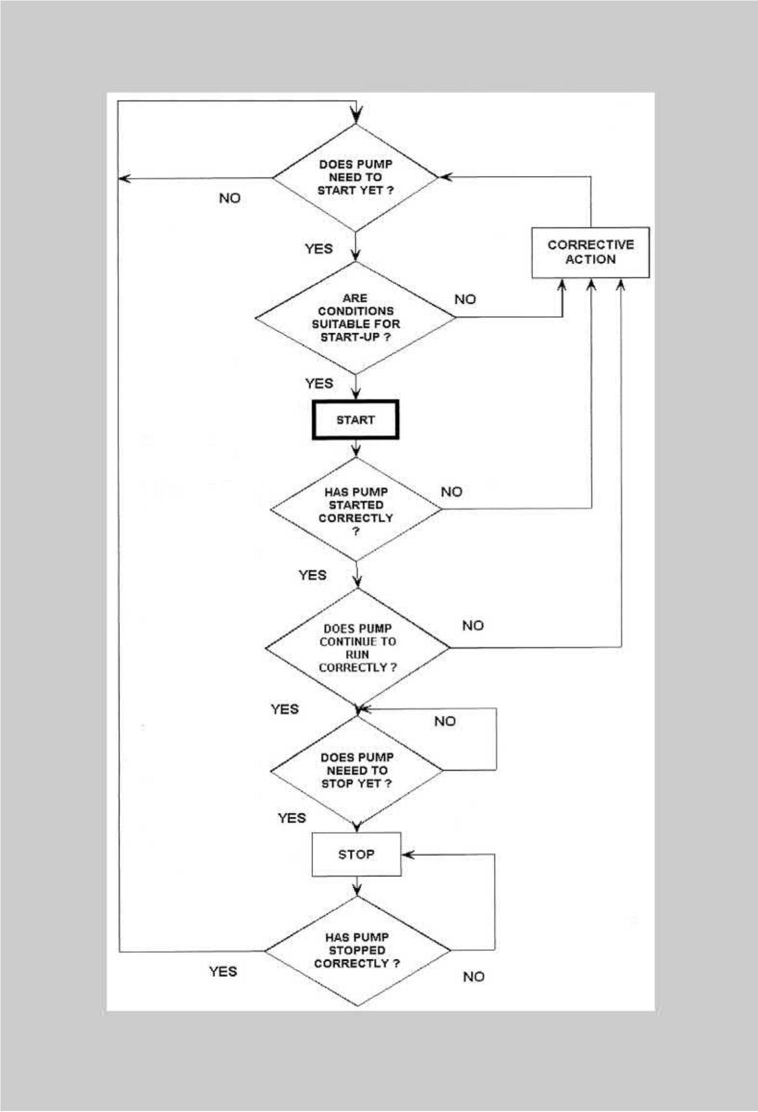
## Endnotes

1. At the time of writing, this method is under review, since it is known to exaggerate the effects on smaller pumps.

2. Note this problem is normally limited to the commissioning phase. Once the system is commissioned and the discharge is maintained in a 'full' state, these problems should not arise.
3. Water hammer, when it occurs, is most likely to originate in the discharge line. Suction lines are rarely long enough to promote these phenomena.
4. A valve that is bypassing often makes a high frequency 'whistling' sound.
5. This feature is particularly applicable to long cantilever shafted pumps. The extended suction nozzle allows a greater range of operating level with increase in shaft overhang.
6. Floating Production Storage and Offloading.
7. This phenomenon is called the Lomakin effect see Appendix F for simple demonstration.
8. The Lomakin effect describes the rotor support created by liquid forced through the wear rings under the effect of pump differential head *see* Appendix F.
9. Or a vertical sump pump.
10. 2.5″ shaft at 3560 rpm.
11. Though even radial flow devices can often seem marginal.
12. Conforming to Lord Kelvin's principle of Kinetic closure.

# CHAPTER 9

# Start pump




*Troubleshooting Centrifugal Pumps and Their Systems*
ISBN 978-0-08-102503-1
https://doi.org/10.1016/B978-0-08-102503-1.00009-8

## Introduction

Starting any pump can be a cause for initial concern. What should be a routine operation, might hide some doubts. Have all the pre-start checks been accomplished, for example? But once start-up is attempted, pumps of different hydraulic geometry will respond differently. Furthermore, different pump system characteristics and pump configuration will respond differently during the start-up process.

## Emergency shut down process ok?

Before pump start, thought should also be given to emergency stops. In the unlikely event that the pump has to be quickly shut down, either during start-up or during normal operation, emergency arrangements should be easily accessible. Events which may predicate such an event might be:

- Excessive vibration
- Unexpected noise
- Liquid leaking from the discharge pipe flanges
- Mechanically incomplete – i.e. loose coupling bolts

A pump that has to rely upon the angular momentum stored in the pump and driver rotor to continue pumping will come to a stop quickly, but not immediately.[1] Typically most of the energy will have been dissipated in to the system in a few seconds, though the rotating element may continue to slowly rotate for much longer. If the pump is operating in a system which consists mainly of static head, then forward flow from the pump will stop quite quickly as the speed drops below that needed to exceed the static. In a frictional system the forward flow drops more slowly since the speed will have to drop more before the head fails to be achieved.

## Start

In all discussions on starting pumps, the mental image is of a stationary pump. However, sometimes the pump may already be turning in the correct direction. There are a few circumstances where this may happen:

- If the pump faces a system with a negative head, then natural flow through the pump may start it turning. This is an outward flow turbine and hence not very effective, but it may arise
- Another case is where a booster pump is pressurising the pump suction. Again this can cause the main pump to act as an outward flow turbine

The speed attained in the condition is seldom very high but it can assist the starting in some small way.

Equally, there are times when an attempted pump start will be made while the pump is already turning in the wrong direction. This can occur if, for example:

- The pump operates on a suction lift and the suction foot valve fails. In this situation, the entire pump line will tend to backflush through the pump. Initially the pump rotor will not turn, but it may eventually do so
- Another possibility is when the discharge Non Return Valve [NRV] fails open on the standby pump of a parallel pair. — Chapter 14.

If the reverse speed is a small fraction of the forward speed, say less than 30%, then it should still be possible to start the pump. But if the reverse speed is high comparable to, or in excess of the forward speed, then the discharge line should be choked to slow the pump down before a start is attempted.

High specific speed pumps can easily attain excessive speed [Chapter 14] so they are most vulnerable. If four quadrant curve data is available for the pump, then the risks can be more readily quantified.

## Starting single pumps

### Speed torque curves

Most pumps are driven by electric motors and their respective characteristics need to be compatible for successful starting. Overlaying their speed-torque characteristics checks this. The motor speed torque characteristics are fairly well defined. For the pump, there is an infinite range of curves depending upon the system resistance characteristics, *see* Chapter 5. More particularly, the route to the operating point depends upon the flow restraint imposed.

- In one extreme, there may be little or no flow restraint, and the pump is allowed to run straight up to its operating flow point. This condition is often referred to as ‘starting against an open discharge valve’. Another definition describes it as starting against an all friction system [Though in fact it is still applicable to systems which also include a good deal of static head]
- At the other extreme, the flow is restricted to virtually zero until full speed is reached, whereupon the flow restraint is gradually removed until the operating flow is achieved. This is referred to as staring against a

closed discharge valve. It is also defined as staring against an all static system, though again, a good deal of friction might exist yet the approach still stands.

These two different routes to the operating flow condition make different demands upon the driver. To understand this requires examination of a series of pump characteristics – snapshots at intervals of time.

Two points are important:

- The driver curve must always exceed that demanded by the driven machine, in order to ensure that start-up is possible
- The difference between the two curves represents the torque available for accelerating the machine up to full load speed (Fig. 9.1).

The speed torque curve for pumps is often dismissed as typical proportionate 'square law' chart. Versions frequently found in the public domain may well hold true for the most popular class of machine, i.e. pumps with *centrifugal [radial flow]* impellers. However, it is as well to appreciate that the curve proportions do change with hydraulic family. Typical proportionate charts are just that – typical and nothing more.

In the following pages are Figures that show a series of instantaneous flow – head and flow torque curves at different fractions of the full load speed. This illustrates differences in the pumps behaviour related to its hydraulic design characteristics.

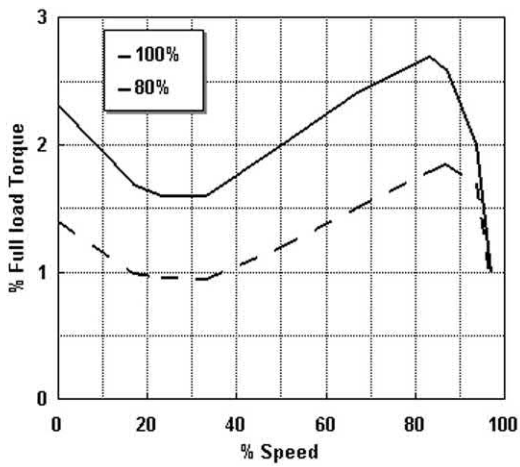



**Fig. 9.1** Electric motor speed torque curve.

### *Low specific speed pump [ns 1400] radial flow*

Beginning with Fig. 9.2 the all-static head case: As the pump starts, its speed builds, but no flow can take place until the zero flow head exceeds the static system head. With this particular pump a speed of over 90% of full speed is reached before system flow occurs. After this the flow increases quite quickly out along the static line to full load-along with its torque demand.

It is more useful to plot the % full load torque against the % full load speed. This is shown in Fig. 9.4. The torque required at zero speed [due to 'stiction' in seals and bearings] is finite. A value of 10% full load torque seems to form a conservative estimate of this. This shows that in any event, starting against a high static-or closed discharge valve induces the lowest load on the driver. This justifies the common advice to start centrifugal pumps against a closed or largely closed valve.

Fig. 9.3 describes the same chain of events for an all friction system. The difference here is that flow proceeds just as soon as the pump is started. The torque demand reflects this — steadily rising to the full load value.

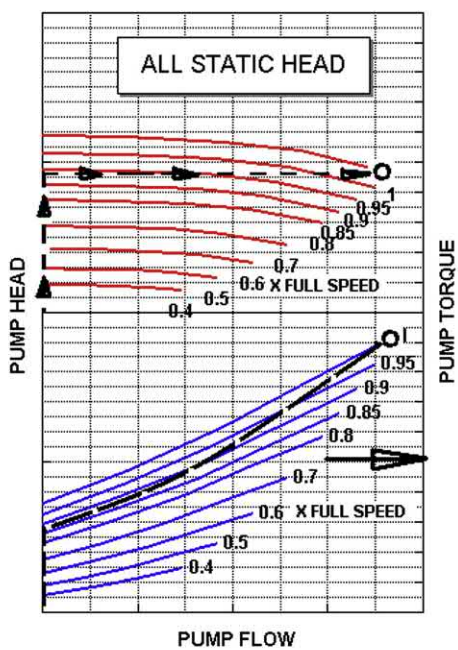



**Fig. 9.2** Torque route to full load, when started against a totally static head [or a closed discharge valve].

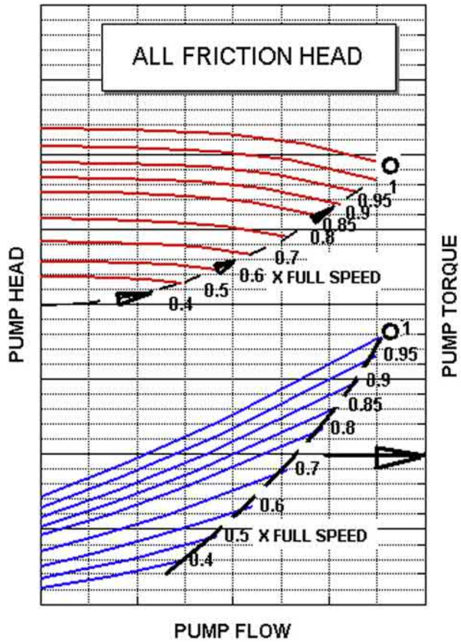



**Fig. 9.3** Torque route to full load when started against a totally friction system, [or an open discharge valve].

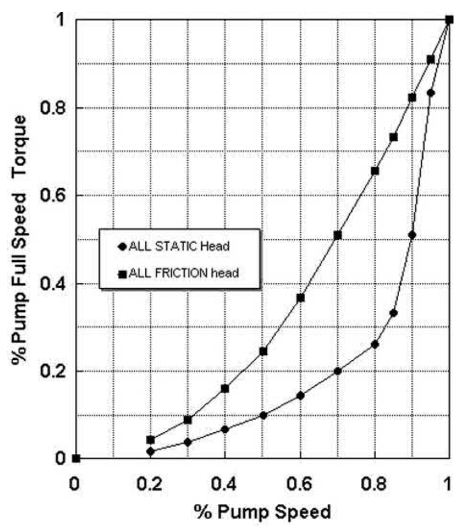



**Fig. 9.4** Traditional Pump speed – torque curve for low specific speed pumps when started against open [friction] or closed [static] discharge valve.

### *Medium specific speed pumps [ns 4000 mixed flow]*

This type of impeller has a rather flat power curve and the power at closed valve may differ little from that at duty point. So start-up against an open or partly open valve is acceptable. This is fortunate because impellers of this family are very unhappy running at low flow for any length of time (Figs. 9.5 and 9.6).

Fig. 9.7 shows that the torque demand is much the same for a static or friction system [open or closed discharge valve].

### *High specific speed pumps [ns 9000 axial flow]*

With this family of impellers, the low flow power may well be higher than at duty point. In fact there may be some difficulty starting such pumps at low flow. The driver may well have been sized for the duty flow and there is insufficient torque to start or accelerate the pump if started against a closed valve.

One way around this problem, if it arises on vertical wet pit pumps, is to inject air into the pump casing or bell mouth and artificially depress the water level so that the propeller is started up in air. Once the machine achieves full speed, the air is slowly released in concert with opening the discharge valve.

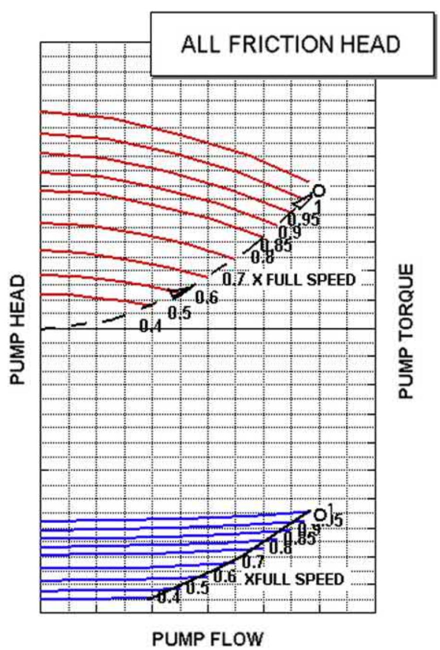



**Fig. 9.5** Torque route to full load when started against a totally friction system [or an open discharge valve].

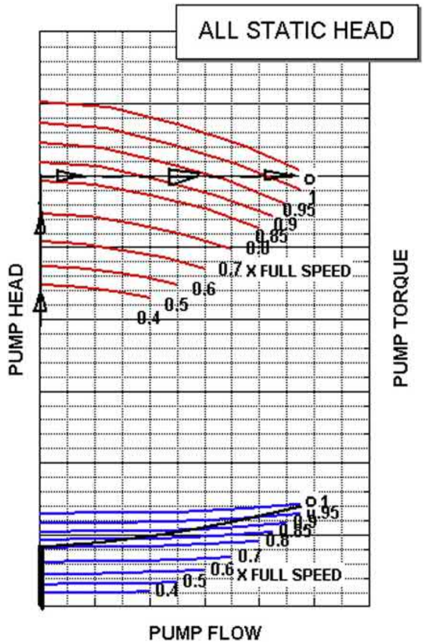



**Fig. 9.6** Torque route to full load, when started against a totally static head [or a closed discharge valve].

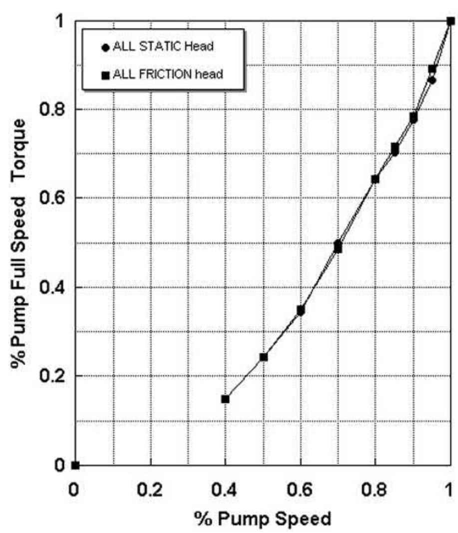



**Fig. 9.7** Traditional pump speed – torque curves medium specific speed pumps when started against an open [friction] or closed [static] discharge valve.

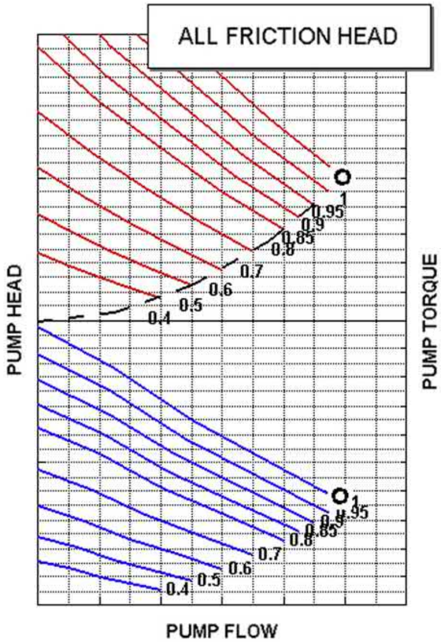



**Fig. 9.8** Torque route to full load when started against a totally friction system [or an open discharge valve].

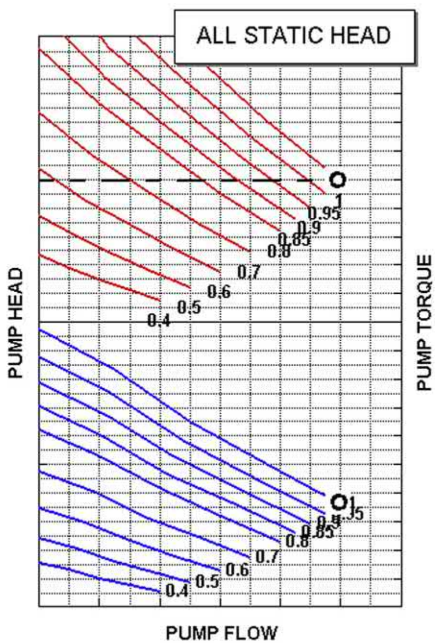



**Fig. 9.9** Torque route to full load, when started against a totally static head [or a closed discharge valve].

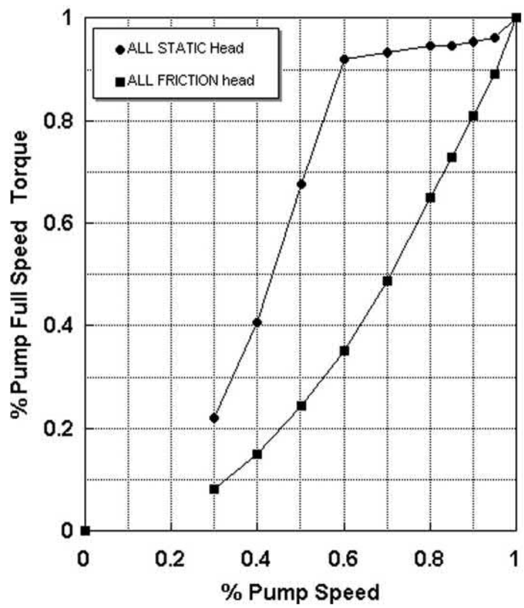



**Fig. 9.10** Traditional pump speed — torque curve for high specific speed pumps when started against an open [friction] or closed [static] discharge valve.

Figs. 9.8—9.10 show that with high specific speed machines, starting against a high static [or against a closed discharge valve] generates the highest torque demands. This is completely at odds with the situation for low specific speed [centrifugal] machines.

## Starting process

The mechanics of starting centrifugal pumps are:

- Open the minimum flow bypass line-where fitted
- Open discharge line less than 20%. [This must be initially closed if pump operates on a suction lift, otherwise the pump would de-prime, then quickly open to 20%]
- Start, having previously checked rotation is correct.
- If VSD ensure that speed is more than minimum rpm [if multistage pump]. This helps establish the Lomakin effect *see* Appendix F
- Pump should develop pressure in five to 10 s At this point, slowly [*see* Chapter 8, to avoid over-pumping] open discharge valve.
- If no pressures in 5—10 s, shut pump down as an emergency, and investigate.

- If pump quickly develops pressure, open valve fully or at least until expected discharge pressure is achieved. Do NOT fully close the discharge valve at this point; as thermal temperature rise will take place see Chapter 10.

## Starting multiple pumps: parallel operation

In many applications, two or more pumps are used in parallel. The idea here is that some degree of flow control is possible. This concept is, of course, best suited to applications where the static head is quite high.

In principle, this arrangement offers some degree of pump control. At times of high demand, multiple pumps can be used, but when the demand subsides, one or more pumps can be shut down. As the demand increases again, pumps can be switched back into the system. This logic is sound because it can keep pumps operating close to their best efficiency flow.

This can be seen in Fig. 9.11, which portrays five identical pumps operating in parallel on a system with very high static head. When all five pumps operate, the system flow is 1500 $m^3/h$ against a system head of 1000 m. When only one pump runs, the system flow drops to about 320 $m^3/h$ against a system head of 900 $m^3/h$. This is just slightly above the best efficiency flow of 300 $m^3/h$.

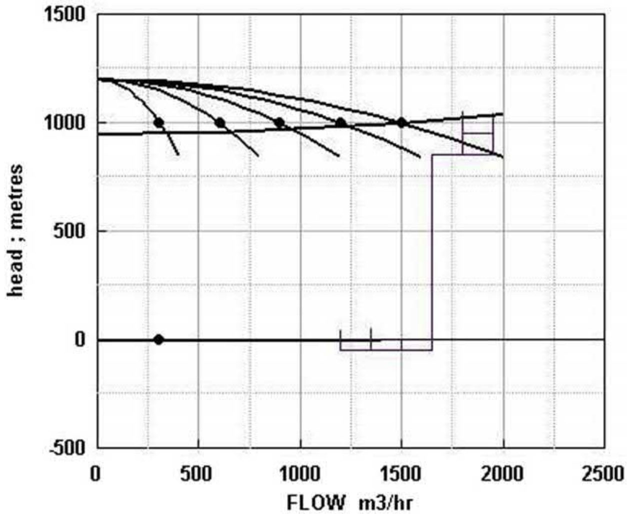



**Fig. 9.11** Multiple parallel pumps in system with high static head.

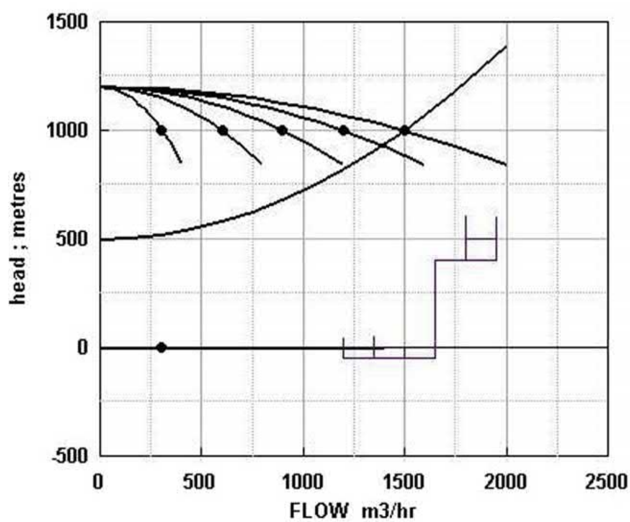



**Fig. 9.12** Multiple parallel pumps in system with combined static and friction head.

In the above case the system resistance consists mostly of static head. Clearly, if the system head had consisted mostly of friction, with very little static, this method of crude flow control could not have worked.

This is shown in Fig. 9.12 where the same pump head is required, but this time the static head has reduced from about 950 m to 500 m. In this case 5 pumps can be easily run, as before. However, if one pump is switched off, the combined pump and system curve intersect at about 1400 $m^3$/h. Now, instead of each pump operating very close to their bep, the pumps are now operating at 15% above bep! This is not efficient; furthermore the high velocities within the machine may become undesirable. There is also a risk that the suction pressure may be inadequate to prevent cavitation. If the next step was to switch off another machine, then the system flow would reduce to 1200 $m^3$/h and so the flow per pump would increase to more than 130% of bep. In fact at such a high flow cavitation would be very likely in most cases. Pumping may not actually be possible in this case. In any event, the pump efficiency would be very poor and the internal wear rates considerable. It's highly unlikely that a single pump could operate. It would be running so far beyond its design flow that the NPSHR would be excessive. In addition, the absorbed power would most likely overload the driver.

The lesson is that parallel pumping is not effective when the static head is low. If, in this last example, the static head had been even lower, then perhaps switching off just one pump might not even have been practical.

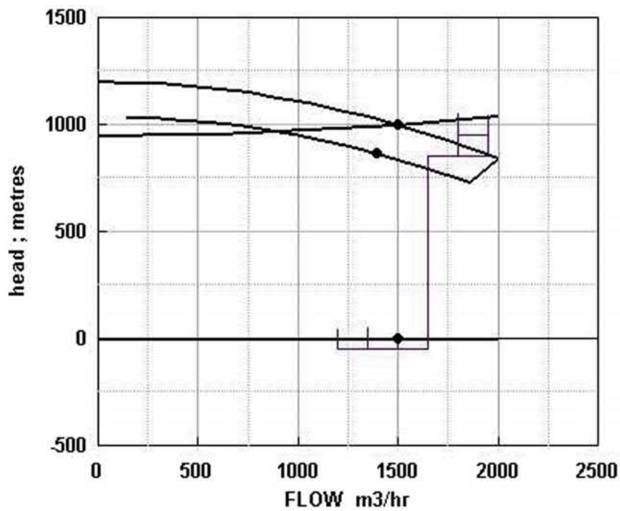



**Fig. 9.13** Sensitivity to speed change as function of curve shape.

The issues of starting multiple pumps will be addressed shortly.

At first sight variable speed drives may seem an attractive alternative. Very, it often turns out that, with high static heads only a small speed change is needed to bring about a large change in flow. Fig. 9.13 shows how much the pump-system intersection moves when the pump speed is reduced by a mere 5%.

Flow in the system is then quite sensitive. It also becomes very sensitive to wear which for small wear rates, has much the same effect on curve shape as speed reduction. Variable speed drives are most suitable for systems that ***do*** have a high friction head ratio.

Though the theory of cascading pumps in and out of action seems sound, the practice may not be actually followed in practice. This is because operators are often acutely aware of the risks involved in starting and stopping pumps. This may not be true of simple prosaic machines where the pre-start checks are minimal. However, if the pump set is large and or complex, with an equally elaborate series of pre-start checks, then the operator concerns heighten. The path of least resistance is then to keep all, or most of the pumps running, and throttle them all back in concert with each other. This immediately causes the pumps to run at much lower flows as the load is now shared between more pumps than the system designer expected.

Of course with this approach, the pumps are immediately available for any load increases. This, and avoidance of start-up risks often becomes the justification of this strategy. The fact that the pumps are now more exposed to low flow operation risks may count for little, not to mention the energy in-efficiency.

Practical examples of this have been found where multiple steam-raising boilers are used each with their own feed pump. The boilers supply industrial complexes where the demand changes frequently and unpredictably. It is not practical to turn boilers systems on and off at will. Dedicating one pump per boiler almost invites low flow problems at some point. There would have been some merit in a ring main feed system for the boilers, serviced by a centralised feed pump cell.

Another example was a steel works complex in which activity tapers off over the weekends. At the end of a working week it was much more convenient to just throttle back all the pumps to minimum flow, rather than shut them all down. That way, they were immediately available for service again the following week. Shutting down the pumps may make more sense to a plant designer, but the start-up risk may not be a factor in his thinking.

## Starting multiple pumps: series operation

Series pumping is particularly appropriate for system consisting largely of friction. As with parallel pump operation, switching pumps off and on can be used as a crude means of flow control. It has the advantage that even if one pump fails, then the pumping scheme will continue to deliver. There is not the same risk of run-out as with parallel operation.

Its main drawback is that the seal chamber of the second [and subsequent pumps] are subjected to discharge pressure from the preceding pump. This is rarely a practical constraint.

Providing the system has been filled, then normally, starting series pumps does not pose the same risks as parallel. In Fig. 9.14, pump 1 is started in the normal way as for a single pump. At that point, it will deliver Q2. This is rarely below the MCSF for the pump. If it is, then of course arrangements need to be taken to side-step this problem. Perhaps a temporary recycle line would solve the problem. Since one pump is frequently used to fill or 'pack' the line in the first place, this might be a prudent provision.

[Very occasionally, situations arise where the shut-off head of the first pump is below the static head of the system. Unless a leak off arrangement is provided, the pump will just churn away, not delivering any flow. A leak

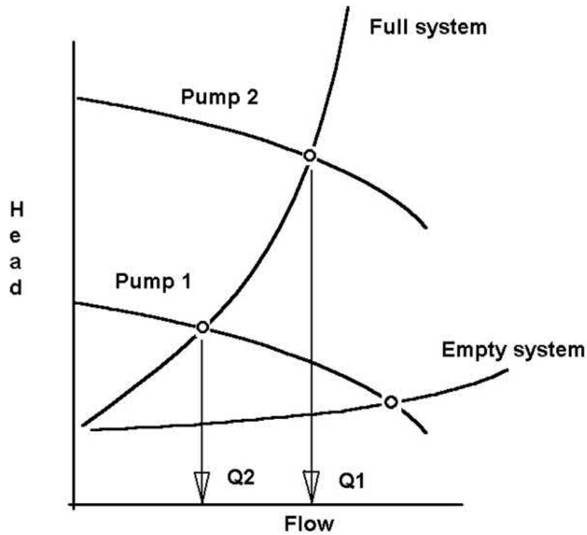



**Fig. 9.14** Series pump start-up should be against a full system to prevent cavitation and motor overload.

off system will allow the pump to start and once the second pump has been started, the scheme will begin pumping. Since this situation can be fraught, it is best avoided of course.]

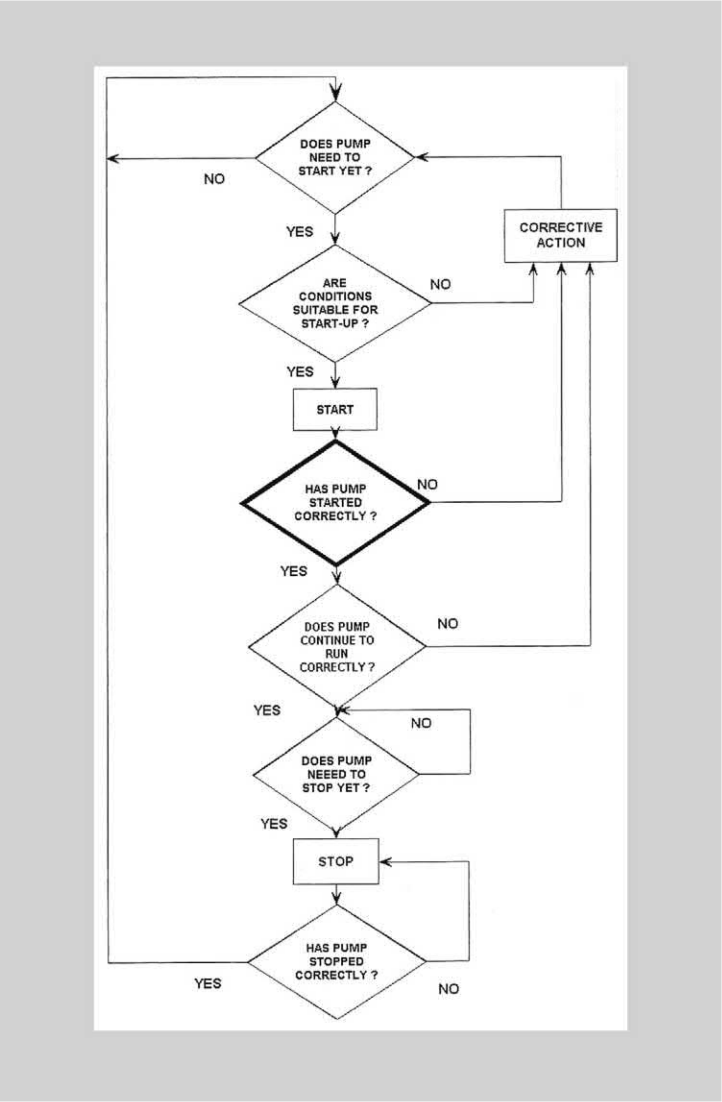
Once the first pump has started, is stable, and the system has been proven full [by assessing Q2 against expectations] then the second machine can be started in much the same way as the first one. As the discharge control valve is opened the flow will move out to Q1.

## Endnote

1. In certain circumstances, it is actually desirable to prolong pumping after power is cut-off. The most usual example is to help suppress liquid surge and 'hammer' in long pipelines.

# CHAPTER 10

# Has pump started satisfactorily?




*Troubleshooting Centrifugal Pumps and Their Systems*
ISBN 978-0-08-102503-1
https://doi.org/10.1016/B978-0-08-102503-1.00010-4

## Introduction

As should be clear by this part of the book, how the pump appears to respond can be greatly influenced by the shape of the system curve. For example, advising that the pump fails to develop enough pressure, ***and*** that it operates on a system with relatively low static head component is much more helpful. By adding more and more elements of knowledge surrounding the application, the root cause of the problem begins to be isolated. Even if no such problems appear, then early signs of mechanical issues might materialise. Reinforcing the point, in certain types of system, a reduction in pump output will appear mainly as a reduction in pump flow, see Fig. 10.1 Pump head will also have been reduced, but not to a significant degree. The same pump in another type of system will signal its wear more by a reduction in head than in flow [3]. So if the pump-system output appears unsatisfactory, the fault could lie with the pump, or it could lie with the system. Indeed, both may be at fault! How to determine which? At one extreme, if measurement of the pump output [head and flow] can be easily measured or inferred, it is quite easy to determine if the pump is meeting the specified performance. Such levels of instrumentation are most likely to exist when the pump forms part of a well-defined process. Here the flow to and from the subject pump needs to be well controlled in order to ensure total system integration. At the other extreme, there may be little or even no instrumentation. But there again, pump performance may not be so critical either. Here it is sufficient if the pump simply exceeds some threshold pumping level. In these circumstances diagnosis is much more difficult.

In this section, various common problems are categorised.

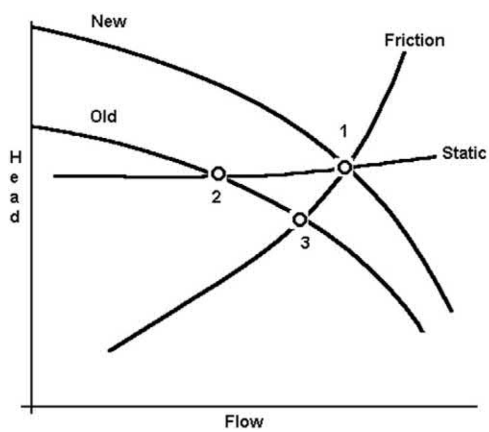



**Fig. 10.1** The impact of pump flow performance decay is felt more in systems with a high static head component than those with high friction.

## Pump develops no pumping performance at all! (Tables 10.1–10.5)

**Table 10.1** Pump absorbs less power, and there is 'some' abnormal vibration.

| Problem |
|---|
| Pump is running dry |
| Pump is air or vapour locked |

**Table 10.2** Pump absorbs less power and there is no abnormal vibration.

| Problem |
|---|
| Poor load sharing, pump is at shut off |

**Table 10.3** Pump absorbs no significant power, and there is 'some' abnormal vibration.

| Problem |
|---|
| Pump is not primed |
| Pump has wrong direction of rotation and is pumping against a high static head (but would absorb some power!) |
| Great amounts of air entering suction line |
| Pump speed is very low |
| Suction lift is too high |

**Table 10.4** Pump absorbs no significant power and there is no abnormal vibration.

| Problem |
|---|
| Slipping drive belt |
| Drive key sheared |
| Shaft, or coupling has sheared |
| Pump is running empty, but has not yet seized |

**Table 10.5** Pump casing gets warm to the touch.

| Problem |
|---|
| Pump power is somewhat correct |

## Pump performs incorrectly, but (Tables 10.6—10.10)

**Table 10.6** Pump emits unusually high levels of noise

| Problem |
|---|
| Cavitation |
| Debris trapped in impeller |
| Air or vapour lock in casing |

**Table 10.7** Pump exhibits both high vibration levels, and power absorption.

| Problem |
|---|
| Internal rubs exist |
| Broken discharge line [pump is at end of curve] |
| Speed is too high |
| Severe wear ring wear |
| Very tight shaft seal packing |
| Pump is operating at too high a flow |
| Pipe loads are too high and cause distortion of pump |

**Table 10.8** Pump exhibits high vibration levels but power is approximately normal.

| Problem |
|---|
| Choked strainer |
| Suction valve partly closed [should never be used for flow control!] |
| Cavitating surge |
| Inlet backflow |
| Operation at MCSF |
| Wrong rotation |
| Choked impeller |
| Pump not primed |
| Pump mis-aligned to driver |
| Loose bearings |
| Mechanical looseness |
| Loose bedplate or foundations |
| Coupling loose |
| Pipe loads |

**Table 10.9** Pump does not show high levels of noise, vibration, or power – pump root causes.

| Problem |
|---|
| Wrong impeller |
| Wrong speed |
| Wrong rotation |
| Air ingress |
| Choked impeller |
| Heavily corroded casing passage |
| Wear ring clearances are very high |
| Wear ring is missing |
| Abnormally high viscosity [power] |
| Balance line leakage too high |
| Not primed |
| Not vented |
| Pump not yet up to speed |
| Pressure gauge is inadequate as flow analogue measurement may be wrong |
| Duty head or flow calculations wrong |

**Table 10.10** Pump does not show high levels of noise, vibration, or power–pipe system root causes.

| Problem |
|---|
| Excess loss in suction line [fouling] |
| Clogged strainer |
| Head calculation is wrong |
| Problem with automatic recirculation valve |
| Air ingress |
| Foot-valve jammed or choked |
| Unexpected change to system [geometry] |
| Excess loss in discharge line [fouling] |

## Pumping performance is correct, but (Tables 10.11–10.16)

**Table 10.11** Pump emits high noise level.

| Problem |
|---|
| Maximum diameter impeller, or even oversize |
| Loose bearings |
| Shaft is bent |
| Cavitation |
| Seal catching internally |

**Table 10.12** Pump displays excessive vibration.

| Problem | Section |
|---|---|
| Pump vibrates initially, but after a while reduces to normal levels | Normal behaviour as temperatures stabilise |
| Pump vibrates initially, but after a while does not reduce to normal levels | Table 10.8 |
| Dynamic balance of rotor | Table 10.8 |
| Pump misaligned to driver, hot alignment notes | Table 10.8 |
| Maximum diameter impeller, or even oversize [explain] | Table 10.8 |
| Suction and/or discharge gauges 'pulse' at low frequency about 0.5–2 Hz | Appendix D |
| Hydraulic cam effect | |

**Table 10.13** There are concerns with the bearings.

| Problem | Section |
|---|---|
| Bearing oil temperature exceeds 80 °C pump product up to 200 °C | Shut down pump |
| Bearing oil temperature exceeds 90 °C pump product over 200 °C | Shut down pump |
| Bearing temperature exceeds 90° pump product up to 200 °C | Shut down pump |
| Bearing temperature exceeds 105 °C pump product over 200 °C | Shut down pump |

**Table 10.14** There are concerns with how shaft packing is functioning.

| Problem | Section |
|---|---|
| There does not appear to be any leakage | |
| Smoke appears from the packing | |
| A few drops per minute leak from the packing | OK |
| A few drops per second leak from the packing. | Loose or worn |

**Table 10.15** There are concerns with how shaft mechanical face seals is functioning.

| Problem | Section |
|---|---|
| No apparent leakage | Might be OK |
| A few drops per minute leak from the seal | Worn |
| A few drops per second leak from the seal | Worn |

**Table 10.16** Absorbed powers appears to be high.

| Problem |
|---|
| Drag from dual seals if pump is small |
| Multistage stacked rotor pump has rubs in the balance device when built or during start-up |

## Effects of surface roughness increase on performance

In earlier Chapters the factors that influence pump performance have been discussed. One important factor is surface finish. Often pumps are constructed of materials, which are prone to corrode on the wetted surface. The degree of corrosion depends upon materials, pumped liquids, and liquid conditions such as temperature. A change in liquid or liquid condition can lead to abnormal levels of corrosion on the wetted surfaces. If the pump has not been run for some time, then unusual levels of corrosion products can form. Cast iron pumps are usually susceptible to this effect.

The result of this is to increase the internal losses within the pump, *see* Fig. 10.2. The original performance decays from [1,2]. This is similar but not identical to the viscosity effect. For all practical purposes, the performance degradation is flow-related and is seen to follow a square law − similar to losses in pipes.

At the same time, the power absorbed by the pump [5] might increase [6]. This is due to the increased liquid drag between the impeller and roughened casing.

## Pump speed is too low or too high

The effect of incorrect speed [usually a problem with variable speed drivers] depends very much upon the system with which the pump is faced. In the worst case, the system consists of significant static head. Chapter 5.

## Speed too low

In this type of application, it distinctly possible that by running the pump at below design speed, there is not enough head generated to overcome the static system head [3]. This means that the pump cannot create any flow in the system and enters a condition of churning. All of the power that goes into the pump is converted to heat and noise, mostly the former. In some cases, the amount of heat is small, and it can be radiated away from the pump casing. In other cases, the heat input can be very substantial, particularly given the small liquid volumes entrapped in a casing. This means that the liquid in the casing is very rapidly brought to boiling point, and the pump

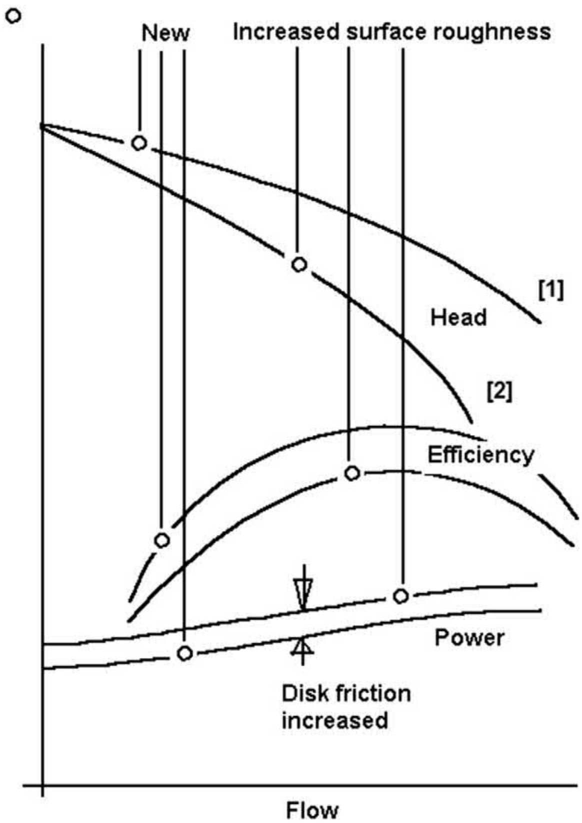



**Fig. 10.2** Effect of increased surface roughness on pump performance is seen least at zero flow.

then proceeds to become Vapour locked. In any event, high vibration is possible as the pump impeller entrains an unbalancing mixture of liquid and vapour. Pumps operating on a largely static system are most susceptible to this. It is less likely to be an issue if the system is mainly friction [2], The higher the static head, the smaller the margin for speed error. Partly because of this sensitivity, Variable speed drivers [VSD] are questionable on high static head systems They are much better suited to high friction systems, where their strengths become overwhelming (Fig. 10.3).

## Speed too high

Again, with high static systems, the pump system curve intersection point is very sensitive. So operating at a slightly higher speed brings a large increase in flow and power. It also means that the system intersects with the pump curve at flows increasingly in excess of its design flow (Fig. 10.4).

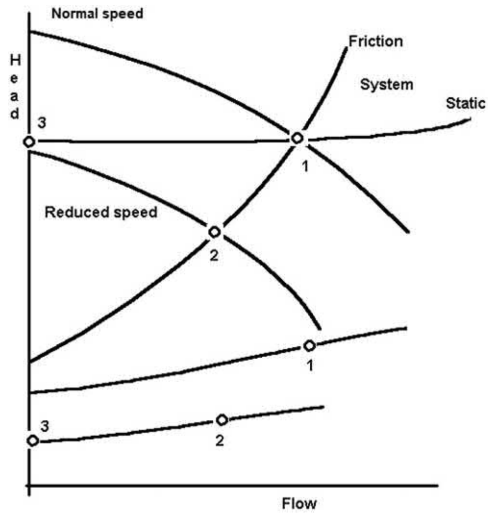



**Fig. 10.3** Effect of speed reduction depends upon system characteristic.

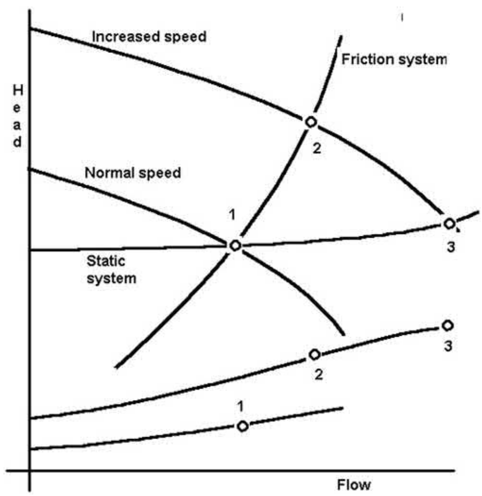



**Fig. 10.4** Effect of a speed increase depends upon system characteristic.

Any operation at above-design flow is associated with increased vibration, but this effect is magnified when the pump gets there by operating at higher speed.

As before, operation on a frictional system can be much less sensitive – this is a further strength of variable speed drives.

## Internal drag or rubs

Sometimes a pump's power can appear high, although all other parameters are acceptable. This can often be traced to internal drag within the pump.

This may be expected, although unwelcome. For example, the power absorbed by dual mechanical seals can be surprisingly high. On a small pump, this may amount to a reduction of perhaps five efficiency points. Such pumps will be difficult to turn over by hand. Pumps of up to 10 kW – absorbed powers might be vulnerable to this condition. This should not be too much cause for concern. It is inherent in the seal design. While this drag also exists on larger pumps, it amounts to a lesser percentage of the overall power and is less likely to be seen as a worrying symptom.

Modern single stage overhung impeller pumps rarely have wear ring rubs although older pumps that have been repaired many times might be susceptible. This is because register fits in the assembly become worn, allowing wear ring contact. This is more likely with commercial pumps since the clearances will be sized for high efficiency. API610 type process pumps have more liberal clearances – the concern being more with reliability than efficiency – hence rubs are less likely with this class of machine.

Single stage overhung pumps equipped with open or semi open impellers are prone to rubs from catching on the casing. The impellers are set with axial clearances in the region of 0.015″, but often there is some 'crush' in the assembly. This materialises when running as a slight reduction in clearance. There are strong motives to set this clearance towards the lower limits, because it can improve the pump performance. As with shrouded impellers, those that are large in diameter, and relatively narrow *[i.e. water passage width is less than 8% of diameter]* are most susceptible. In a practical sense, large diameter impellers always pose more difficulties than small ones, due to the greater difficulties involved in maintaining squareness.

Multistage pumps may rely upon the so-called 'Lomakin effect' [Appendix F] to centralise their rotors when running. The shaft slenderness ratio allows the shaft to adopt a catenary shape when static and this may entail slight contact between the rotor and the casing at some points [usually mid span]. However, when running, the pressure differentials acting over the internal wear rings and bushes drive the leakage flows, which then create Lomakin restoring forces to centralise the rotor. Such pumps may be quite stiff to turn by hand, due to the slight contact or rub. However, once running, contact is avoided, so no internal drag effects would be felt.

All multistage pumps are capable of generating large axial hydraulic thrusts. Some configurations attempt to counterbalance these thrusts internally *[the 'back-to-back' configuration]* Others, having so-called 'stacked rotors', make no attempt to dilute the thrust, and allow it all to be absorbed by an external thrust bearing. This approach is restricted to relatively small pumps.

Medium and large sized stacked rotor pumps need some form of internal thrust compensation. A whole family of devices exist – ranging from the straight balance drum or piston, to the balance plain disk.

These devices are usually located at the high-pressure end of the machine because this makes it easiest to channel discharge pressure into the area.

Device [6]; [*balance drum]* relies only on an **axial** clearance creating a true piston effect in which the compensation depends on the piston diameter – all other things being equal. Clearly, this is a non-contacting device and this has attractions.

At the other end of the scale is device type [1] *[balance disk]*. This relies upon high-pressure liquid forcing the disk apart and flowing through a **radial** clearance. Clearly, this can be a contacting device if there is not enough pressure to maintain separation of the faces.

Between these two extremes is a whole family of devices with different nuances. However, those, which rely in whole or in part on axial clearances, are susceptible to internal rubs. This possibility may be extremely remote. Rubs would not occur in operation unless there was insufficient head **difference** across the device to keep the faces separated.

- This would happen if the pump were operated at so high a flow that the generated head was not enough. Typically, when the flow is so

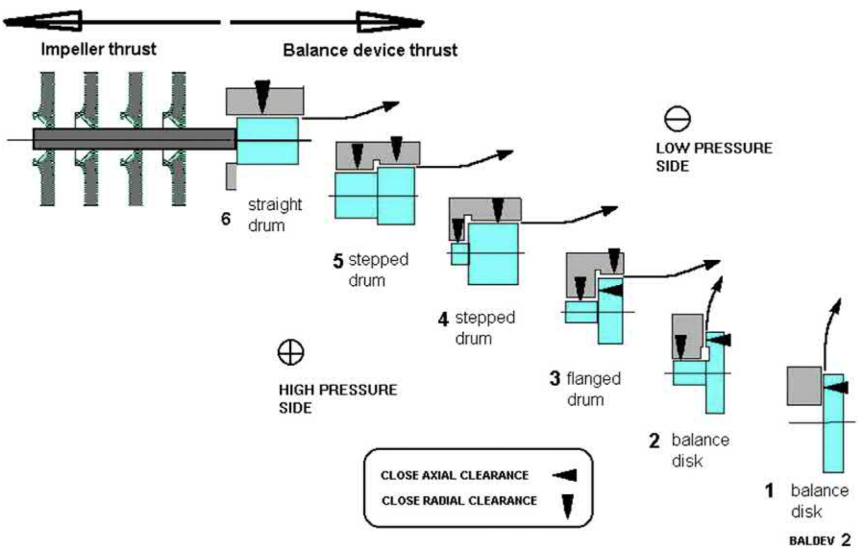



**Fig. 10.5** Hydraulic balance devices range from those on the right with purely axial gap faces [drums, pistons] to those on the left with purely radial gap faces [disks].

high that head is less than 50% of the bep head, and then there will be a risk of balance disk faces contacting.

- Another way that the difference could fail to be established is if there is problem with the line [*often called the balance line*] that connects the pumps internal balance chamber with a low-pressure source. That may entail discharge to atmosphere, discharge to a suction vessel, connection with the suction pipe or in internal connecting pipe. Any restriction in this line will cause a pressure backup that simultaneously helps reduce the differential head acting across the compensating device. This will tend to destroy the axial balance in favour of the impeller thrust and result in face contact. Interestingly, the balance line flow can be used as a general measure of all the internal clearances (Fig. 10.6).
- It is worth noting the last problem can also affect balance piston configurations, since they also rely on the level of differential pressure to achieve axial compensating balance. If this differential is reduced, due to pressure backup, then the compensations are reduced and the thrust bearing is more loaded.

One great strength of balance piston devices is that normally there is no way that the piston can contact with its stationary liner. However, were it to do so, then the very large contact area would exert a considerable rub or internal drag. That is one reason why balance pistons have a heavily serrated surface – it reduces the contact area very considerably. It also reduces the wasteful leakage, which, in a balance piston, tends to be higher than with the balance disk family. Balance pistons would normally only contact if during assembly, the rotor was incorrectly positioned in the hydraulic bundle. Rubs are most likely to arise during shop build, or start-up. Once up to speed, the Lomakin forces tend to be so strong that they centralise the piston in the sleeve.

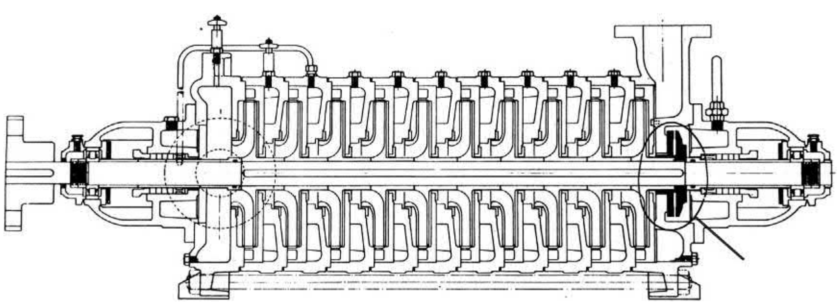


**Fig. 10.6** Location of balance device at high pressure end of pump. Illustration: Mackley Pumps.

## Type of rubs

If there is evidence of serious rubbing either during the pre-start checks or during start-up, then the pump should be dismantled for checking. Rubs usually fall in to three distinct categories.

The rotating component shows evidence of contact at one spot, but the stationary component shows contact on all or most of its periphery.

These symptoms are typical of:

- Shaft that is bowed
- Components that are not concentric with the shaft
- Mass unbalance in the impeller [Debris]

The rotating component shows evidence of contact on all or most of its periphery but the stationary component shows contact at only one spot.

These symptoms are typical of:

- Rotor is not correctly aligned
- Excessive pipe loads
- Low flow operation [high radial loads]

Both stationary and rotating components show contact on all or most of their periphery.

This is typical of:

- Loose or worn bearings
- Large amounts of abrasive debris passing through fine clearances

## Wrong rotation

### Axial flow pumps

Fig. 10.7 shows the performance of an axial flow pump running in the wrong direction of rotation. The system curve is also shown. Like most axial flow pump applications, the system curve consists almost entirely of friction losses.

The pump vibration and noise levels increased significantly, but the machine still pumps, and would generate a flow in the system. In the example shown, the pump would displace about 6400 $m^3/h$ compared to the design flow of 7250 $m^3/h$. This shortfall might not be immediately obvious and the pump could run some time before the problem became apparent. If despite verifying the driver rotation, the problem still persists, a deeper root cause may exist. With pure axial flow pumps,[1] it may well be possible to machine the casting the wrong way round. This will ensure wrong hydraulic rotation with the correct mechanical rotation (Figs 10.8 and 10.9).

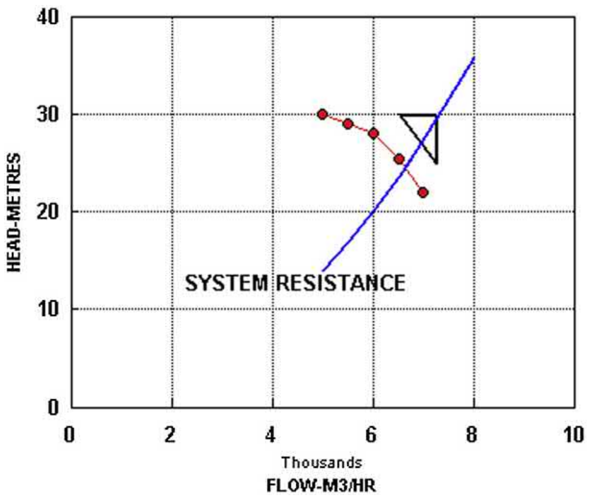



**Fig. 10.7** Characteristic of an axial flow pump where the impeller casting has been machined to run in the wrong direction.

**Fig. 10.8** The correct rotation for this impeller is clockwise as viewed. Generally, the convex blade surface faces the approaching flow.

## Mixed flow pumps, centrifugal pumps

There is much less scope for the manufacturer to incorrectly machine or even incorrectly cast these sorts of impellers but it is still possible. Therefore, if a driver rotation check proves correct, it is worth trying to glimpse either the impeller inlet or outlet and doing a reality check.

This is most unlikely if the pump is single stage. It might be possible if the pump is a back-to-back rotor [where the impellers are 'handed' for rotation] and spare impellers have been bought – untested.

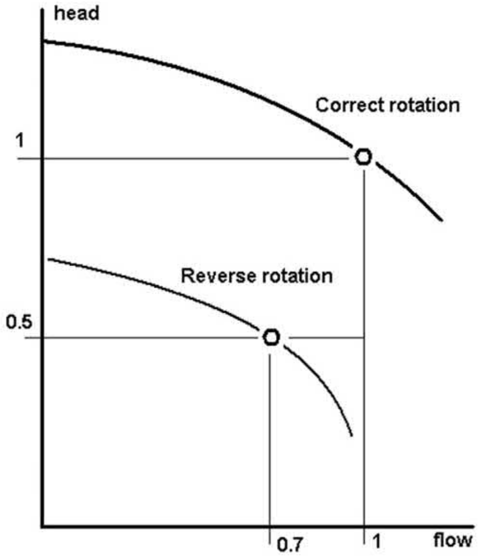



**Fig. 10.9** (above right) Performance of centrifugal pump when run in reverse, compared to correct rotation. This is broadly applicable up to specific speeds of 2000.

## Cavitation [see also Appendix E]

### Is the suction pipe exposed to solar heating, or heating from other sources?

In some industries, liquids are pumped while close to their vapour pressure. In the extreme, this means that with a little extra warming, the liquid will flash to gas. The pump is then presented with a liquid/gas mixture [Few centrifugal pumps are happy with more than 4% or 5% gas by volume. Such warming can occur if the suction line is long and can be exposed to solar heating.].

At a more basic level, if the margin between NPSH [A] and NPSH [R] is very small, then these sorts of circumstances can push the pump into cavitation noise emission, without cavitation damage becoming too serious.

If the pump suction line leads from [say] a cool pond and travels a long distance where it can be subject to solar gain, then the liquid temperature rises. Cool liquid can carry more dissolved gas than when it is warmer, so the gas is released. This is more acute if, in line with good practice, the suction velocities are quite low. If too low the gas accumulates into slugs of gas. As above, most pumps dislike such two-phase liquids. If the two phases remain well mixed then the pumps can tolerate this better.

An extreme case is on record of pumps handling LPG. Though the liquid was homogenous at the supply vessel, the accumulated solar gain in the long suction line raised the liquid vapour pressure and caused flashing in the suction line. Sun shading solved that particular problem.

### Is the barometric pressure unusually low and you are in northern hemisphere?

Look outside! If it is raining, the chances are that the barometric pressure is below average and in marginal cases, can trigger cavitation.

## Cavitation noise

In the section on Pump Basics [section 3, 4], the fundamentals of cavitation in pumps are described. So far as the pumping system as a whole is concerned, the most obvious symptom of cavitation is a reduction in system flow. However, another outward symptom is of noise and vibration.

Conventional wisdom has it that cavitation is accompanied by a significant increase in noise emissions. As often occurs, there is rather more to it than that.

In Appendix E, it is mentioned that the anatomy of a cavity reveals at least two major zones: 'sheet cavitation' and 'cloud cavitation'. It turns out that the stable 'sheet cavitation' component is relatively quiet. However, the unstable 'cloud cavity' that forms along the downstream boundary of the sheet emits much more noise. That's because this is where the cavity is collapsing, and where implosions are taking place.

It was mentioned that the length of sheet cavitation grows when the pump operates at flows greater or less than the inlet design flow.[2] In these cases, the liquid flow angle and the impeller blade setting angle differ, this enhancing sheet cavity growth.

It grows proportionately more at high flows than at low flows. The reason for this is bound up in the internal fluid velocity vectors, but it stems from the fact that liquid velocities experienced by the impeller are greater at high flows than at low. A longer sheet cavity usually results in more intense cloud cavities. On this basis alone, more noise would be expected at either side of inlet design flow. However, a further factor is that noise emission also increases with liquid velocity.

The net result is that, taking noise emission at the inlet design flow as a benchmark:

- As flow is reduced, down to [say] 40%, the cavitating noise emission does not increase very much. In fact, it can appear to increase slightly in certain circumstances.

- Below about 40% [in the inlet backflow regime] the noise takes on a different characteristic. If NPSH [A]/NPSH [R] approaches 1, then surging noise at frequencies of 0.5–2 Hz may be experienced
- As flow is increased above the benchmark, cavitating noise emission intensity increases very substantially.

To complicate matters further, even at a fixed flow, the noise intensity depends upon the ratio of *NPSH [A]/NPSH [R]*. As this value reduces to a value of 1.0, the sheet [and associated cloud] cavity grows longer and so emits more noise. At values much less than one, the impeller passages 'choke' and the neat distinction between sheet and cloud cavity becomes lost. Equally, the noise intensity will most probably reduce.

For the most part, traditional NPSH [A] curves reduce in value as flow is reduced. Similarly, NPSH [R] curves increase with flow. This would suggest whatever the [NPSH [A]-NPSH [R]] margin is at design flow, it will be even greater at low flow.

It may not be immediately obvious why, despite this apparent extra margin, a pump can become noisier as flow reduces. This should be even truer if the NPSH [A] curve contained a good deal of friction in its makeup.

First a note about the term 'traditional NPSH [R] curves: There is a grudging agreement between manufacturers and users that significant cavitation exists at that NPSH [A] where the pump total head has decayed by 3% [In multistage pumps this definitions modified to say 3% of the head per stage.] [Appendix E].

However, cavitation does not just suddenly appear below this line and just as suddenly disappear above it. At a fixed flow, cavitation commences with barely visible bubbles. This takes place at NPSH [A] much higher than the 3% decay line. This progressively develops as NPSH [A] continues to be reduced and on the 3% head decay line there will already be extensive vapour within the impeller. The line defining commencement of the 'barely visible bubbles' lies substantially above the 3% line. This line is often referred to as the Cavitation Inception line. For most practical purposes, this also defines the commencement of significant cavitation noise emission, though strictly speaking noise begins at even higher levels of NPSH than cavitation inception.

While the 3% curve tends to continually fall as flow is reduced, the inception/noise line demonstrates very different behaviour. Firstly it comprises of two parts:

- The first part has a horseshoe shape centred more or less on the b. e.p at full impeller diameter [*see* Fig. 10.10]. The trough of the horseshoe corresponds with the optimum inlet flow condition. Here the liquid flow angles best match the impeller vane setting angle. At greater or lesser

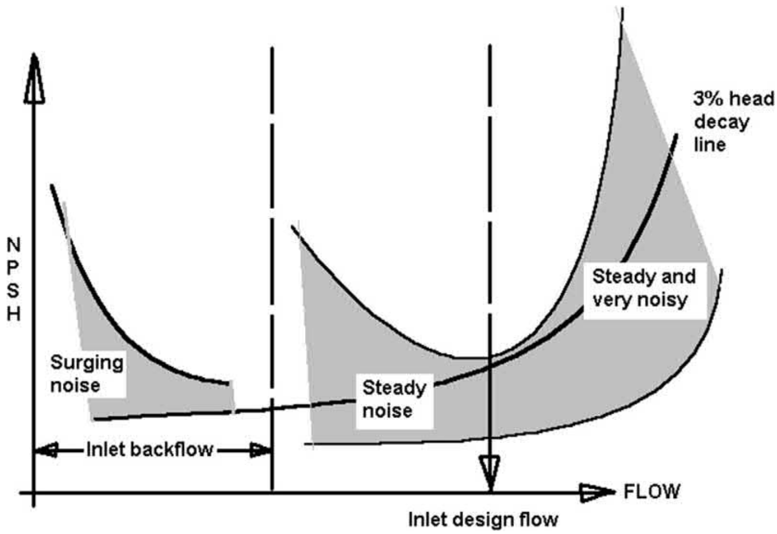



**Fig. 10.10** Cavitating noise regimes.

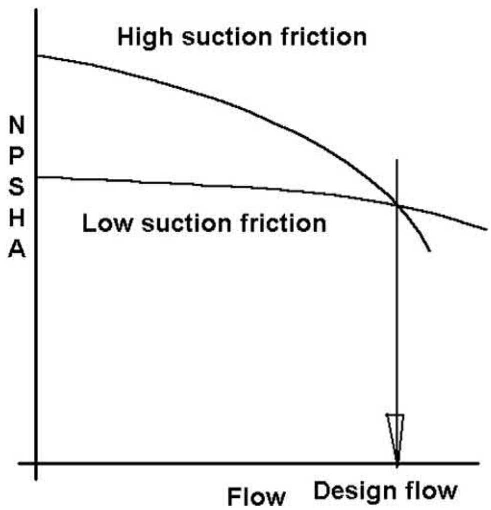



**Fig. 10.11** NPSH [A] curves comprised largely of friction, will tend to escape the noise zones, compared to those with little friction.

flows, there is some mismatch and this causes an additional increase in the noise emissions.

- At higher flows, the inception and 3% curves converge. At low flows they diverge. This difference is amplified as pump speed increases. It is also more pronounced in pumps of relatively low head, due to the comparatively short blade length and low inlet-to-outlet diameter ratio. Pump designers describe such pumps as 'High Specific Speed' machines.

Below a certain flow, the curves converge and this is associated with a collapse of the inlet flow field known as 'inlet back flow' *see* Appendix D. The risk of noise here will depend on the shape of the site NPSH[A] curve. Such curves that include a large frictional component will be less susceptible (Fig. 10.11).

Hence over a certain low flow band, the inception NPSH [R] may exceed the NPSH [A]. The pump is then susceptible to increased noise in this zone. That does not automatically mean that the cavitation is damaging to the impeller. The cavitation may be collapsing in the inter-vane space, rather than onto the blade itself.

## General pump noise

The flow picture in an impeller is not the uniform pattern often imagined. In fact, the flow tends to separate into two conceptual zones, the so-called **jet** and **wake** regimes. Such segregation is caused by the way work is done on the liquid as it passes through the impeller passages, and is inevitable. This results in an irregular flow pattern leaving the impeller. The collector that wraps around the impeller *[volute or diffuser]* experiences the same velocity and pressure changes as each impeller vane passes by any fixed point, creating a sort of siren effect. This picture blurs with radial distance from the impeller rim until at [typically] a distance 20% greater than of impeller radius, the distortions 'mix-out' to a practically uniform flow field.

While overall noise emission is a practical symptom, it can tell us very little about the conditions of the pump. It is more useful to disassemble the noise signal into component emissions at a range of different frequencies.

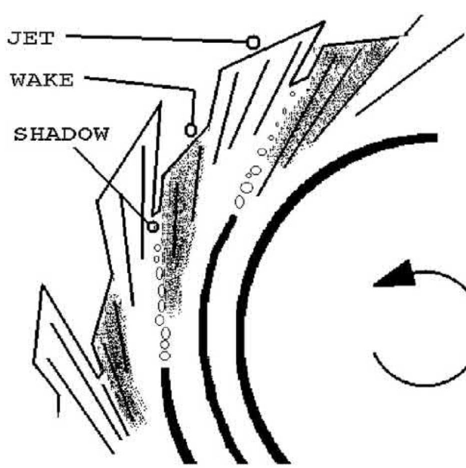



**Fig. 10.12** Idealised flow picture at impeller outlet.

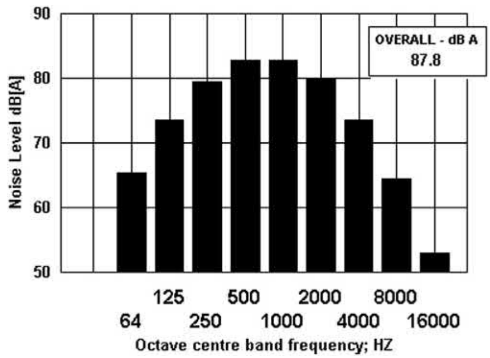



**Fig. 10.13** Typical analysis of pump noise emission broken into octave bands [*Refs 8 and 14*]. This generalised signature can be used as a benchmark to help pick out the root cause of abnormal noise levels.

This can be easily accomplished using an electronic measuring device that is widely available nowadays. It is common to break down the noise signal into octave bands, *see* Fig. 10.13, though finer resolution [half or third octave bands] is possible. Nevertheless, whole octave bands are the most widely used and can yield useful analyses.

Fig. 10.14 attempts to portray the major influences on pump noise. Well behaved pumps operating at or near to their design flow will tend to emit most noise in the octave band containing their vane passing frequency.[3]

The noise level at that frequency is largely dictated by the impeller tip speed. Note that the levels will be artificially exaggerated if the volute cutwater or diffuser bore radius is less than 2% larger than the impeller diameter. This is because the flow distortions mentioned above have had

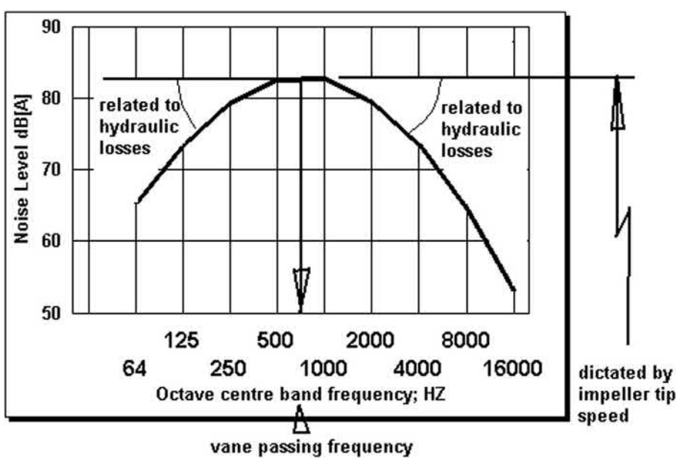



**Fig. 10.14** Influences on shape and scale of pump noise emission.

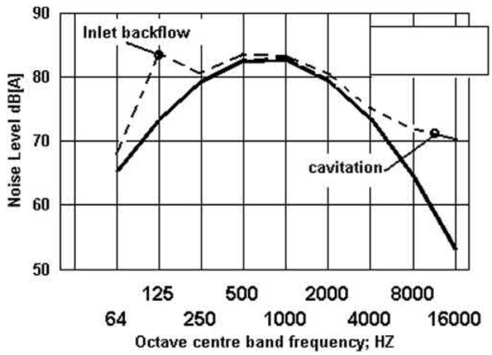



**Fig. 10.15** How some pump problems disclose themselves in the pump noise signature as departures from a standard benchmark.

hardly any chance to mix out and the volute or diffuser bear the full brunt of it. At greater or lesser frequencies, the noise emission usually reduces.

Creating and using this idealised noise curve as a benchmark, can then give clues as to the nature of the abnormal pump noise. It turns out that pumps operating at low flow and within the backflow regime often display a large 'spike' when compared to the ideal. This spike occurs in the octave band containing the inlet blade passing frequency. [Ref 10]. Typically, this will be in or around the 125 Hz octave band. On the other hand, pumps that are suffering cavitation show departures in the very high frequency zone.

These tendencies can often be used with other indirect evidence to help pin-point the root cause of abnormal noise, as well as identify the causes of performance departures (Fig. 10.15).

It is worth noting that some liquid passages within a pump can behave acoustically like 'organ pipes'. A small periodic excitation may bring the passage to resonance. This can significantly amplify the effects. This behaviour not only depends on the pump geometry, which is generally fixed, but also the liquid properties. So a variation in liquid properties might bring about an unexpected change in resonant behaviour [Ref 32].

## Vibration

Vibration can be sensed [by touch] and an opinion formed as to its severity. With [considerable] experience, personal acceptance standards can be established. However, these are practically impossible to convey to a second person in a consistent manner. The only way round this consistency problem is to use dedicated devices for vibration measurement.

A choice of measurement parameter exists, vibration displacement, vibration velocity or acceleration. All of these are in current use and acceptance standards exist for each. Acceptance standards mostly have little analytic basis, but are rather consensus opinions among machinery experts [users, manufacturers] based on their accumulated experience.

Simple measurement devices monitor the vibration motion in real time and convey the behaviour as an overall value, *see* Fig. 10.16. The real time vibration signal taken from a pump is rather complex and normally yields little direct information. Any complex signal can ultimately be broken down into a series of simple sine waves having different frequencies. Examining the signal in such a way is called Fourier analysis and is a widely practised technique, using specialised instrumentation (Fig. 10.17).

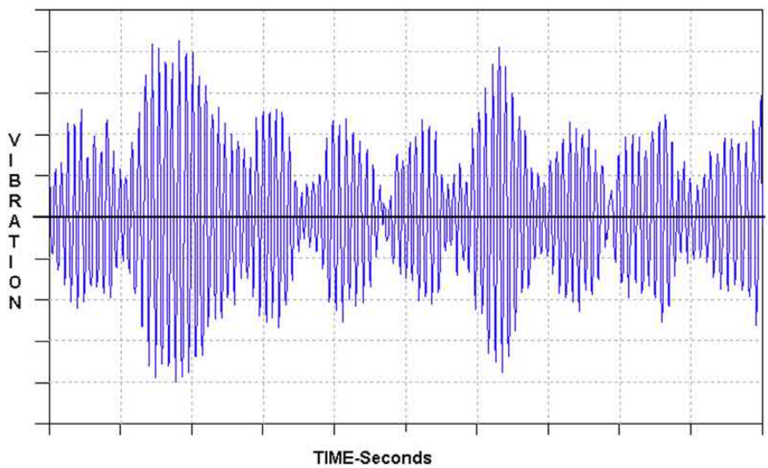



**Fig. 10.16** Bearing housing displacement, measured with respect to time. Any information within these data is less than obvious.

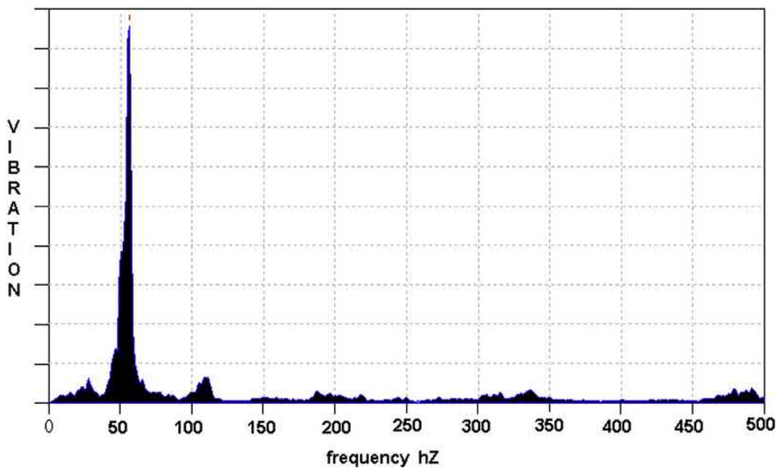



**Fig. 10.17** A Fourier analysis of the raw signal reveals that the biggest component of the overall vibration occurs at 50 Hz, [*or running speed in this example. The most common cause for this is mass unbalance somewhere in the rotor.*] In contrast, all the other components at, say, 110, 200, 325, and 475 Hz contribute little to the overall vibration level.

Without the capability of subsequent Fourier analysis, overall vibration data is of little value. True it may be used as a measure of long-term trends. It can also be used to put the overall levels into context with similar machines. Nevertheless, Fourier analysis permits a much deeper insight to the mechanical and hydraulic state of health. As has been mentioned before, all generalisations are dangerous, and in respect of vibration analysis, this is particularly true. However, some behavioural patterns are beyond much doubt. The common ones are listed in Table 10.17.

Vibration can have several different root causes. Common vibration problems in pumps can be simplified into two categories (Fig. 10.19):

- Where a large force within the machine 'shakes' the pump, [and or piping]
- A small force within the machine excites some part of the pump set to resonate

**Table 10.17** Some common vibration cause and effect.

| Cause | Vibration symptom | Remarks |
|---|---|---|
| Mass unbalance in rotor | 1× running frequency | Can often be corrected by external trim balance |
| Pump to driver misalignment | 2× running frequency | — |
| Hydraulic 'cam' effect[a] | 1× running frequency | Can rarely be corrected by external trim balance [*see* Fig. 10.18] |
| Inlet backflow | In the range of 0.5–2 Hz | Frequency too low for normal vibration meters. Severity decreases as suction pressure is raised |
| Impeller too loose on shaft | 1× running frequency | Impeller should ideally be a shrink fit to shaft |
| Impeller diameter too large for casing[b] | Vane pass frequency | May be accompanied by abnormal noise levels |
| Pipe work poorly supported | Very low frequency – up to 30 Hz | Energy will be spread over large range, little or no 'spike' in Fourier |

[a]Exaggerated Illustration of the 'Hydraulic Cam' effect where the hydraulic centre is eccentric to the mechanical centre, see Fig. 10.18.
[b]High vibration occurs because the uneven flow field discharging from the impeller has no chance to 'mix out'. The collector experiences associated hydraulic excitation. Increasing the clearance between impeller and collector allows the exiting liquid streams to merge and reduce the levels of excitation. This effect can be quite dramatic. Fig. 10.19 shows test data taken from a large between bearings process pump.

## Where a large force within the machine 'shakes' the pump, and possibly piping

### Mechanical issues

#### *Static and dynamic rotor balance*

Balance is the most obvious cause and is the most easily solved. It shows as vibration with a frequency of once per revolution.

However, there are hydraulic causes, which will also demonstrate vibration at the same frequency. If the impeller has been machined [or re-machined during repair] such that the mass centre differs appreciably from the hydraulic centre, then a **'hydraulic cam'** is created, *see* Fig. 10.18. This appears largely at once per revolution when the pump runs, even though it check balances correctly in air. Since the impeller is mechanically 'in balance', this problem is very difficult to balance out. Often the only solution is to weld up the impeller bores and re-machine.

#### *Wear ring rubs*

Wear ring rubs may occur in older machines that have a relatively slender shaft and/or close component fits. Over time, the fits deteriorate with each repair and eventually there is a risk of rubbing due to misalignment. In most cases, misalignment shows as contact marks on 360° of the impeller wear

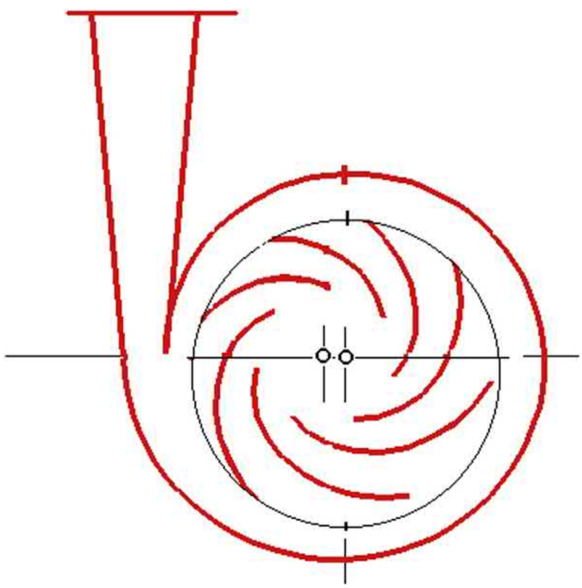


**Fig. 10.18** If poor machining places the hydraulic passages eccentric to the mechanical centre, pump will exhibit operating unbalance symptoms when operating, even if apparently perfectly balanced when checked in air. This effect cannot easily be balanced out. Normally re-machining is required.

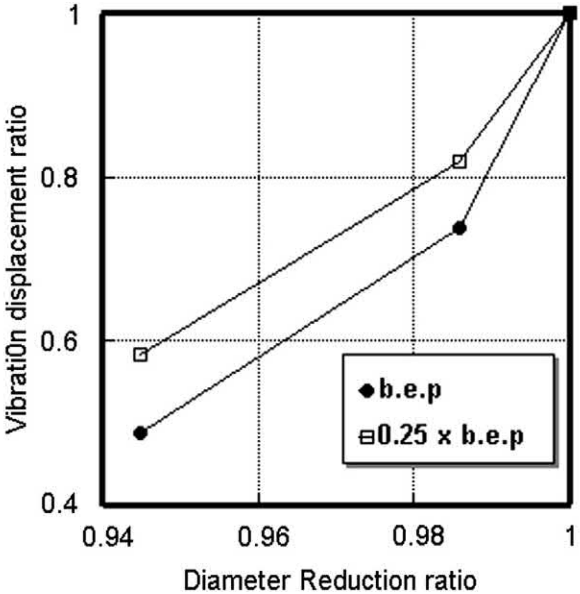



**Fig. 10.19** Effects, on vibration, of increasing clearance between impeller outside diameter, and volute cutwater. Achieved by impeller diameter reduction. Small increases in gap can produce a surprising reduction in vibration.

ring, but only a part of the case wear ring. Vibration occurs at a wide range of frequencies, many at harmonics, but rarely at an integer of running speed.

Re-machining of the fits is the obvious answer, along with perhaps larger wear ring clearances where permissible [small pumps], or maybe different materials such as non-galling/non-metallic.

### *Bent shaft*

Most likely appears as vibration at running speed.

### *Alignment*

This can appear as vibration at frequencies of running speed and twice running speed.

### *Loose impeller fit*

With the drive to achieve lower and lower vibration levels, the practice of shrink fitting impellers to shafts has grown. There is no doubt that slack impeller fits contribute to poor vibration on larger two-pole single stage machines, even though balanced well as individual components. Furthermore, there is always the temptation, during assembly, to 'ease' the fit in order to improve assembly. Poor impeller fit also appears as vibration at running speed. However, other factors [described above] can also produce the same symptom. The downside of this process is the difficulty of removing the impeller. This may be acute if the impeller is large in diameter, a situation that is exaggerated if the impeller – shaft materials may tend to gall.

## Hydraulic issues

### Impeller design

Certain impellers can be an inherent source of vibration. The flow leaving an impeller is not as uniform as most people might imagine. Generally each impeller passage is obliged by the laws of nature to issue a high velocity stream and a low velocity stream-often defined as a jet and wake.

As a generalisation, the more impeller blades there are, the smaller the difference between the two velocities. These velocity differences have associated pressure pulsations because of their interaction with the casing cut water [or diffuser blades]. In some machines, it is not practical to have a large number of blades; for example, sewage pumps, and some other solids handling pumps. These designs may therefore have an inherent vibration at a vane passing frequency.

### Impeller/casing clearance

The impeller/casing interaction described above largely depends upon the gap between the impeller rim and the casing. The bigger the gap, the weaker is the interaction, and the lower the pulsation/vibration levels (Fig. 10.20).

A further complication is that the levels of pulsation are also dependent upon where the pump is operating on its curve. Near to the best efficiency flow, expect the pulsation levels to be low. As flow reduces, the pulsation levels increase. So, in a relative sense, the most favourable arrangement from a vibration standpoint is a pump with a reduced diameter impeller operating near to its best efficiency flow. The least favourable is a maximum diameter impeller running at minimum flow (Fig. 10.21).

### Low flow operation

The preceding paragraph described how low flow operation would produce the highest levels of vibration as far as the impeller outlet conditions are concerned. In addition, impeller inlet flows can collapse adding to the problem. This is covered in more detail under Appendix D. At flows below *about* 25%, depending on pump size and speed, the impeller flow pattern

**Fig. 10.20** Conceptualized velocity distribution leaving an impeller at its best efficiency flow.

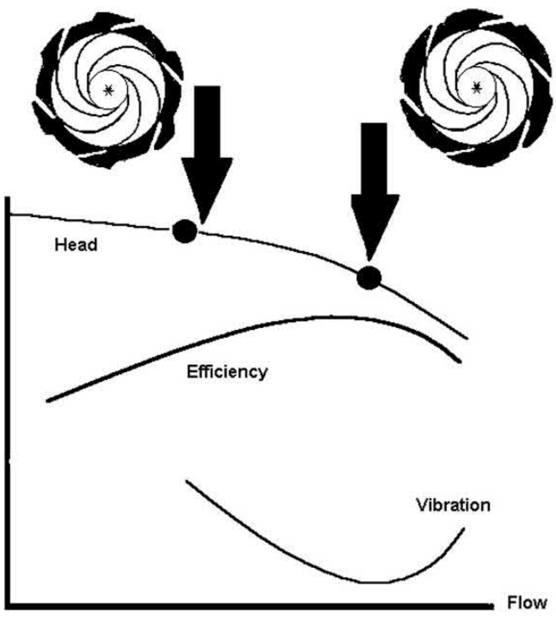



**Fig. 10.21** Flow related change in velocity distribution leaving an impeller.

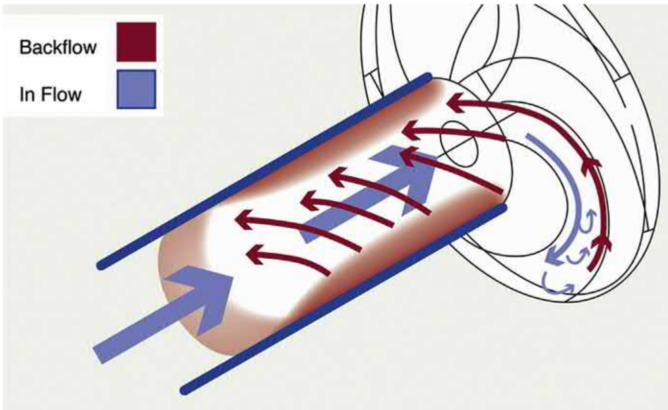



**Fig. 10.22** Inlet backflow pattern conceptualized.

collapses and some flow actually **leaves** the impeller inlet. To escape the impeller it has to travel in excess of the impeller inlet peripheral velocity. This bestows considerable kinetic energy on the liquid. If this high energy, swirling liquid interacts with stationary protrusions in the casing of the pump, or in the suction pipe work, then the resulting torque reaction will induce vibration, If the ratio of NPSH [A]/NPSH [R] is close to 1.0, then cavitating surge may occur *see* Appendix D (Fig. 10.22).

## A small force within the machine excites some part of the machine to resonance

We have seen above that the pump internals can generate cyclic forces from a number of sources. Though relatively small, they can sometimes excite surrounding components to resonance and possibly even fracture. Typically susceptible areas are:

- Bearing housing and bearings
- Impeller shrouds, if too much material has been removed in rebalancing
- Piping [On-skid, as well as system pipe]
- Seal components
- Bedplates mounted on poor foundations

These are all structural elements of the machine, and typically possess little mechanical damping to vibration. So only, a small excitation is needed in order to produce quite tangible results.

These elements tend to vibrate at one fixed frequency, plus harmonics. The pump excitation frequency varies with speed. So on start-up or run down it is not unusual for these elements to be briefly excited as the pump excitation frequency approaches, then coincides, then exceeds the response frequency. In rare cases, the response frequency corresponds to the running speed excitation and problems ensue.

A common case is of auxiliary piping exhibiting significant response at running speed. This is easily solved by adjusting or adding pipe supports in order to shift the response frequency. However, if more major elements are involved, such as bedplate pedestals or top-plates, the problem is more difficult to solve. And of course if the pump is driven by a variable speed driver, the scope for coincident frequencies is much higher.

## Pump casing gets warm to the touch

Fig. 10.23 illustrates that not all of the power input to a pump appears as work in the liquid. Some of it, [the **in**efficiency], appears as heat, sound, vibration. Of most concern is the temperature rise of the liquid entrained in the casing. This thermal temperature rise relates to the pump flow, and the inefficiency at that flow.

At zero, or small flow, the heat output is less than the heat input, so the liquid entrapped in the casing gradually gets hotter.

Low power pumps may heat up slowly, at reduced flow whereas high power pumps may churn and vapourise the entrained liquid in a few seconds.[4] Fig. 10.24 illustrates how the temperature rise relate to flow [see Appendix H].

Some multistage pumps are equipped with hydraulic balance pipes. The flow they pass helps to reduce the internal temperature rise by dissipating

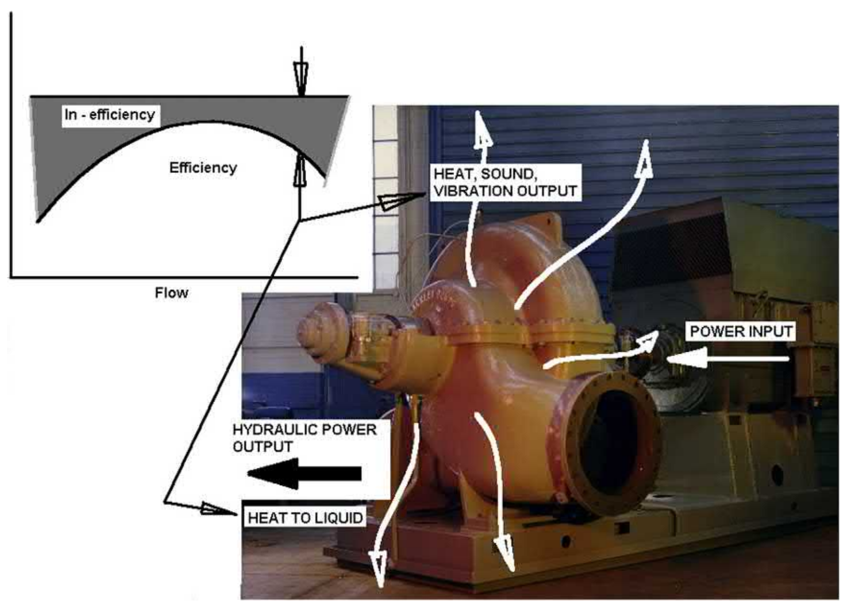



**Fig. 10.23** Pump inefficiency leads to heating of the pumped liquid. Photograph: Mackley Pumps.

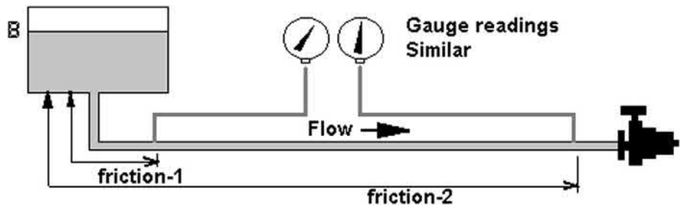



**Fig. 10.24** Temperature rise as a function of pump flow.

the heat away from the pump body. Some pumps even have a permanent leak-off feature[5] which means that a fixed amount of liquid is always bled off from the main flow and back to the source [some distance away] *see* Fig. 10.25. The amount is sufficient to never let the temperature rise exceed some set value. This is an effective protection against vapour locking. Even if the discharge valve is fully closed the liquid cannot exceed the set value. However, since the power put into this flow does little or no useful work in the process, it represents a total loss, so the overall efficiency of the pumping system suffers [Multistage pumps are sometimes equipped with a balance line that is not internally re-circulated, but fed back to suction or source. In this case the balance line flow constitutes all or part of a re-cycle flow.]. If the pump power is quite low – say less than 100hp, then this loss can often be tolerated. If the power exceeds this, then some more elaborate arrangement needs to be made, such as Automatic Recycle Valves. They perform in a similar way, by diverting flow as forward flow is reduced. The main benefit is that they only operate as low flow is approached. At normal flow, they are inoperative, so efficiency is maintained.

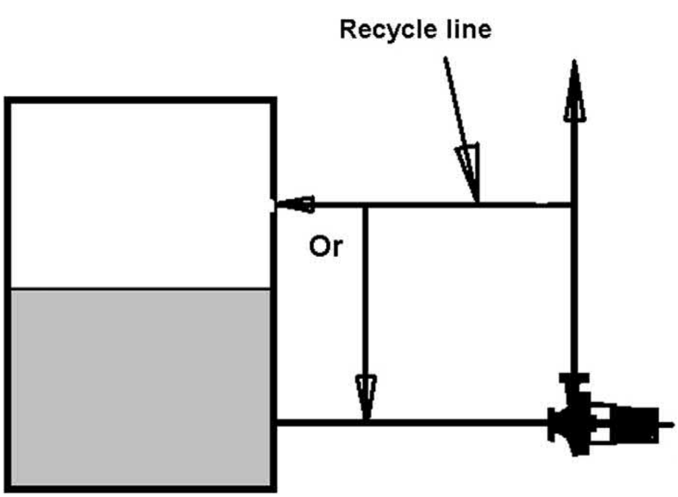



**Fig. 10.25** Permanent re-cycle (by-pass) line prevents low flow temperature rise.

The flow at which the liquid temperature rise becomes undesirable is defined as the Thermal Minimum Flow. This is always less than, and should not be confused with, the Minimum Continuous Safe Flow.

## Run dry?

Dry running [see also Table 10.2] can occur for many reasons. It is often the easiest problems to diagnose, because the symptoms are all too obvious. Most typically, there will be evidence of rubbing contact in the fine clearances. This may become quite severe with some material combinations. There will be evidence of frictional heat and blackening of the components. With other more tolerant materials [example] the damage could be slight. Cracking or crazing of the mechanical seal faces [if fitted] might arise.

If the shaft system is very rigid, then the pump can often tolerate the 'vapour locked' conditions described above, without harm. The only susceptible item would be the mechanical seals, where fitted. Pumps having special seals and lubricant can tolerate extensive dry running quite happily.

Pumps that can and will operate in a 'dry run' condition quite satisfactorily are:

- So-called self-priming pumps
- Heavy duty end suction/overhung shaft pumps
- Cantilever shaft pumps

Others are less likely to tolerate such abuse. Typically this will include:

- Most multistage pumps [They rely on fluid passing through wear rings to create a pseudo bearing property – the so-called *Lomakin* effect.]
- Older between-bearings pumps, because of their slender shaft and large bearing span [designed for packing]

If the pump relies upon the *Lomakin* effect to any large degree, then stator-rotor contact will result if this is lost. This is then often followed by seizure.

## Does the liquid contain free gas, or gas dissolved in it, but not chemically bonded with it?

Manufacturers of conventional[6] centrifugal pumps always caution against pumping liquid that contains 'significant' amounts of free gas in the liquid. Such a pumpage is often called a 2 phase liquid. There are at least two major categories, those possessing either:

- Free gas, or
- Dissolved gas

Gas can get into the fluid in a number of ways:

## Free gas

It may result from air leaks in the suction line if the pump is operating on suction 'lift', or the suction pressure is otherwise below atmospheric.

On the other hand, it may become accidently entrained from a free surface vortex close to the suction line inlet *see* Fig. 10.29.

Occasionally, gas may be deliberately introduced to the suction line in order to help mask the noise of cavitation and suppress its damage to some extent.

Free gas in the pumpage is rarely a deliberate part of the pumping process. The pump and its system will not usually have been designed on this basis and so performance will suffer.

## Dissolved gas

Dissolved gas, on the other hand, may be an endemic property of the liquid being pumped i.e. sour crude oil, raw sewerage.

On the other hand, process gas is often cleansed and filtered in equipment called 'scrubbers'. Here the unwanted gas constituents are either absorbed into the liquid physically, or by chemical reaction.

Occasionally, liquid has been stored in contact with a gas on its surface at a temperature/pressure that encourages absorption.

Dissolved gas evolves out whenever the liquid pressure is reduced sufficiently, in a similar fashion to Champagne when the cork is released! In the internal flows of centrifugal pumps, there are often sufficient local pressure gradients to encourage similar gas evolution. Inevitably, the liquid pressure does reduce, the closer it gets to the pump. This is because it experiences friction losses during the flow process. Dissolved gas coming out of solution is more likely to have been anticipated than free gas so the in the pump and system design would reflect this.

## Operation on a suction lift

Pumps that operate on a static suction lift are vulnerable to air or gas in the suction flow. The effect depends upon the basic pump configuration.

## Self-priming pumps [including assisted prime]

These pumps are designed from the outset with impeller passages that have a higher than normal tolerance to free gas. In such a case, free gas/air just slows down the priming rate.

## Externally primed pumps

Because the liquid pressure reduces with horizontal distance from the free surface, the volume of any free gas increases. At best, the free gas appears as a homogenised mixture and only segregates in the internal pressure fields of the impeller.

At worst, the gas agglomerates into 'slugs' of gas in the pipe leading to the pump. In the extreme, these slugs can cause the suction lift 'column' to collapse, thus de-priming the pump and leaving it running dry (Fig. 10.26).

When a pump starts and stops frequently, this is an excellent chance for liberated gases to slowly agglomerate, particularly if the suction line is long, or contains high spots or air trap loops.

While the liquid is flowing, air is most likely to become, or remain, entrained in the liquid. Upon shut down, the static pressure rises because the kinetic energy [velocity head] is regained. The gas is then less prone to come out of solution.

## What are effects of gas in the liquid?

Gas in the fluid has a modifying effect on the pump's performance characteristics. To understand why, needs a brief understanding of how centrifugal pumps work?

Fundamentally, pump impellers create a forced vortex within an enclosing casing. In achieving this, a pressure differential is created across each blade. Pump designers naturally want this differential to be moderately

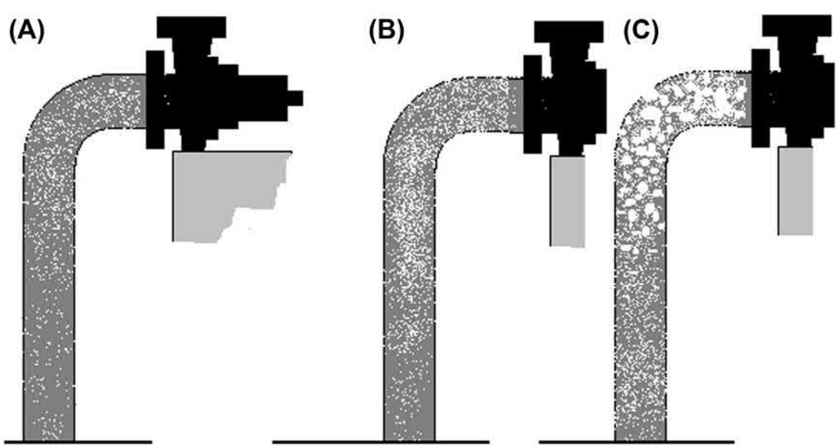



**Fig. 10.26** Effect of dissolved gas on priming; [A] Low gas content – no significant effect, [B]Medium gas content – slight effect on performance, [C] High gas content – pump on point of depriming.

high in order to accomplish size-effective machines. Often this results in significant pressure gradients across the passages of an impeller [at right angles to the passage flow]. The lowest pressures usually occur on the suction [or concave] surface of traditional designs. Here, pressures may actually become low enough for dissolved gas to evolve and collect. Furthermore, any free gas will also collect in these low-pressure areas (Fig. 10.27).

Gas agglomeration causes the passage areas to become partially blocked. This restricts the flow capability of the pump relative to its single-phase liquid performance. Furthermore, since free gas modifies the passage velocity distribution, the head generated by the machine is also reduced. In this respect, the external symptoms are very similar to cavitation, i.e. decay in overall performance.

At low throughput, the impeller passage flows are sufficiently disordered to generate several areas of high-pressure gradient. The resulting high gas concentration seriously inhibits the impeller's ability to generate and sustain the forced vortex, and so its generated head is significantly reduced. Fig. 10.28 shows a series of test data.

Around about design flow, the impeller passage flows are more organized. Whilst pressure gradients still exist, they are smaller in magnitude. Consequently, gas has a less disruptive effect on the head generation mechanism than at low flow.

To put the effects into perspective, 'conventional' pump impellers will not tolerate more than about 5% of gas by volume at the inlet flange. In this

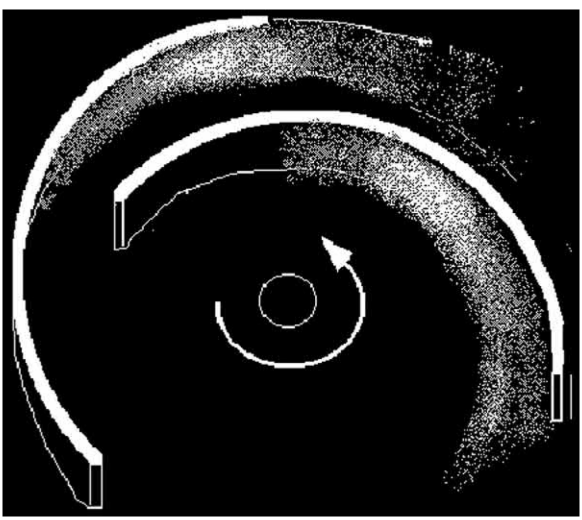


**Fig. 10.27** Gas tends to agglomerate onto the low pressure [or concave] surface of the Impeller blade.

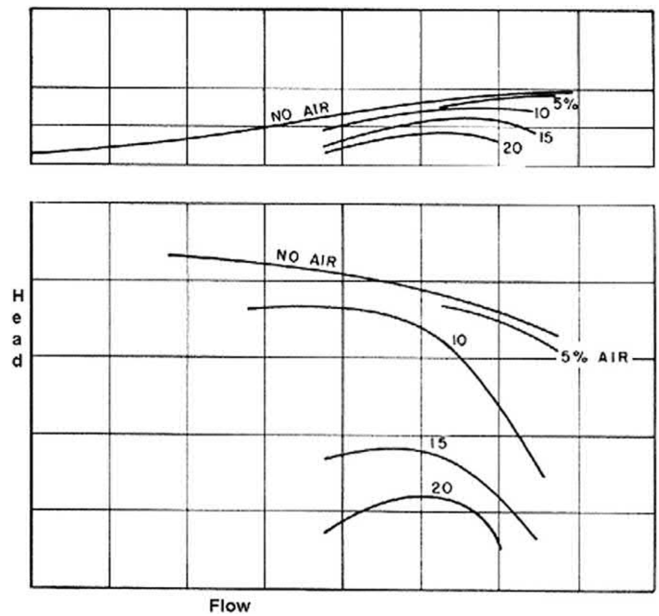



**Fig. 10.28** Effects of air content [see *Ref 16*].

condition, the head generated by the machine will have deteriorated by more than 10%. The effect of this would be more severe at low flow. For example, more than 15% head decay at 20% of design flow.

When the gas is finely dispersed in the inlet flow, any effects on reliability are minimal. The main effect is on performance output. If the pump operates on a system which consists largely of static head, *Chapter 5*, the overall effect on system throughput will be much more profound than on a system consisting largely of friction.

It is when the gas is presented to the pump in discrete slugs that reliability suffers most. Non-uniform liquid mass distribution occurs in the impeller with one passage handling more than another does. This can cause serious, if short lived, unbalance problems. The consequent vibration can accelerate seal face wear, but is not so likely to make much impression on the bearing life.

## How could air or gas enter the suction line at source?

The commonest cause is air leaks into the suction line when the suction pressure is sub-atmospheric. Every flange joint, or gauge connection represents an opportunity for air to enter the system. To combat this particular entry route, there is no substitute for common sense.

However, under certain conditions, air or gas can become entrained into the liquid even before it enters the suction pipe. If the free surface of the liquid is in close proximity to the beginning of the suction pipe, then air or gas can be drawn into the liquid though the mechanism of a surface vortex.

In the core of this forced vortex, the velocity increases and pressure decreases. This may permit a void to appear in the vortex core, and gas can come out of solution. In this void, air can also be entrained. This entrained air can then pass into the suction pipe where it can either:

- Homogenise, to a finely dispersed mixture
- Or it may accumulate into distinct 'slugs'

Such vortices are easy to spot on an accessible free surface such as a lake or river, but in an enclosed system such as a tank or vessel, existence of a vortex may be less than obvious.

The likelihood of such vortex formation can be substantially reduced by adherence to the excellent guidelines laid out in [Ref 17]. This defines sound engineering practice in the art of sizing pump sumps to suit a wide range of installations. Some might argue that these guides are a little on the conservative side, but that needs to put in to the context of the cost and inconvenience of correcting a poor design.

In REF 11, Prosser tacitly assumes that there are no restrictions in the layout of new sumps and that an ideal arrangement is the goal. While this assumption is often valid, there are occasions where substantial constraints appear to exist. These constraints may be physical in the sense that an ideal layout is untenable in the space available. This may be particularly true when an existing station is being expanded. On the other hand, the original capital budget may have been insufficient to allow a near ideal layout. Such cases can represent substantial reliability risk.

Air entraining vortices can often be very difficult to remove because they are related to the basic hydraulic design of the sump. This shape may have been largely driven by civil engineering issues, particularly if the designers were unwilling, or unable to acknowledge Prosser's guidelines. Local vortices may sometimes be suppressed by 'add-on' fixes. However, it is the sump shape itself that dictates the macroscopic features of the flow and the effects of such microscopic changes are not often useful. The best insurance against such problems is to conduct model testing of the sump before it is built. Model tests may also be used to help correct problems which otherwise seem intractable.

**Fig. 10.29** Underwater view of air-entraining vortex forming at liquid surface. When fully developed, such a vortex allows air direct entry to the suction bell mouth [out of camera and to the right]. Photograph: SPP.

Fig. 10.30 shows a sump model test being conducted. In such studies, a scale model of the sump is made, complete with all its feeds and pump intakes. For the most part, the sump is constructed with timber, with transparent panels located at critical points [i.e. where vortices might be expected]. A circulating pump generates flow in the model. It is a matter of judgement to determine how much of the system is modelled upstream of the pump intakes. Since vortex growth is strongly influenced by the degree of swirl in the liquid, any upstream feature that might remotely impart swirl should be included in the study.

In terms of physical scale, the larger the model the better. Hydraulically, the model should be run at Reynolds numbers equal to that of the full-size sump. This will model the viscous forces that govern turbulence in the flow. It should also be run at equal Froude number. These attempts to model the wave patterns of the full size sump. The flows that satisfy these two criteria turn out to be different. Fortunately, experience shows that running at equal Froude number is conservative, *see Ref 10.*

Testing should be conducted at all extremes of the anticipated operating window, high then low level, maximum then minimum flow. The timber construction allows the macroscopic shape to be easily modified iteratively. This is repeated until both:

**Fig. 10.30** Sumps model testing can help eliminate air-entraining vortices [The author in his younger days]. Photograph: Sigmund Pumps.

- Satisfactory flow conditions are measured at the pump intake
- A vortex, free surface is accomplished

As noted earlier, presenting the pump with a homogenised mix is preferable to intermittent slugs. Although commercial machines do have limited air-handling capacity, presenting it as 'slugs' seriously reduces this. Furthermore, if prolonged, slugs can impact on pump reliability.

Another way in which air can be unwittingly introduced is by cascading feed liquid onto the surface close to the suction pipe entry point *see* Fig. 10.31. Air bubbles introduced in this way naturally take a finite time to surface and disperse. Their buoyancy may not be sufficient to resist entertainment into the downward suction pipe flow. Wherever possible, liquid should be introduced at a great distance from the suction pipe entry, in order to allow for complete bubble dispersal. This may not always be possible: For example, in many air-draft cooling towers, the water cascades straight down from the spray bars into the cooling pond, entraining air in the process. Often the pump intake is very close to the cooling tower base, to minimise civil costs. This makes cooling water pumps on this service particularly vulnerable to performance variability a factor often incorrectly attributed to cavitation (Fig. 10.32).

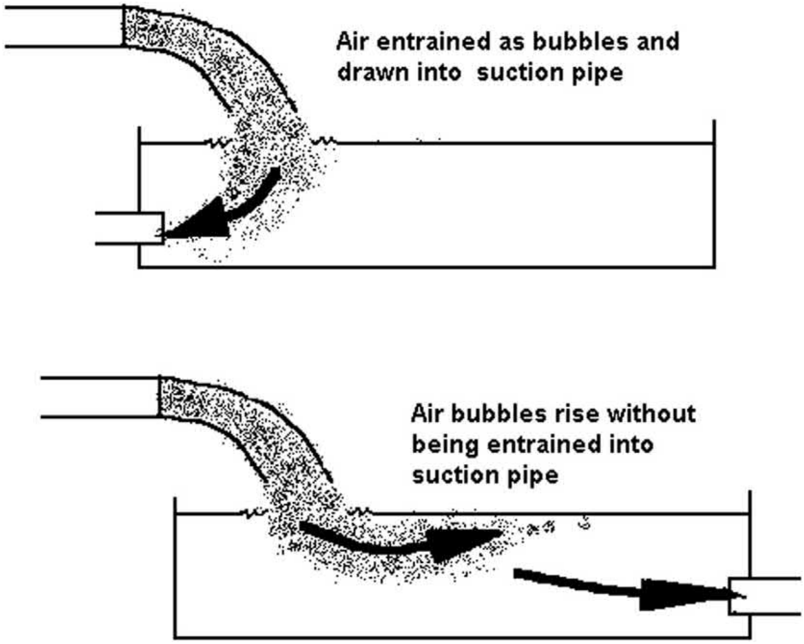



**Fig. 10.31** Good and bad suction pipe location when surface aeration is present.

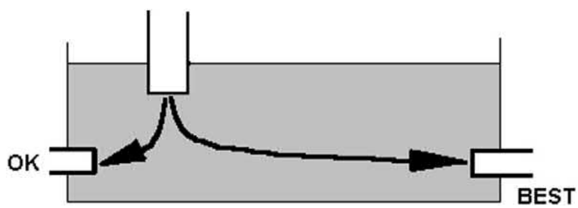



**Fig. 10.32** Preferred sump feed arrangement is below liquid level to avoid air entrainment.

Prevention is always better than cure, and the air entrainment issue can be avoided by introducing the feed flow below the surface, at the same time taking care that the feed flow does not promote swirl. Swirl is best avoided otherwise air entraining vortices, of the type discussed earlier, may be generated.

Also, take care that an underwater discharge does not create a syphonic effect and, by backflow, empty the sump. Non-return valves can help prevent this.

## Checking suction line for blockage

Since suction line problems are the most prolific, it may be useful to delve into what some simple measurements might tell us. We have earlier seen how interrogation of the suction pressure gauge might give a good indication of whether there is liquid in the pump or not. This study applied to the stationary pump. Remarks relative to the pump when being operated are made in *Chapter 8*. However, it is worth discussing the same item here under suction pipe issues.

In anticipation of having to do some trouble shooting once the pump is running it is prudent to set up provisions for a pair of pressure gauges at each extreme of the suction line. When there is no flow in the system, then there should be no difference in the two gauge readings. If there is a difference most probably this is because one gauge is defective. This will make subsequent analysis difficult if not impossible (Figs 10.33 and 10.34).

Once the pump is running and flow takes place in the suction line, then there should be a difference between the readings of the two gauges. Normally the liquid will experience pressure loss the further it moves away from the suction source. This is due to friction forces on the pipe wall. The pressure just ahead of the pump will be the lowest in the system. With piping systems designed along conventional principles [*see Chapter 2*] the friction effects in the suction line will not be great and so the two gauges will give similar readings, with the gauge closest to the pump having the lowest value. The overall levels should be close to that measured in the no flow condition. If the distance between the two measuring points is small [say less than 50 diameters] then the gauges should read within a few

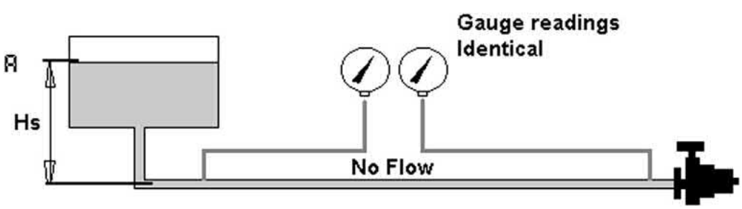



**Fig. 10.33** When there is no flow in the pipe, both gauge readings should be identical [Note clockwise movement shows increasing value].

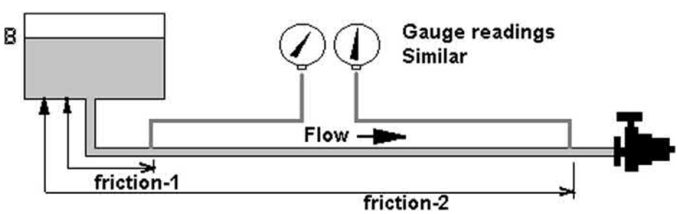



**Fig. 10.34** When flow is established, the two gauges should display similar readings taking into account the separation of the gauge points and the scale of pressure drop that is likely.

percent of each other. At large distances [say more than 500 diameters] then the difference might exceed 10%.

These sort of readings indicate that the suction line is probably clear and not prone to causing difficulties.

When the pump is static, a suction pressure gauge mainly registers the static suction head which may be negative or sub atmospheric. As flow develops, the liquid will experience head loss due to friction, expansion, turning, etc., Fig. 10.35 indicates the readings given by a suction pressure gauge as a function of pipe flow. The departure from a constant value is partly because of the liquid velocity head and partly from losses experienced in the journey from source to measuring point.

This leads to some important conclusions.

- These losses virtually follow a square law, and consequently diminish the static pressure in the pipe.
- At a ‘no flow condition’, the pressure gauge reading will be higher than when flow takes place.
- The higher the flow, the higher will be the difference/loss
- The severest restrictions will cause the most difference/loss
- High flow with normal losses, can give the same pressure gauge reading as low flow with abnormal losses. So the suction pressure gauge is of little help in determining pipe flow rate.
- Abnormal losses may result from:
  - Detached sheets of pipe lining
  - Debris admitted while strainer being cleaned

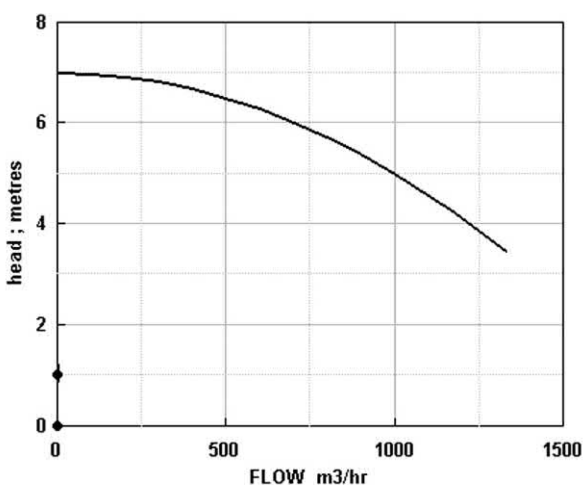



**Fig. 10.35** Suction pressure change due to blockage.

- Suction booster pump wind milling because it has not yet been started, or cannot pump [*Chapter 11*].
- Suction hose is not reinforced and collapses under vacuum

Locating a blockage can be a problem, but here again a pair of pressure gauges can help pin down the likely area.

Generally, the blockage location falls in to one of two groups (Figs 10.36 and 10.37).

1. If the blockage exists between the two gauge points, then they will show very different readings. Again, the gauge closest to the pump will register the lowest pressure of the two. The gauge closest to the source will show a reading similar to that measured in the no – flow situation. Blockages result in very high local velocities and high losses. This appears as an abnormal reduction in pressure.
2. In the second category, the blockage lies between the gauging point and the suction source. In this case, both gauges will read substantially lower than when no flow was taking place.

## Does the liquid contain abrasive particles?

In some circumstances, pumps are required to handle abrasive particles suspended in the product. This may have been a planned part of the pumps

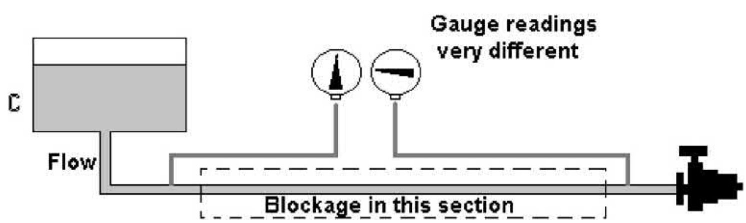



**Fig. 10.36** A blockage between the two gauge points will cause a significant difference in the readings.

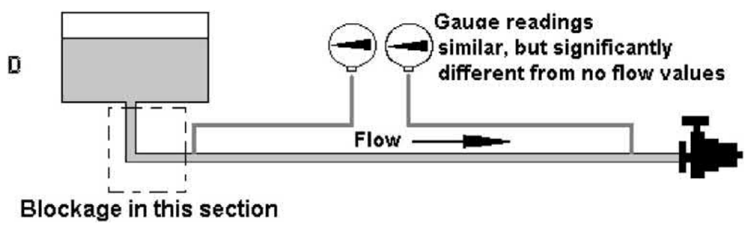



**Fig. 10.37** A blockage before the gauge points will result in similar gauge readings – taking into account the separation of the gauge points and the scale of pressure drop that is likely.

function, or it may be unexpected. This subject is so important that a separate section [Appendix J] is devoted to it.

## Minimum operating flow and 'operating comfort zone'

### Why the need for a definition?

All centrifugal pumps are designed for a certain set of fluid conditions such as flow, generated pressure, and minimum suction pressure. Some latitude exists to regularly operate at off-design conditions, but this is restricted by the absorbed power of the machine. In order to help define this limit, a minimum flow is often declared for the pump. Operation below this flow will bring with it possible pitfalls. Terms in common use include;

### Thermal minimum flow

Historically, low flow was meant to define the point where fluid temperature rise within the machine exceeded some set value. This definition of thermal minimum flow might be in the region of 5–10% of the pumps design condition.

### Minimum flow

Experience showed that prolonged operation just above the thermal minimum flow did indeed avoid temperature rise problems. However, it was still associated with a diverse range of bad habits, so the definition of low flow was often arbitrarily raised and renamed minimum flow. This might be in the region of to 10–15% of pump design flow. The pump may be operated 'occasionally' during unforeseen or unscheduled operating upsets at this flow. It is *not* normally the flow that the manufacturer would be happy to guarantee the pump for continuous operation.

## Minimum continuous stable flow [MCSF]

This is understood as the flow, which the manufacturer would be happy to guarantee the pump for continuous operation. Minimum Continuous Stable Flow or MCSF may be between 15% and 60% of design flow, depending on the pump power, size and speed.

It is perhaps worth exploring the background to these concepts.

The designer of any device will normally be given some performance targets that his creation is expected to accomplish. He in turn may list the restrictions to subsequent operation of his device. In some cases, there will be little or even no restrictions, beyond the obvious safety caveats. In others, the complexity of the device may generate a significant list.

If the device owner chooses to subsequently operate the device within these design limits, then he should reasonably expect it to give satisfactory service.

Some devices, once activated, require little or no further attention. Examples might include light bulbs, air circulating fans, refrigerators. At the other extreme are systems that have complex and restrictive limitations on their use. They require continuous assessment. Examples might include; power generation stations or jet airliners and so on.

All and any of these systems will possess a set of conditions which the designer might label 'Normal use'. Operation outside of these conditions would be neither recommended nor even tolerated. In some cases operating equipment outside their normal design conditions might induce an immediate and tangible complaint. In other cases, the equipment distress might be less than obvious. In discussing how such constraints might apply to centrifugal pumps, it might be helpful to introduce the concept of an operating 'COMFORT ZONE'. This describes the set of conditions within which a device's behaviour can be considered by the user to be unremarkable. Operating outside of this zone may introduce some peculiar and often unwelcome characteristics to its behaviour.

Centrifugal pumps are often considered to be 'plug and play' machines since any complaints they make might, when asked to operate outside their respective COMFORT ZONEs, can go largely unnoticed. And since the way they work is invisible externally, any distress can be less than obvious. But all centrifugal pumps do actually possess a COMFORT ZONE. If in service, the pump operating flow is never allowed to deviate significantly from the nominal design flow, then acceptable pump life should result. This is an overly harsh position because it is known that such operating restrictive constraints can be relaxed, and acceptable life is still achieved. Rather than limiting pump operation to a single specific point, each pump has its own performance output band within which the pump will still operate well. I define this as the pump's **COMFORT ZONE**. The owner should have little cause for concern provided pump operation is managed to within this zone. By not operating within this zone, the owner accepts that some accelerated damage or decay of the pump is implicit-however small. Pump owners will want to know the proportions of this COMFORT ZONE, in the form of some easily remembered rule. This is not possible unfortunately. Small and or low power pumps have a COMFORT ZONE that is very wide indeed and is almost unrestricted. Equally, large and or very powerful pumps may have a COMFORT ZONE that can be surprisingly narrow and very restricted.

- There are a range of factors that help decide the pump's COMFORT ZONE and they include;
- Low flow liquid heating due to internal 'churning'.
- Onset of inlet backflow and potentially, cavitating surge.
- Effect of pressure pulsations that result from rotating impeller flow field interacting with stationary collector components.
- Margin of NPSH [A]/NPSH [R] and suppression of cavitation at an intensity that causes performance decay. Conventional wisdom would exclusively view this as a high flow concern It is less widely understood that it can be just as significant at low flow

Fig. 10.38 lists the chief hydraulic factors involved in defining the COMFORT ZONE. Other less common factors might include radial bearing loads and or dynamic shaft deflection. To this might be added cavitation damage on either convex or concave vane surfaces.

I will look at the elements that help make up the COMFORT ZONE.

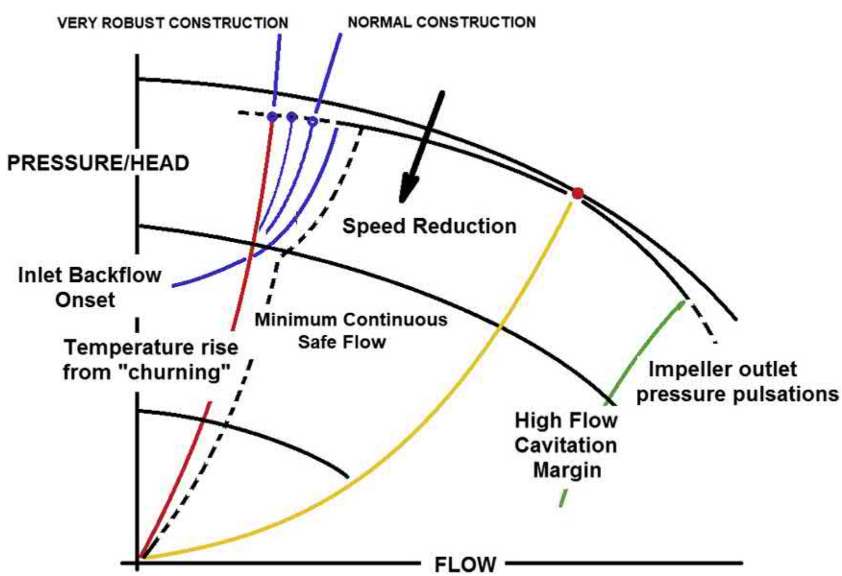



**Fig. 10.38** This figure illustrates how the chief COMFORT ZONE components change with running speed. The low flow constraint flips from Churning to Inlet Backflow once inlet tip speeds exceed 20–25 m/s. Heavy duty 'ROBUST' pumps can tolerate these effects of inlet backflow better than pumps of 'NORMAL' construction.

## Minimum flow boundary

The flow boundary of most interest is usually that of low flow. It is widely known that operation below this flow may bring pitfalls. In order to help, a Minimum Flow is often sought for each pump. Historically this has often seemed a rather vague area to many users. In particular, the term Minimum Flow and Minimum Continuous Safe Flow was often muddled.

In simple terms, **Minimum Continuous Safe/Stable Flow** describes that lowest value which the pump can be operated continuously and above which wear/decay rates should be tolerable. This flow value will be a little higher than either the low flow dictated by churning temperature rise, or by inlet backflow effects. The first is chiefly a function of the power absorbed by the pump. The second is influenced by the impeller inlet tip speed and the overall robustness of the pump's mechanical design.

The term, '**Minimum flow**' is lower than the Minimum Continuous Safe Flow, and is a value at which the pump can operate at 'Occasionally' or in an emergency. Manufacturers used to assume the user would be aware that operation here would be associated with accelerated wear and decay of the pump *[In the early years, the Thermal limit was often used as a description of Minimum Flow. However, the 'Low Flow Problem' soon became well known in its various forms. In order to acknowledge its existence the Thermal limit value would be arbitrarily increased so as to provide some safety margin.]* (Fig. 10.39).

## Thermal/churning minimum flow

The phenomenon of low flow liquid heating was touched upon earlier. Historically, Minimum flow was often taken to infer that point where liquid temperature rise within the machine exceeded some set value. This definition of Minimum flow might be in the region of up to 10 or 15% of the pumps design flow. This percentage increases as the pump size, speed or power increases. Fig. 10.38 shows this as a 'churning' limit line.

## Onset of inlet backflow

All centrifugal pumps will experience this phenomenon, described in Appendix D. Its intensity is chiefly related to the peripheral velocity of the impeller inlet rim [impeller eye] small and or slow speed pumps will be largely immune to its effects. In the case of large and powerful pumps

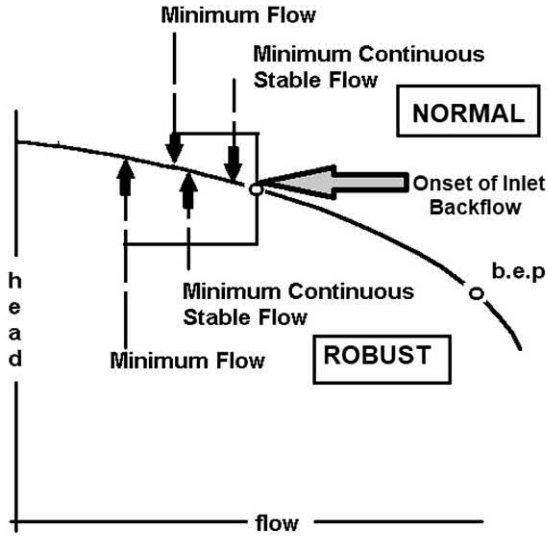



**Fig. 10.39** this is an abbreviation of Fig. 10.1 for pumps with an inlet tip speed in excess of 25 m/s. It attempts to show conceptually how MCSF and Minimum Flow are influenced by the robustness of the mechanical design.

however, this limit will override any restrictions related to the above mentioned thermal effects. Backflow effects will include;

- Liquid pressure pulsations in the inlet duct. These can radiate out in to the general pipework system, causing damage to connected equipment.
- Increase in;
  - Casing vibration levels-[particularly if the pump includes any flow straightening ribs close to the impeller inlet region].
  - Shaft vibration particularly at vane inlet pass rate if straightener ribs exist.
  - Cavitating surge if the ratio of NPSH [A]/NPSH [R] becomes too small [see Appendix E]. This surge frequency will typically lie in the region of 0.5–2 Hz. In some cases, a lower frequency 'beat' may also be experienced. These surges can become violent and have been responsible for damage to piping as well as bedplate foundations. Fluctuating noise is the most common and often the only outward sign that the pump is in low flow distress. Even so, this is rarely a really prominent sign, except for the case of pumps of rather low shaft speed pumps, [which would otherwise be relatively quiet].

Cavitation damage to sites in the impeller region that seem illogical and inconsistent with the normal 'assumed' liquid flowpath.

So the low flow boundary is a composite, relating to inlet tip speed, machine construction and/or absorbed power. For this reason, it is unwise to try and allocate a single simple definition *[Though that won't stop people from trying!].*

- With inlet tip speeds in excess of 20–30 m/s, inlet backflow becomes a more suitable limit. However even this is an over simplistic statement. Some of the inlet backflow effects relate to the mechanical response of the pump structure.
  - For any given impeller and inlet tip speed, a lightly constructed foot mounted pump construction intended chiefly for water service *[at say, temperatures below water boiling point]* is likely to respond more violently than a robust centre-line mounted hydrocarbon processing pump construction designed to operate over 350 °C.
- Inlet backflow forces relate to the kinetic energy of the rejected liquid. So pumps handling dense liquids will experience higher forces than those handling lower density products.

Hence some pumps can operate with a degree of established inlet backflow provided the product is not too dense and their mechanical design philosophy exceeds the needs of the application.

So both **MCSF** and **Minimum Flow** will usually be lower for machines of robust construction.

As Fig. 10.40 shows, the 'Thermal' limit [maybe with some small margin] is quite suitable for small pumps or pumps of low inlet tip speed. Fig. 10.41 shows an equivalent diagram for pumps with an inlet tip speed in excess of 30 m/s.

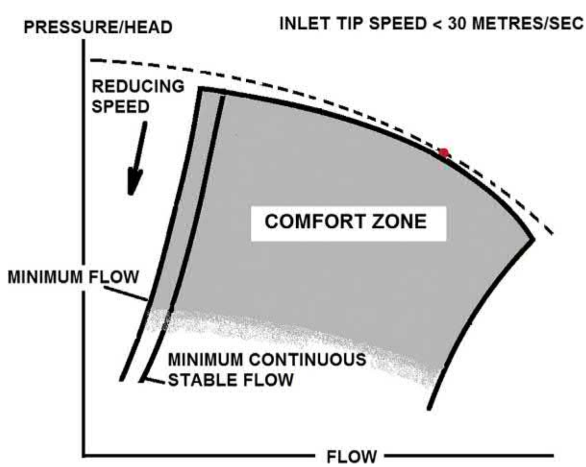



**Fig. 10.40** The Comfort Zone minimum flow boundary for pumps having an inlet tip speed less than 25–30 m/s is almost entirely based on the Thermal/Churning criteria.

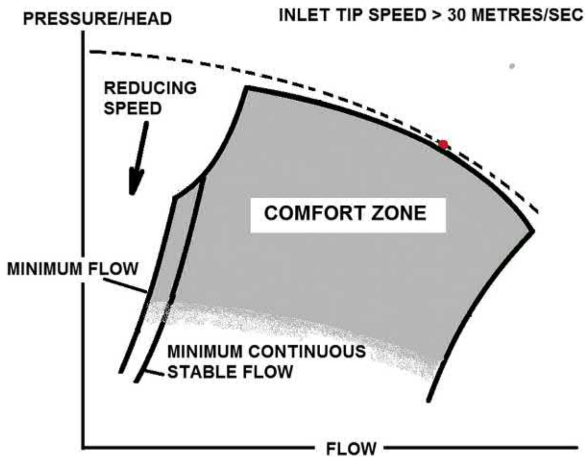



**Fig. 10.41** The Comfort Zone minimum flow boundary for pumps having an inlet tip speed greater than 25–30 m/s is initially based on Inlet Backflow criteria, but as speed reduces below that figure it becomes entirely based on the thermal/churning criteria.

## Maximum flow boundary

The most significant contributor to the high flow boundary of the COMFORT ZONE definition is the onset of cavitation noise and impeller damage. Unlike low flow cavitation, high flow cavitation is very prominent and more difficult to overlook. And its value seems less dependent on pump size and speed. Experience would say that a value of 120% of design flow for small pumps decreasing to 110% for larger more powerful pumps would be typical for a pump at its design speed. As speed is reduced *[but NPSH [A] stays constant],* this margin value can often be increased.

At present, there is no universally agreed definition for 'Maximum Continuous Safe Flow'. The only formal definition that comes close is the term 'End of curve' a reference to the highest flow shown on a pumps performance curve. However, this is literally an opinion of the curve-producer; there is no unanimity within the pump community.

Being pragmatic, the power of the driver is often the limiting factor in determining the maximum flow possible from a pump.

## Maximum head boundary

It goes without saying that physical size factors will limit how large an impeller can be installed in the collector. But this physical limit is rarely if ever fulfilled. Earlier, the effects of a rotating impeller flow field interacting cyclically with the collector were described. The intensity of such behaviour is largely governed by the gap between the impeller and casing – [see Fig. J66]. This gap has become known as the 'B gap'.

Industry practice has evolved two general criteria for the minimum B gap ratio of D3 to D2; 1.03 for multivaned diffuser collectors and 1.1 for spiral volute collectors.

For pumps with collectors of 'conventional' proportions, this will help avoid high cycle fatigue damage and also help suppress pressure pulsations to tolerable levels. It is known that pressure pulsation intensity increase at off-design flows so theoretically pumps selected for such duties should be constrained to a slightly lower pressure head in these cases. This would lead to a slightly smaller impeller diameter. But in practice, the normal mechanical designs will be robust enough to accommodate the extra forces. An exception to this would be in the case of pumps handling liquids prone to gas evolution [Such as gas sweetener/stripper pumps]. If the radial clearance gap is too small, then gas evolution and collapse in this zone can cause damage to the impeller peripheral tip [see Figs J.63–J.69].

The available driver power will also constitute a factor. But this is much more difficult to generalise, since the choice of driver margin over required is merely a matter of choice. A common strategy is ensuring that the driver power exceeds the pump absorption by 10% more than that required at rated flow. This might be applied to all centrifugal pumps on a project. However, as Chapter 9 shows, certain pumps have a power requirement that actual reduces as flow increases beyond design. Such pumps are deemed to have a 'non-overloading' power curve. In these cases, a 10% margin might to be quite conservative.

## Minimum head boundary

It is common practice, for manufacturers to also show on their general performance predictions charts, a line for minimum performance. This should not be interpreted as a firm line, which cannot be crossed under any circumstances. Appendix C explains that is generally intended only as a guide as to where a smaller machine would be a more sensible selection. However, if instead a performance below this line is accomplished by reducing the impeller diameter – *[at a constant and fixed speed],* then a minor irritation may arise. If, or when, the reduced impeller outlet diameter begins to approach that of the inlet annulus [eye] then performance becomes less and less predictable without confirmatory testing. The Affinity Laws no longer apply [see Appendix C]. But at no point does the pump suddenly stop working; it just becomes less and less effective.

## Choke ring

Older designs often had vanes with exaggerated inlet angles, since this appeared to be an easy route to good NPSH performance. Sadly, as we now know, this also exaggerated the flows at which inlet backflow occurred. One rough and ready way to make these older designs more respectable is to fit a 'choke ring'. This is a rotating orifice plate, attached to the impeller eye. It functions by increasing the axial velocity in the vicinity of the orifice, while reducing the associated peripheral velocity. The net effect is to reduce the zero incidence flow, and correspondingly suppress the inlet backflow point (Fig. 10.42).

The price to pay for this is an increase in NPSH [R], but sometimes the site NPSH [A] can tolerate this. If so, then the choke ring may push inlet backflow to below the rated flow and improved behaviour will result (Fig. 10.43).

Some test data also suggests that the ring energises the shroud-side boundary layer on low specific speed impellers resulting in improved vibration results. Outline details for emergency sizing of a choke ring are given in Appendix K.

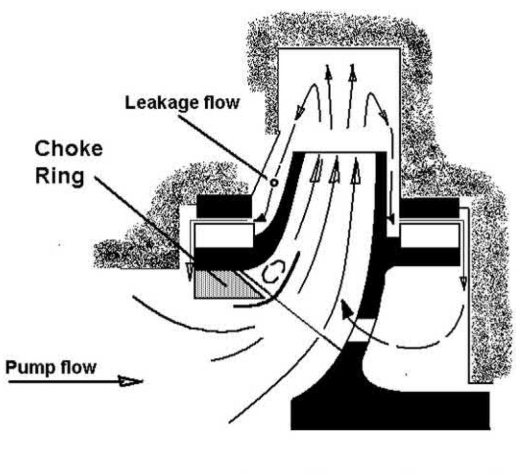



**Fig. 10.42** Choke ring concept.

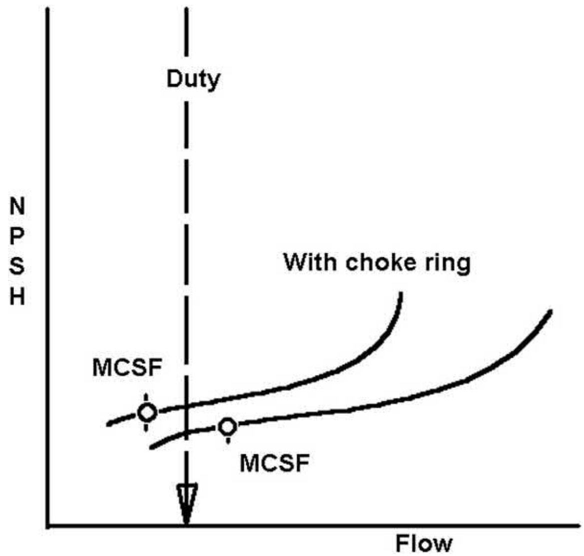



**Fig. 10.43** Effect of choke ring on NPSH.

## Alternative impellers

Manufacturers will often have alternative impellers available for a particular pump. The *normal* impeller will be designed to achieve as much output from a given weight of pump, in order to be competitive in the pump market. These pumps will have relatively flat pump curves, and relatively steep power curves. We have seen that in certain cases, a flat pump curve can be a disadvantage. It will appear to wear out more rapidly if simultaneously, the system curve is quite flat. For these and some other applications, a steeper curve is preferred – accepting that such impellers are inherently less efficient. In addition, because less output is being accomplished from a given weight of pump, the machine is more expensive. However, there is always a finite demand for such alternatives, and many manufacturers provide them (Fig. 10.44).

Typically, such *low flow* impellers will have a design flow of about 70% of the normal impeller. The head generated will also be a little less than the *normal*. These two factors combine to provide a steeper output curve, which we have seen to be often an advantage. Furthermore the power curve will tend to be non-overloading, or more so than the *normal* impeller. The perils of operating *normal* impellers at low flow are described elsewhere [Appendix D] and when prolonged operation is envisaged, it may be worthwhile considering a low flow alternative.

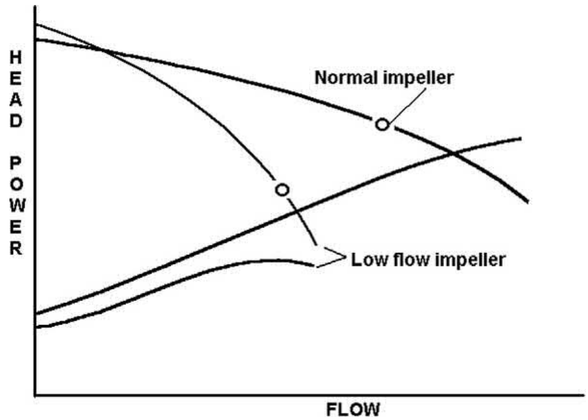


**Fig. 10.44** Differences in output curves for alternative impellers.

In terms of troubleshooting this is a point to be wary of. Low flow impellers are usually retrofitted long after the pump was bought and started up. This might mean the records are incomplete. Attempting to conduct hydraulic audits of such a pump, inadvertently using the *normal* impeller output curve as a reference is bound to be confusing.

Less frequently, manufacturers will also provide impellers with higher output than the *normal* one. The scope for this is constrained. Having to achieve this within an existing casing and shaft/bearing system is restrictive. Typically, such designs have even flatter output curves[7] and steeper power curves. They have limited suction performance due to the casing suction branch restriction. However, in the right circumstance these impellers can give a useful increase in output where, for example, a process throughput has been increased by de-bottlenecking.

## Choked impeller

Despite efforts, sometimes solids get into an impeller and choke the flow passages. It hardly matters whether the solids are hard and inflexible, or fibrous and 'stringy'. The effect is just about the same.

Such choking solids will affect the pump in two important ways.

## Suction performance

It has been reiterated many times that the most sensitive and susceptible part of any centrifugal pump is the suction side. Unless the liquid can get into the pump, no work can be done on it. Debris that chokes the impeller inlet has precisely this effect.[8] Generally, the impeller only becomes partially choked and in this case, the effect is to increase the NPSH requirements of the pump. If this increased.

NPSH [R] is still less than the NPSH [A] then the pump will run, albeit noisily and perhaps with some vibration due to the mass unbalance. This will be revealed by an FFT analyser as having a strong component at running speed (Fig. 10.45).

If the NPSH [A] exceeds the NPSH [R] then the pump will naturally run at flow [4]. In fact, the pump could even run at flow [1] if the pumping scheme would allow it. At this increased flow the pump would then cavitate.

As the impeller begins to choke, there is little external effect. The NPSH curve intersection will modify from [1,2]. Until the intersection, drops below flow at [4] there will be no performance change.

Once the NPSH intersection drops below [4] the pump will cavitate more and more. At about this point, the head generated by the pump will begin to decay. There may be a slight increase in noise and vibration at this point.

As the impeller chokes more and more, the performance also decays more. At flow [3] there will be clear outward signs. Not only will there be performance decay, but it will certainly be accompanied by increased vibration [mainly at running speed] and most probably higher noise levels.

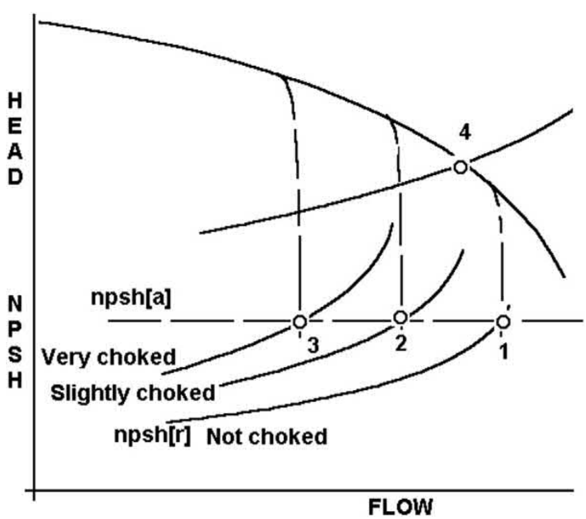



**Fig. 10.45** Effect that choked impeller inlet has on performance.

## Discharge performance

### *Blocked impeller outlet*

As mentioned in the footnote, impellers typically choke in the inlet, but occasionally the outlet may become blocked. In this case, the output performance of the pump becomes modified. The modification is a function of impeller flow (Fig. 10.46).

At zero flow, there is little if any effect. As flow is increased, the performance departs more and more from the normal curve. The blockage simply reduces the impellers capacity to do work on the liquid. In many respects, the outward effect is similar to that of a restricting discharge orifice, though the mechanisms are completely different. Increase in noise is unlikely, but some change in vibration may be evident [at running speed] due to the mass unbalance.

### *Blocked casing passages*

The effect here is much the same as a restricting discharge orifice. In this case the impeller can do work on the liquid, but some of it gets lost. It gets lost in the high velocity area created around the blockage. The increased local velocity might also create a slight increase, but this may not be obvious. One would not expect any vibration increase.

## Maximum flow

Few pumps will operate at more than, say, 125% of their design flow without complaining in some way, though the effects may take time to

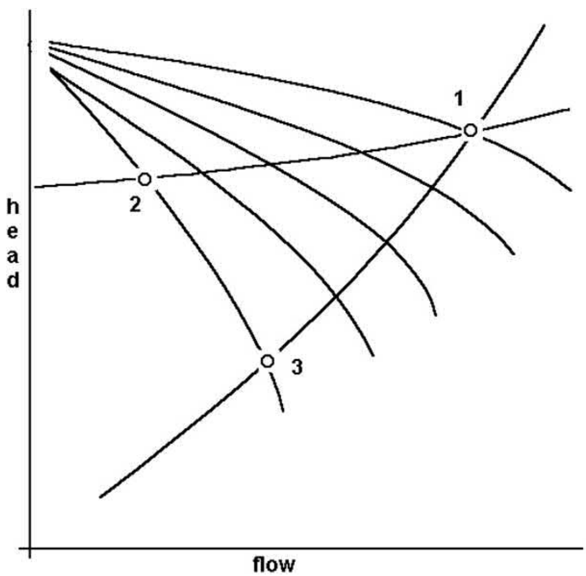



**Fig. 10.46** Blockage in either impeller outlet or casing passages show very similar effect.

materialise. Many low flow problems arise because fixed speed pumps can only be designed for one flow. At other flows, the geometry of the pump components such as impeller and casing becomes mismatched.

The same situation exists when the pump runs at too high a flow. Unlike low flow, it will often be much more apparent that the pump is in distress when operating at high flow. Perhaps the most significant aspect of high flow operation is that the fluid velocities are much higher.

At present, there is no universally agreed definition for 'Maximum Continuous Safe Flow'. The only formal definition that comes close is the term 'End of curve' a reference to the highest flow shown on a pumps performance curve. However, this is literally an opinion of the curve-drawer; there is no unanimity within the pump community. But 110% of bep seems to be emerging as the *de facto* standard for maximum allowable flow.

Being pragmatic, the power of the driver is often the limiting factor in determining the maximum flow possible from a pump. [Unless the pump exhibits a non-overloading power curve] This means that the pump shaft is most highly stressed at high flow. Operation at high flow is more likely to reveal any deficiency in the shaft coupling condition.

If this absorbed power obstacle were removed, then other constraints would be:

## Erosion due to suspended solids

If suspended solids are a potential problem then risks increases dramatically at high flow. This is because the rate of wear varies at least as the cube of velocity. In other words, a pump operating only 14% above its design flow will wear out about 50% more than at design flow. This will materialise as a general reduction in impeller wall vane material thickness, loss of wear ring material, and reductions in casing wall thickness. Local losses due to impact or turbulence damage may be greater, [Appendix J].

Pumps designed for low flow and high head will show more damage in the casing than in the impeller. With pumps designed for high flow and low head, the reverse is likely.

## Unfavourable bearing and seal environment

Centrifugal pump impellers all exert a radial load on the shaft. It is caused by mismatching of the casing areas with the flow picture emerging from the impeller. A mismatch arises at flow above and below the design flow and the radial load is transferred to the pump bearings. In addition, the loads result in shaft deflections, making the task of the mechanical seal more difficult.

Such a problem can be diminished by choice of collector design. Multiblade diffusers are the best from this standpoint with so-called double volutes next best. Single volutes are the worst, but are most prevalent due to their low manufacturing costs.

## Increased noise emission

Noise in pumps originates from several sources. The principal source is hydraulic interaction between the rotating impeller blades and protruding parts of the stationary pump casing, *see* Fig. 10.12. In large or slow speed pumps, local Strouhal type vortex shedding can compete with this vane interaction noise. High pump flow is associated with higher vortex shedding levels. In most pumps, this vortex shedding noise appears in the low frequency end of the spectrum.

## Increased pump vibration

Modern pump designs have the rotors balanced to exceptional levels. Consequently, the hydraulic disturbing forces often outweigh mechanical forces such as rotor balance. These hydraulic forces arise from non-uniform flow distributions in the impeller, mentioned above. This non-uniformity is then discharged out into the collector where it causes unsteady interaction. This interaction is related to the flow and so its contribution increases rapidly above design flow. However, some test data shows the vibration levels reducing again at extremely high flows [say 180% of bep]. Presumably, the pump is just not generating enough head to establish any significant hydraulic disturbance.

## Loss of balance disk force

Multistage pumps can generate large axial forces due to the unbalanced areas inherent in most designs. To an extent, this can be nullified by a back-to-back rotor design. Other pumps use a stacked rotor layout. Large pumps of this type combine sophisticated external thrust bearings with internal hydraulic balancing or compensating devices. Many small pumps use very simple external bearings, but may still employ balance devices.

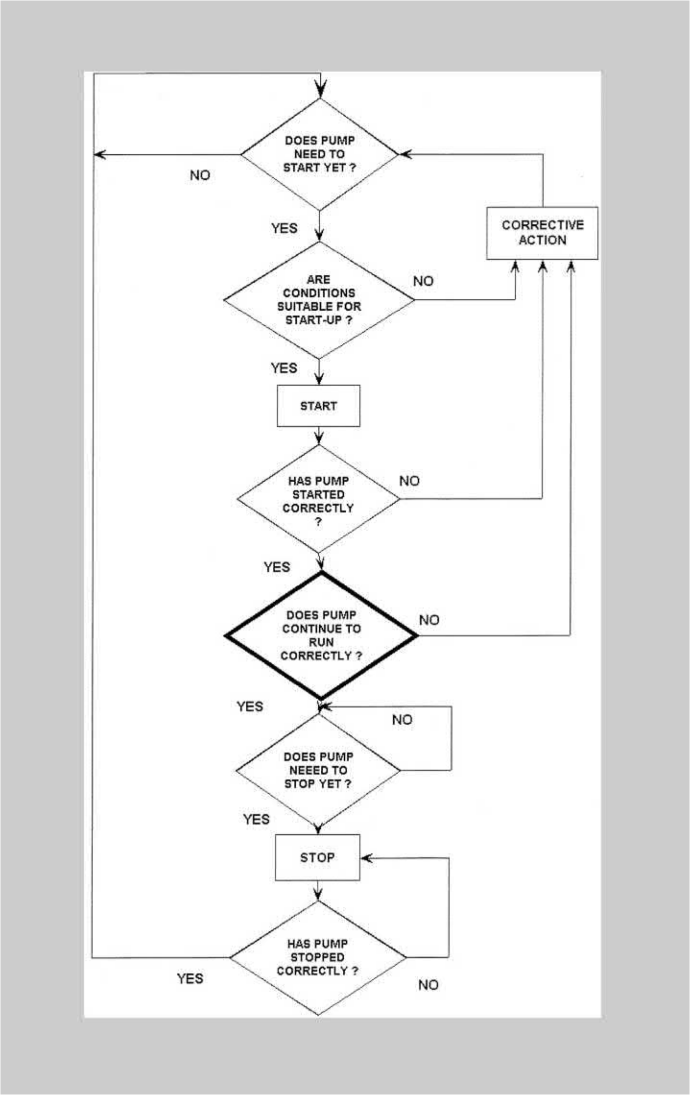
These devices capitalise on the pressure differential between suction and discharge. By applying this differential across a pair of radial faces, [one rotating with the shaft and the other in the casing], the entire axial unbalance can be compensated, *see* Fig. 10.5. These devices are known as balance disks, or flanged drums. However, at high flow, the pump head reduces, and so does the differential head across the device. Consequently, the gap between the two faces also reduces and may even result in contact.

## Endnotes

1. That is where the hub is parallel.
2. Inlet design flow is not obliged to be the same as the outlet design flow. Older machines were often had over designed inlet flow in an attempt to demonstrate low NPSH levels at design flow.
3. Vane pass frequency relative to stationary outlet protrusions such as diffuser on volute tips.
4. This more or less also describes the apparatus that James Wall used to determine the mechanical equivalent of heat.
5. Sometimes called re-cycle lines, bypass line, or kickback lines.
6. i.e. pumps in which no special measures have been taken to tolerate large quantities of gas [say] more than 5% by volume at the pump inlet flange.
7. That might not constantly fall as flow is increased. This could inhibit their use in parallel pumping schemes, though they can be used on single pump schemes.
8. Because impeller passage areas generally diverge, it is unusual to find debris choking the impeller outlet. However, pumps of low specific speed have passage widths [b2 in Fig. 10.29] that narrow appreciably towards outlet. In these pumps inflexible solids may become jammed. The effect then is similar to discharge side choking.

# CHAPTER 11

# Does the pump continue to operate well?




*Troubleshooting Centrifugal Pumps and Their Systems*
ISBN 978-0-08-102503-1
https://doi.org/10.1016/B978-0-08-102503-1.00011-6

## Introduction

The content of this section leans unashamedly towards aspects of pump and system interaction. It does so because, in the writer's experience, this aspect is most often misunderstood. In contrast, many of the mechanical aspects of pump design are not unique to that industry. A wider body of experience exists to solve such problems, which in any case are often intuitive.

There are two segments to this Chapter.

- **Hydraulic problems** – looking at combined pump and system interactions. Both pump and systems characteristics may appear to slowly or suddenly change. This will appear as a corresponding change in the performance of the pumping scheme.
- **Mechanical issues** – looking at the pump in isolation. The mechanical aspects of pumping machines are also prone to change and degradation. This might have little if any effect on the output of the overall scheme, though its reliability may be at risk.

## Hydraulic problems

Having got the pump up to speed and giving its required performance, one hopes the pump will continue to do so, until its role is fulfilled. All pumps 'wear out' over a period of time. We have seen earlier that this 'apparent' wear may be hastened by the shape of both the pump and system curve.

A decay in pump performance may take place progressively and measurably, in which case there will/should be few surprises. Appropriate steps can be taken to minimise the impact. Frustrating as this sort of wear might be it is nothing as compared to unexpected changes in performance.

But of course, the piping characteristics can also change with time and it is clear that the overall performance of the pump scheme is as dependent on this as it is on the pump.

Most pumps are procured in order to create a liquid flow.[1] They do this, indirectly, by generating pressure to meet the demand of the system, at that desired flow. Therefore, you might imagine that any shortfall in pump performance is best detected by a reduction in flow. Sadly, there is more to it than this.

Imagine that a pump operates in a system that largely consists of friction. As it wears, the intersection of the two curves shows that the effects on head will be much more apparent than the reduction in flow. In an 'all-friction system' its characteristic is well described as a square law.[2] In such a system, the head might reduce by 81%, but the flow only reduces by square root of 81%, i.e. 90%. In this situation, the pump would most probably be reported as low in *head see* Fig. 11.1.

Next let's look at the same pump in a system that largely consists of static head *see* Fig. 11.2. As it wears the intersection of the two curves again

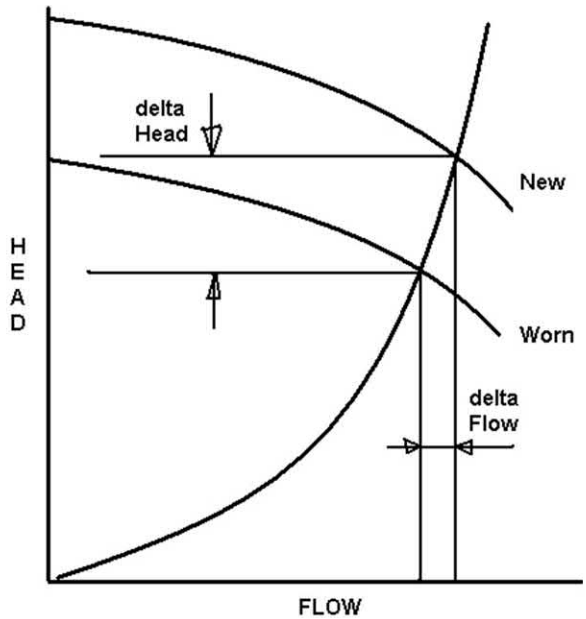



**Fig. 11.1** Pump will be reported as low in head.

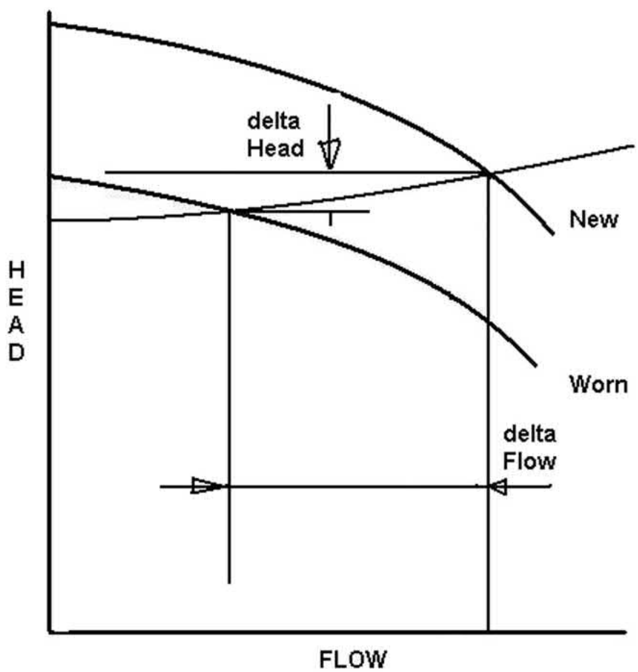



**Fig. 11.2** Same pump will be reported as low in flow.

moves. This time it becomes more apparent by a reduction in flow than by a reduction of head.

In this situation, the same pump would most probably be reported as low in *flow*.

This illustrates that the same pump, with the same 'worn' features would either show symptoms of flow decay, or head decay, depending upon the characteristics of the system in which it operates. This is a point that is almost always overlooked – particularly in diagnosis.

Stating the pump problem as 'Low in flow' is not at all helpful unless some indication of the system characteristic is also known.

This example also highlights another major problem to those involved in pump troubleshooting. In most practical cases, it is not possible to assemble data at more than one flow. Such a single 'snapshot' of the pump system interaction makes diagnosis very difficult, even when the pump characteristic is known and documented. As we have already seen, and will see later, several different root causes can have similar symptoms. This further complicates matters.

If the flow in the system can be varied to some extent, then a much better picture of the system characteristic can be assembled experimentally. We will come back to this later in the Chapter.

## Fault categories?

Pump performance problems fall into four major fault categories,

**Category 1:** Liquid can't get into pump

**Category 2:** Liquid can get into the pump, but impeller cannot do enough work on it

**Category 3:** Liquid can get into the pump, and impeller can do work on it. However, abnormal amounts of work then are lost or absorbed inside the pump

**Category 4:** Liquid can get into pump and gets work done on it without abnormal losses, but it cannot get away from the vicinity of the pump

Exploring each of these in a little more detail:

**Category 1** – Liquid cannot get into pump

Any problems of this sort should have been eliminated in the section Chapter 8 'Is the pump ready to start?' This is particularly true when the majority of pump problems relate to this issue in one way or another. But just reiterating a selection of likely causes:

- Strainer blocked
- Suction line heavily fouled, blocked or even frozen.

- Suction line not connected to liquid source
- Suction isolation valve closed
- Debris in impeller eye
- Liquid viscosity has increased very significantly

**Category 2** – Liquid can get into the pump, but impeller cannot do enough work on it.

The most common causes of this are either cavitation, or gas/air entry. The detailed issues of cavitation are discussed in Appendix E. The causes and effects of air/gas entry have already been discussed.

Nuances of this problem category are (Table 11.1):

- Pump is not vented
- Pump is not primed
- Air or gas is mixed with liquid, but not dissolved in it
- Blocked impeller liquid passages
- Damaged impeller blades
- Diffuser tips severely damaged
- Air leaks at shaft seal
- Wrong impeller rotation

**Table 11.1** Cause and effect.

| Cause | Flow | Head | Power | Notes |
|---|---|---|---|---|
| Pump not vented | Low | Low | Low | High vibration |
| Pump not primed | None | None | Very low | Low noise – initially |
| Air or gas in suction flow | Low and surging | Low and surging | Low and surging | High vibration |
| Blocked impeller | Low | Slightly low | Low | High vibration |
| Damaged impeller vane inlet | Slightly low | Slightly low | Slightly low | May be noisy. Some vibration |
| Diffuser tips damaged | Changeable | Changeable | Changeable | May be noisy. Some vibration |
| Shaft seal leaking | Low | Low | Low | No significant vibration increase |

**Category 3** – Liquid can get into the pump, and impeller can do work on it. However, abnormal amounts of work also are lost or absorbed inside the pump

This refers to internal losses such as (Table 11.2):

- Increased wear ring clearances of shrouded impellers or vane tip clearance in the case of open or semi open impellers
- Increased roughness of wetted passage walls due to corrosion or erosion.
- Abnormal internal flow re-circulation losses – maybe thorough seal piping or ill-fitting interstage O rings
- Internal mechanical rubs within the wetted end [as opposed to within bearings]
- Higher liquid viscosity than expected

**Table 11.2** Cause and effect.

| Cause | Flow | Head | Power | Notes |
|---|---|---|---|---|
| Wear of internal clearances | Low | Low | High | Increased NPSH [R] |
| Internal roughness increased | Low | Low | Normal | |
| Internal re-circulation path | Low | Low | Normal | |
| Internal mechanical rubs | Normal | Normal | High | Detectable vibration |
| High viscosity | Low | Low | High | |

**Category 4** – Liquid can get into pump, and gets work done on it without abnormal losses, but it cannot readily flow away from the pump

Circumstances that could cause this would be (Table 11.3):

- Fouled discharge piping [May be temperature dependent, such as waxing up, or part-freezing]
- Discharge isolation valve not fully open
- Stuck Non Return Valve
- Control valve not fully opening
- Head requirements underestimated [very unusual]
- System resistance higher than visualised
- Pump operates in parallel with another that has a 'stronger' characteristic.

**Table 11.3** Cause and effect.

| Cause | Flow | Head | Power | Notes |
|---|---|---|---|---|
| Fouled discharge | Low | High | Low | |
| Isolation valve stuck | Low | High | Low | Vibrations might increase |
| Control valve not fully open | Low | High | Low | |
| Head requirements understated | Low | High | Low | |
| System resistance higher than visualised | Low | High | Low | |
| Poor load sharing | Low – may surge | High | Low – may surge | Vibrations might increase |

These examples give some general thoughts on performance problems, but to make further progress in diagnosis requires either trial and error, or more depth to the analysis.

And do not lose sight of this; it is natural to presume [Palgrave's third Rule] that the fault always lies with the pump. But it is worth restating that the overall performance is dependent upon both the pump *and* system characteristics. In any problem solving, segregating the two influences is important in establishing root causes.

## Site testing

The above summary tables give generalised statements and show how, in many cases, the same effect can have several different causes. Now let us look more explicitly.

What is known about the *real* pump or system characteristics? Clearly, this has to be determined experimentally. Ideally, system flow, pump head and pump power would be measured. However, this is not often possible, though the pump contribution is usually known.

It is not often that a pump installation is comprehensively instrumented, hence the difficulty in drawing conclusions as to the root cause.

Most often, the instrumentation is only provided for monitoring purposes rather than diagnosis. In monitoring, it is only necessary to examine relative changes in the measured parameter. The absolute values are of less interest, so less accuracy may be acceptable *see* Appendix I.

Since most pumps are purchased as flow movers rather than a liquid pressuriser, it might be assumed that flow is usually measured. If the pump forms part of some more complex process, then flow is most likely measured, but very little else.

Despite this, more often than not, pump discharge pressure is monitored as the only means of assessing a pump health. We have seen earlier, that pressure measurement is not a reliable or very accurate method of assessing a pumps flow. However, it is a much cheaper form of instrumentation than a flow meter, hence its prolific use. But in fact, with some intelligent analysis, even quite prosaic instrumentation can yield constructive data.

What instrumentation would be useful? [Assuming a valid set of performance test curves is available].

**Table 11.4** Relative value of different measuring instrument combinations.

| Suction head | Discharge[a] head | Flow | Power | Comment |
|---|---|---|---|---|
| Suction head at natural flow | | | | Useless[b] |
| Suction head at natural flow + | Discharge head at natural flow | | | Useful |
| Suction head at natural flow, and zero flow + | Discharge head at natural flow, and zero flow | | | Quite useful |
| | | Natural pump flow[c] | | Useful |
| Suction head at natural flow + | Discharge head at natural flow + | Natural pump flow | | Very useful |
| Suction head at natural flow, and zero flow + | Discharge head at natural flow and zero flow + | Natural pump flow | | Very very useful |
| Suction head at natural flow + | Discharge head at natural flow + | Natural pump flow + | Absorbed power | Nearly ideal |
| Suction head at natural flow, and zero flow + | Discharge head at natural flow and zero flow + | Natural pump flow + | Absorbed power | Nearly ideal |
| Suction head at specified flows + | Discharge head at specified flows + | Specified pump flows + | Absorbed power at specified flows | Ideal |
| | | | Absorbed power | Useful |

[a]Head is used instead of pressure because this simplifies matters. It eliminates the liquid density from the discussion.
[b]Except where cavitation studies are involved.
[c]This is the natural equilibrium flow where the pump and system characteristic curve naturally intersect.

The instruments need not be laboratory standard at this level of problem solving, but should be good quality, functional, and indicate zero when isolated.

## Interpreting data

Before looking into the different ways in which pump performance decay is diagnosed, it is worth bearing a few useful practical rules in mind.

**Palgrave's first rule of pumping:** over time, a pump's performance characteristic does not normally improve.

For example, the head at zero flow does not naturally increase with time. As pumps get older, all the loss mechanisms will tend to reduce the zero flow head. If the head does increase with time, then that is symptomatic of a reduction in internal running clearances. This is in itself a fault, and not natural.

There are cases where the pump has been handling an extremely fine abrasive suspended in the liquid. This has polished the cast surface and initially reduced the casing/impeller frictional loss. An improved performance results, for a while.

The author is also reminded of early attempts to use non-metallic material in high-pressure multistage pumps. The material was used as a replacement in wear rings. However, the high-pressure differential acting across the components caused them to slowly creep and flow — resulting in a progressive clearance reduction and an associated performance increase.

Pumps with 'unstable'[3] performance curves can also be an exception. Under certain circumstances, the pump head can *appear* to rise, as the pump wears. This is just an illusion, but can delude the unwary.

However these and similar circumstances are quite eccentric and it traditional to say, 'With time, pump performance always decays.' If the performance *does* appear to improve it will be the result *of some change to the pump, or its speed.*

**Palgrave's second rule of pumping:** When pumps performance decays, the slope of the characteristic curve always increases [curve becomes steeper].

Sometime the slope increase[4] is so small as to appear as a parallel shift at first glance. In other cases, the slope increase is quite profound.

There are few, if any, circumstances where the pump characteristic slope decreases as the pump wears. Note however, that when pumps handle liquids containing free gas, the curve slope may decrease significantly at low flow. Traditionally if it becomes less steep, then there has been a physical change to the pump.

**Palgrave's third rule of pumping:** The pump is always the guilty item, until proven innocent!

At this point in the book, it should be quite apparent that in many cases the system characteristics have equal or perhaps more responsibility in determining the overall performance of the combination. But this is not always self-evident and the path of least resistance is to blame the pump.

**Palgrave's fourth rule of pumping:** Pumps are never sized or bought for the real duty requirements.

They are always rated either for some fictitious future conditions, or to cover some outrageous contingency. This has the effect of obliging the pump to operate at a less favourable part of its performance curve, unless other steps are taken. The causes include

- The real duty is not yet known at the order stage, but will be firmed up later
- The real duty cannot be known until the pump is installed
- The real duty has been calculated, but 10% has been added as a contingency
- There is a future long term duty and a present short term duty
- An overage is required to permit some degree of process control across a control valve
- The pump is to replace, or compliment an existing [oversized] pump

Over-sizing the frictional component of system head is a defensible measure. Some of the factors used in head loss calculations are empirical and may not relate exactly to your circumstances. Some contingency in this respect [typically 10%] is understandable, so long as the compromises in operation are understood and allowed for. The static component is a direct measure and should have very much less uncertainty.

But it can get out of proportion. For a while, the author was involved with pumps used to drain very deep underground mines. In this case, the frictional resistance was typically only 5–8% of the overall head. An additional 10% contingency on the frictional head would be realistic though small, compared to the overall head. The main head component was the static lift from the underground pump station to the surface. However, the industry frequently applied the 10% margin rule to the **overall head** not just the friction head! This meant that the estimated head requirements were massively overstated. It was not uncommon to find that the pumps required either de-staging or [more usually] tiny breakdown orifice plates fitted to the pump discharge. The orifice dissipated the surplus head. On a more serious note, it would be hard to argue that having a latent surplus pumping capacity was not a good idea in this unpredictable and critical application.

Over-sizing means that the pump may perform at flows in excess of requirements. Most pumps have their inlet/outlet pressure routinely measured – because it is cheap and easy to do. Since only process[5] pumps routinely have their flow measured with any great precision, the *true* operating conditions of many pumps is not known.

Thus a manufacturer may receive a call from pump operator saying that on site the pump is generating a head lower than ordered. Whilst this may be true, it will also be true that the pump flow is *greater* than ordered. But as mentioned earlier, flow is less often measured. More often, pressure is used as an analogue via the performance curve. Unfortunately, due to the relatively small difference between head at zero flow, and head at b.e.p coupled with the questionable accuracy of site pressure gauges, this is not a very accurate analogue.

The consequences of this over specification are many:

- Possible cavitation noise and damage
- Driver overload
- Excessive control valve wear.
- Accelerated wear if abrasive solids present

The easiest solution is to bite the bullet and reduce the impeller diameter.

**Palgrave's fifth rule of pumping:** The degree of owner respect is proportional to the pump power.

As mentioned earlier, large and powerful pumps generally get a lot of attention during installation and operation. Small pumps are generally taken for granted on both counts. But when a small pump fails, it often has the capability to disable the plant, usually by upsetting some element of the infrastructure. This may be Fire-protection, or some Health and Safety related service.

## Interpretation of results

What can the foregoing instruments tell us? Remembering that most pumps are acquired to generate flow, any shortfall might become quickly apparent. However, this knowledge of flow does not always help in diagnosis.

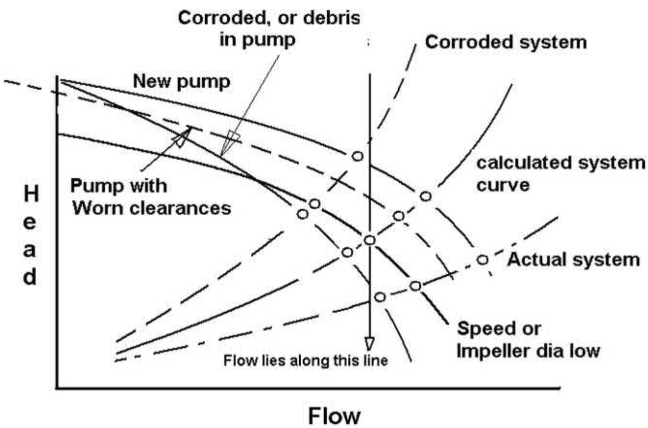



**Fig. 11.3** Inadequacy of only using flow for diagnosis purposes.

The pump could be operating anywhere along the line shown in Fig. 11.3. If the flow agrees with the design expectation, then this might seem nothing to worry about. However, recalling Palgrave's Rule 3 & 4, this might indicate there is actually a problem – the pump should appear to be delivering more!

Moreover, if there were a problem, we would still not be able to discriminate between the ailments described. So paradoxically, even although pumps are usually thought of as flow generators, knowing flow alone does not help much in problem diagnosis.

The problem has to be viewed from another direction in order to pinpoint the problem area. This involves measuring another performance parameter other than flow.

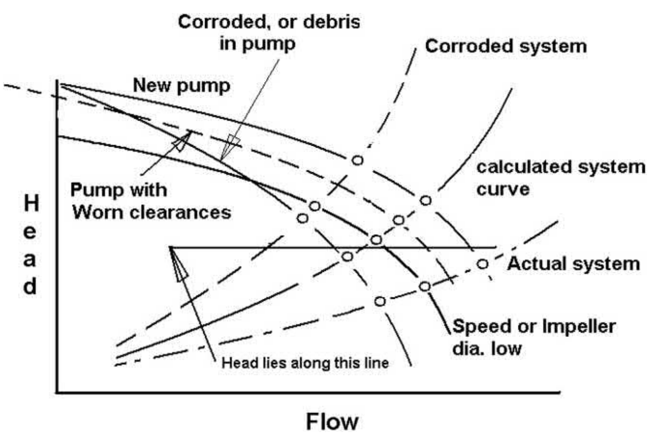



**Fig. 11.4** Inadequacy of only using head for diagnosis purposes.

If we only know the pump head while it is pumping, but nothing else, then this is also useful but not definitive. Referring to Fig. 11.4 it only tells us that the pump is operating somewhere along the shown line. We do not know what the flow is [Though we may be able to infer this if absorbed, power readings are available along with a test curve.]. We would not be able to differentiate between:

- A pump that is operating at its duty point
- A pump operating on a pipe system suffering from corrosion or fouling that has also afflicted the pump liquid passages causing its output to fall
- Pump that is drawing air into its suction.
- Pump speed low
- Clogged impeller
- A pump that is performing correctly in a system where the head has been overestimated [Palgrave's fourth Rule]

In all but the first example, the pump flow would be lower than desired. From one head point alone, this would not be obvious.

Let's look at the different analyses possible with the instrument scope defined in Table 11.4.

### *Case 1 You only know the suction head while operating at its natural flow*

This information is not very helpful. If the pipe diameter is known, then the velocity head [the difference between the static and the dynamic reading] does allow the pipe flow to be inferred. In most circumstances, this will be a very crude estimate, but in some situations, may be better than nothing.[6] Otherwise, suction head yields little information.

### *Case 2 You only know pump differential head generated while operating at its natural flow[7]*

This will have one of three possible outcomes:

### *The differential head at natural flow is higher than expected*

In accordance with the First Rule, if the head is high, it must have been high all along. It could not have progressively got higher. Alternatively the system curve may have got steeper [but observance of the Third Rule will hinder this possibility being considered of course].

Say that point [1] is the expected intersection between pump and system characteristic curves.

If the operating speed is too high, the pump intersection moves from point [1] to point [3]. The differential head would be perceived as higher than expected, which is true. However, the pump flow will also be higher, as would be the absorbed power [Since the NPSH [R] increases with speed, the pump might move into a cavitation situation.]. In this case, the differential head would probably appear less than expected (Fig. 11.5).

If the impeller diameter was oversize, then more or less the same picture would be presented.[8] This situation can sometimes arise when a spare

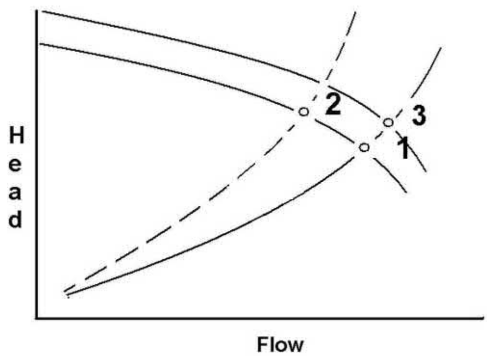



**Fig. 11.5** Possibilities for apparent high differential head readings.

impeller is fitted that has not been reduced to suit the duty. Instead, the maximum diameter has been fitted. The outward signs, apart from a high head, might be motor overload as well as flow increase – if they were measured.

In almost every practical case, the actual system resistance curve turns out to be flatter than that calculated [Palgrave's fourth Rule]. This brings with it other problems – see later. It is possible, but unusual, for the actual system curve to be steeper than calculated. In this case, the actual intersection of the characteristics occurs at point [2], instead of point [1]. This is at a higher head than expected [but at a lower flow!].

With most centrifugal pumps, this will be associated with a reduction in absorbed power.

With some mixed flow pumps, and almost all axial flow pumps, the power will increase as flow is reduced in this way.

### *The differential head at natural flow meets expectations*

In many cases, this might be indicative of a problem! Bearing in mind Palgrave's fifth Rule of pumping, one would expect the head at natural flow [1] be lower than calculated [2], with the flow and power correspondingly higher. This statement excludes systems where a feedback control valve governs pump flow. If the control valve is assumed to initially dissipate about 10% of the total friction head [a typical value used in system design] then in many cases it will be dissipating more than this. If it were set to only dissipate the 10%, the natural system flow would almost certainly exceed specification.

In systems without an automatic control valve, the most probable causes of extra system resistance are (Fig. 11.6):

- Fouled strainer
- Isolation valves not fully open
- Debris in pipe work

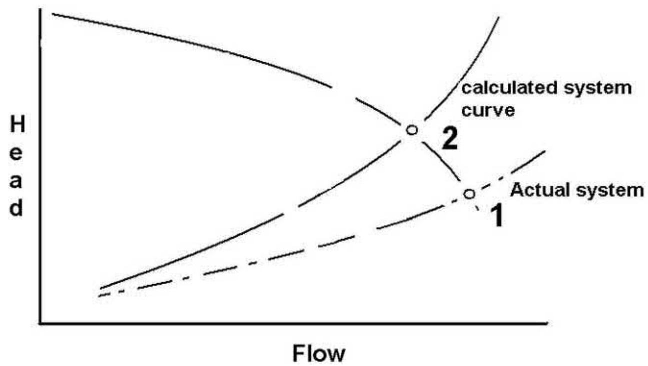



**Fig. 11.6** Actual and calculated system heads may differ, mainly due to contingencies In the calculations.

### *The differential head at natural flow is lower than expected*

Following on from the above, if the natural differential head is lower than expected, it may indicate that everything is satisfactory! This would be supported if absorbed power were slightly higher than expected.

If on the other hand the power is also lower than expected, this most probably indicates a problem.

- Choked impeller
- Speed
- Wrong diameter

The First and Second Rules apply in this situation.

### *Case 3 You know pump differential head at both the natural flow, and at zero flow*

If a second data point can be taken, then that increases our understanding of the pump problem significantly. The most useful point is at, or near to, zero pump flow. Two data points allow you to form some view as to the actual pump curve shape.

Normally, operation at zero flow is always discouraged. The shaft bending forces are at a maximum here and axial loads are likely to be high too.[9] The disturbing forces of inlet backflow will assume significance Appendix D and the liquid trapped in the casing will steadily warm due to the churning effect. Appendix H.

In practice, pumps of a few horsepower [under 10] can be run at zero flow for some little time [up to 30 s]. Of course, the life of bearings and seals will be slightly eroded, but not seriously. Moreover, 30 s is unlikely to be enough for the trapped volume to heat up dangerously. On the other hand, pumps of high horsepower [several hundreds, or more] can sustain significant damage even if run at zero flow for just a few seconds. This is particularly true of multistage pumps where hundreds or maybe thousands of horsepower are channelled into each stage. The stage itself might contain little more than a couple of kettle-fulls of liquid, so boiling and vapourisation might occur in just a few seconds!

However, all is not lost – running these high power pumps at minimum flow[10] can still yield useful data. In most cases, the head at minimum flow is not greatly different from that at zero flow.

The outcome of a zero flow [or near zero flow] test will fall into one of three groups, each with three subdivisions.

### *Case 4 The zero flow differential head is higher than expected*
### And the differential head at natural flow is also higher than expected

The Second Rule eliminates this being a wear-related effect. It must mean a constructional change. The Third Rule also applies (Fig. 11.7)!

Among the possible causes for this set of conditions are:

- Speed too high – Point [2]
- Impeller diameter too large – Point [2]
- All the above, plus a fouled system – Point [3]
- Head Gauges not zeroing or wrong in other ways – i.e. calibration error or gauge fault

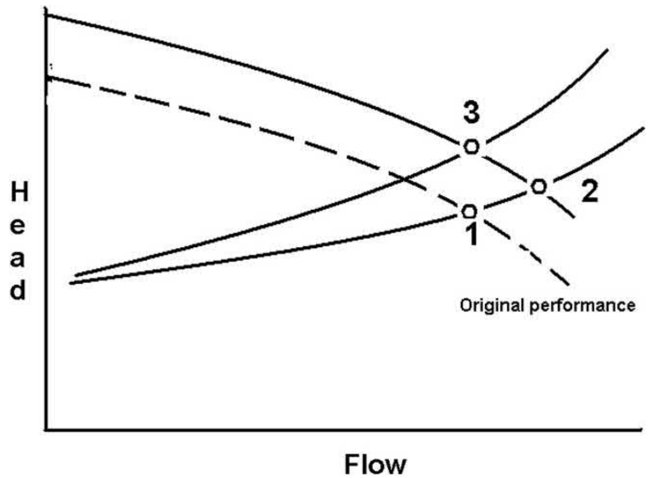



**Fig. 11.7** Some causes for high head at natural flow, and zero flow.

## And differential head at natural flow is about as expected

The implied increase in curve slope is in accordance with the Second Rule, but the First Rule means that there must have been a constructional change (Fig. 11.8).

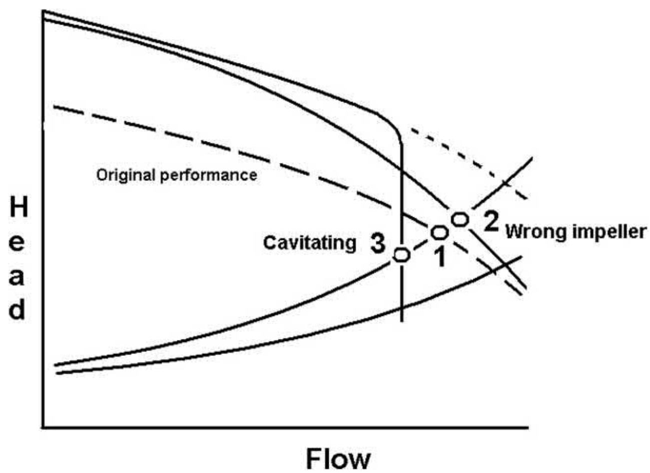



**Fig. 11.8** Possible conditions leading to correct head at natural flow with high head at zero flow.

Possible causes for this are:

- Alternative low capacity impeller, but diameter too large – Point [2]
- Correct impeller but speed too high – causing cavitation and head decay – Point [3]

All the above would be exaggerated if, as often happens, the actual system curve lies below that assumed.

**And differential head at natural flow is lower than expected**

Again the Second and Sixth Rule apply.

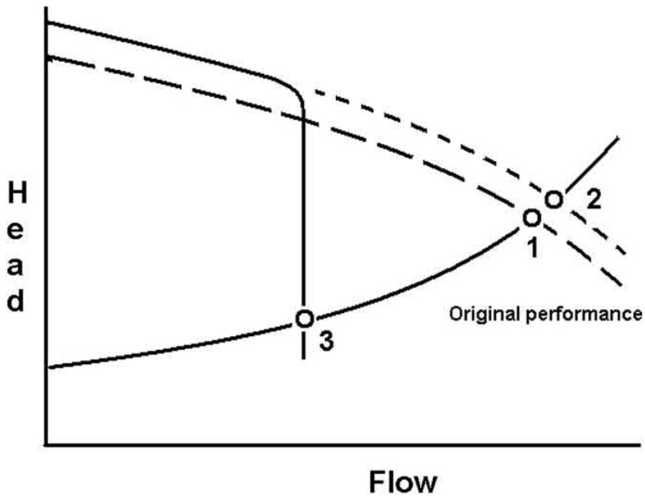



**Fig. 11.9** Cavitation can cause low head – irrespective of head at zero flow.

The root causes outlined above may all become magnified, which will cause an even greater head variation.

For example, an even more severe reduction in NPSH, or increase in speed will cause the effect shown in Fig. 11.9. The pump flow will be reduced further.

***Case 5 The zero flow differential head is close to expectations***
**And differential head at natural flow is higher than expected**

This violates the Second and Sixth Rule, meaning that this is not the effect of wear, but some constructional change to the pump.

Possibilities are:

- Wrong impeller [High output version] – Point [2] in Fig. 11.10
- Impeller is correct, but system is fouled, or valve partly closed – Point [3]

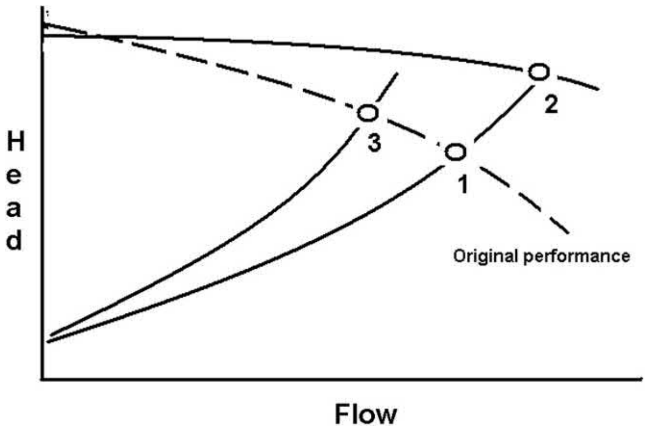



**Fig. 11.10** Causes of high head at natural flow coupled to normal head at zero flow.

## And differential head at natural flow is about as expected

If head at natural flow is also close to expectations, then chances are high that the pump is delivering the correct flow [Point 1]. The pump is likely to be in good condition *see* Fig. 11.11.

However, as noted earlier, this situation can, paradoxically, because for some mild concern Palgrave's fourth Rule implies that most pumps should disclose low head at natural flow, and be correct at or near to zero flow. To have the correct head at natural flow would be fortunate, but most probably there is another minor fault.

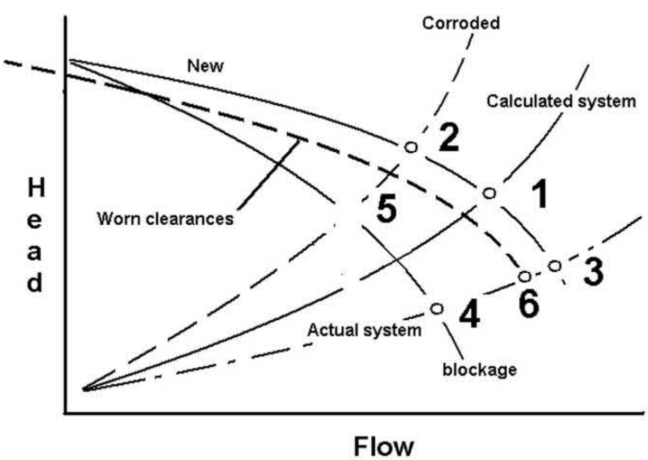



**Fig. 11.11** Correct head at natural flow.

For example:

- Fouled or corroded system
- Unexpected loss in pipe system.

This may be in conjunction with

- A blocked impeller
- Worn clearances

These issues would mean that the natural flow would tend to reduce and may even approach the design flow!

### And differential head at natural flow is lower than expected

Referring to the previous Figure:

- The pump is in good condition, but the original head requirements were excessive [Palgrave's fourth Rule].
- This means that the natural flow occurs at higher than the design flow – Point [1]
- The pump is in good condition, but there is debris jammed in the impeller reducing its flow capacity – Point [4]
- The pump is suffering front worn wear rings [increased clearance if an open impeller] Point [4]. Flow is low.
- Pump is cavitating and not achieving its design flow, *see* Fig. 11.12

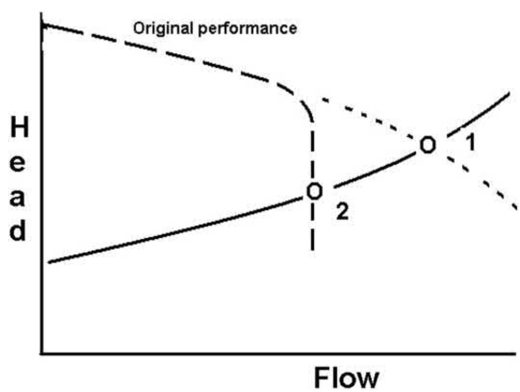



**Fig. 11.12** Cavitation reduces performance from point [1] to point [2].

### The zero flow differential head is lower than expected and differential head at natural flow is also higher than expected

This situation violates the first and second Rule, so the pump construction must have been changed in some way.

Possibilities include (Fig. 11.13):

- Wrong impeller fitted [2]
- System fouled [3]

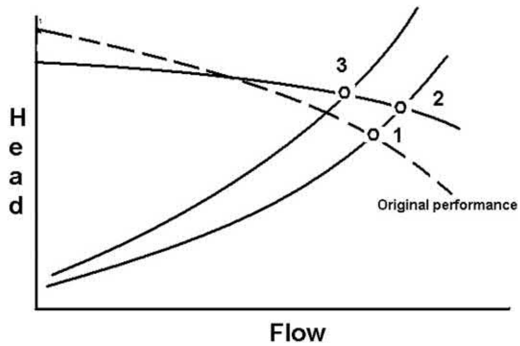



**Fig. 11.13** Some reasons for high head at natural flow.

### And differential head at natural flow is about as expected

This situation is a milder version and much the same analysis applies. The conditions also overlap those below.

### And differential head at natural flow is lower than expected

Some possibilities include (Fig. 11.14):

- The pump speed is low – Point [7] as is flow
- The pump impeller diameter is too small – Point [8] as is flow
- Air/Gas is getting into the suction pipe – Point [9]. The pump flow will be low.
- Air or gas evolution

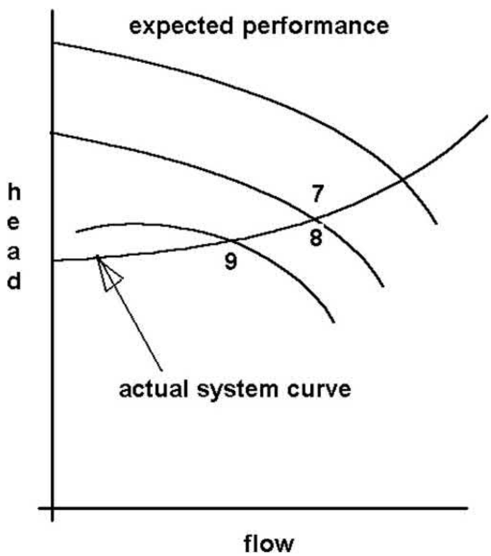



**Fig. 11.14** Some possible reasons for low pump output.

- Air entraining vortex

In all the above examples, there could be other root causes, but the diversity already evident in even these few should be enough to convince sceptics that using head alone [even if two extreme points are taken] is not a very powerful analysis tool.

### *Case 6 You only know the natural system flow*

Fig. 11.3 indicates that knowing flow only allows you to gauge whether the pumping scheme is working well or not. On its own it helps very little in diagnosing problems. To reiterate again, a statement of 'low in flow' or 'high in flow' does not help much unless some sense of the system resistance curve shape has also been conveyed. There are just too many reasons why flow might be wrong and without knowing at least one more parameter, the root cause cannot be pinned down. Ideally the second parameter should be pump head.

### *Case 7 Only know absorbed power [or its analogue]*

Power is the least satisfactory measurement. A graph of power as a function of flow is quite flat, in most cases his means that unless the power can be read very precisely, it is not possible to infer the pump performance very accurately. It would be uncommon to find that anything more than rudimentary power measurement facilities exist (Fig. 11.15).

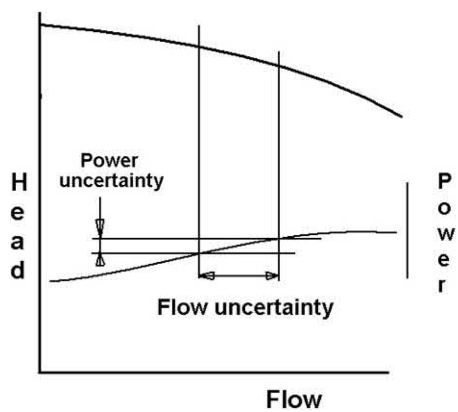



**Fig. 11.15** Unsuitability of power as an accurate flow analogue.

In any event, the same constraints apply to differentiating common performance problems.

## Don't use pressure gauge as a flow analogue because it introduces uncertainty

Although pumps are frequently equipped with means of measuring discharge pressure, they rarely have direct flow measurement. The logic here is that if the pump develops correct pressure, then it is passing the correct flow. The flaw in this logic is that many pumps have so called 'flat' or 'shallow' characteristic curves (Fig. 11.16).

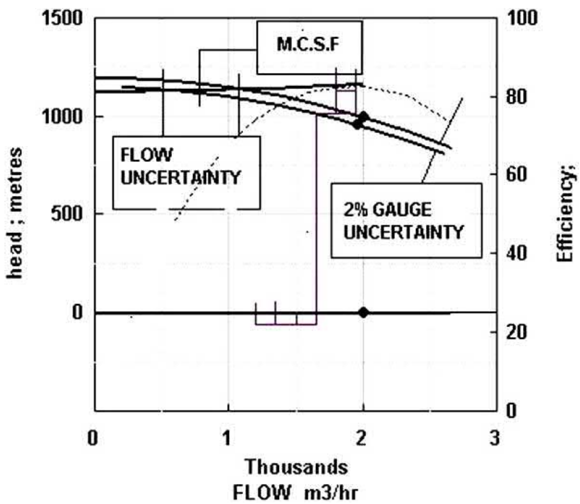



**Fig. 11.16** Inadequacy of using pressure gauges to infer flowrate.

This means that the pressure seems to change very little over a wide range of flows. Furthermore, the pressure gauges are often small and make accurate reading difficult. This means that the pump flow can be too high or too low, yet this is not immediately obvious. The operator can conclude that the flow is approximately correct.

In some cases, pumps are selected where the rated flow is near to minimum continuous flow, *see* Chapter 10. This occurs with difficult services – particularly where the flow is low and the head is high. Such apparently poor selections, might in fact, be the only practical choice.

Near minimum flow, the pump curve shape is flat anyway, so using discharge pressure as the only analogue of flow is prone to vast error. In fact, the pump may well be operating well below MCSF. The operator has to look for other signs to feel confident that the pump has started satisfactorily. The only real corroboration can come from the absorbed power, for which an ammeter is the normal tool. Even so, site instruments are nearly as difficult to read and interpret as the pressure gauges. But if both analogues come to similar conclusions on flow, that should be enough to bolster confidence that the pump flow is correct, or otherwise. An important exception is when this is the first time the pump has been started in the system. In this case, Palgrave's fourth Rule may be invisibly at work.

In the fullness of time, over or under pumping will become apparent in other ways and allow a more accurate assessment to be made. If the pump forms part of a process, then the process flow as a whole may be monitored. For example the rate of level change in a supply tank is a very useful and accurate method of flow measurement, but it takes some finite time to conduct such measures.

## System resistance changes

During problem solving it is traditional to first assume that the issue lies with the pump [Palgrave's third Rule]. However, it should have become evident that the system also plays almost as large a role in determining the overall system [performance].

In the foregoing, some standard system effects have been alluded to. It would be worth summarising them in one short section.

System faults fall into two main groups:

- System head demands change slowly
- System head demands suddenly change

## System head demands change slowly

### *Head demands slowly increase*

It is quite normal for the system head requirements to change with time. Normally it increases slowly. Reasons for this includes:

- General ageing and corrosion
- Pipe bore deposits that reduce the cross sectional area and increase the velocity [Wax deposits, Biological growth, Freezing]
- Static level difference between source and destination may increase with time.
- Product viscosity increases with time

### *Head demands slowly reduce*

This is quite unusual. When it happens, it is most likely due to:

- Static level difference between source and destination may reduce with time
- Pipe system leak develops close to the pump.

- Product viscosity reduces.
- Changes occur that will discourage pipe wall deposits [i.e. temperature increase reduces waxy deposits on pipe wall]

## System head suddenly changes

### *Head demand suddenly increases*

- Isolation Valve partly closed inadvertently
- Pipe internal liner becomes detached
- Static level difference between source and destination suddenly increases
- Additional pipe loop added or included in system
- Symphonic recovery effect lost at discharge [due to lowering static levels at discharge uncovering discharge pipe]

### *Head demand suddenly decreases*

- Symphonic effect created at discharge [due to submerged discharge]
- Discharge line fractured
- Static level difference between source and destination suddenly decreases

## Increased wear ring clearances

In Category 3 problems, work put into the liquid by the impeller is wasted before it even leaves the machine. One example of this is wear ring or wear plate leakage.

Traditionally centrifugal pumps have small clearances between the stationary and rotating parts. This largely isolates and seals the high-pressure zones of the machine from the zones of low pressure. The detail design relates to the type of impeller (Fig. 11.17).

With fully shrouded – closed impellers, these sealing clearances take the form of cylindrical gaps or complex labyrinths. Because of their torturous passage shape, labyrinths are more effective at containing the internal leakage and this is reflected in higher pump efficiency. But they are more costly to produce and more applicable to machines installed in applications where Life Cycle Costs are a major consideration. Most clearances are usually made renewable since the clearances will gradually widen. This results from any abrasive particle entrained in the pumpage, or even due to liquid velocity erosion.

With open or semi open impellers, the sealing clearances are arranged axially (Fig. 11.18).

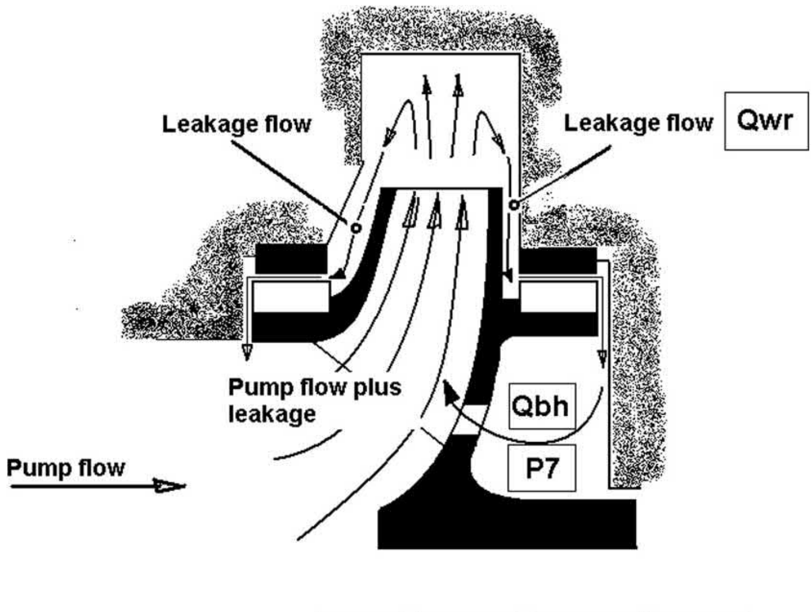



**Fig. 11.17** Wear ring and leakage flow in a fully shrouded impeller.

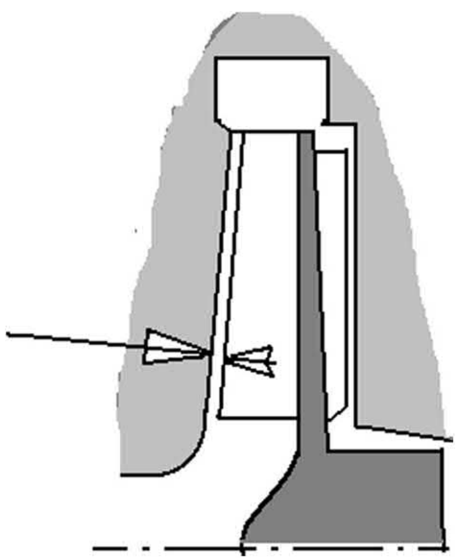


**Fig. 11.18** Leakage gap In open and semi open impellers.

In some cases the mating surface of the casing are also made renewable. This is prevalent if the product is abrasive, or contains fibrous material. Clearance between stationary and rotating parts can be restored by axial resetting of impeller.

To understand the effect, it is perhaps helpful to explain one aspect of pump performance curve composition [The following description is developed and based on a shrouded impeller with wear rings. However, the logic and effect is the same for open or semi open impellers. Only the mechanism of leakage changes.[11]].

Since all executions are designed to be non-contacting, they cannot seal perfectly[12] and some pumpage escape to be re-circulated. One outcome of this is that not all the flow generated by the impeller appears at the discharge. Irrespective of impeller type, this re-circulated leakage has the same effect on performance. Formally, this leakage is described by the term volumetric efficiency.

The normal test curves that we are familiar with only represent the performance as indicated by the amount of liquid issuing from the pump discharge nozzle. This is not the same performance as that of the impeller! The impeller discharges more flow that this, but it does not all appear at the discharge nozzle. Even when the wear rings are new, a small percentage of liquid is wastefully re-circulated by the impeller. So impeller flow is greater than the pump flow.

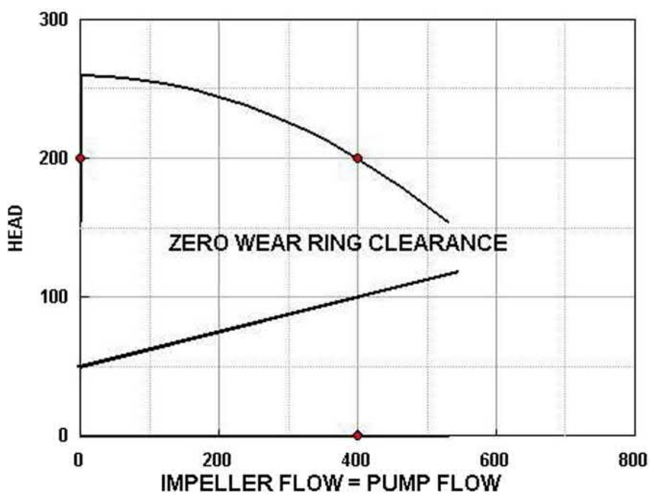



**Fig. 11.19** Pump performance and impeller performance are the same at zero leakage losses.

In Fig. 11.19 is shown a performance curve for the machine with zero re-circulation leakage [i.e. volumetric efficiency is 100%]. This is not a normally achievable value, but is used for illustrative purposes. In this particular case, the performance curve for the pump is then the same as that for the impeller.

The graph for impeller flow and pump flow are superimposed, but since there is no leakage flow, they appear to be one and the same (Fig. 11.20).

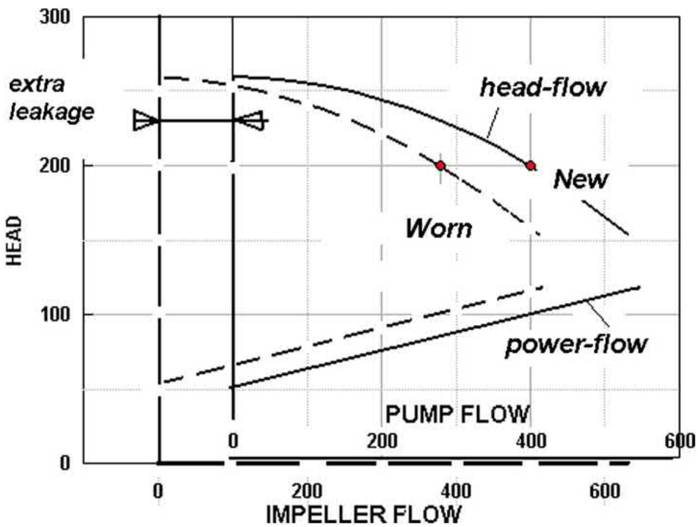



**Fig. 11.20** Effect of wear ring leakage is to displace the impeller curve leftwards relative to the zero leakage curves. This yields the familiar pump curve.

Now imagine a more practical case where the re-circulation – leakage is finite – [say 5%, which represents a practical value for many pumps in the 'new' condition].[13] The flow passing through the impeller is greater, by that amount, than that discharging through the pump. Therefore, the hydraulic performance for the impeller and pump differ. The difference is the re-circulation – leakage flow. Even when the flow from the pump is totally throttled to zero, the impeller still passes the leakage flow – conceptually, the impeller curve is displaced to the left of the pump curve by the amount of leakage. Projecting the displaced impeller curve back onto the pump curve gives a comparison of the leakage effect. The impeller flow has not changed, but the flow from the pump seems to have reduced. At the same time, the absorbed power seems to have increased.

The leakage shift is often portrayed as a constant displacement of the curve – an approach I have used in this explanation. This is a simplification.

The leakage flows are actually influenced by the head differences within the machine. In turn, these are related to the differential head of the pump. In practice the amount of leakage flow reduces as the pump head reduces. This means that the leakage flow reduces as the pump capacity increase [This simplification is rarely a practical issue, but needs to be stated]. For most practical purposes, the curve for 100% volumetric efficiency can be assumed to shift left by an amount equal to the leakage flow.

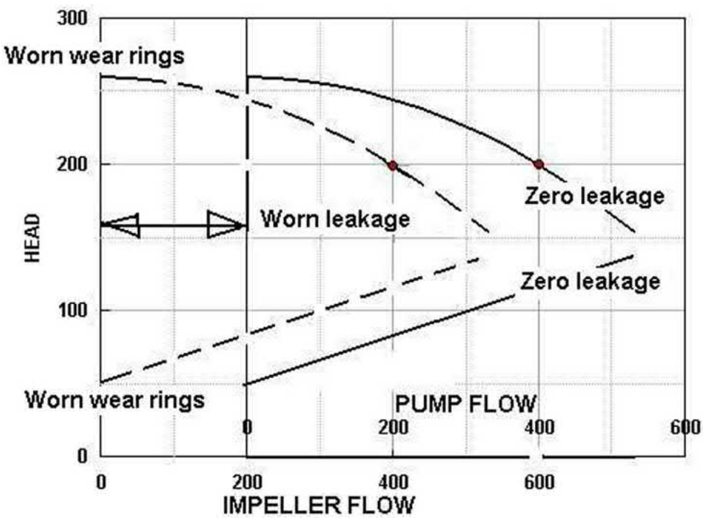



**Fig. 11.21** As rings wear, curve is displaced more and more to the right.

Fig. 11.21 portrays the effect of extensive wear. Again, the pump performance seems to have decayed further, while the absorbed power has increased further. But so far as the impeller is concerned, nothing has changed. It still passes the same flow as when it had new clearances. It's just that less of that flow goes down the discharge pipe.

For clarity, the pump efficiency has been omitted. But clearly since the head has reduced and the power has increased, then the efficiency will have deteriorated quite substantially.

## External symptoms

One outward effect of wear is to alter the pump curve shape. This is often mooted as a useful symptom of increased wear. However, the precise nature has been frequently reported incorrectly. On the one hand, the effect of wear has been described as a [superficially] parallel shift of the performance curve. On the other, the effect is described as more or less pivoting the curve about its closed valve head value. Actually, both are possible, but which one depends upon the initial pump curve shape.

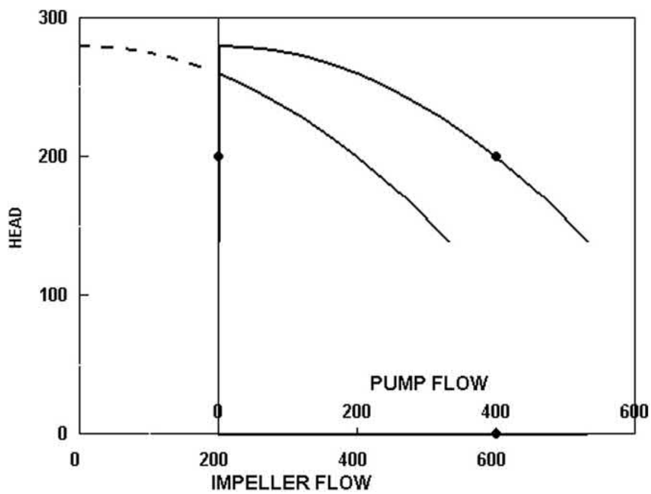



**Fig. 11.22** Effect of wear on a pump with an inherently steep curve shape.

Fig. 11.22 shows the effect of wear on a pump that inherently has a rather steep characteristic [In this sense the term 'steep' means that the closed valve head is more than 20% greater than the head at the best efficiency flow of the pump.]. This does indeed show a superficially constant shift that might seem parallel to the 'as new' curve over much of the useful pump flow range.

But beware, a pump which had a slightly undersized impeller diameter, would give an outwardly similar symptom, or where the pump speed was a little reduced. Further study would permit both these root causes to be distinguished. In both these cases, the pump power would be slightly reduced with respect to flow — whereas in the case of leakage, the absorbed power would appear to increase.

This is a good general example of conditions were completely different causes appear to have much the same overall effect. Deeper investigations would be needed before the root cause could be then ascertained. This sort of deeper investigation is not easy to accomplish in site testing, because the scope of instrumentation is rarely extensive enough. In many cases, the range of flows possible is also quite limited. Only having such a narrow 'window' on the problem, might not lead to a conclusive view on the root cause.

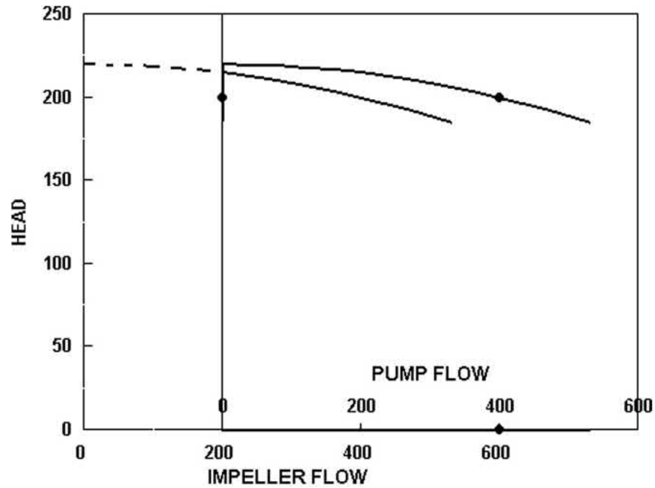



**Fig. 11.23** Effect of wear on a pump with an inherently flat curve shape.

Fig. 11.22 describes the effect with a steep pump curve. On the other hand, Fig. 11.23 shows the effect on a pump that has a rather flat curve [Both the steep and flat curves examples have had the same leakage flow shift subtracted.]. In contrast, this reveals a curve that, at first sight, shows little change at closed valve, but a head departure that seems flow-related.

This describes very different behaviour compared to the pump with a steep curve. Furthermore, this sort of head-flow behaviour might easily be confused with, say restriction in the casing causing Category 3 or 4 loss [Chapter 5] such as an orifice-type of loss, or an increase in impeller or casing surface roughness [*see* next section].

As before, the characteristics of the absorbed power curve would help distinguish the root cause. Performance decay due to clearance increase would have an associated increase in absorbed power, whereas an orifice effect, or increase in internal friction would show no power change [with respect to flow].

These two examples should help show that the way the curve shape appears to change is largely influenced by the initial curve shape. In addition, these changes might look similar to the changes brought about by other defects, until further factors are considered.

This discussion has centred upon wear in rings and bushes. Previous Chapters described how many multistage pumps utilise one form of balance device or another to help offset the large inherent hydraulic thrusts. Such devices, involve bleeding high-pressure liquid back to the suction pipe[14] through fine clearances. Hence these devices are just as vulnerable to wear as rings and bushes – maybe more so. Wear in these devices has much the same effect on performance.

- Some pumps have the so-called balance flow piped internally back to suction
- Others have it piped back to the source – in which case it partly constitutes a permanent leak off protection
- Still others dump the balance flow to drain.

The last two examples offer scope for directly measuring the leakage. Since this can be considered as an analogue of the general condition of all clearances within the machine it is useful data. The rate of change with respect to time can be a good guide as to the wear rate within the machine clearances in general.

## Summary

This complex leakage and curve shape relationship can all be summarised in a chart.

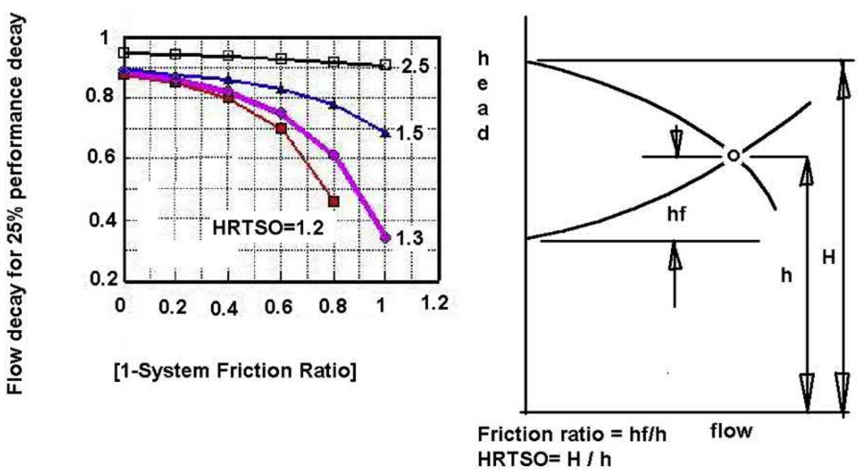



**Fig. 11.24** Apparent wear sensitivity as a function of pump curve shape and system characteristic.

On the x-axis is shown the static head ratio, as defined in Fig. 11.24. The Y-axis shows the expected reduction in system flow due to a 10% reduction in pump performance. The various curves represent different pump HRTSO.

For example if the friction head ratio is 0.6, and the HRTSO is 1.2, then the system flow will reduce to about 70% if the pump wears 10%. If the pump HRTSO could realistically be increased to 1.5, then the system flow would only reduce to 82%.

## Unstable performance curves

Even if the pump has an unstable performance curve, then as it wears, it will behave in the same manner as a pump with a stable performance curve (Fig. 11.25).

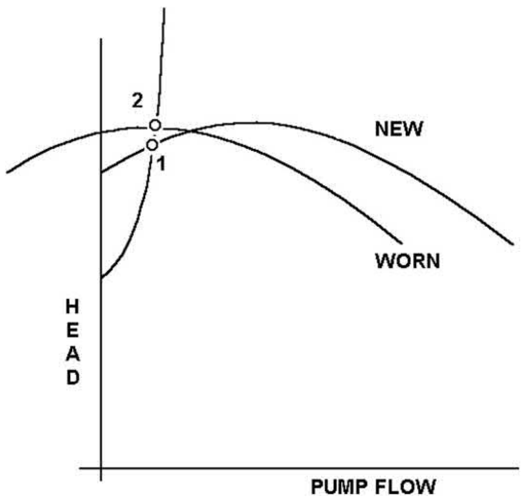



**Fig. 11.25** Unstable curves can appear to create anomalies.

However, there is one set of circumstances where this is not true, and it appears to violate Palgrave's First Rule. This happens if the system characteristic is very steep, and the pump is operating on the part of its curve where head increase with flow. In these circumstances, the pump will initially operate at point [1] but as the pump wears [and the curve appears to move to the left] the intersection moves to point [2], This creates the false illusion that the pump head is increasing, and apparently violating Palgrave's First Rule.

## Impeller balance hole effects

Impellers with back wear rings are usually also equipped with balance holes. Their purpose is to bleed back-wear-ring leakage back to the impeller suction. In doing this, they ensure that the pressure in the impeller rear chamber is almost at suction pressure. Almost, but not quite!

In order to drive the liquid through the balance holes, it is necessary to have a small but finite pressure difference across them, which means that the chamber pressure exceeds suction pressure. Since the [inboard] seal environment is closely related to this chamber pressure, it is a parameter that evokes a lot of interest.

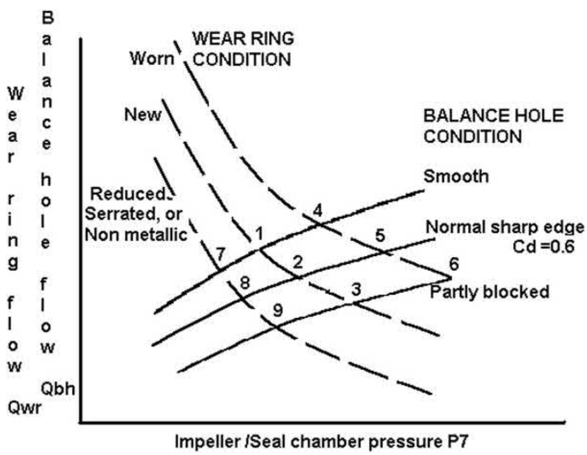



**Fig. 11.26** Pressure in the seal chamber area has complex dependence upon wear ring design/condition & impeller balance holes.

Let's consider the dynamics of the situation, and let's look at the wear ring and balance holes separately.

The flow across the wear ring annulus can be calculated by any number of standard methods, once the differential head is known. The main uncertainty is in estimating the head just before the wear ring. This is somewhat less than the pump head — being reduced by the velocity head in the volute, and the liquid whirl in the impeller-casing sidewall gap. Typically, this head will be in the order of 60% of pump head. If the head in the impeller cavity is known, the differential head can be used to estimate the leakage flow. Of course, the chamber head is not known at this stage, so a range of values is used yielding a range of flows. This gives the 'New' curve shown in Fig. 11.26.

The balance hole flow is estimated from simple orifice theory. In this case, the differential head is between impeller chamber and suction pressure. In a range of tests, the author found that a discharge coefficient of 0.6 was a representative value for drilled holes with sharp edges. Again, the chamber head is not known, but balance hole flows can be evaluated for a range of values. This gives the 'Normal sharp edged' curve in Fig. 11.26.

These two curves intersect at a certain flow and chamber pressure Fig. 11.26 [Point 2]. This chamber pressure is the only value that gives equilibrium, i.e. where the wear ring flow equals the balance hole flow.

Now lets us look at what happens when things change:

If the wear ring clearances increase, then the equilibrium point moves from point [2] to point [5]. In other words, the leakage flow increases, and the impeller cavity pressure increases.

If the wear ring clearances are reduced [say, by the use of a non-galling material such as PEEK, or carbon] then the equilibrium intersection moves from point [2] to point [8]. In other words, the leakage flow reduces, as does the chamber pressure. Much the same happens if the wear ring clearances are serrated, or replaced with a complex labyrinth design.

If the balance holes are not sharp edged, but instead are smooth, or chamfered, the intersection moves from point [2] to point [1]. Since this increase the equilibrium leakage, there would seem little point in doing this. However, simultaneously reducing the chamber pressure might be useful in certain axial thrust situations.

If the balance holes become partly blocked or plugged, the equilibrium point moves from point [2] to point [3]. This increases the chamber pressure and decreases the leakage flow. Increasing the chamber pressure increases the component of axial thrust that is tending to push the impeller away from the seal and towards the suction branch. This can be of benefit in cases where the suction pressure is very high. In such cases, the piston effect of suction pressure acting on the shaft section that passes through the shaft seal exerts a high axial thrust. This thrust tends to push the impeller away from the suction branch, and towards the seal. Consequently, the countering thrust resulting from partly blocked balance holes would lower the overall load on the thrust bearing. In fact sometimes, on high suction pressure service, the impeller balance holes are deliberately blocked, in order to encourage this countering force and reduce bearing load.

On the other hand, if the suction pressure is not very high, the resultant thrusts are already towards the suction branch [away from the seal]. This means that any thrusts resulting from partly blocked balance holes would increase the bearing loading in this case.

## Internal performance head loss

In Category 3 problems, work put into the liquid by the impeller is wasted before it even leaves the machine. An example of this is internal friction loss.

Some fraction of the work put into a pump, via the impeller, is immediately dissipated in fluid friction within the impeller passages and casing channels. This loss is related to the roughness of the wetted surfaces, and the liquid velocity. The surface roughness or 'finish quality' can have a profound effect on performance and maintaining it is often a matter of some urgency. Certain pump designs are more susceptible than others.

The performance of pumps which are relatively narrow [impeller water passage width much less than 10% of the diameter] are quite sensitive to roughness in the casing, while the impeller surface is not so critical. This is average because velocities in the casing will be quite high, whereas in the impeller they are somewhat lower.

With wide impellers [much more than 20% of the impeller diameter], surface finish is equally important on casing and impeller.

Surface roughness is dictated by the manufacturing process and the susceptibility to surface corrosion. Dealing first with the manufacturing process:

## Influence of manufacturing methods

Most pumps are just a collection of castings, mouldings, or pressings. The roughness of mouldings and pressings are, in pump designer's terms, very low. Castings, on the other hand, demonstrate a whole range of finishes, depending upon the manufacturing process. Precision process such as lost wax [investment castings] produces the best finishes, but fall short of mouldings or pressings. Sand castings have the worst finishes, though the use of ceramic cores can improve matters. In some cases the surface finish of sand castings is manually improved by the manufacturer, using techniques such as filing or grinding or abrasive slurries. This is useful if the base material is corrosion resistant.

Often though the surface is left unattended – either because it raise the costs [of prosaic machines], or the materials is too hard to yield any improvement attempts [Nickel irons]. With other materials, such as cast iron, any manual surface improvement is futile because it immediately begins to rust when run for a short period. Impellers in these materials rarely have surface upgrades – they are just installed 'as cast'.

Another method of surface improvement is to use surface coatings, *see* Figs 11.27 and 11.28. This is useful with materials such as cast iron, since the improvement in surface finish is more permanent than just grinding and filing.

Modern machines tend to automatically have good surface finish in order to demonstrate high efficiency, in this energy conscious climate. But some older machines are capable of useful performance improvement by attention to the surface roughness.

Surfaces can be improved by grinding and filing – just as the manufacturer would do. As mentioned above, this is only likely to have long term benefit on non-corroding material.

## Surface corrosion

Materials such as carbon steel and cast iron slowly corrode in water – roughening the surface in the process. Flake Cast iron in particular corrodes differentially at the flake boundaries creating an effect called tuberculation, which greatly enhances surface roughness. Hence the friction losses within the pump slowly increase. Less and less of the power provided appears as useful work at the pump discharge. To an outside observer, it is hard to differentiate between performance drop off caused by friction, and that caused by restriction or fouling in the pump discharge line. They both induce friction loss though seldom of the same degree. Chapter 10. Surface roughness loss appear as a 'square law effect'

- If the casing is of a corroding material but the impeller is not, then though the performance reduces, the power stays the same. It's just that less and less of the power results in useful work on the liquid. More and more of it is absorbed. This is typical of Category 3 problems.
- If the impeller is also of corroding material, the power will also increase. This is particularly true in those pumps where the impeller is quite narrow compared to its diameter [see above]. In such cases the disk friction loss also increases. This is the power required just to spin the impeller disk in the liquid. The power levels to do this are quite surprising. A 12″ impeller running at 3600 rpm consumes about 12 bhp.

A new impeller may siphon off some 5% of the total drive input just to overcome disk friction. By the time it has severely corroded, that may increase to more than 10%.

If only investigated over a narrow a band of flow, a corroded pump and a pump with worn clearance can demonstrate similar behaviour.

## Surface coatings

As noted earlier many older machines can benefit from treatment to restore the surface finish. Such 'efficiency enhancement' coatings are commercially available. It is worth noting that they can have two different overall effects. This mirrors the corrosion effects described above.

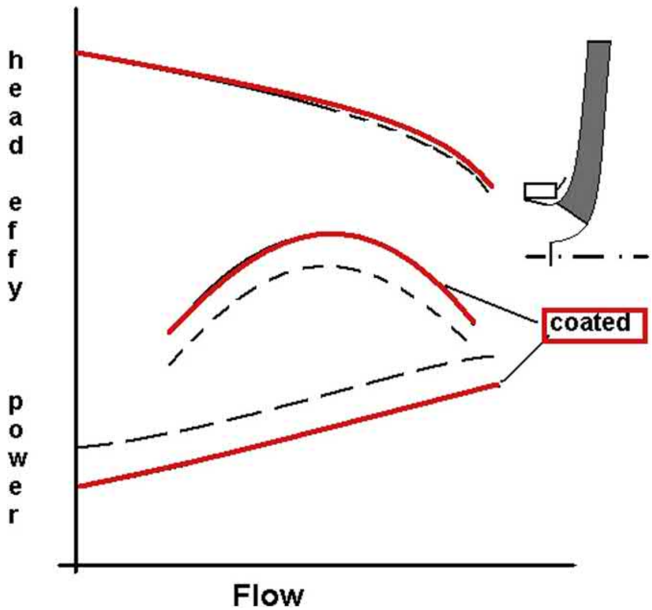



**Fig. 11.27** Coating narrow impellers mainly increases efficiency by reducing the disk friction power. The generated pump head changes very little.

If the pumps have relatively narrow impellers, then the main benefit of coatings is to reduce the power [absorbed in disk friction]. There will also be some improvement in head due to reduced casing friction loss. This sort of impeller usually shows the greatest efficiency improvement.

If the pumps have relatively wide impellers, the power remains much the same [There is not enough basic disk friction to materially affect performance]. As before, there will be some head increase due to reduced losses. However, it is unreasonable to expect the same level of efficiency improvement as with narrow impellers.

## Mechanical problems

The mechanical elements of pumps, such as bearings, couplings, gaskets, and to some extent shaft seals are not highly specific to the pump industry.

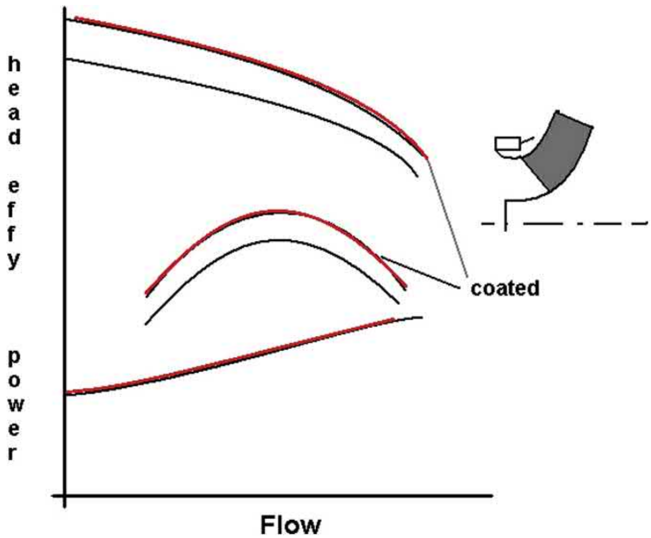



**Fig. 11.28** Coating wide impellers mainly increases efficiency by increasing head through reduced hydraulic losses. The absorbed power changes very little.

Most of them are found in several other areas of mechanical engineering. This means that a broad knowledge base already exists for the behaviour of these elements.

Here we will touch on a few areas that are more pump industry-specific (Tables 11.5–11.9).

**Table 11.5** Vibration levels increase.

| Cause |
|---|
| Sudden air entrainment<br>Pump – driver become mis-aligned<br>Shaft becomes bent<br>Bearings wear<br>Debris gets into impeller<br>Internal rubs develop<br>Coupling develops a problem<br>Foundation block develops a crack<br>Thermal shocks<br>Shaft 'shuttles' back and forth [usually caused by inlet backflow]<br>Loss of grout adhesion<br>Pump has handled significant amounts of abrasive solids pump is inadvertently operating at low flow NPSH [A] has reduced<br>The acoustic properties of the liquid had changed<br>Pipe work strain has caused excessive forces and moments to be applied to the pump flanges – Impeller or rotor is rubbing.<br>Pump has been started and stopped too frequently<br>Product temperature has risen significantly above design value. |

**Table 11.6** Noise levels increase.

| Cause |
|---|
| Air entrainment, or air entry [pipe flange, gland, etc] |
| Debris jammed inside of impeller |
| Fault with seal [dry run] |
| Operation at low flow |
| NPSHA reduced |
| The acoustic properties of the liquid had changed |

**Table 11.7** Seal fails.

| Effect |
|---|
| Fretting under dynamic O ring |
| Seal chamber not vented |
| Seal flush blocked |
| Dynamic O ring damaged during assembly |
| Seal runs dry |
| Leak under seal sleeve |

**Table 11.8** Bearings show distress.

| Cause |
|---|
| Pump and driver misaligned |
| Shaft bent |
| Dirt in bearing |
| Dirt in oil |
| Moisture in oil |
| Grease over-packed |
| Overloaded or approaching end of life |
| Fitting faults |
| Black oil |
| Hot, but may be inherent due to speed |
| Radial load vector is up! |
| No oil, or wrong level |
| Incorrect assembly |
| High pump vibration |
| Cavitating surge |
| Wrong oil or grease |
| Journal bearing fit too tight [low vibes but high tempo] |
| Fit to housing or shaft is too tight [reduces internal clearances] |
| Line bearing cannot float |
| Oil has absorbed atmospheric moisture and its properties have decayed. |
| Clogged balance holes |

**Table 11.9** Reasons why cavitation might develop.

| Cause |
|---|
| Sun warms suction line |
| Waxes form in suction line increasing the pressure drop |
| Pipe liner detaches in suction |
| Wear rings clearance increase |
| Suction strainer has become blocked |
| Static head reduces and pump over-pumps |

## Oil ring lubricators

Many pumps still use oil lubrication furnished by oil pick-up ring [oil ring]. A lot of myth has grown to surround how these devices work, and what the effects of changes are. The following notes, extracted from *Reference 21* will be useful in improving the general understanding of what affects oil ring performance. That paper contains much more detail.

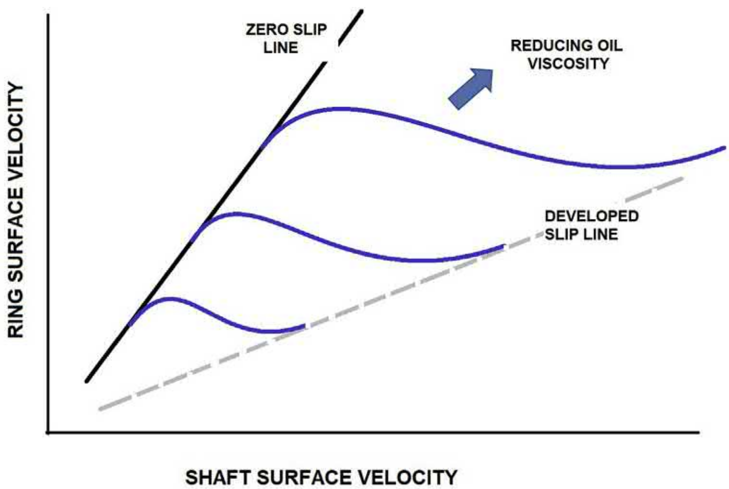



**Fig. 11.29** This shows how ring slip occurs at relatively low speed, depending upon oil viscosity [*Ref. 21*].

Fig. 11.29 shows test data plotting the shaft rubbing speed against the ring rubbing speed. At low speed there is negligible slip between the oil ring and the shaft. As the shaft speed increase the oil ring initially keeps pace with the shaft surface, but at a certain speed, the ring starts to slip. A further increase in shaft speed brings only a small increase in oil ring speed, and of course quantity of oil transported. This need not cause concern, if the temperature is deemed satisfactory. Above a certain speed the oil ring slips more, and not enough oil is being transported into the journal to transfer the heat away. At some extreme the oil supply is depleted so much and the temperature rises so much that the viscosity is critically reduced. The oil ring ceases to function effectively and the bearing fails.

Experience in pumps says that, at shaft surface speeds in excess of 14 M/S[15] [45 ft/s] the problems mentioned above become increasingly likely. The oil ring is very likely to 'skid' on the shaft causing markings and producing micro-swarf. On the other hand, electric motor designers seem able to routinely run at speeds in excess of this – generally with rings of heavier cross section.

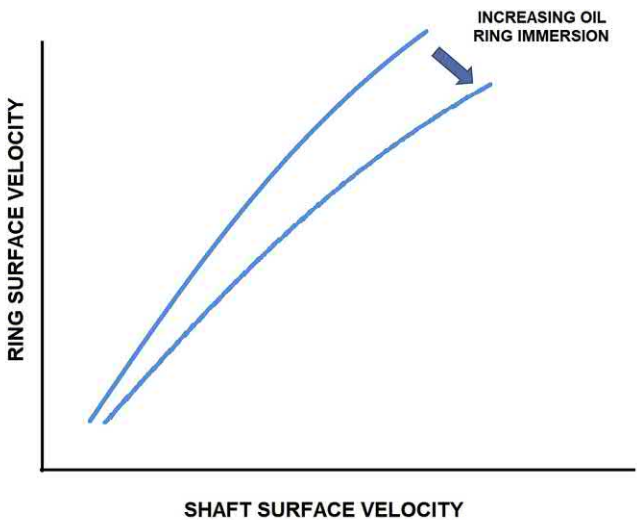



**Fig. 11.30** Effects of different oil ring immersion on ring speed [*Ref. 21*].

Another issue relates to the depth of immersion. Fig. 11.30 gives some idea as to how the ring speed is affected by the depth of immersion. Too much and the viscous drag slows the ring – too little and the ring does not pick up enough. Clearly greater immersion is associated with more viscous drag on the ring itself, and this results in lower ring speed. The oil ring itself will normally swing away from the vertical 'rest' position by around 15°, in the direction of shaft rotation. Higher immersions damp oscillations in these swings.

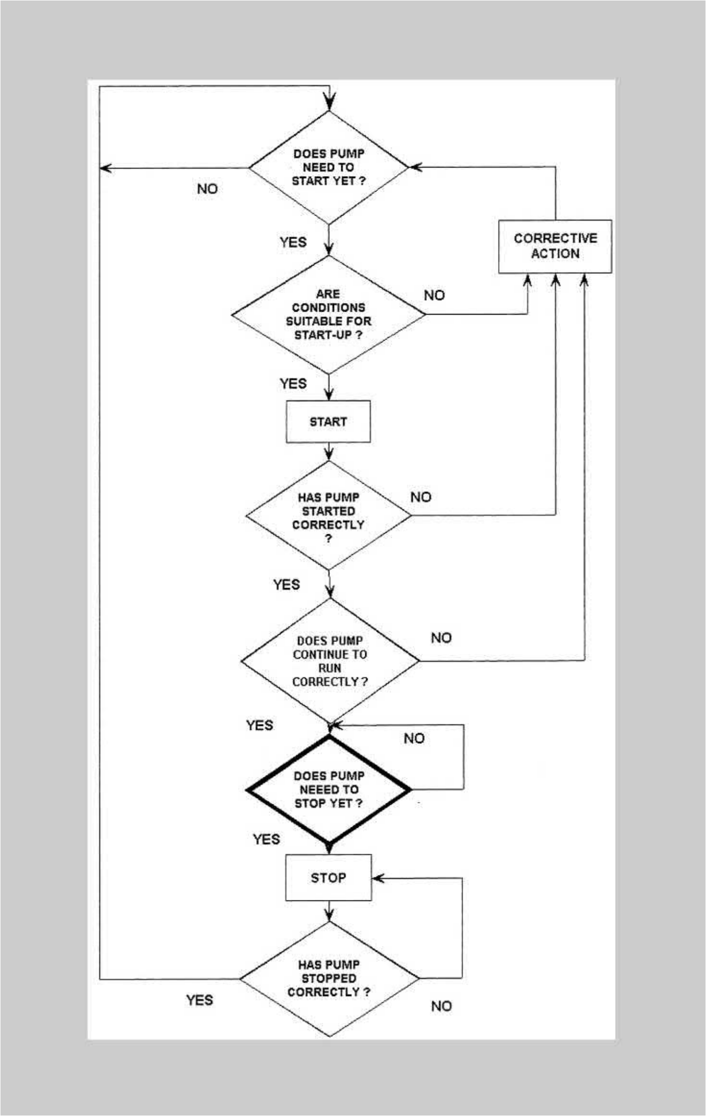
Another factor, which can influence performance, is of course the oil viscosity. Some hint of this is already given in Fig. 11.29 above. This emphasises the importance of using the correct oil viscosity grade. Oil rings can easily become dislodged during transportation unless facilities exist for limiting their movement. It is well worth checking that they are visible through the inspection holes provided, particularly if the pump is being commissioned or re-commissioned after repair. If the motor rotation is checked by 'bump starting it', then it also pays to check that the oil rings are actually working.

## Endnotes

1. Only in a minority of cases, are centrifugal pumps are bought to pressurise the liquid or process. Any pump flow at that pressure is only to replace internal flow losses.
2. If Reynolds numbers are above the laminar flow zone.
3. Unstable here means head curves that rise, then fall as flow increases.
4. Here slope refers to the curve gradient as flow is increased. With flat curves, the gradient does not increase very much as flow increases. With steep curves, the slope increases a lot as flow is increased.
5. Here the term process is being used very generally to encompass any activity where the pump flow rate is critical to that activity-boiler feed, reverse osmosis, oil and chemical processing etc.
6. Since it will also include any flow-related losses, its use is questionable.
7. This is the flow at which the REAL pump curve intersects with the REAL systems curve, and it may differ from the design flow in most cases.
8. Except there would be no NPSH [R] change.
9. Unless the pump is operating under abnormal suction pressure.
10. Not to be confused with minimum continuous safe flow.
11. Open impellers do not have wear rings. The leakage in such cases occurs between the impeller and the casing faces.
12. The so-called volumetric efficiency is the ratio of pump flow to impeller flow.
13. Although 5% is a representative value, the figures show a much larger value, just to illustrate the point.
14. Or in some high temperature cases, the suction source vessel — in order to avoid 'flashing' in the suction pipe.
15. To put this in perspective, this corresponds to a shaft diameter of about 3.5″ running at 2900 rpm. At 3600 rpm shaft would be about 3″.

# CHAPTER 12

# Does pump need to stop yet?




*Troubleshooting Centrifugal Pumps and Their Systems*
ISBN 978-0-08-102503-1
https://doi.org/10.1016/B978-0-08-102503-1.00012-8

## Introduction

Pump shutdown is a simple decision when only a single pump is involved. When however, several pumps are involved, the question is more complex.

## Category 1: single pump

If only a single pump operates, then the decision to stop is rarely tinged with complexity. There may be valid questions such as 'does it stop too frequently?' but rarely does it need to go beyond that.

Another tier of questions surfaces when more than one pump operates;

## Category 2: parallel pumping

This type of multiple pump scheme is favoured when:

- It is not possible to pump efficiently with one machine [flow too high for optimum specific speed].
- Some incremental flow steps are desired.
- The system resistance consists largely of static head.

In neither case would it be normal, or even desirable to switch all pumps in or out at exactly the same time. More often, the concept of progressively switching pumps in or out is presumed. This allows some measure of flow control albeit in discrete steps.

However, is that concept applied in reality? Often operators succumb to the temptation to keep all pumps running, but then simultaneously throttle them all back to the desired flow. This also achieves flow control, but in an inefficient manner and in a way that accelerates pump wear.

It has the advantage that extra flow capacity is almost instantly available without too much preparatory work, when compared to switching in and out. It also avoids sending the pump round the operating loop – each cycle of the loop having associated risks.

Furthermore, when only one pump operates on a parallel pump scheme with some significant friction ratio, then there is a risk that the pump will run out to end of curve and begin to cavitate.

So the pragmatic answer to the question 'does the pump need to stop?' might be ***yes***, but that does not mean the pump ***is*** stopped.

## Category 3: series pumping

This concept is most suitable for systems where;

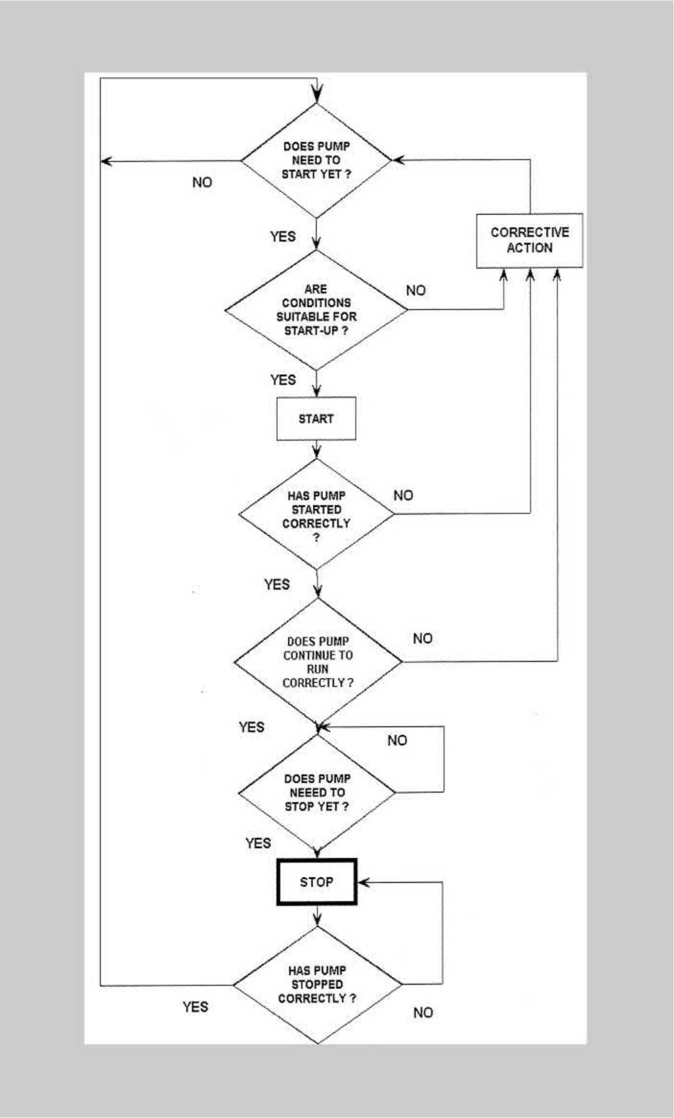
- It is not possible to pump efficiently with one machine [head too high for optimum specific speed].
- Some very limited flow steps are desired.
- The system resistance consists largely of friction head.

One of the dangers of running a solo pump on a parallel pump system is the risk it will run way out on its performance curve, cavitating or possibly overloading the driver.

With series pumping the risk is reversed. If the pumps have been sized to deliver required flow with two or more pumps, then with only one pump there is no risk of cavitation or overload. In fact, if the pump curve is flat, and the system static head high [low friction ratio] the pumps may not develop enough head at zero flow to break into the system. The pump will run 'dead-headed' and the entrained liquid will become heated, maybe to flash point. Such risks should have been foreseen and eliminated at the scheme design stage of course!

# CHAPTER 13

# Stop pump!




*Troubleshooting Centrifugal Pumps and Their Systems*
ISBN 978-0-08-102503-1
https://doi.org/10.1016/B978-0-08-102503-1.00013-X

## Introduction

Stopping a pump is not without its potential troubles either. Once again, we see that the piping system can influence the pumps behaviour.

## Pump shutdown

There are two main categories to this discussion:

- Shutdown against an open or 'free' discharge line. [i.e. little artificially imposed friction resistance]
- Shutdown against a 'closed' discharge line. [i.e. substantial artificially imposed friction resistance]

## Shutdown against an open discharge line

### Sudden shutdown

A pump discharge line is like a potential organ pipe. Just like an organ pipe, or any other wind instrument, it can acoustically resonate in response to a periodic signal located at one end. Acoustic mechanics tells us that the resonant frequency is a function of pipe length. If the line is very short, the resonant frequency is high – usually far too high to be excited by anything within the pump.

As the line gets longer, then it is possible for the pump to generate vane-passing pulsations at frequencies corresponding with the pipe acoustic resonant frequency. Some liquids, such as hot water, have so little damping that classic sound wave reflections occur along the pipe length. Other liquids may contain entrained or suspended gas that slow the wave propagation and inhibit, or at least discourage reflection.

If the pipeline is very long [more than a thousand discharge diameters in length, say] then the pump is normally incapable of continuously generating frequencies low enough to resonate.

If a pump that is steadily running is suddenly stopped, or its discharge valve very quickly closed, then a negative wave or pulse is propagated down the pipeline away from the pump. Upon reaching the extremity, the pulse changes sign to positive and then reverberate back down the line towards the pump. Whereupon it is again reflected, and so on. At each reflection, the pulse changes sign.

While all this is happening, another set of events occurs in concert. Before shut down, the head generated by the pump caused the pipe bore to swell ever so slightly. A certain amount of strain energy is stored in the pipe walls. When the pump stops, and the negative wave propagates down the line, this stored energy is of course released. Upon reflection at the downstream extremity, the situation is reversed and the positive pulse restores energy back into the pipe wall. The time for the wave or pulse to leave the pump and return may be several seconds or even moments depending upon pipe length. This rapid release from and gain to the pipe walls each time the wave reflects from the pump is usually audible as a sharp 'crack' sound. This has earned the phenomena the name of 'water hammer'.

The frequency and severity of water hammer is, in part, dictated by the pipe system, its length and wall thickness. The water hammer may take several moments to die away completely during which time the pressure history at the pump may well display some remarkable swings. Pump casings of brittle material such as cast iron might be vulnerable to fracture by these water hammers. Ancillaries such as seals and pressure gauges may also suffer.

Without going any further, it should be apparent that suddenly stopping a pump that is operating on a very long pipe system may expose it, and the pipe, to mechanical shock. This situation is best avoided. Good practice dictates that the pump is hydraulically isolated from the system before it is stopped, as described below. The most convenient method of doing this is to shut the pump discharge valve before shutting down the pump. That way no pulses propagate out into the system and water hammer is avoided. Obviously, this is much less critical if the discharge line is quite short.

If controlled valve closure cannot be affected, or guaranteed, then other approaches are needed.

The kinetic energy locked up in a pump & driver rotor is quite small and is quickly dissipated out into the system by pumping against the system resistance. In a few seconds, the pump will have stopped-initiating water hammer in the process. One method of protection is to arrange a flywheel somewhere in the pump drive shaft system (Fig. 13.1).

## Gradual shutdown

From the foregoing, it is evident that no water hammer trouble would be expected on an initially closed or open discharge line, provided that the pump was gradually shut down.


**Fig. 13.1** Mounting a flywheel in the drive-train can help extend the run-down time in the event of an unexpected power failure. This proven solutions can help to avoid water hammer problems [Weir Pumps].

This can store enough energy to prolong the slowdown and greatly reduce the severity of shut down pulses. The drive must be capable of overcoming this additional start-up inertia. In any event, pumps should be shut-down quickly and decisively. With VSD, or steam turbine drive it is unwise to prolong speed reduction below that at which the *Lomakin* effect vanishes (see Appendix F). This would increase the risk of internal seizure in the fine clearances. So turbines should be stopped by manually activating the over-speed trip.

The mechanics of good shut-down are as follows:

### *Flooded suction*

- Quickly close the discharge valve
- De-energise the driver

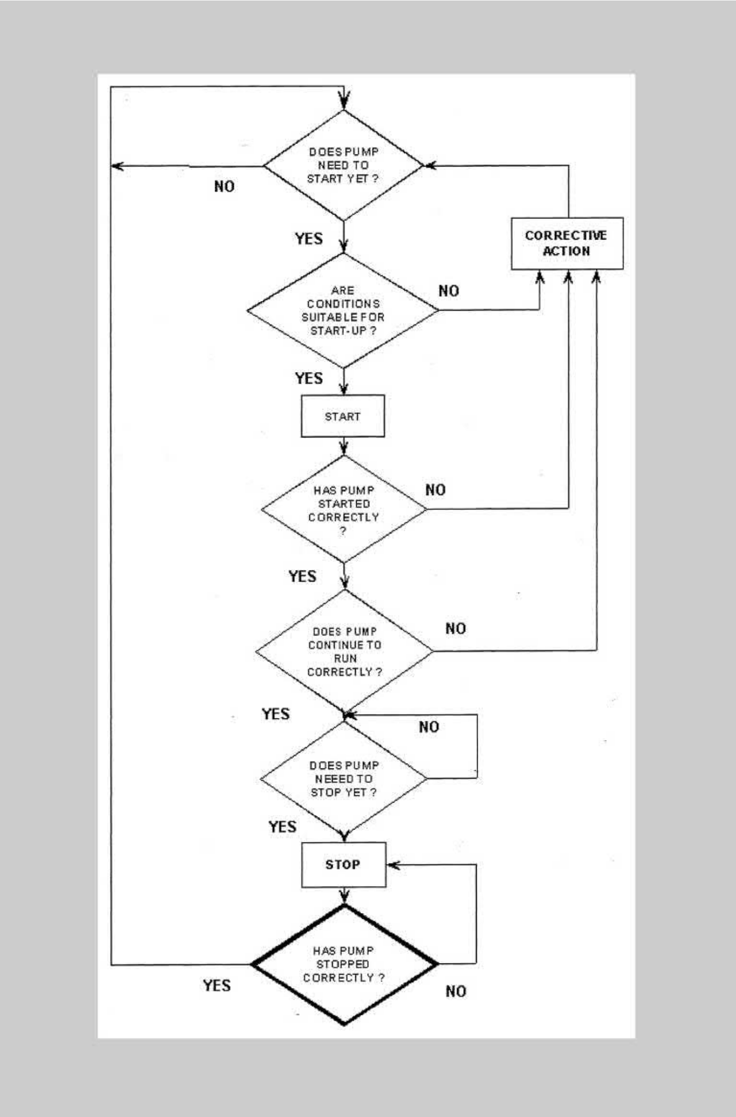
### *Suction lift*

- Quickly close the suction valve
- De-energise the driver

Both approaches ensure that the pump is primed ready for the next start-up. However it may slowly de-prime on a suction lift so a primed state should not be automatically assumed.

# CHAPTER 14

# Has pump stopped correctly?




*Troubleshooting Centrifugal Pumps and Their Systems*
ISBN 978-0-08-102503-1
https://doi.org/10.1016/B978-0-08-102503-1.00014-1

## Introduction

Logic would say that, if the pump has been de-energised, then it should slow and stop. However, this cannot be routinely assumed, unless there are some positive signals to confirm this. But why would there ever be any doubt?

## Common situations

Well there are some circumstances that will cause the pump to continue to turn; even no power is applied to it. This can happen if it becomes possible for flow to pass through the ostensibly inactive pump. Flow may be forward, in the conventional sense, or the reverse of that.

If the pump has a forced lubrication system, this may become inoperative when the main pump is de-energised. This leaves the shaft rotating without any lubricant supply so the journal bearings may eventually fail.

## Reverse speed and flow at zero torque

There are a number of situations when a pump can stop, then run in the reverse direction, mainly due to the discharge line back flushing through the pump. Fig. 14.2 gives some general indication of the reverse speeds that might be achieved. Some idea of the reverse flow achieved is shown in Fig. 14.1.

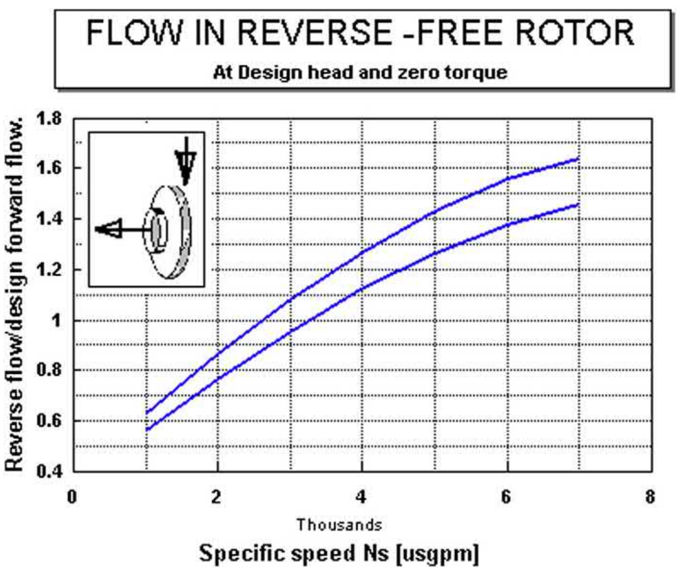



Fig. 14.1 Reverse flow under full discharge head.

In extreme cases [high specific speed designs], the turbine speed is significantly in excess of the forward speed. This has been known to damage the rotors of large squirrel cage motors, for example.

## Reverse flow from parallel operating pumps

Normal pipe design practice for pumps opening in parallel is to install NRV [Non Return Valve] on each pump discharge. This allows one pump to be temporarily stopped, if necessary. If it is necessary to shut the pump down for long periods, than it will be isolated. But in situations where the demand has only reduced for a short time, it may be prudent just to stop the pump briefly. An NRV allows this to happen.

There is a slight danger that an NRV can 'stick' open – especially if it has been running in that condition for a very long period of time. In that situation, flow from operating pump [or pumps], can be split between the system and recirculating back through the idle pump.

When this happens, the idle pump slows down, stops, and then reverses its direction of rotation. This happens because the pressurised liquid in the discharge manifold has a route back to the lower pressure source liquid, via the unprotected pump. The pump then behaves as a largely unloaded turbine. The speed, which the reverse running pump can attain, is dependent upon the specific speed of the impeller.

## Reverse flow in systems with high static head components

If a pump system characteristic contains a perceptible static head component and the suction 'lifts' from a free surface, then it is quite normal to fit a NRV to the suction leg. This NRV [frequently described as a foot valve in this case] may be combined with a coarse strainer. The role of this device is to prevent the liquid in the discharge pipe back-flushing through the pump. This is even more important if the pipe discharge at the destination is submerged in such a way as to permit continuous syphonic backflow.

The foot valve-strainer is intended to prevent the discharge line emptying back through the pump. It tends to keep the complete system full of liquid and hence ready primed. Of course, even the newest NRV will imperceptibly leak and 'pass' liquid back to source. With time, the NRV ages and is less successful in holding back the static column of liquid. At some point, the flow may be sufficient to spin the pump in reverse, though not at full speed. If however, the NRV fails to close, and backflow occurs for more than several seconds, full reverse speed may be accomplished in accordance with Fig. 14.2.

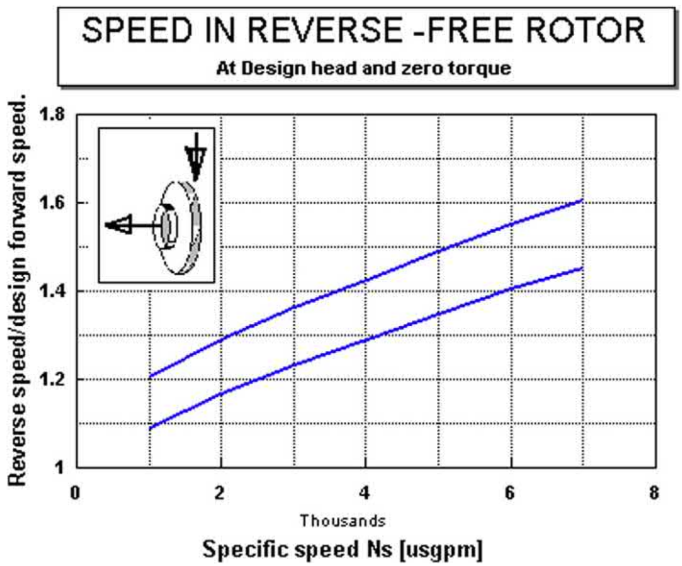



**Fig. 14.2** Reverse speed with full discharge head imposed. [Data for volute and diffuser pumps].

If the discharge is over the liquid surface, then the situation will play itself out. As the liquid contained in the discharge column flows back, the static head decreases and so the pump will slow down – ultimately stopping as the column completely empties. If the discharge is submerged, then the machine will run in reverse just as long as there is liquid in the destination vessel.

## Forward flow from booster pump, or downhill pumping

The previous examples covered cases where the flow across the machine reversed when the driver was de-energised. In the next example, the flow direction does not change.

In some cases it is necessary to furnish pumps with booster pumps upstream and in series. Typically this occurs when there is otherwise insufficient natural NPSH in the system to avoid unusual or oversize pump selections. Or in branched systems, a booster may be needed to help the main pump satisfy one or more operating system combinations. Most often the booster is started before the main pump.

Similarly, the booster is shut down after the main pump has done so. However, should the booster continue operating, for whatever reason, the main pump is then susceptible to 'wind-milling'. The pump becomes an outward flow turbine.

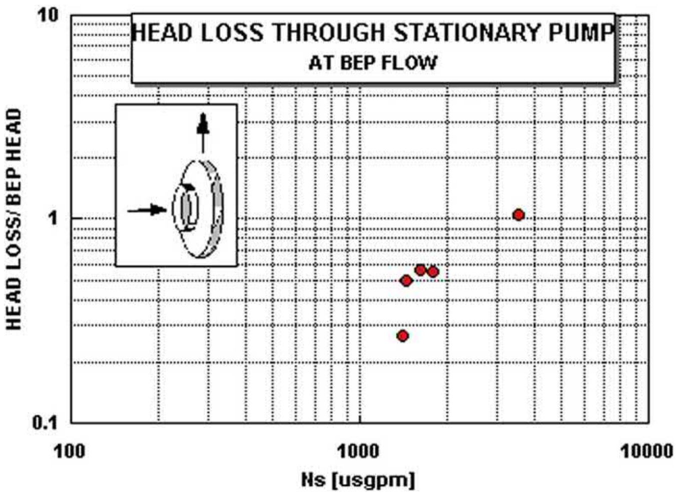



**Fig. 14.3** B.E.P Head loss across a stationary pump with locked rotor and flow imposed upon it in the normal pumping direction. [Single stage volute pumps].

Downhill pumping schemes are also susceptible in this respect, unless positive steps are taken to prevent it. Before 'wind milling' sets in, there will be some appreciable loss across the stationary rotor. An indication of this loss is shown in Fig. 14.3.

## Locked rotor loss

Sometimes it is of interest to know how much head is lost across a stationary pump with a locked rotor that has flow externally imposed upon it. Fig. 14.3, summarises some data taken from a number of single stage machines with flow imposed in the normal pumping direction. The data give some broad idea as to the losses involved, but no more. The data represents loss at the pumps original design flow. At other flows, the loss seems proportional to the square of flow i.e. at half the flow[1], the loss is one-quarter that at full locked rotor flow. Impeller diameter reductions of up to 90% of full diameter do not seem to show any significant change in loss.

Appendix G gives more information on pumps in reverse.

## Endnote

1. Assuming turbulent flow in accordance with Eq. (5.1).

# APPENDIX A

# Pump selection guidelines

In carrying out a pump audit, it is often useful to know if the layout will most probably be satisfied with a commercial pump design, or whether some degree of custom engineering will be required.

Assume that only required head and flow is known. Selection is then often an iterative process in which the suction constraints are balanced with the discharge constraints until a balance is achieved.

The general aim, for clear liquid pumps, is to select a pump speed that gives a Suction Specific Speed of less than 7000 and 11,000 (Eq. 3.2) while yielding a stage discharge specific speed of between 600 and 3500 (Eq. 3.1). In this process one starts with the assumption of a single stage single entry pump. If one exists, this represents the lowest cost solution.

However, it may then prove necessary to consider a double entry impeller design, and/or multistage layout[1] in order to reach compatibility with the suction and discharge specific speed criteria.

## STEP 1.0: ESTIMATE THE MAXIMUM PUMP SPEED

In most cases it is preferable to run the pump as fast as reasonably possible, in order to reduce its size and cost, as well as maximise its efficiency. *[This would not be the case when there are significant suspended solids in the liquid.]*

This step of the process is mainly concerned with the inlet pressure conditions and the pump inlet design geometry. Generally a solution can be found based upon a single entry impeller design. In more difficult applications, it will be necessary to invoke the double entry concept of two impellers operating in parallel (Fig. 4.21)

**1.1** Estimate the suction pressure available to the pump

**1.2** From this calculate the NPSHA. This is the total head at the pump inlet above the liquid vapour pressure.[2]

**1.3** Use Fig. E.21 to estimate a margin − this then helps establish the maximum pump NPSHR under the given site NPSHA.

[1] Double suction impellers are rarely available, nor worthwhile on multistage pumps with suction branches less than 4 inch.

[2] Or more exactly, above the impeller centre-line because in some cases the impeller may be located away from the suction nozzle. The "NPSH[A]" must acknowledge this.

**Table A.1** Approximate motor speed for initial selection.

| Motor poles | 50 hz | 60 hz |
|---|---|---|
| 2 | 2950 rpm | 3560 rpm |
| 4 | 1470 rpm | 1780 rpm |
| 6 | 980 rpm | 1190 rpm |
| 8 | 730 rpm | 890 rpm |

**1.4** Substitute this NPSHR value into the Suction Specific Speed calculation; Eq. (3.2); to get an estimate of the maximum allowable running speed. Unless the calculated speed is within 15% of a normal electric motor speed round down the estimate to the next lower speed.

Note that different suction specific speed criteria should be invoked, depending upon the type of application (see Table A.1 and Table A.2).

**1.5** With this first estimate of speed, a realistic impeller inlet configuration has emerged. It is now necessary to confirm that at this speed, realistic impeller outlet geometry is also possible.

## STEP 2.0: CALCULATE THE IMPELLER DISCHARGE SPECIFIC SPEED FROM EQ. (3.1)

In this second step the remaining aspects of the pump hydraulic design are considered. In most cases a solution can be found that is based upon a single stage pump [one impeller]. In more difficult applications, it becomes necessary to invoke the multistage concept, where impellers are operated in series. Obviously this simple selection description cannot cover all the nuances involved, but some sense of the decisions and logic will be gained.

**2.1** If Specific Speed turns out to be less than 600, consider increasing the speed, i.e. from 4 poles to 2 poles or from 6 poles to 4 poles. To do this it will probably be necessary to consider a double suction impeller. Revisit step 1.4, but this time uses half of the pump flow in the calculation. With this new speed [if any] go to step 2.2.

**Table A.2** Preliminary values of suction specific speed for use in initial speed estimate.

| Application | Initial Nss value |
|---|---|
| Cold water | 7000 |
| Hot water | 8000 |
| Hydrocarbon | 11,000 |
| Sewage | 5000 |
| Abrasive Solids handling | 5000 |

**2.2** If Specific Speed turns out to be less than 600, and speed cannot be increased any further, a multistage pump will be required. Go to step 2.3.

**2.3** Divide the head by 2, and recalculate Ns. If Ns exceeds 600, a commercial solution might exist – otherwise increase the stage number again, until agreement is reached. If a stage number of 12 still fails to produce acceptable Ns, then it may be that a commercial solution does not exist. In which case, consider a series-pumping scheme [i.e. sharing the head between two or more pumps]. If this still fails to produce acceptable Ns, then a reciprocating pump is probably indicated [Note this step has been considerably simplified at this level. For example small multistage pumps of say 2″ discharge nozzle size might be available with stages of 14 or 15. On the other hand, pumps of 10″ discharge might be limited to only 6 stages.

**2.4** If Ns exceed 3500, then a specialist machine is most probably required, and manufacturers should be consulted.

Fig. A.1 gives some idea of the domain of feasible machines, but this is only a guide. The chart is drawn for 50 Hz operation.

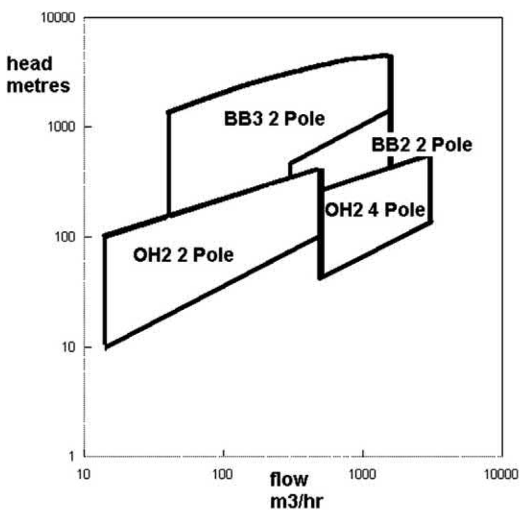



**Fig. A.1** Approximate domains of feasible commercial machines [50hz]: **OH2** – **O**ver**H**ung impeller, end suction, single stage pumps; **BB2** – **B**etween **B**earings, double suction, single stage pumps; **BB3** – **B**etween **B**earings, single or double section, multistage pumps.

# APPENDIX B

# Guides to estimating a pumps performance, if its hydraulic sizes are given

Pump hydraulic design is a combination of art and science. Quick and simple rules that claim to accurately establish pump performances should be viewed with great suspicion. However, very rough rules giving broad estimates are sometime useful. Such a rough guide is detailed below, and is based on easily obtained hydraulic measurements.

The following data is needed see Fig. 3.21

- Impeller outer diameter
- Impeller outlet passage width
- Shaft rpm

## STEP 1

From the impeller diameter, use Fig. B.1 to estimate the likely head b.e.p if the pump was to run at 2950 rpm.

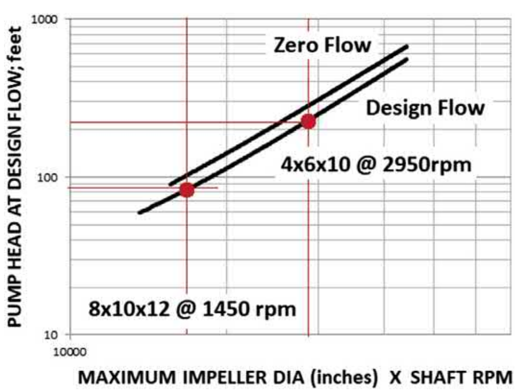



**Fig. B.1** Head generated per stage as a function of impeller diameter, running at 1450 rpm. Chart applies to machines up to **Ns 2000**, which covers most commonly found pumps. Above this value the chart will provide an underestimate.

## STEP 2

Correct the estimated head to the actual running speed.

H = Head from Fig. B.1

'h = Head at actual running speed

N = 2950 RPM

'n = actual rpm

'h = H × $[n/2950]^2$

## STEP 3

From the outlet width ratio chart, Fig. B.2 estimate the pump specific speed

## STEP 4

By re-arranging the specific speed Eq. (3.1)

$$\text{Flow @bep} = \left[\left[\text{Ns} \times \left[h^{0.75}\right]\right]/\text{rpm}\right]^{0.5}$$

## STEP 5

Use Fig. 3.36 for a rough estimate of efficiency.

For a more approximate approach, consider the process outlined in Chapter 3; [performance estimate, unknown pump]

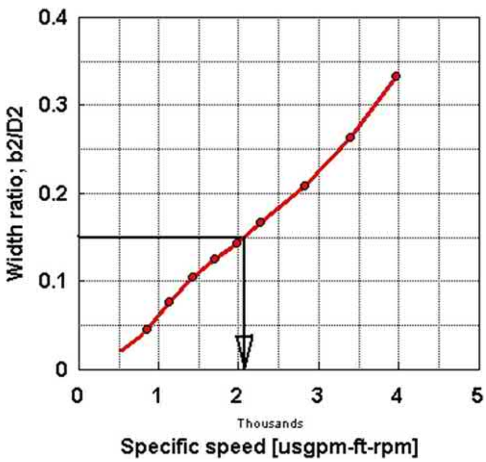



**Fig. B.2** Specific speed related approximately to impeller width ratio. As outlined in Chapter 3, pumps having the same design point performance, and hence same specific speed will exhibit a range of width ratios. This reflects the curve shape – in particular the ratio of head at design flow to that at zero flow – the so-called Head Rise To Shut Off.

## APPENDIX C

# The affinity laws, and breaking them

It is not currently economic to engineer specific pumps for each and every application. Instead, most manufacturers produce a series or range of pumps in different sizes. Each pump is bigger than its predecessor and covers a unique output 'patch'. By adjustment to the respective impeller diameters the performance within each patch can be infinitely varied. In this way, standardised pumps can be adequately tailored to each application. The same technique can be used in the field to make adjustments to a pump performance. This can be particularly attractive where pumps have been oversized and are consuming more power than expected (Fig. C.1).

At various point in the main text I refer to the so called Affinity Laws. This is a set of equations that can help predict the change in behaviour if either the shaft speed or impeller diameter is changed. They are derived

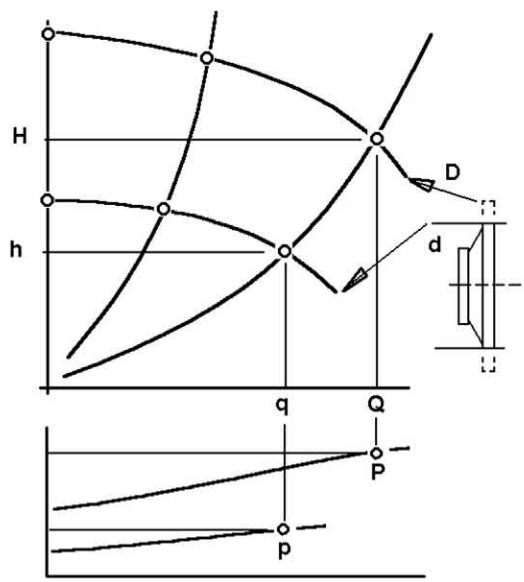



**Fig. C.1** This shows the general effect of changing impeller diameter. Superficially, any points on one curve closely follow a square law as diameter is changed. Small departures exist due chiefly to the inevitable failure to maintain exact geometric similarity with each change.

from the application of Fluid Mechanics to the liquid conditions at the impeller rim. In effect they relate to the tip speed of the impeller — which is a function of impeller diameter and/or shaft speed. They are normally written as;

New flow = old flow × TSR

New head = old head × $[TSR]^2$

New abs power = old power × $[TSR]^3$

where 'TSR' is the Tip Speed Ratio;

TSR = New speed/Old speed, or

TSR = New impeller diameter/Old diameter.

If both are changed simultaneously, then

TSR = [New Speed/Old speed] × [New diameter/Old diameter].

The laws presuppose that internal/hydraulic efficiency does not change — which as we shall see, is never completely correct. However this assumption only makes a few percent difference to the calculations. For troubleshooting purposes it can be factored in with sufficient accuracy (Fig. C.2).

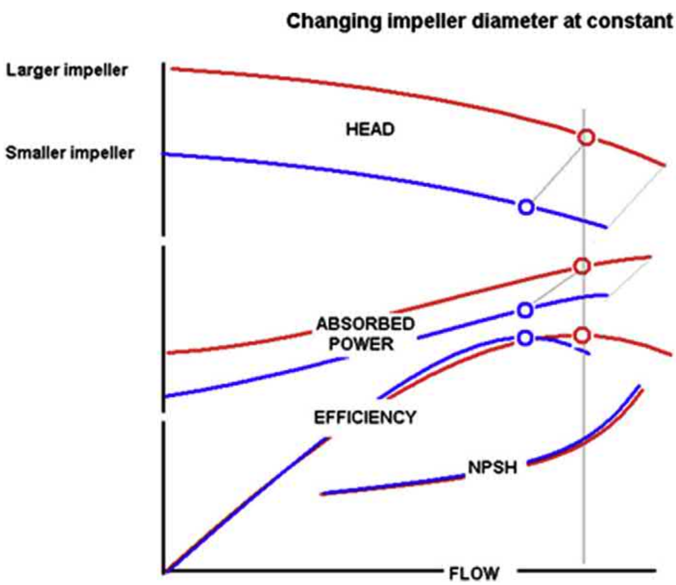



**Fig. C.2** Changing impeller diameter, also reduces impeller tip speed. However, dynamic similarity is not maintained due to, for example — the changing gap between impeller and collector. So additional losses appear. The input power does not change, so the nett effect is an efficiency reduction over and above any Reynolds Number effect. NPSH[R] is chiefly related to the inlet tip speed. Since speed has not changed, neither does the NPSHR curve.

Any point on the full diameter performance curve follows a square law as the impeller diameter or speed is reduced. Coincidentally, if the pump is operating in a system where the resistance consist entirely of frictional losses *[circulators for example]* the affinity laws also describe how the pump output will appear to change. This is because in such systems, the resistance losses also follow a square law. *[On the other hand, in cases where there is also a significant static component to the total head, the affinity laws will not adequately describe how the pump and system interact.]*

This next Figure summarises the nett effect of speed change (Fig. C.3).

The effects on **NPSH** need to be considered separately. Simplistically, this is **only** affected by speed change, not by changes to impeller diameter. But in reality, there is an additional slight speed effect over and above that forecast by the Affinity laws. Furthermore there is actually an additional effect independently related to the impeller diameter change. This is not directly predicted by the Affinity laws.

The laws assume complete and total dynamic similitude is maintained when the impeller diameter is changed. This is clearly impossible, *[internal hydraulic efficiency changes slightly for example]* and that is why the laws do not hold fully. I will discuss these and other minor Affinity Law deficiencies later.

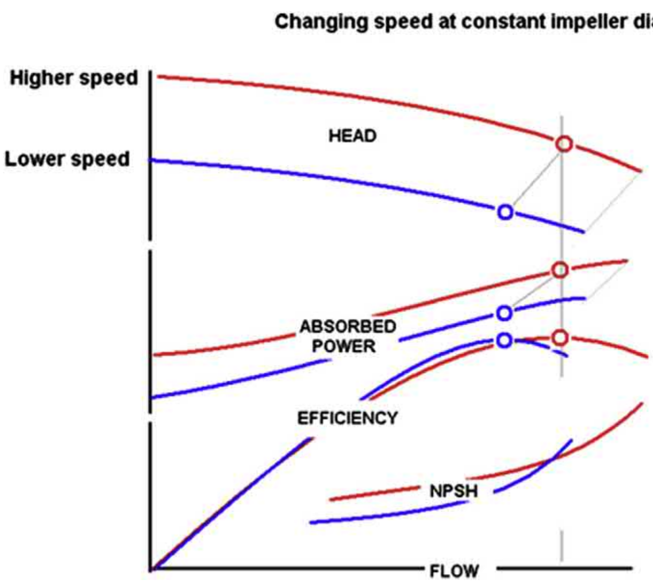



**Fig. C.3** Only changing speed, retains much of the dynamic similarity of the internal flow patterns. Hence significant change in shaft speed will produce significant changes in HEAD, POWER and NPSH[R]. A small change in maximum efficiency might be experienced due to Reynolds Number effects on the internal hydraulic losses. [Reynolds Number is velocity related].

If for example, the affinity laws suggest that a centrifugal pump impeller diameter should be **reduced** by 10%, then the pump performance would actually be reduced too much. [*Due to additional losses resulting from failure to maintain full geometric similarity.]* In practice a diameter reduction of only about 8% would be implemented. *[The extra 2% performance reduction comes from increased internal losses that are induced.]*

Similarly, if the affinity laws suggest that a centrifugal pump impeller diameter should be **increased** by 10%, then the pump performance would actually be increased too much. [*Due now to the reducing/absence of losses resulting from failure to maintain full geometric similarity.]* In practice a diameter increase of about 8% would be more prudent. This type of discrepancy increases with impeller specific speed.

As just described, this form of the laws, though not exact, is sufficient for most general troubleshooting purposes. But if great precision is required, due to [say] external/critical constraints, then the OEM should be consulted.

Where the Affinity Laws would be used in Troubleshooting? The main application is when the performance of an existing pump needs to be reduced. The result is a more or less parallel downward shift of the performance curve. If it is required to, say significantly 'pivot' the curve about the zero flow head, so as to increase or decreases the general curve slope, then additional hydraulic modifications need to be considered. These are discussed in Appendix K (Fig. C.4).

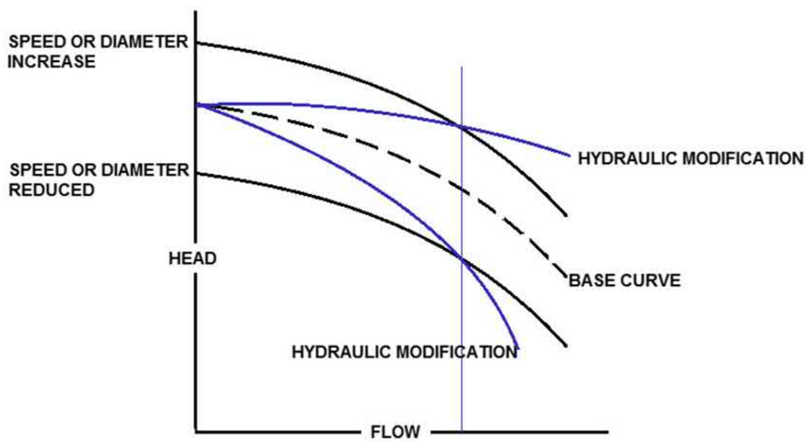



**Fig. C.4** Speed or diameter changes do not significantly change the pump curve, at least inside its "Comfort Zone". If concurrent curve shape changes are required, then additional hydraulic modifications will be called for.

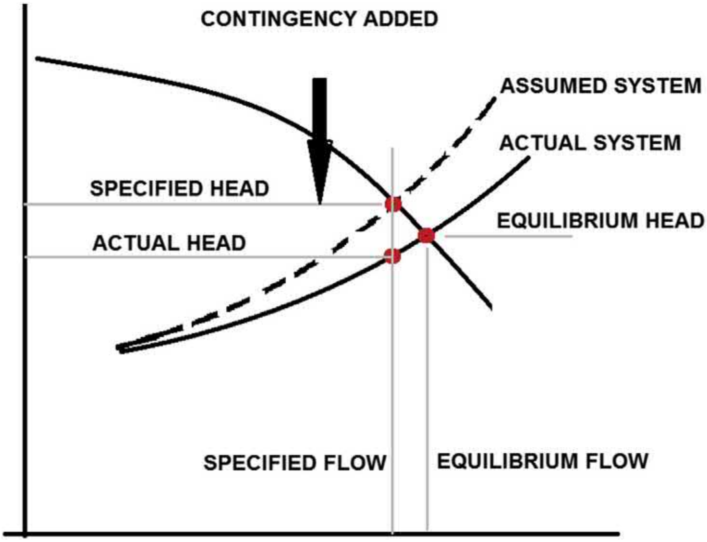



**Fig. C.5** Almost all pumps are initially ordered with a contingency allowance on the system head. This is quite understandable but such over-specification may result in over pumping when the pump is installed. It can lead to reduced margins on the driver as well as on NPSH.

A simple worked example might help illustrate the use of the laws in practice. Pumps are most often procured against duty requirements that are typically higher than actually required. On site, this can result in increased flow, coupled to higher than expected power consumption – even overload (Fig. C.5).

Excess flow can often be tolerated, but not the associated excess power consumption. [Power will also increase if the liquid density changes – for example pumping seawater instead of potable water]. So we will examine the case where we need to reduce the output a little.

If we reduce the impeller diameter, then the pump performance will also reduce but will require less power to do so! How do we estimate the new diameter required?

To conduct this process rigorously, requires some knowledge of the pumping system resistance curve. *[This is discussed in earlier Chapters.]* In many cases this will be known with some certainty. For example it will often be obvious from the installation that friction losses will dominate in the resistance curve makeup. Transferring liquid over a relatively long distance will yield such a 'Steep' curve. Elevating or 'Lifting' over large level

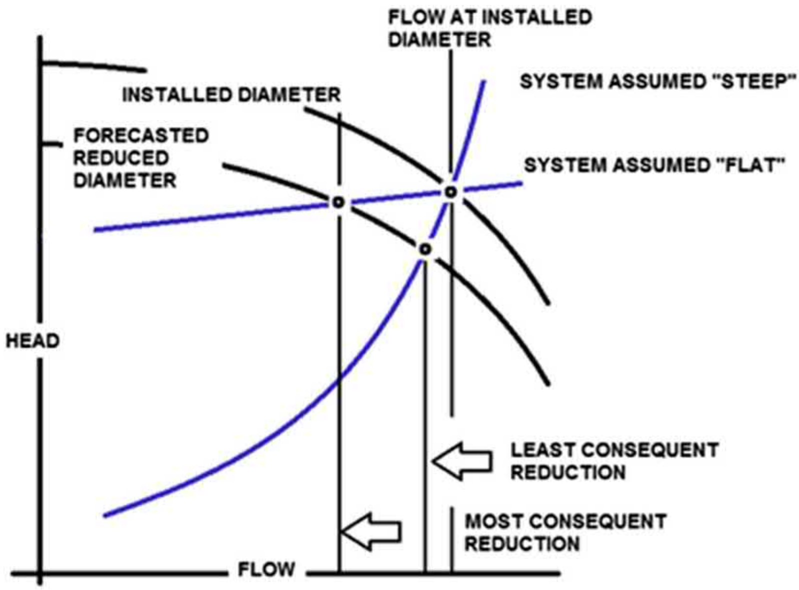



**Fig. C.6** If it is desired to reduce flow, but the system curve make-up is unknown, then it is fail safe to assume that it consists entirely of static lift head. If it turns out to be steeper, then an additional trim might prove necessary.

changes will often point to a system curve in which static lift predominates. This will yield a 'Flat' system curve.

Where little or no knowledge of the system appears to exist, one can still take advantage of the fact that almost all centrifugal pumps are purchased in order to stimulate flow in a system. As a conservative approach and in the absence of knowledge to the contrary, one must assume that the system curve is completely flat (Fig. C.6).

If the system is truly flat [*consisting almost entirely of static lift* Category 2 Section 5], then the following process should not result in this lower flow threshold being broken. If, on the other hand, the system is truly steep [*consisting almost entirely of frictional resistance Category 1 section 5*] then the expected flow reduction will be approached, but not exceeded. A further diameter/speed reduction may be needed. A flat system curve assumption is therefore safest bet.

The motivation for performance reduction will come from either over pumping or driver overload. For clarity, I will break the analysis into two distinct steps.

Let's consider the case in which the machine is known to 'over-pump' to an unacceptable level. I will make the tacit assumption that the driver becomes overloaded as well. . We will estimate how much the impeller diameter needs to be reduced in order to reduce flow to an allowable level. Then we will estimate how much the absorbed power has been consequently reduced. If the power has not been reduced enough, then we will estimate how much further the allowable flow has to be reduced in order to meet the power consumption target. When power absorption is the only issue, then the first step can be omitted.

As a reminder – these calculations are based on the worst case assumption that is that the system resistance curve is flat. Significant departures from this assumption will result in lower than expected reductions.

1 The first step is to project a line back from the original duty point, marked onto the original performance curve. This line should cut a vertical line that indicates the New Maximum Allowable Flow (Fig. C.7).

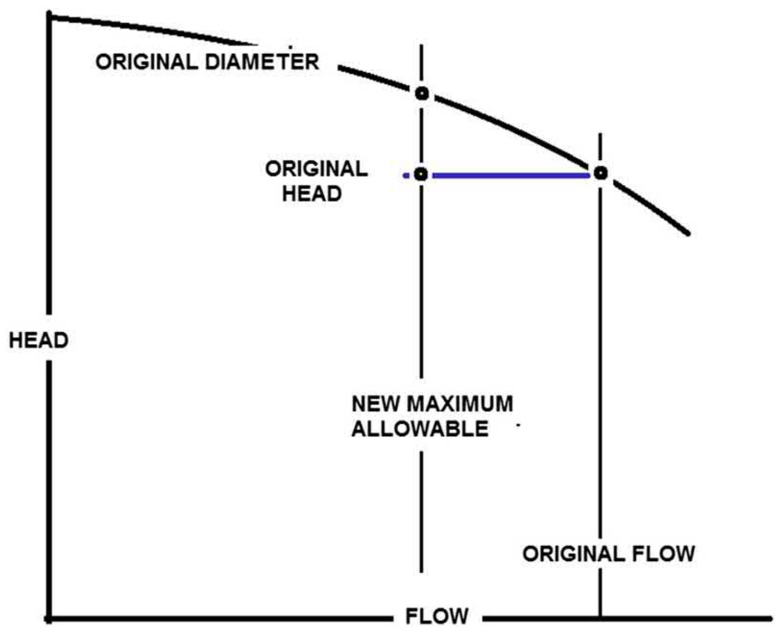



**Fig. C.7** Following on from Fig. C.2, the first step is to project back from the original flow to the new flow.

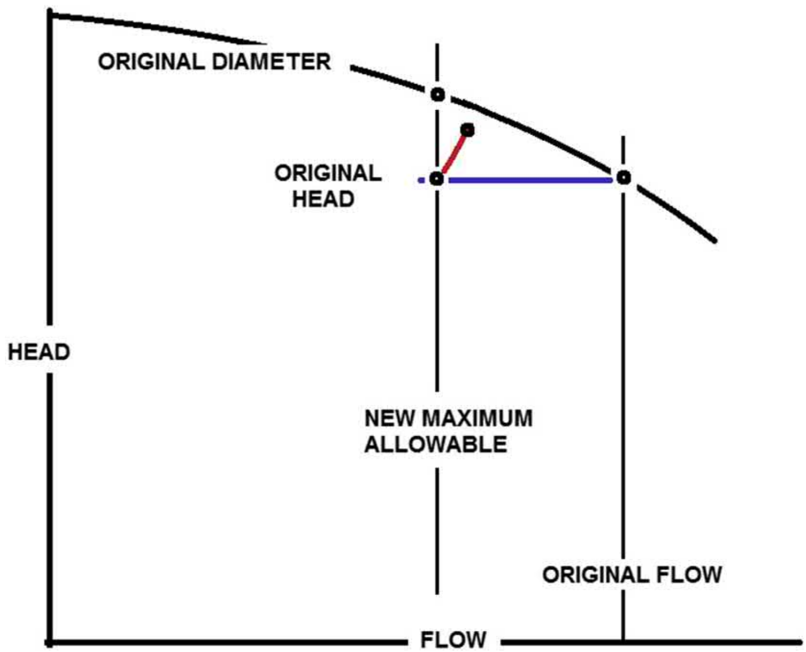



**Fig. C.8** Using the Affinity Laws are next used to construct a parabola up from the point derived in Fig. C.7.

2 Now choose an arbitrary Tip Speed Ratio [**TSR**] – *[say 0.95 if examining a performance INCREASE or 1.05 in the case of performance DECREASE]* (Fig. C.8).
Then, taking the values at the intersection, plot a new point where;

X axis = New Max Allowable Duty Flow × TSR1

Y axis = New Max Allowable Duty head × $[\mathrm{TSR1}]^2$

3 Repeat these steps with different increasing values of TSR1 until the derived curve crosses the original performance curve (Fig. C.9).

Next, we read off the head [H2] at the new intersection point (Fig. C.10).

The TSR for the trimmed impeller will be given precisely by

$\mathrm{TSR2} = [\mathrm{H1}/\mathrm{H2}]^{0.5}$

The required new impeller diameter is given by;

New dia. = Original dia. × TSR2

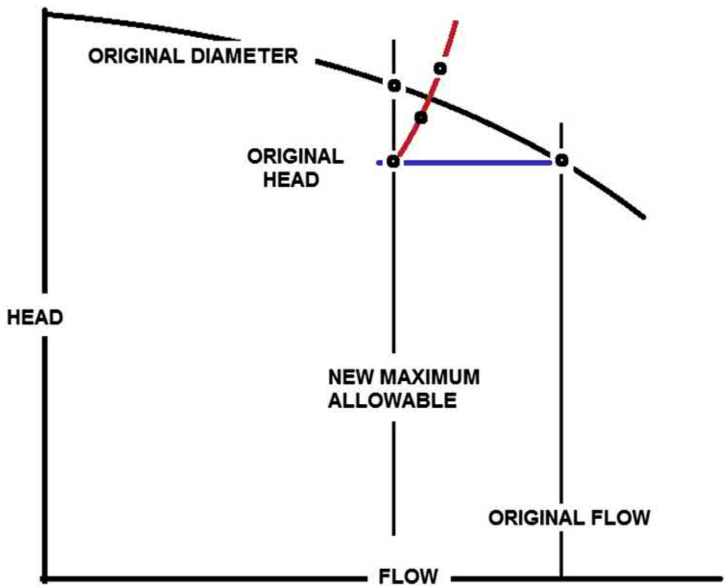



**Fig. C.9** Using a range of TPR values this parabola is extended above the original curve.

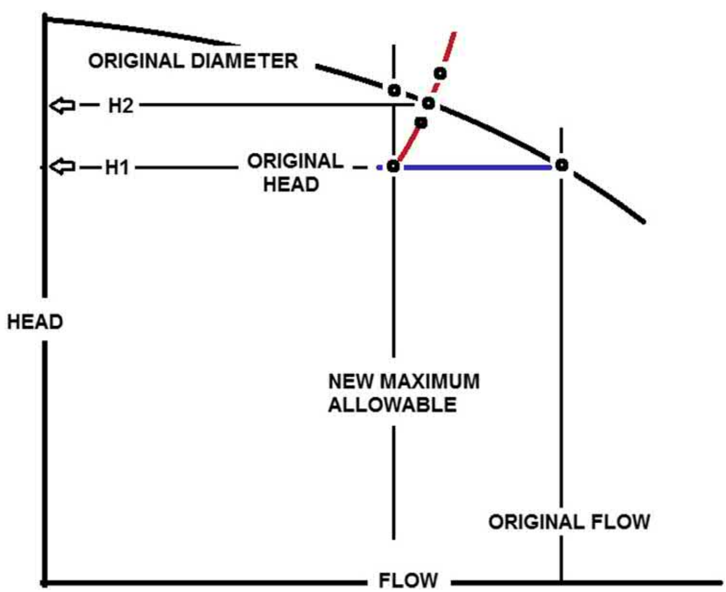



**Fig. C.10** This parabola intersects the original curve at a certain Head value. Using the equation in the text, a value for TPR can be derived. This needs slight correction, for the reasons outlined in the text.

**4** The next step is to reduce this new diameter trim forecast by about 2% for every 10% predicted. This correction helps offset the increase in hydraulic losses resulting from an impeller diameter trim. . So if the Affinity Laws forecast a 10% reduction in impeller diameter, only about 8% will actually be required.

**5** The new absorbed power is similarly calculated, in accordance with the Laws.

This worked example covers a change in impeller diameter, but could just as well apply to a change in shaft speed. Since they relate to impeller tip velocity, the laws apply equally to a change in speed – i.e. a 10% speed change has the same predicted effect as a forecasted 10% diameter change. The only difference now is that the hydraulic geometry has not changed, So unlike in impeller diameter change, a 10% speed change **does not** normally require any further corrections for diameter-dependent losses – *[unless the Reynolds Number changes very significantly – that is the speed changes much more than 50%]* (Fig. C.11).

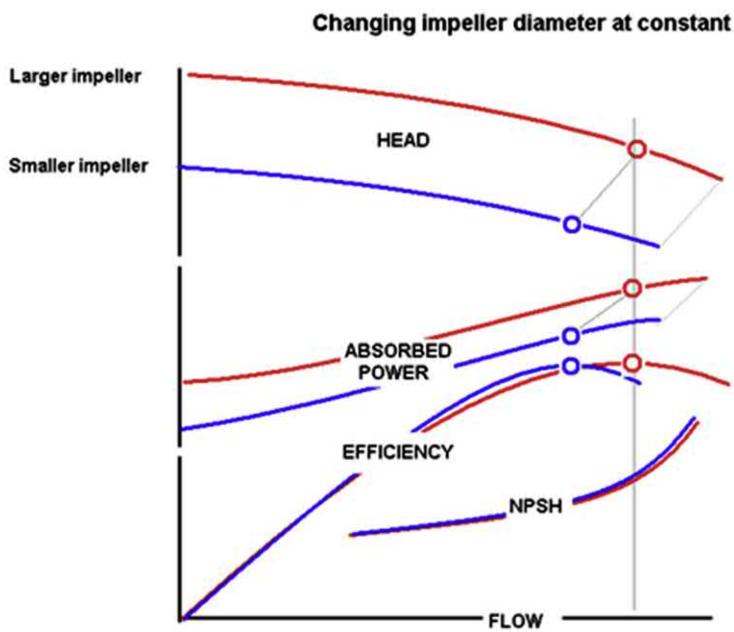



**Fig. C.11** Fig. C.2 explains that when performance is reduced by impeller trimming, but at a constant speed, the NPSH[R] curve appears to changes very little. That is because the inlet tip speed has not changed. Any apparent NPSH[R] change is partly a result of the impeller vane length becomes shorter and partly the way the curve is derived – see text.

But there is a big difference when it comes to forecasting the effect on NPSHR curves. Cavitation performance is chiefly related to the geometry of the impeller inlet annulus. There is only a very weak connection with the impeller vane and the outlet geometry.

So the Laws would predict that changing performance by diameter reduction would have little if any effect on the NPSHR curves. But because hydraulic similarity is not maintained [*vane length is reduced for example]* we will see this is not completely true. Reducing the impeller diameter appears to actually slightly increase the NPSHR *[At least so far as curves determined by the 3% head decay criterion].*

On the other hand, the Laws predict that changing performance by changing speed results in the NPSHR curve being displaced in the same manner as the pump output head curve. Reducing speed reduces NPSHR (Fig. C.12).

But again, similarity is not actually maintained − for example, the residence time of the growing then collapsing cavity bubbles changes as speed reduces. The nett effect is that the NPSHR curve does not appear to

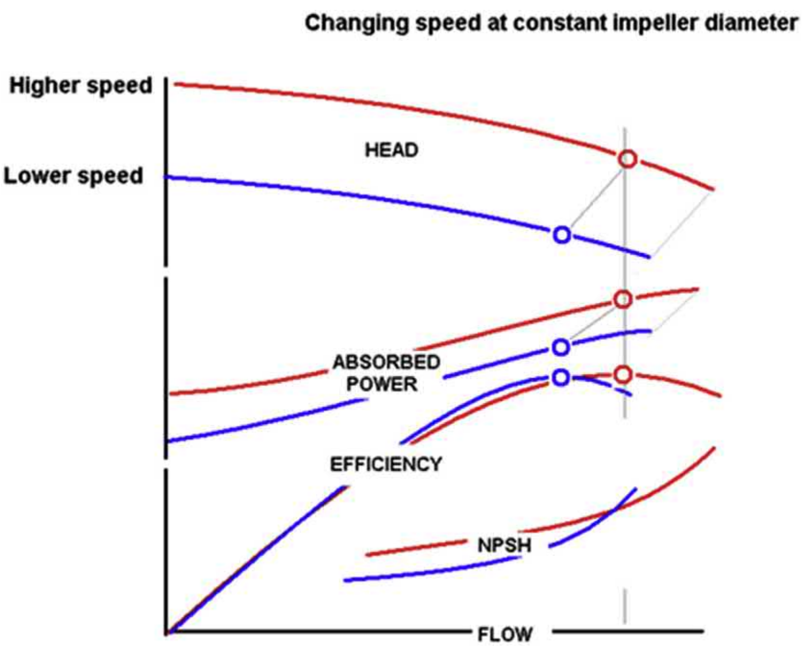



**Fig. C.12** Reducing speed consequently reduces inlet tip speed and hence NPSH[R].

reduce so much as predicted by the Laws. With speed increase, the NPSHR does not increase as much as the Laws predict.

So, if the NPSHA/NPSHR margin is small, the OEM should be involved.

The Affinity Laws predict that the absorbed pump power will change in accordance with the cube of **TSR**. In practice, the pump efficiency deteriorates slightly as speed is reduced and *vice versa*. So the absorbed prediction will be optimistic and vice versa. But the deterioration generally becomes more significant when impeller diameter is reduced. For several reasons, the internal losses increase because hydraulic/geometric *similarity [upon which the Affinity Laws are based]* cannot be maintained. This makes the absorbed power prediction even more optimistic. If the impeller diameter is being changed *[reduced]* only in order to reduce absorbed maximum power, then the OEM should probably be involved in checking the calculation.

## OTHER FACTORS

I have tried to explain why manufacturers rarely if ever use Affinity Laws without some minor correction. I noted that dynamic similarity cannot be maintained as an impeller has its diameter reduced. The most obvious departure is that the gap between impeller and collector changes and with it, the magnitude of associated hydraulic losses. But there are other reasons, which I glossed over at that point For example;

- In real impellers, the passage width at outlet *[b2]* increases as diameter is reduced.
- If the impeller has been designed to operate under very low suction pressure conditions, then it will have an unusually large 'eye'. This will imply a certain area distribution between impeller inlet and outlet. If an impeller of otherwise identical performance is designed instead for maximum efficiency, it will have a smaller inlet 'eye'. The area distribution will be correspondingly different and as the impeller diameter is reduced, the area distribution of each impeller will change at different rates.
- For similar reasons, the impeller vane angle may not be constant as diameter is reduced [though the designer will try to minimise significant departure]. Impellers operating under low suction conditions will tend to have lower vane inlet angle though may have identical outlet angle. Hence the rate of angle growth from inlet to outlet will differ.

- Another factor is the ratio of the inlet 'eye' diameter to the impeller outlet diameter. To an extent, this ratio defines the blade length that is available to do work on the liquid. In those pumps where the ratio nears 1.0, then the blade length will be short and consequently more heavily loaded. Impeller diameter reduction has more influence on such short heavily loaded vanes.
- A related design parameter is the ratio of vane length to the peripheral spacing. This is termed the vane 'solidity' High solidities tend to give better fluid guidance, though perhaps at the expense of efficiency. The special extreme where solidity = 1.0 is mentioned later.
- Impellers with high radius ratios [Inlet eye diameter approaches outlet diameter] also tend to low solidities. Impeller diameter reductions affect fluid guidance more on such impellers.

Factoring all these together gives us some insight as to affinity law departures. But this should be kept in perspective; they are of greater significance on pumps of several megawatts power than those of a few kilowatts. This might all sound as if the affinity laws are of limited use. In fact they are extremely useful, once their limitations are clearly understood.

Other significant aspects include;

The closed valve head does not always change in accordance with the affinity laws. As the gap between impeller and collector increase, secondary recirculating flows may develop. Hence the head at and close to zero flow may reduce more than the affinity laws would predict. Impellers that already have a flat or marginally stable curve can then become unstable at low flows-simply due to these secondary flows. This effect can be sidestepped by machining the impeller rim at a conic angle, rather than cylindrical − see Appendix K (Fig. C.13).

Up to specific speeds of 2000 [units of Usgpm and rpm] the pump NPSHR does not appear to change appreciably. Beyond this level, there does appear to be some effect. In part this relates to the change in blade length previously mentioned. But another cause relates to the way that NPSHR is currently measured (Fig. C.14).

At Specific speed levels less than 2000, the shape and proportions of the Head v NPSHA curve does not change much as the impeller diameter is reduced. For the purpose of this illustration, assume it does not change at all. Then in absolute terms, 3% decay in the low head of a trimmed impeller intersects that curve at a higher value of NPSH than 3% of the head of a maximum diameter impeller. So the NPSH appears to increase as impeller diameter is reduced.

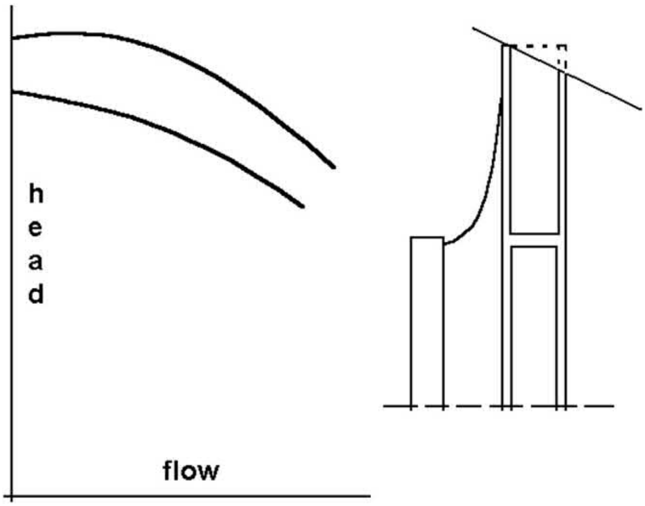



**Fig. C.13** Trimming an impeller at a conic angle can help to make an unstable pump curve more acceptable. This is discussed in Appendix K.

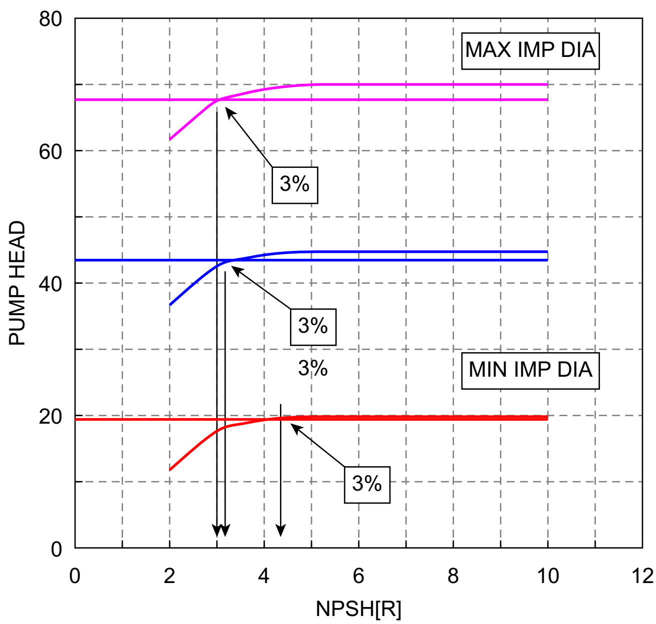



**Fig. C.14** The International definition of cavitation is at a 3% head drop. Often, the decay curve shape hardly changes with impeller reduction. So the 3% intersection with the head at Maximum diameter occurs at a lower NPSH than for Minimum diameter.

Incidentally, as noted earlier, NPSH[R] does not exactly follow the affinity laws when speed is changed.

- When a pump is slowed down, the NPSHR appears to increase slightly relative to the affinity law predictions.
- Similarly, when a pump speed is increased, NPSHR appears to decrease relative to the affinity law prediction.

Both these effects relate to the change in pressure gradient along the flowpath. Furthermore, the bubble residence time changes with speed though the bubble growth rates are more or less constant.

## MINIMUM IMPELLER DIAMETER

The concept of Minimum Impeller diameter is worth discussing. By how much may an impeller diameter be reduced? Manufacturers show minimum impeller diameter on their curve *but only as a guide*. The main intent is to show at what point a smaller more efficient pump should be considered. But that does not mean there is a hard limit to impeller reduction. All that happens is that its effectiveness as a pumping machine decays more and more. But nothing sudden or serious occurs once that threshold has passed. It is only when the vane solidity approaches 1.0 or less in radial flow machines that the decay becomes severe. At a solidity of 1.0 in radial flow machines, when looking radially inward on the impeller rim, one can see through the blades to the shat. This illustrates how far one could go.

In fact some pumps are obliged to have such low solidities, by design. Sewerage and other solids handling pumps adopt this inefficient, but effective strategy on order to enhance their choke resistant properties.

## NOTE

In Appendix K, methods of modifying the performance of a pump re described. Some of these can also be implemented along with a change in impeller diameter.

APPENDIX D

# Inlet backflow

NOTE; Chapter 1 explains that the concept of a centrifugal pump has been in existence for a very long time. In that period, different manufacturers have made many different interpretations of the concept. Consequently it is unrealistic to suppose that the outcome of any of the general advice given in this volume could be guaranteed in all circumstances. This advice is given in good faith, but is only intended to illustrate the likely scale of any outcome.

## WHAT IS BACKFLOW?

Almost every centrifugal pump impeller will possess an inlet backflow regime and consequently can be vulnerable to its damaging effects. The onset of inlet backflow brings with it the potential for several unwelcome matters. In many ways, backflow acts as a silent killer. It is only because we cannot readily see inside a working pump that this phenomenon is not more widely recognised. So it might be useful to begin this Appendix with an illustration of inlet backflow in action.

The next few pictures attempt to convey what happens in the backflow regime. A commercial end suction pump was equipped with a short section of clear suction pipe. On the walls of this pipe were glued several tufts of wool. The wool was free to move with the liquid flow and would disclose the flow angle at any instant. In order to improve the clarity and contrast, a little air was introduced to the pipe far upstream. This gives the opaque appearance to the liquid in the pipe. In each case, two pictures are shown. The first shows the pump, pipe, and a conceptual graphic. The graphic portrays the pump operating flow relative to the pump design flow. In each set, a second picture zooms in on the transparent pipe to give more detail of the flow patterns (Fig. D.1).

- In the first pair of pictures, the pump is operating at or near to its best efficiency flow. The tuft directions indicate that the flow is generally proceeding parallel to the pipe axis [at least on the visible inner wall]. Any direction differences are due to the random nature of turbulence.
- In the second pair of pictures the pumps flow has been reduced to the point of backflow onset. This is implied by the tufts closest to the impeller. They have swung through nearly 90°, in the direction of

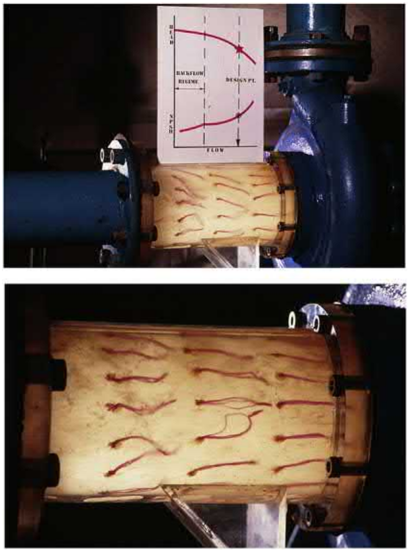



**Fig. D.1** Pump is operating close to its design point and no backflow is exhibited. Not that the direction of the tufts is more or less uniform and into the impeller [Ingersoll Rand].

impeller rotation. The explanation here is that flow is just starting to emerge out of the impeller eye at its largest diameter. The flow pattern within the impeller had already begun to collapse at a slightly higher flow, but this was contained. As flow reduces, this collapsed pattern begins to spill out into the suction pipe**. In order to escape from the impeller, the water particles must travel at a velocity higher than the impeller peripheral velocity.** Even for a small pump such as this, that can imply relatively high velocities. This, in its turn, can lead to high kinetic energies in the liquid particles (Fig. D.3).

- In the final set of pictures, the pump flow has been reduced even further and it is now deep into the backflow regime. The tufts have swung more than 90 degrees away from their position in the first set of pictures. This shows that flow is actually **leaving the impeller** with the high velocity described above, and is spiralling down the pipe and against the main inflow. Measurements show that this swirling flow may extend up to 15 pipe diameters upstream and away from the pump before it is overcome by viscous drag and is turned back towards the pump.

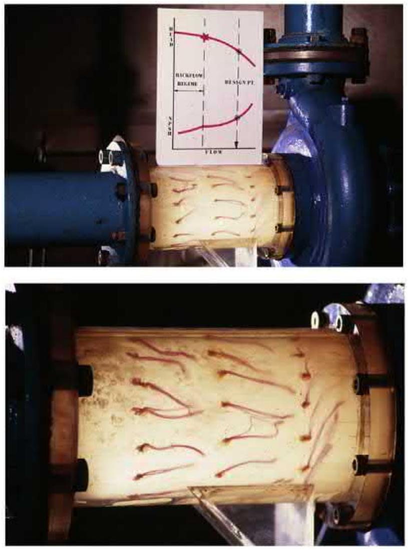


**Fig. D.2** pump flow has been reduced down to the point of inlet backflow onset. Note that tufts close to the pump are just beginning to reverse direction. The impeller flow picture has already collapsed [Ingersoll Rand].

- Separate studies show a complex picture; an annulus of swirling back-flow surrounding a core of axial inflow.
- Inlet backflow can have a direct effect on the MCSF, *see* Chapter 7.

If there are some fixed elements within the 15 diameters, such as bends, straighteners, or ribs, then there is a torque reaction as this swirl is more forcibly dissipated. The swirl is cyclic and unsteady, and this results in cyclic and unsteady torque reaction, leading to low frequency vibration

## WHAT CAUSES IT

There are two co-existing phenomena that result in an impeller ejecting flow back out of the eye and against the main inflow:

The first relates to flow conditions local to the impeller vane inlet edge. In most modern cases, the vane angle is set to more or less match the direction of incoming flow, as seen by an observer sat on and rotating with the impeller (Fig. D.4).

Obviously this setting angle can only be correct at one flow. This is normally at or about the design best efficiency flow of the machine at full impeller diameter. At all other flows there will be a mismatch. The mismatch is greatest at very low or very high flows.

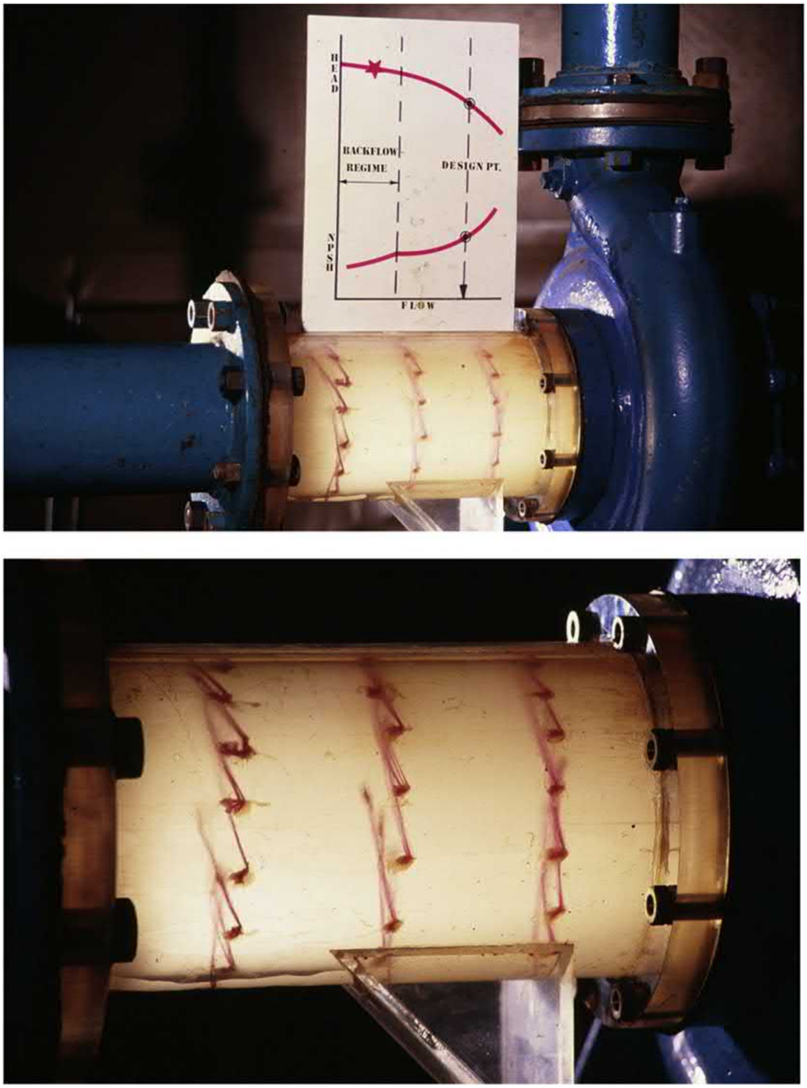



**Fig. D.3** pump flow has been further reduced relative to Fig. D.2 and this causes backflow to become very well established. Note the tufts now show flow leaving the impeller and along the pipe wall with high swirl velocity but low swirl angle [Ingersoll Rand].

[The most prominent exceptions would be pumps designed to handle suspended solids with or without fibrous material. Such impellers are usually configured to strongly encourage inlet backflow as part of a 'self-cleaning inlet' design feature. The backflow helps to backflush any

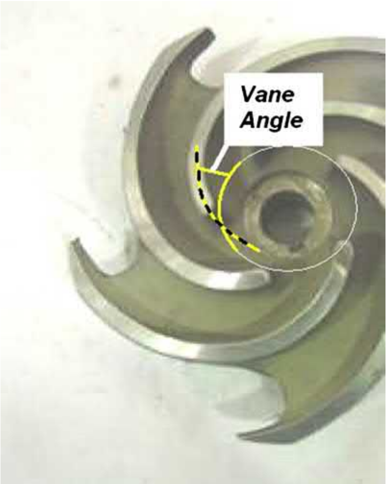



**Fig. D.4** The setting angle of an impeller vane will more or less match the liquid approach angle at the design flow of the pump. At flows greater or smaller than this, the flow will approach at a greater or lesser angle. Thus the blade setting angle can only be correct for one flow. This is known as the 'Zero incidence' condition.

fibrous mass that might otherwise collect on the impeller inlet edge. This strategy depends on the vane angle and flow direction **NOT** being matched.]

At the flow chosen for design setting conditions, the flowpath is deflected very little before proceeding on and into the vane-to-vane passage space. If flow through the machine is reduced, the flowpath angle also reduces. Consequently, the flowpath angle and fixed vane setting angle begin to differ. This difference is called the 'angle of incidence'. Zero incidence will normally, [bit not invariably] occur at or near to the design condition.

At first, the flowpath liquid is able to adjust to the now larger vane setting angle. It is still able to closely follow the blade surface and make the turn into the vane-to-vane space.

With further flow reductions, the flow, the angle difference becomes larger. A point is reached where the liquid becomes unable to make the turn and instead separates from the vane surface. For the purposes of illustration, this takes place whenever the difference between the two angles exceeds about 8—10 degrees. A strong cell of unsettled and detached flow will now form downstream of the inlet edge. But further down the flowpath, this low energy liquid becomes gradually entrained back into the main flow.

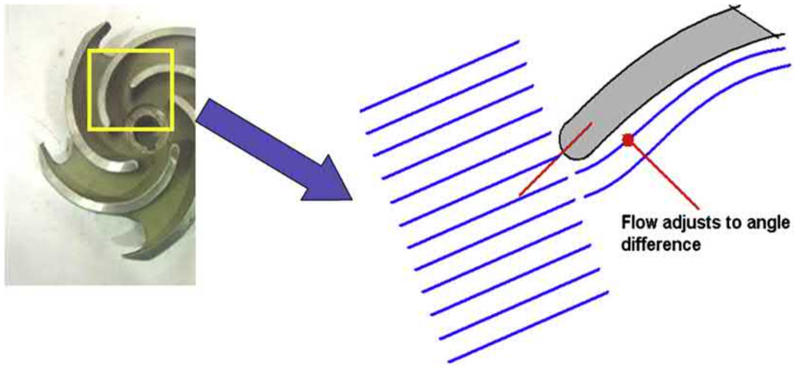



**Fig. D.5** Where the flow angle and the vane setting angle only differ by a few degrees, the liquid is still able to re-adjust and closely follow the surface of the vane.

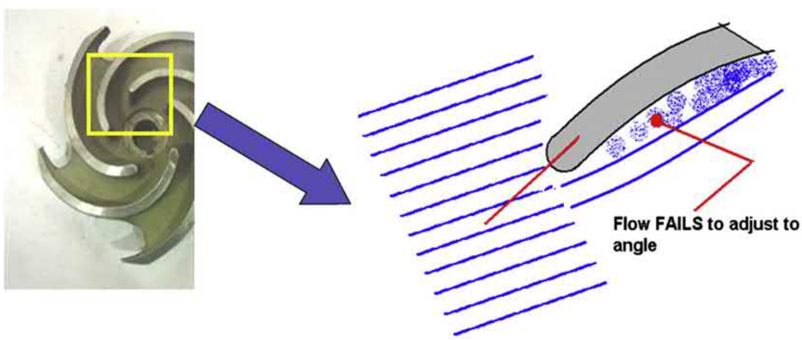



**Fig. D.6** If the difference between the vane and flow angle becomes too great, then the liquid fails to follow the vane surface. Instead it separates and forms a cell of stalled liquid.

If flow is further reduced beyond this point, then the size and extent of the cell continues to increase, chiefly in the direction of flow.

Secondly, conditions at the impeller rim change, as pump flow is reduced. This picture has already been described previously, but, to reiterate, consist of low energy 'WAKE FLOW' attached to the concave face of the blade while higher energy 'JET FLOW' tends to follow the convex surface. Conceptually, at least, a boundary can be imagined between these two regimes. Even at the flow of best efficiency, the Wake Flow exists but for the purposes of this illustration only becomes significant at about 2/3 along the impeller flowpath.

As pump flow reduces, the Wake Flow onset point occurs at an earlier and earlier upstream point along the flowpath. With reducing flow, there

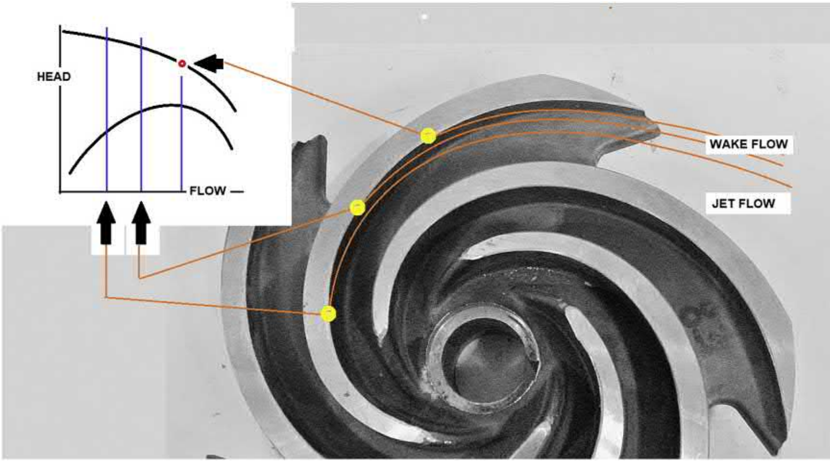



**Fig. D.7** Conceptually, the impeller passage flowpath divides into two zones, a high velocity Jet and a lower velocity Wake. The onset point of the wake flow moves upstream as pump flow is reduced. The wake also thickens in the process.

comes a point where the inlet and outlet flow disturbances can connect and conceptually merge into one. At first, this condition remains completely contained within the rotating impeller.

It seems that, within the geometry range of normal commercial machines, the backflow picture remains chiefly inside the impeller, provided that the inlet tip speed is less than 20—25 m/s. It remains largely contained down to at least Minimum Flow.

If the inlet tip speed exceeds this threshold, then backflow breaks out and a flow 'spike' will start to emerge from each impeller passage. This is mostly ejected from the concave face and is directed out against the incoming flow. Re-stating the obvious; in order to escape the impeller, the particles must attain a velocity in excess of the inlet tip velocity. In the case of an end suction pump the only way this swirl can be dissipated is by viscous drag on the pipe walls and against the incoming flow. This is a weak mechanism which, if not assisted by anti-swirl ribs may then extend upstream more than 10 pipe diameters before being entrained back into the incoming flow.

This is a two dimensional description of a mechanisms which is three dimensional in nature. Merging of the two patterns occurs first and most intensely along the impeller outer shroud flowpath. That is because the inlet flows will normally first 'stall' along this streamline. This Figure shows [conceptually] the flow pattern in two views.

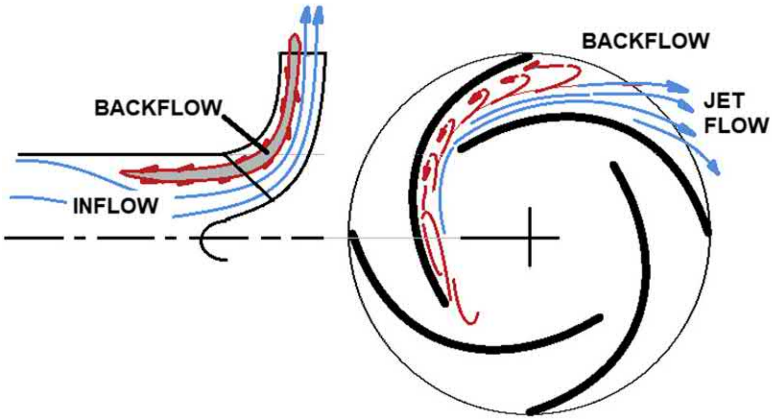



**Fig. D.8** This figure attempts to illustrate the flow patterns of inlet backflow. Referring to Fig. D.7, a point is reached where some of the wake flow reverse and connects with the inlet edge stall cell pictured in Fig. D.6. The consequent backflow forms flow spikes at each passage throat. If the inlet tip speeds are high enough, these spikes can escape from the impeller and into the inlet duct. They may then extend up to and beyond 10 pipe diameters.

One might conclude from this that those impellers where the outlet diameter tends towards that of the inner [eye] diameter would be more susceptible to the flow patterns merging, than those where the difference is great. This turns out to be the case. In practice, as the impeller inlet to outlet radius ratio approaches, then exceeds 0.5 the effects seem to become more noticeable (Fig. D.9).

There is a small but very important flow detail not easily shown on either Figs D.5 or D.6. These are the filamentary shear vortices that form along the interface between the inflow and core and the backflow annulus. This is easiest explained by an analogy:

It is believed that when the Pyramids of Egypt were built, the large stone blocks were moved across the ground in wooden log rollers. In this backflow analogy, the ground represents the main inflow, the stone blocks as outgoing backflow and the wooden rollers the filamentary shear vortices.

Since the backflow must exceed the impeller inlet tip speed in order to escape, it means it is capable of imparting a high tangential velocity to the vortices. Consequently the pressure in the vortex core can go below the vapour pressure and cavitation clouds can form. It is the subsequent collapse of these clouds that sometimes gives backflow its characteristic crackle. Most of these collapses occur harmlessly in free space. But some cloud

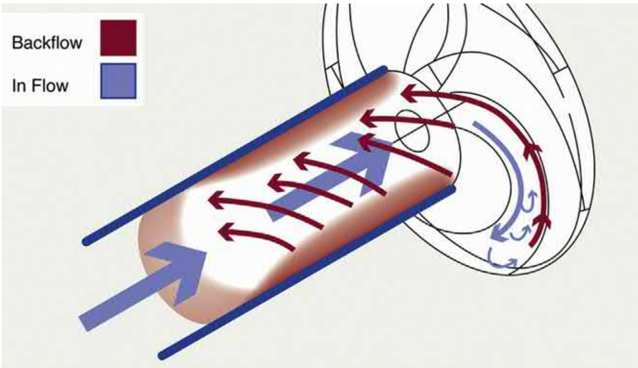



**Fig. D.9** This is a simplified and idealised three dimensional sketch of the complete mechanism. It should be read in conjunction with Fig. D.8. The core liquid moving up the centre of the pipe will typically be travelling with a velocity of only a few m/s. The escaping backflow will be spiralling out along the wall of the pipe with a velocity 10−20 times higher. Filamentary shear vortices will form all along the interface between these two flow regimes [see text].

bubbles can become entrained into the main flow where they then collapse in unexpected places, not predicted by classic theory. That might be on the impeller or perhaps on casing surface near to the impeller eye see Figs J.25, J.92 and J.93.

## WHAT ARE EFFEFTS OF INLET BACKFLOW?

Backflow has associated Hydraulic and Mechanical side effects.

### Hydraulic

#### *Performance curve shape*

Fig. D.8 illustrates the backflow pattern extending less than one pipe diameter upstream. This reflects the situation at flows a little below the onset point of an impeller where the inlet tip speed exceeds 20−25 m/s. If the flow in such a pump is reduced down to only a few percent of its bep flow, then this swirling flow annulus may extend more than 10 diameters upstream.

Fig. K.32 shows how inlet backflow will reduce the apparent head of a pump. Work that would otherwise be channelled into producing head is instead absorbed in driving the backflow mechanism. It might be interesting to note a further nuance. At or about the point of inlet backflow onset, a Head Hysteresis zone will be detectable in the performance curve. This will be located at the point where the effects of inlet backflow cause a detectable change in pump curve slope. This effect was first highlighted by Gongwer in [Ref. 25] (Fig. D.10).

It is probably academic to note that most pumps can exhibit this hysteresis zone, but to reveal it requires a very large number of closely spaced test points. The normal densities of test points are most unlikely to show it *[other than maybe a 'bad' test point*]. In passing through this zone, pump head will generally appear slightly higher if test flow is reducing than if it is increasing.

In many cases, there will be a corresponding zone in the overall efficiency curve shape (Fig. D.11).

The changing conditions at backflow onset also affect the NPSH [R] curve shapes. The curve based on 3% head decay will often show a small but detectable change in curve slope below this point. But any curves based on lower head decays will exhibit more profound curve shape changes. Typically this will appear as a greater discontinuity. However it is the onset flow value that is of greater significance because it signifies the upper limit of the cavitating surge zone.

It is worth reiterating that, the presence of inlet backflow is a prerequisite for cavitating surge to exist. Furthermore, all and any pump becomes vulnerable if the NPSH margin becomes small enough

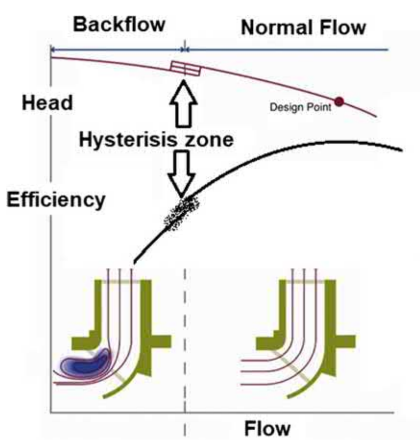



**Fig. D.10** Onset of inlet backflow is often signalled by a slight hysteresis loop in the performance curve. Inside this loop the pump head will be higher if flow is decreasing than if it is increasing. A corresponding 'kink' will be identifiable in the overall efficiency curve.

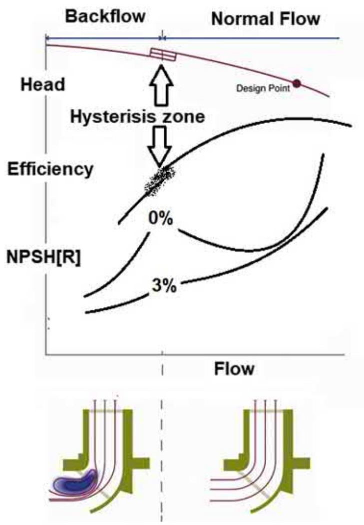



**Fig. D.11** Typically, the hysteresis zone closely corresponds with a discontinuity in any NPSH [R] curve determined on the basis of very low head decays, such as 0% or 1%. Any accompanying discontinuity in the 3% curve will usually be less than obvious.

### *Inlet surge*

If a pump simultaneously operates in the backflow regime and the NPSHA is reduced down to the 3% head decay line, then the machine is prone to exhibit flow surge characteristics. With further reductions, the surge noise intensity becomes more pronounced. The phenomenon is characterised by flow and head surges at frequencies between 0.5 and 2 Hz [Ref. 26].

With machines of low intrinsic noise level [less than 80 dB A], even the early stages of surge will be self-evident-over and above the crackle from the filamentary vortices. . In noisier machines, the initial surge levels will be swamped by general machine noise [Ref. 20].

A typical surge cycle proceeds as follows (Fig. D.12).

- [A] The cavity presence progressively chokes off the outer annulus of the impeller inlet passage flow that is back flowing.
- [B] The backflow pattern grows along the inner wall of the inlet pipe — in oppositions to the incoming main flow.
- [C] This blockage effect reaches a point where the axial inflow velocity is high enough to unstall the previously stalled inlet edge, and so the complete backflow patterns suddenly collapses.
- [D] Without the backflow blockage, the axial inflow velocity drops and once again the flow angles differ from the blade setting angles, so stall

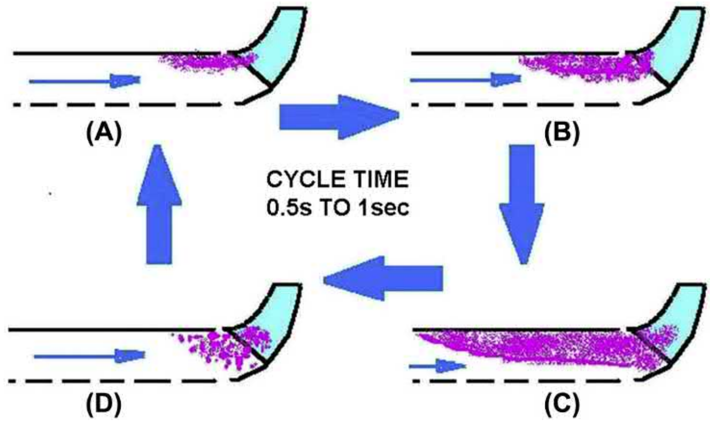



**Fig. D.12** Typical cavitating surge cycle. Backflow repeatedly grows and collapses. There will be associated surging noise.

conditions reappear all along the edge. . Backflow intensity grows again and the cycle returns to [A].

The cavitating surge regime is closely related to the NPSH [R] curve and is shown in the Figure.

In normal circumstances, sites which have a flat NPSH [A] curve would normally be preferred [curve A], since they will permit the highest run-out flow. On the other hand, and for the same reason, NPSH [A] curves which are rather steep would usually be deemed undesirable. However, to help avoid surge, steep curves [curve B] are actually more helpful.

As with the backflow phenomenon itself, inlet tip speeds need to be in excess of 25–30 m/s for surge effects to become serious. Below these levels, the effects are mostly likely confined to a reduction in long term life. But with levels significantly above this limit, quite serious mechanical damage can be created.

## Mechanical

Appendix K describes how the presence of flow straightener ribs, either integral to the pump, or in the closely adjacent pipework might help the curve shape. If the inlet tip speeds exceed the 20–25 m/s threshold, they can also be responsible for creating low frequency vibration. Each back flowing 'spike' can produce a torque reaction with these vanes at inlet vane-pass frequency. Evidence that this occurs is shown in Fig. J.95.

Even where or when such ribs do not exist, the pipework design may still induce unwanted effects. Since the backflow pattern can extend several

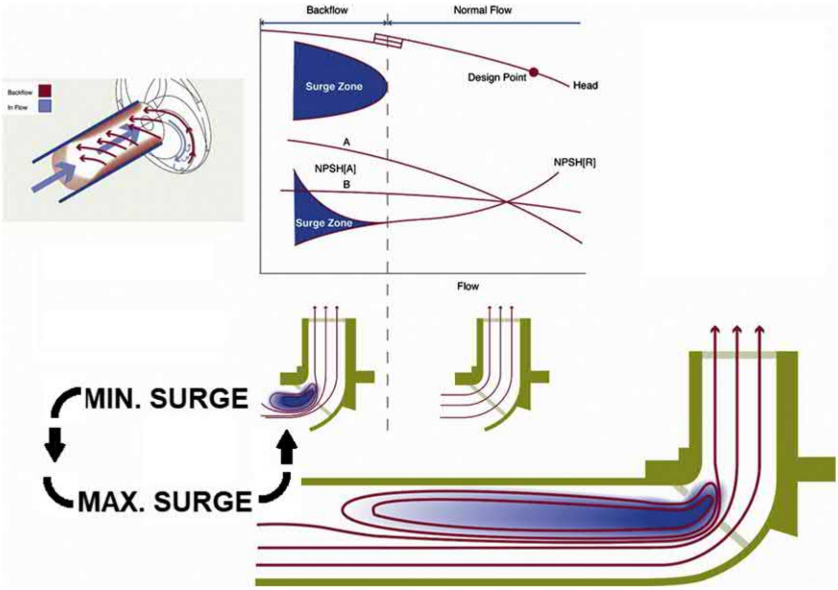



**Fig. D.13** The risk of cavitating surge occurring depends on the site NPSH [A]. Paradoxically, those with a high friction component [steep curve] are less likely to suffer than the more traditional flat curves.

pipe diameters upstream, and if there are instead flow discontinuities such as sharp bends or tee pieces, then a dilute form of interaction is likely.

The flow momentum changes brought about by each surge cycle can manifest itself in significant long term mechanical damage. There are examples of pipe and or bedplate holding down bolts shearing. Small bore service pipework on the pump can also suffer.

In some cases, the pump rotor can be seen to 'shuttle' within the limits of the axial location device. This occurs at the same low frequencies of 0.5–2 Hz.

## BACKFLOW PREVENTION

Methods of diluting the effects of inlet backflow are outlined in Appendix K. But methods of prevention also exist.

The first concept is particularly suited to machines in which the inlet flow approaches from an axial direction. As the impeller enters the backflow regime, high velocity spikes trace out an annular volume at the eye diameter. These spikes leave with a high swirl angle. To capture this swirling liquid, a coaxial upstream annulus is arranged. Cambered vanes within this annulus remove the swirl and reintroduce the now-benign fluid back to the inlet duct (Fig. D.14).

The second concept is more applicable to machines in which the flow approaches rom a radial direction. As in the previous example, the backflow spikes leave with a high swirl velocity and trace out an annular volume. Again a coaxial upstream volume is arranged to capture the liquid. Unlike the first device, there are no internal vanes of any significant. Instead, the liquid loses angular momentum as it spins radially outwards. At the outer rim of the volume, most of the swirl has been dissipated so, again, the liquid is re-introduced benignly to the inlet duct (Fig. D.15).

Not only is the surge intensity now drastically reduced, but the curve shape is dramatically improved.

Both these devices can largely mitigate the potential for backflow to generate any significant level of swirl. They involve some amendment to

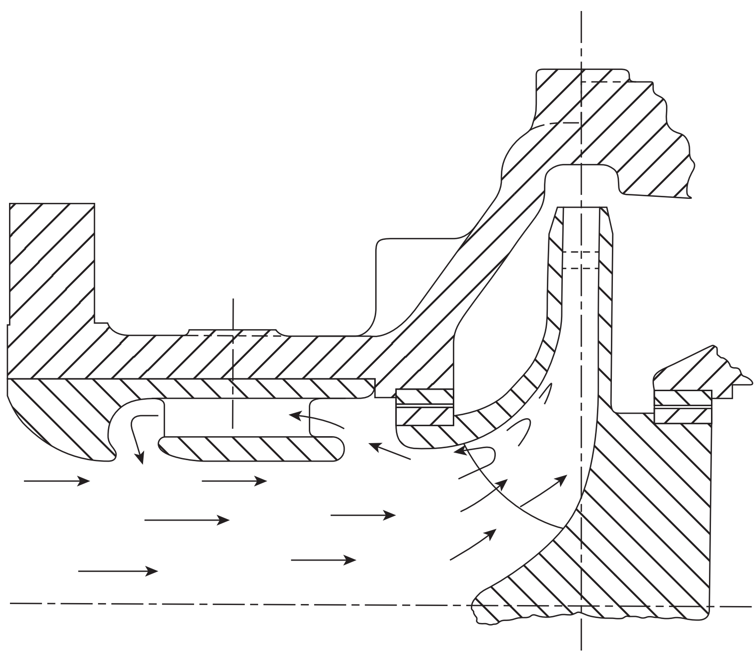


**Fig. D.14** This device segregates the Backflow into a co-axial chamber where it is de-swirled by a cascade of cambered vanes before being re-introduced to the inlet duct. It is most suited to machines where the flow approaches from an axial direction [Ingersoll Rand].

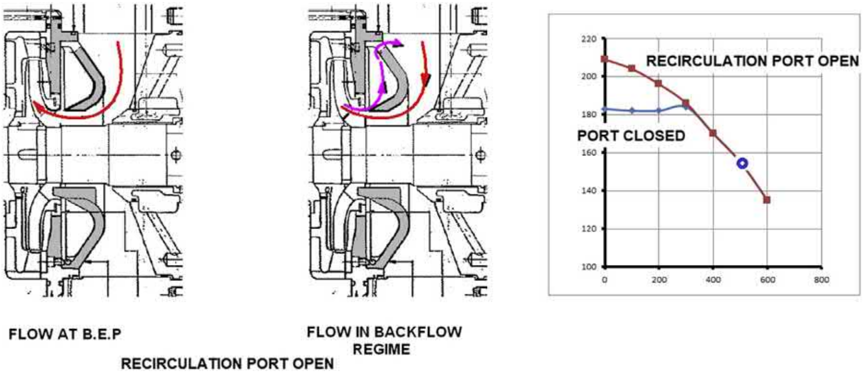



**Fig. D.15** In this device, the segregated backflow is lead in to a more radial chamber. The liquid loses much of its angular swirl velocity as it gains in distance from the axis of rotation. It has become quite benign by the time it is reintroduced to the inlet duct at a high radius. This arrangement is most suited to machines where the liquid approaches from a radial/side direction. Without the recirculation ports, the performance curve becomes quite flat inside the backflow regime. With the ports open, the curve is much more suitable [Mather+Platt].

the basic machine layout. If or when the pump is installed in conditions where backflow *[and by implications surge]* is not a risk, they only improve the characteristics of the pump – at some cost of course.

## ESTIMATING INLET BACKFLOW POINT

Those engaged in troubleshooting may wish need to know the backflow onset point for a particular machine. This can often be determined accurately during a performance test. But in the field, this information may not forthcoming, so a method of estimation will be useful. The trouble-shooter may not have access to do this. Nevertheless, with some approximation, helpful results can be yielded [Ref. 22].

- Not surprisingly, knowing the inlet tip speed is a key requirement. The tip speed [eye peripheral velocity] [UI] is related to the eye diameter of the impeller. The impeller can be measured to determine this. If that is not possible or convenient, then the suction branch diameter can be used as a **very rough guide** to the eye diameter.
- Equally, the impeller inlet blade angle will rarely be known to the user. If a sample impeller exists, then an estimate of the blade angle can easily be made. This process is outlined later. For the most part, angles impeller inlet setting angles will lie in the region of 14 degrees–22

degrees. Pumps with high values of suction specific speed [exceptionally low NPSH [R]] will have values nearer to 14 degrees.

- Knowing these two pieces of data allows Fig. D.16 to be entered and the value of backflow estimated. Let's look at an example:

Example 1-Process pump, single stage single entry, end suction, 6″ [0.150 m] suction nozzle, 2950 rpm, Suction specific speed 11,000 [Usgpm units]. Liquid SG = .88, flow at bep is 850 Usgpm. The ratio of NPSH [A]/ NPSH [R] is 2

1 Inlet tip speed [U1] is approx.

$$U1 = 3.142 \times 0.150 \times 2980/60 = 23.4 \text{ m/s}$$

2 Assume the outcome of a measurement attempt on a similar spare impeller [*see later*] yields an inlet angle of 17 degrees.

3 From Fig. D.16, ratio of Qc/Qi is 0.52.

Therefore, the backflow onset point QC = 0.52 × Qi.

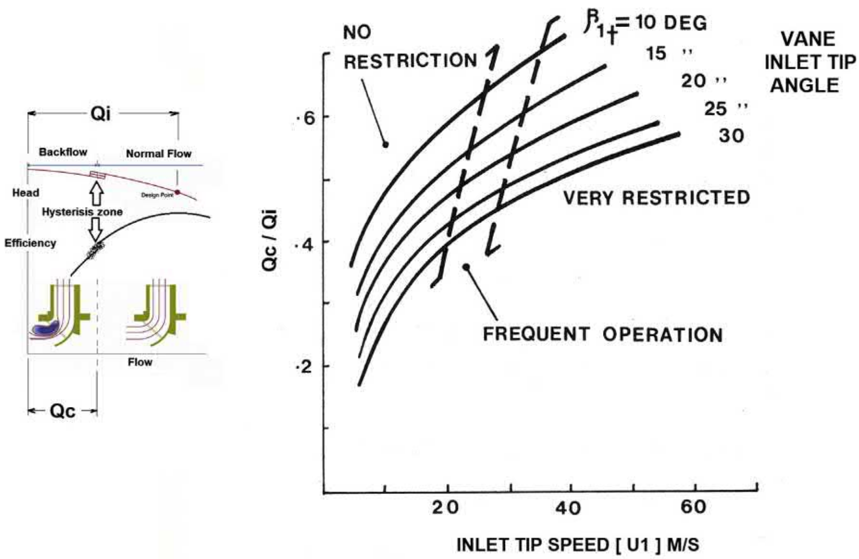



**Fig. D.16** This figure relates the point of inlet backflow onset to the inlet tip speed and the vane inlet angle. Methods of estimating the inlet angle of an existing impeller are described in the text. This figure shows how the onset intensity increases with tip speed. As a rule, impellers with low values of inlet angle have low NPSH [R] requirements and accordingly - high Suction Specific Speed [Nss]. But they also have higher tip speed because the eye diameter will be larger. That is why impellers of high Nss will also TEND to have high values of Qc/Qi and hence narrower Comfort Zones.

**4** Now Qi lies between 1 and 1.2 of the bep flow for the pump [in most cases–depending on design philosophy].

So backflow onset [Qc] occurs at $0.52 \times Q$ [bep] $\times 1.1$, = 486 Usgpm

**5** Now since inlet backflow effects relate to kinetic energy, and since the data upon which this approach was based was gleaned from water test, then Qc can be multiplied by the SG to obtain Qc on final product.

This gives a value of 427 Usgpm and puts it [Fig. D.13] in the zone of **'Frequent operation'**. Pumping in the backflow regime is thus permitted, but not encouraged.

**6** Furthermore the ratio of rotor mass to casing mass influences the general response of the machine**.** Table D 1 can be used as a guide in this respect.

For a pump of robust Oil industry construction, operating in the 'Frequent Operation' zone, and then operation at $0.8 \times$ Qc is allowable.

**7** With this adjustment, the MCSF turns out to be $0.8 \times 427$, = 342 Usgpm.

**8** Now looking at the avoidance of cavitating inlet surge [see Appendix E], the NPSH margin must be explored. From Fig. D.17, and with an NPSH [A]/NPSH [R] margin of two, the margin factor is given as 0.66. This means that the pump can be operated at $0.66 \times 342$ before inlet surge might become an issue [with a pump of this construction]

Therefore, MCSF would be $0.66 \times 342 = 226$ Usgpm

Some interesting observations come from this review. For example, it is quite possible to configure pumps for the same hydraulic design conditions,

**Table D.1** MCSF adjustment based on pump stator-to-rotor mass ratios.

| **Pump category** | **Operating inside the 'very restricted' zone** | **Operating inside the 'frequent operation' zone** | **Operating inside the 'no restriction' zone** |
|---|---|---|---|
| Casing: rotor ratio is high, i.e. Oil industry pumps, Solids handling pumps | 1.0 | 0.8 | 0.6 |
| Casing: rotor ratio is low i.e. Industrial pumps, Chemical pumps | 1.0 | 1.0 | 0.7 |

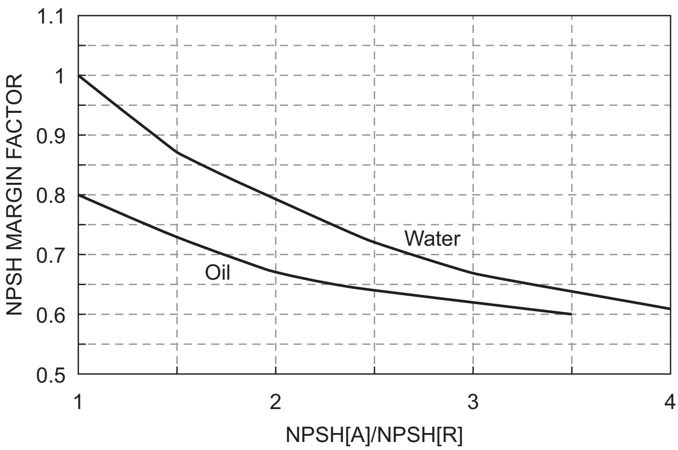



**Fig. D.17** Cavitating surge may occur if, and only if, inlet backflow is present [see Fig. D.13 and Appendix E]. However, the surge intensity will reduce [and may be insignificant] if the NPSH [A] exceeds NPSH [R].

but destine them for two different industries. A pump destined for the oil industry will be of a heavier duty construction than that expected by the water supply industry. This means its stator-to-rotor mass ratios will be different. Furthermore, the product densities will usually differ. Therefore, the same hydraulic design would be assigned a different MCSF if mechanically configured for [say] the oil industry than for the water industry.

## METHOD FOR MEASURING APPROXIMATE VANE INLET TIP ANGLE

Knowing the vane setting angle is an essential part of the Inlet Backflow calculation described above. However, if troubleshooting in the field, this value might not be available. The OEM may be unwilling or even unable to disclose this data. On the other hand, if the OEM has gone out of business, getting this data might be doubly difficult.

The following outlines a measurement process that will allow the angle to be determined with enough accuracy for troubleshooting purposes.

There are two impeller categories; In the first, the impeller has vanes that are only twisted in two dimensions (Fig. D.18),

For these impellers it is often more convenient to take a 'rubbing' of the vane onto paper–in the same manner as archaeologists.

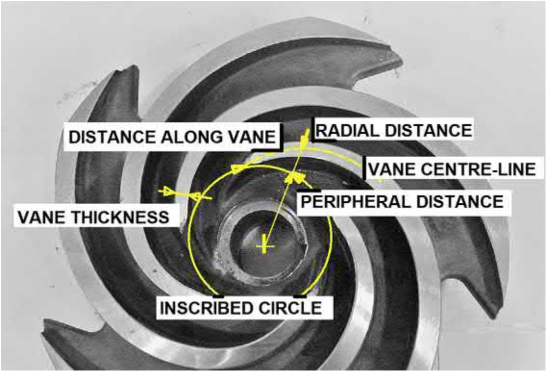



**Fig. D.18** These measurements can help to yield the vane inlet tip angle. It relates to vanes that are twisted in only two dimensions. Fig. D.19 shows the equivalent for vanes twisted in three dimensions.

- On the rubbing, mark a circle that inscribes the vane inlet tip.
- Mark a distance along the inscribed circle equivalent to typically eight [8] times the vane thickness at inlet.
- Through this point, draw a radial line that cuts the vane centre line.
- Measure off this distance along the vane of the intersection point. Then, from this, the vane tip setting angle can be assessed.

Cos [vane setting angle] = Peripheral Distance/Distance along Vane

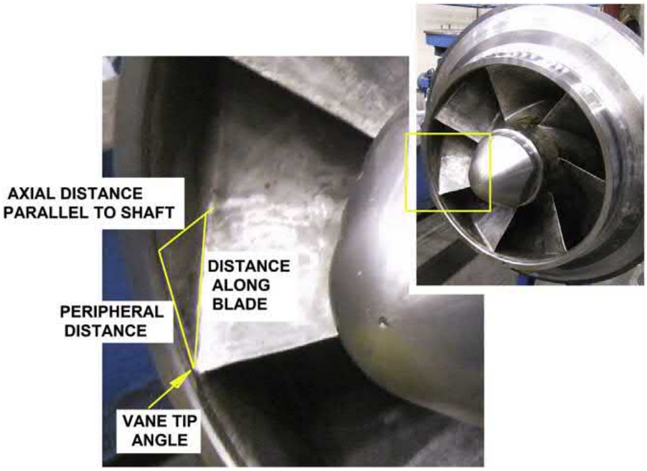



**Fig. D.19** These dimensions can help to estimate the vane inlet angle of vanes twisted in three dimensions. Often there will be a fillet radius where the vane and from shroud inner surface merge. Thus some judgement will be needed.

The second category has vanes that are twisted in three dimensions. Taking a 'rubbing', as with two dimensional vanes, is seldom practical.

- In this case, a Peripheral Distance of typically ten [10] times the vane thickness is marked of on the actual component.
- Then a line is drawn at right angle and parallel with the shaft axis.
- Finally, the Distance along the Blade is measured.
- With this category of vane, the casting foundry will usually add smoothing fillet radii to intersections. This will make assessment of these two dimensions more subjective.
- Because this process measures the approximate angle of the, concave vane surface, not the vane centre line, one should add [say] 2 degrees before entering Fig. D.16.

## APPENDIX E

# Cavitation

Previously, I noted that under certain conditions cavitation can develop inside a pump. Pragmatically, many pump designers and users are more interested in how to avoid cavitation, than the physics of the phenomenon. But there is an old saying; 'Know your enemy!' So some insight to the phenomenon might be useful. The subject is covered very thoroughly in Ref. [30]. What follows here, is a more superficial description.

Just to repeat, pumps do not suck liquid. A rotating impeller creates a low pressure zone in the vicinity of its eye (Fig. E.1).

It is actually the higher pressure acting on the feed vessel surface that pushes the liquid towards this low pressure region pump impeller eye. If the impeller wants to displace more flow than this mechanism can provide, then cavitation occurs. Pumps that are connected to pressurised feed vessels can provide liquid more readily than if the surface pressure is [say] atmospheric. Hence they are less susceptible to cavitation phenomenon. So clearly, pumps can work better hydraulically with higher suction pressure. At lower and lower suction pressure there comes a point where, as described above, the rotating vanes want to displace more flow than the suction pressure can provide and it is then that cavitation occurs.

Conventional wisdom will avoid cavitation at all costs. Even if the effect on performance is ignored, there is a widespread belief that all cavitation

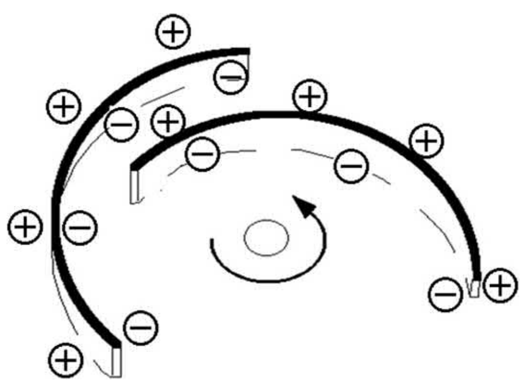



**Fig. E.1** Conceptual pressure distribution on impeller blade of a pump operating near to its design flow. The differential pressure acting over the blade wetted surface area is responsible for creating most of the torque that the pump drive must overcome.

should be avoided. However there are some cases where the existence of cavitation will be almost inevitable. But it might be quite benign. Even though the impeller will suffer damage it occurs so slowly that the impeller will often fail by other means, such as erosion, or corrosion. Avoiding cavitation completely in such cases will incur unnecessary costs. It will solve a problem that does not really exist in practice.

In the past, cavitation was the sometimes blamed as the root cause when all other explanations failed. Nowadays we better understand both the cause and effects of cavitation and it is not a subject worthy of over-simplification. Outwardly it makes its presence felt in a number of ways, depending on the severity or intensity.

At first, I will describe cavitation in simple terms. Then I will elaborate a little on its effect, both in terms of machine performance and then on impeller life.

## CAVITATION PART 1 – SIMPLE DESCRIPTION

Cavitation describes the occurrence, in a liquid, of vapour bubble. We are familiar with the 'singing' sound of a boiling kettle. This is actually the noise of cavitation developing at local hot spots. In a kettle containing water, this boiling point occurs at 100 deg. C if, and only if the local atmospheric pressure equates to a standard set of values. However, if for example the local atmospheric pressure is reduced, then the water will boil at a lower temperature. It is for this reason that it can be impossible to produce a good cup of tea or coffee on a mountain top, for example. This is because the water appears to boil at less than 100 °C, due to the lower local atmospheric pressure. But the local atmospheric pressure can reduce in other ways. If for example the velocity of a liquid stream is accelerated, then its local static [or atmospheric] pressure, as measured by a gauge, will reduce [Bernoulli's Law]. Such a local acceleration occurs at sudden and significant area changes along the flow path. This can occur in orifice plates, for example and may be detected by noise emission. Another qualifying area change would exist in a [say] flow valve that is nearly, but not quite closed. In this case, the apparently closed valve emits a high frequency noise and is often described as a 'passing' valve – meaning some through-flow still exists.

Yet another way in which liquid can be [very] locally accelerated is where it encounters an obstruction in the flow path. This will occur at the upstream edge of a vane or rib placed in the flow stream. In having to divert slightly around the body the flow must accelerate. Here the acceleration can be very local indeed. One purpose of streamlining is to help reduce the intensity of such local acceleration. However, the outcome is just the same as in a kettle: wherever the local pressure becomes low enough, local cavitation/boiling can occur. Obviously, streamlined bodies will cavitate the least.

This last condition can obviously occurs in a pump, at the point where the flow first encounters a rib or vane. While stationary ribs in the pump casing entry duct are susceptible to this pressure drop, the impeller vanes are far more susceptible because the liquid encounter velocity is very much higher. This explains why boiling or cavitation normally occurs first and is most intense, close to the entry edges of an impeller blade. In abnormal circumstances, cavitation might occur at other locations. This will include cavitation possibly forming in the core of any vortices that might form somewhere further along the flowpath. But any vortices that form in the low pressure region of the pump will be the most susceptible.

But the presence of cavitation is not necessarily always a matter of concern. That is because it exists in various scales or intensity. As the boiling point is approached but not reached, then the scale of cavitation is microscopic. In fact it is most easily detected by its noise emission, though this is occurs at frequencies beyond the range of commercial detectors. This condition is called the 'acoustic inception' point. As the boiling point continues to be approached, the boiling bubbles, though microscopic, begin to be visually detectable. This point is called 'visual cavitation inception', as opposed to the acoustic detection point previously described. Right at the boiling point, the bubbles become quite apparent.

But, as we shall see, it is not so much the presence of vapour bubbles, but how and where the bubbles collapse that is significant. In a conventional kettle, the bubble rises away from its creation point and it collapses, or condenses harmlessly in the liquid space. In an impeller, the bubble may collapse on or close to the blade surface itself. On the other hand it may also collapse harmlessly in free space. In both cases, the collapse will emit noise, but damage is only likely in the first case.

Unfortunately, it is rarely possible to see inside a pump that is operating [Ref. 19]. This makes it difficult to define a 'hard' flow-related threshold for cavitation onset. In the 1950s the issue of pump cavitation seemed less of a concern. It was often possible to meet the manufacturer's requirement by plant design. Even then, the issue of cavitation was an emerging discipline. In the years since, plant design has become lighter with less margins and this exposed some gaps in the pump state of the art.

The problem begins with defining cavitation. There are a number of states, some of which we have already mentioned, such as acoustic or inception. The definition is tailored to the circumstances. The pump community is chiefly concerned with defining cavitation intensity that does not cause the performance to decay adversely. But there is an almost equal concern with defining cavitation intensity levels low enough to ensure that the impeller will probably fail by other means. The first criterion can be predicted, then directly measured by testing and so can be proven quite quickly. The second criterion can also be predicted, but needs time to be proven.

Historically there has been a somewhat grudging agreement between pump makers and pump users that cavitation is understood to be already well established when the head generated by the pump has decayed by 3%. *[With multistage pumps this relates only to the head of the first stage].* The pump makers produce flow-related tests curves that show the NPSHR where the performance has decayed by 3%. But this has little or no direct obvious connection to damage rates in impellers. It is left to others to apply some safety margin on such data in order to ensure an adequate life and resistance to cavitation damage.

Earlier I briefly outlined, the concept of NPSH 'Required' as a means of defining cavitation onset. This is a property that relates specifically to each pump at each flow. It is determined experimentally during the course of a pump performance test. Ideally, this is accomplished by changing the pressure at which the inlet flow is supplied, while all other parameter such as pump speed and flow, are held constant. The pressure at which cavitation is perceived to induce performance decay is determined. This test is conducted at several points across the flow range.

Each physical pump installation also has a property called the 'NPSH Available' This is dependent on conditions on site and is independent of the pump. This value must not only equal the 'NPSH Required' by the pump, but it must exceed it by some degree, if both cavitation is to be avoided **and/or** satisfactory life achieved. If the pump is relatively small or of low shaft speed, then this margin/excess is very small. In the case of larger and or faster pumps, this margin can become quite significant.

## CAVITATION PART 2 – EFFECT ON PUMP OUTPUT PERFORMANCE

Earlier I described cavitation as occurring when the impeller blades want to displace more flow in to the collector than suction pressure can provide to them. I left the impression of the blades tearing the liquid apart at their leading edge. Here I want to elaborate on both the shape/positon of the cavitation occurrence and explore its component parts a little more. It will be easier to illustrate these effects by first making reference to the performance of a simple orifice plate installed in a pipeline. Its behaviour can then be mapped to the behaviour inside a normal pump impeller.

Orifice plates are widely used in industry for flow measurement. The pressure drop can be used as an analogue of flow. Such devices are also useful as a primitive pressure dissipation device. They are can be employed whenever excess liquid pressure needs to destroyed (Fig. E.2).

At some point [1] upstream of the orifice, the pressure is above atmospheric [say]. As the fluid approaches and accelerates into the reduced area of the orifice [2], its velocity increase and, in accordance with Bernoulli principles, its static pressure reduces. Near to the orifice [3], the liquid velocity reaches its maximum and accordingly its pressure minimises. The degree of pressure reduction depends on the orifice bore compared to the pipe bore. Smaller diameters infer higher velocities, and from Bernoulli,

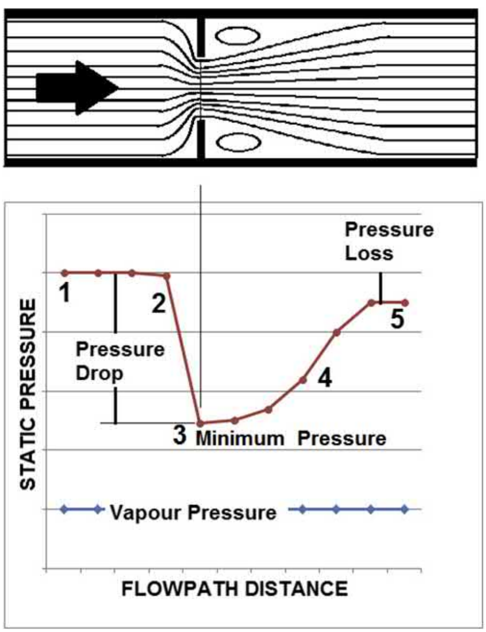



**Fig. E.2** Flow and pressure history of liquid particles passing through a pipe mounted orifice plate. If the minimum pressure goes below the vapour pressure, then cavitation will occur and the device is no longer useful or reliable.

lower static pressure. If the orifice is small enough, or the flow high enough, the pressure here can get low enough to locally cavitate/boil, even at ambient temperature. Downstream of the orifice [4], the fluid velocity decreases, corresponding to the increased pipe area. Beyond the orifice [5], the pressure recovers, but never quite to its original value. Some of it is lost in driving the standing ring-type vortex that forms in an annulus around the orifice. The measured pressure difference between [2] and [4] is very useful. It can be used to calculate the actual flow across the device at any instant.

If the upstream pressure is [1] now reduced then the pressure at [3] also reduces. At some point the pressure at [3] will drop below the liquid vapour pressure at the current liquid temperature. At that instant, the liquid will cavitate/boil locally. Beyond that point and as the static pressure increases again, the vapour bubbles will recondense back into the liquid. Heavily cavitating orifice plates can be detected in industrial sites because of their noise emission. Obviously, they cannot then be a reliable flow measurement device. *[One way of solving this problem is to create downstream backpressure that moves the whole pressure profile line back up above the vapour pressure line and so cavitation is removed.]*

How does this relate to cavitation in pumps?

Well as liquid approaches then passes around an impeller blade/vane placed at its minimum disturbance angle, then it also accelerates. This is due to the blockage effect of the vane presence (Fig. E.3).

And just like in an orifice plate assembly, if the upstream pressure is reduced enough, then cavitation can occur on and around the blade edge. It does so because the presence of the vane causes a blockage effect. Thick vanes cause more blockage than thin ones of course. Hence, thick vane edges are more prone to cavitation than thin ones. In this example, the flow is impacting the vane at what is known the zero incidence angle. This angle will have the lowest losses associated with it. Naturally, pump impeller vane are usually arranged to achieve this zero incidence angle condition at some flow close to its best efficiency point (Fig. E.4).

Boiling/cavitation first occurs when and where the local static pressure goes below the vapour pressure line. The inlet pressure/NPSHA at which cavitation first becomes visible is called the 'visual inception' point. Cavitation at this intensity rarely if ever causes damage to impellers of normal construction. *[As noted earlier, acoustic inception tales place at even higher levels of suction pressure/NPSHA. However, the noise emissions at this condition will not be detectable by the human ear.]*

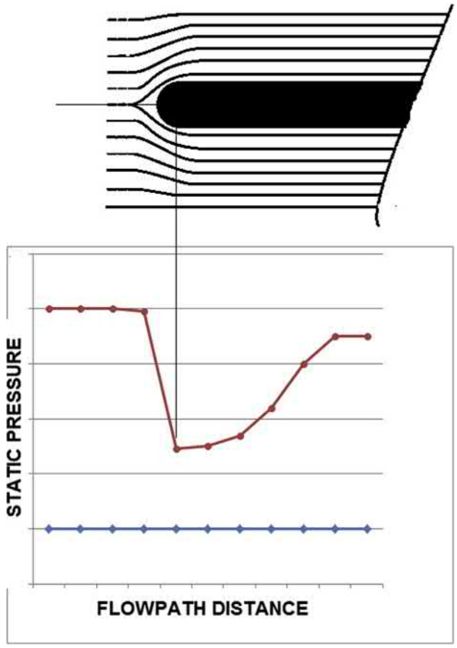



**Fig. E.3** Equivalent flow and pressure history of liquid particles passing over an impeller blade set at an angle to create minimum flow disruption.

Fig. E.5 describes what happens when the impeller vane is closely aligned to the approaching flow. This will normally coincide with the design flow of the pump at its maximum impeller diameter. At all other flows, the approach angle is not zero. What effect does this have?

On one side of the vane, the minimum pressure is lowered even further, due to the **INCREASED** liquid turning losses. In Fig. E.5 this is shown as the full line compared to the dotted line which represent zero incidence angle.

On the other side of the vane, pressure increases slightly due to the **REDUCED** turning losses. [This has been omitted from the Figure for clarity.] The same cavitation conditions are therefore reached at a higher inlet pressure NPSHA than would be the case at zero incidence.

In any case, if the inlet pressure/NPSHA are further reduced, then the cavitation becomes more extensive and more intense. This is because more of the pressure profile dips below the vapour pressure line. Even so, the level of cavitation might still be insufficient to cause detectable performance decay. However it might be enough to initiate component damage.

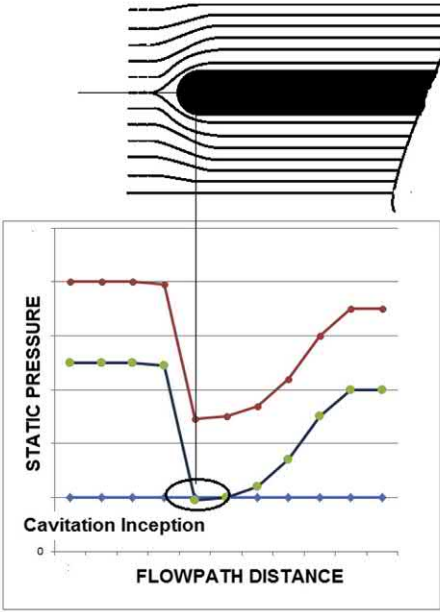



**Fig. E.4** By reducing the upstream flowpath pressure in Fig. E.3 the minimum pressure point can go below the vapour pressure line and cavitation will occur. Because the flow picture is symmetrical, cavitation will form on both sides of the blade.

So far as centrifugal pumps are concerned, there are three progressive stages of cavitation intensity. One state merges seamlessly into the next, but for the purposes of discussion here I will discuss them in isolation.

## Condition 1 no outwardly detected cavitation – [inception]

Cavitation would first be detected, at a microscopic level, on the rounded part of the blade edge profile. This is the location of visual inception. But actually cavitation can be detected acoustically at higher pressures than it can be detected visually. As already mentioned, early acoustic detection is beyond the capabilities of the human ear, and in fact most common industrial measurement devices

Incipient cavitation commences at the point of lowest pressure. As the flowpath moves into the regions of higher pressure, the bubble collapses by being recondensed back into the bulk flow (Fig. E.6).

With the inlet part of the impeller vane set to more or less the minimum loss angle, then these inception bubbles will form on either side of the vane edge. If the flow is increased/decreased and approaches at greater or smaller angle, then the inception bubbles will grow simultaneously grow in size on

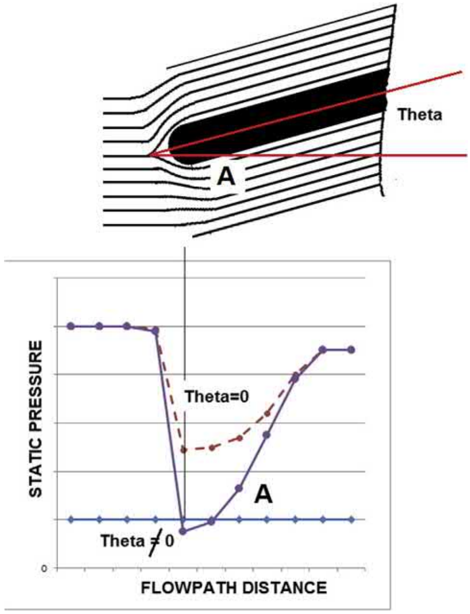



**Fig. E.5** If the blade is set at an angle that does not correspond to minimum disruption, the minimum pressure can go below the vapour pressure line, even though upstream flowpath pressure is higher than shown before in Fig. E.4. Since the flow picture is now skewed and not symmetrical, the cavitation will most intense on face 'A'.

one side and shrink on the other. This leads to an important characteristic, that the NPSH [R] at which inception cavitation JUST appears is flow-related and exhibits a vee form. The minimum 'notch' point is associated with the point of lowest inlet loss [nominally zero incidence angle]. The change in curve slope at flow ratios of 0.4 or less is due to the onset of a particular flow regime. This 'Backflow regime' discussed in Appendix D.

Although I mention the cavity length here, it will become more useful in the subsequent discussion to replace this definition with the suction pressure/NPSHR at which the cavity just appears/disappears. This is because NPSH can be readily measured during any routine pump tests. But, determining the cavity length is not something that that can be easily achieved (Fig. E.7).

At low Flow Ratios [less than 1.0 but greater than say 0.3 to 0.4] cavitation develops most intensely on the concave face of the vane – often called the Suction surface. This is the surface that would be most easily visible when looking into the eye of an impeller. The impeller will require a high inlet pressure NPSHA in order to suppress visual inception.

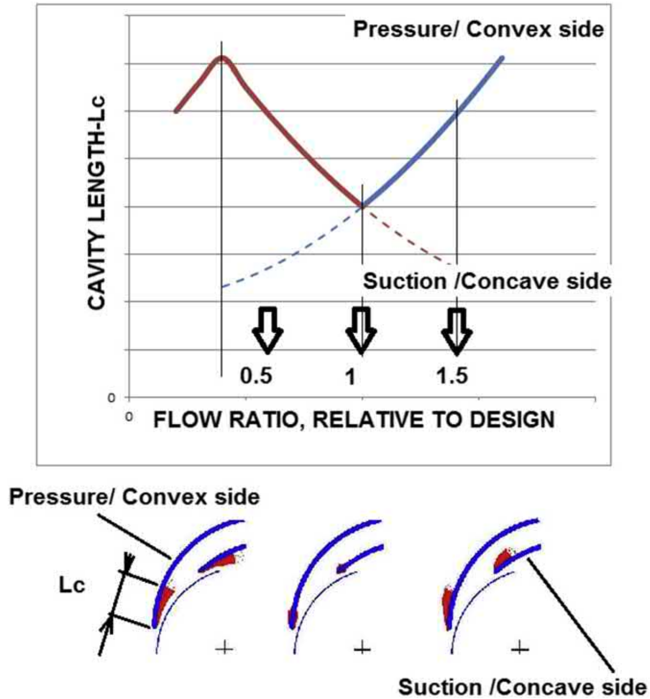



**Fig. E.6** Illustration of cavity length development with respect to pump flow. This is due to asymmetry of the approaching flow angle, relative to the fixed impeller blade setting angle.

At or near to design flow, cavitation occurs on both concave and convex sides of the vane. This is often described as the shockless entry flow or zero incidence condition. Flow angle and vane entry angle closely match each other. The cavity length on the Suction surface has reduced, while that on the Pressure surface has increased to be of a similar magnitude.

At high flow ratios [greater than 1.0] cavitation develops most intensely on the convex face of the vane. This is the surface normally invisible when looking into the impeller eye. Again, the impeller will require a high inlet pressure/NPSHA in order to supress cavitation inception.

Under normal circumstances and with most common liquids or impeller materials, inception cavitation is almost harmless. *[Some pumps may operate in this state all of the time without distress.]* With any further reduction in free-stream pressure, inception cavitation grows in size. This is accompanied by an increase in high frequency noise emission, still outside the range of the human ear.

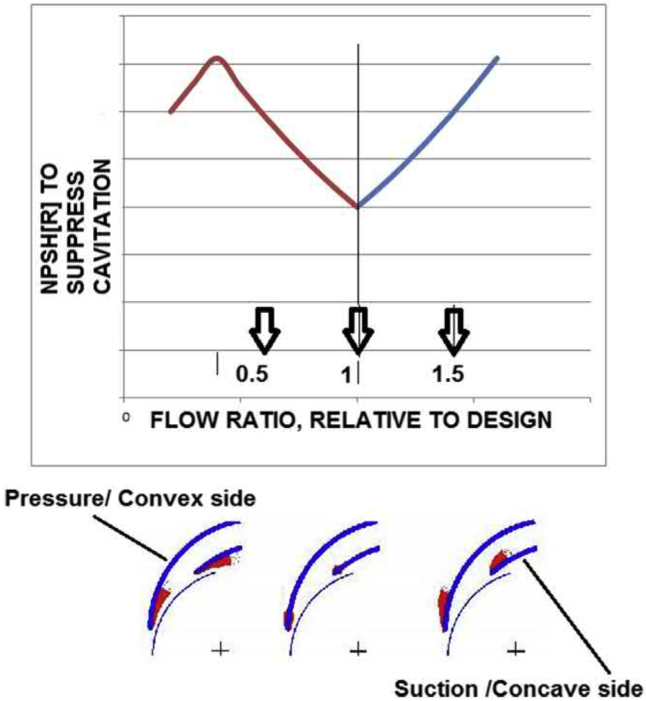



**Fig. E.7** Outside of the laboratory, it is not useful to employ the cavity length as a criterion. That is because visual access does not normally exist. In practice, it proves more convenient to use NPSH [R].

It is also useful to note that cavitation inception occurs almost in isolation in an impeller of 'normal' design. Any isolated cavity occurrence is little influenced by the presence of adjacent vanes on the impeller.

## Condition 2-detectable head decay – [sheet and cloud cavitation]

If the freestream pressure is further reduce, then the inception 'bubbles' continue to grow beyond the microscopic scale, chiefly in the direction of flow. They become readily visible. At some point in this growth, their relatively stable shape is upset by the emergence of unstable bubble clouds at the downstream boundary. Thus the cavity now consists of two components (Fig. E.8):

- A stable and relatively quiet zone of 'sheet' cavitation.
- A vigorous unstable and noisy zone of 'cloud' cavitation where the bubble contents recondense.

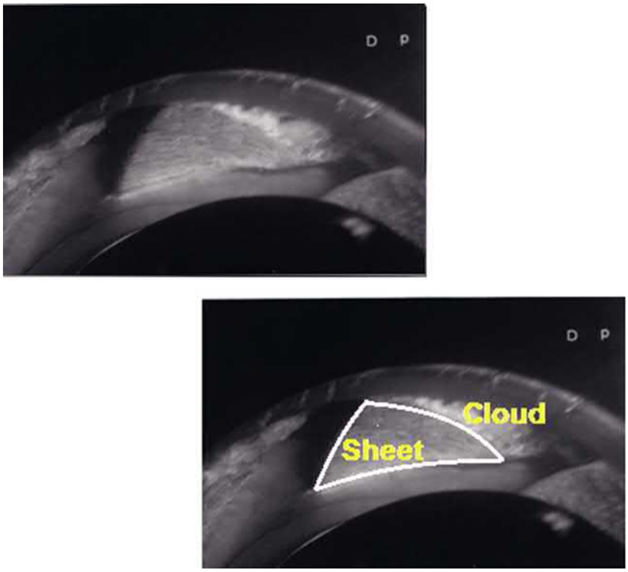



**Fig. E.8** Once the suction pressure has been reduced sufficiently to create fully developed cavitation; visual access would show two zones [see text]. Even in this advanced state, decay in the pumping output will barely be detectable [Dr Paul Cooper].

In the early stages, as suction pressure/NPSHA continues to reduce, this sheet/cloud cavity still has little if any, effect on pump performance. However as it continues to grow, a point is reached where its presence **JUST** begins to influence the liquid flowpath and consequently performance of the pump. For convenience, this is described as the 0% head decay condition, though it is very difficult to conclusively determine. Plotting the corresponding NPSH values reveals a similar but significantly different curve to the vee inception curve described earlier. We can continue and plot curves for just 1%, 3% and 9% head decay. These adopt different shapes but cluster about the same point of Relative Flow (Fig. E.9).

It is known that cavitation damage is chiefly caused by the collapse of the downstream 'cloud' bubbles. It is also known that in pumps, little if any material damage occurs in the region enclosed by sheet cavitation. Factors affecting damage rates will be discussed in the next section. But the more immediate effects of established sheet/cloud cavitation are on pump performance. Once the inception phase is exceeded, the existence of other adjacent vanes begins to [eventually] influence the overall cavitation behaviour of the pump.

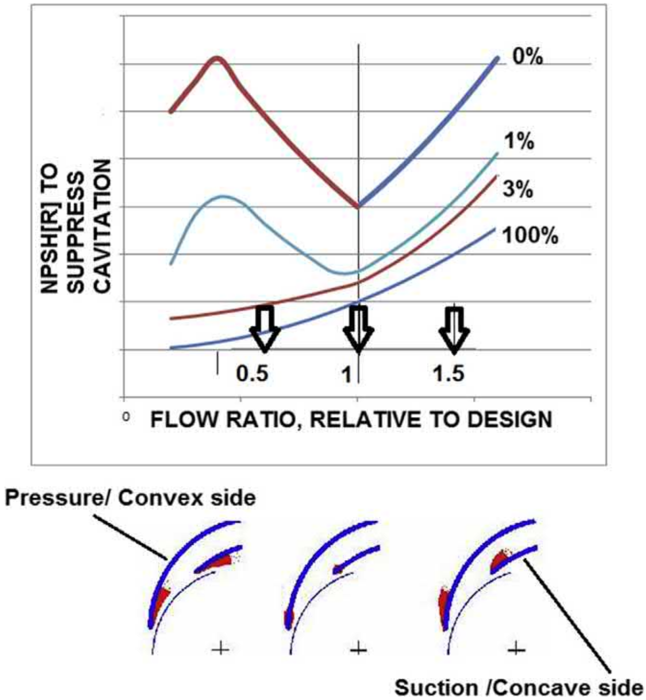



**Fig. E.9** Fig. E.7 was a plot for cavitation intensity not severe enough to induce decay in performance, even though its presence may be substantial. This figure shows plots for the intensity levels just sufficient to cause 1% decay in performance. To this is added a typical curve for 3% head decay-this corresponds to the most widely used criterion for cavitation presence. The 100% line indicates NPSH levels at which performance completely breaks down and the machine becomes ineffective. The curves form a continuous family centred on the Relative Flow of 1.0.

If for example, the impeller has perhaps only one vane [typical of some solids handling pumps] then it is easy to visualise that as the freestream pressure reduces, the length of the combined sheet & cloud will also increases, in the direction of flow. Fig. E.10 and Fig. 11 shows conceptually how pump output, at a given constant flow, would decay as NPSH [A] is progressively reduced. Starting at say point [1], the combined sheet/cloud cavity will increase in length to [2], and then [3] and so on, as suction pressure/NPSH is reduced. Its height/thickness will also increase in some way. The presence of this cavity volume will obviously disturb the way in which the impeller can do work on the liquid. Evidently, as the freestream pressure reduces, then the work done in generating output pressure head will also reduce in some way (Fig. E.11).

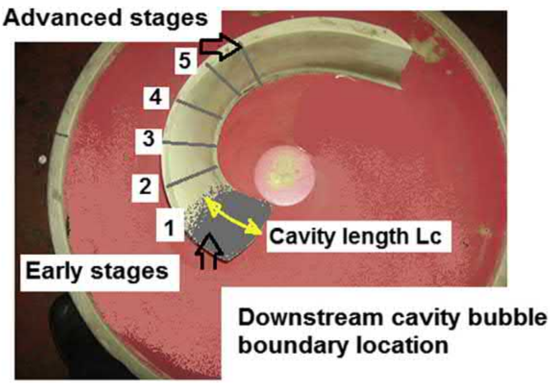



**Fig. E.10** This figure illustrates, on this single blade, that the cavity shape outline-shown in Fig. E.8, grows in the direction of flow as suction pressure/NPSH [A] is reduced.

The shape of this relationship is actually characteristic of many high Specific Speed/axial flow impellers. Such designs often have few vanes and are so widely spaced that the above picture of a solitary isolated vane applies quite well. In such cases of impellers with widely separated vanes *[low vane solidity]*, the head output of the pump seems to just steadily and progressively decay as suction pressure/NPSH [A] is reduced. There are no sudden changes of slope. Simultaneously, the cavity total length increases.

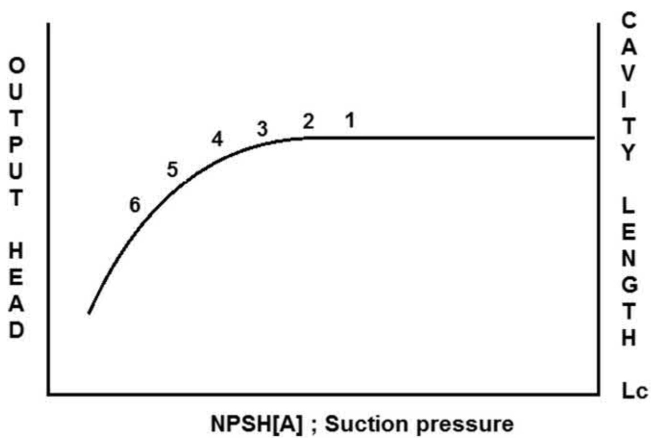



**Fig. E.11** This figure shows how the head output of the single blade impeller of Fig. E.10 will continuously and progressively decay as suction pressure/NPSH [A] is reduced.

If the impeller used in the above mental experiment now has additional vanes added, then the dynamic behaviour changes significantly. As before, the combined cavity length grows as a function of freestream pressure. But now there comes a point where the cavity eventually forms a flow blockage in the liquid passage between vanes. At this point, Point 4 in Fig. E.12 the cavity begins to 'choke' the impeller passage flow and causes a more severe disturbance to the work transfer process.

Clearly this choking effect becomes more severe as the vane number is increased. *[Because the vane-to-vane gap tends to get smaller.]* The choking point will consequently move closer to point [1]. This partly explains why impellers capable of operating under low suction pressure conditions or low NPSHA generally have fewer vane than those designed for high efficiency. The cavitation intensity is not significantly reduced, only the outward effect. All other things being equal, impellers with many vanes will have a more severe horsetail 'cut-off' than those with few vanes. They will appear to have higher cavitation presence, whereas in fact it will probably be less (Fig. E.13).

This highlights a weakness of the 3% head decay criterion. 50 years ago, it was common practice among designers to produce impellers with as few as three vanes whenever low NPSHR designs were required. By delaying the choking point to lower NPSH, this helped reduce the APPARENT effects of cavitation on performance. It helped create the illusion of apparently superior designs on the testbed. But in the long term, these designs were more susceptible to cavitation damage. Many of these designs subsequently operated on cavitation-tolerant liquids, such as hot water or hydrocarbon products. And they were installed in plant that, for the most part, would be operating at near to design capacity. So the design shortcomings were masked. It was only when the Energy crisis of 1973 forced a

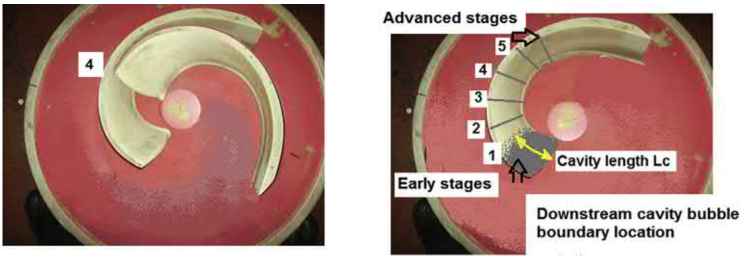



**Fig. E.12** The presence of an adjacent blade forms a finite constriction point on the flowpath. This modifies the progressive head decay profile of Fig. E.11 into two distinct regimes, as shown in Fig. E.13.

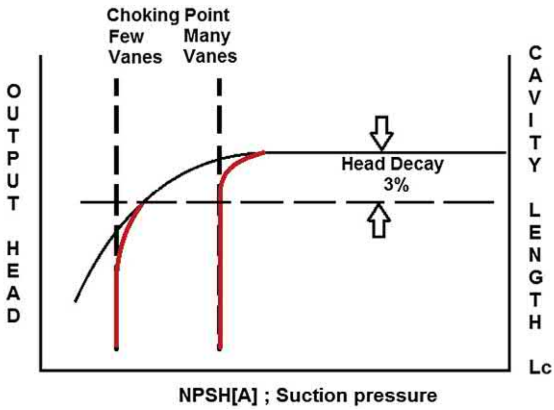



**Fig. E.13** This figure illustrates the modifying effect that an adjacent blade has on the behaviour of an isolated blade. The overall behaviour curve has two parts; one related to the individual vane profile and one to the presence of an adjacent blade. This is why re-profiling a blade inlet sometimes has a significant NPSH [R] improvement, yet in other designs it does not.

lot of these plants to operate at much reduced throughput that an unexpected consequence of these design weaknesses appeared [Ref. 7].

So far, the description of cavitation behaviour has been restricted to pump flows near to its best efficiency flow/lowest inlet loss, that is, when the flow approach-angle is close to the vane-setting angle. Fig. E.14 attempts to summarise.

- Imagine that we conduct a series of performance test where we progressively increase flow from near to zero, out to some maximum value which is in excess of the pump design flow. Initially, the suction pressure/NPSH [A] is extremely high: so high as to suppress all practical evidence of cavitation. After each performance run the NPSH [A] is reduced and another performance tests conducted, and so on.
- .The initial, non-cavitating performance curve *[NPSH [A] is very very much greater than NPSH [R]],* is taken as the reference. There is no detectable evidence of cavitation presence at any point on this curve.
- As the NPSH [A] is reduced, cavitation will first become detectable from acoustic emissions. These are at very high frequency and cannot be registered by the human ear. The general shape of the acoustic emission curve is not shown, for clarity. It follows a similar shape to that for cavitation inception, though at much higher values of NPSH [A]. The performance curve will remain unchanged.

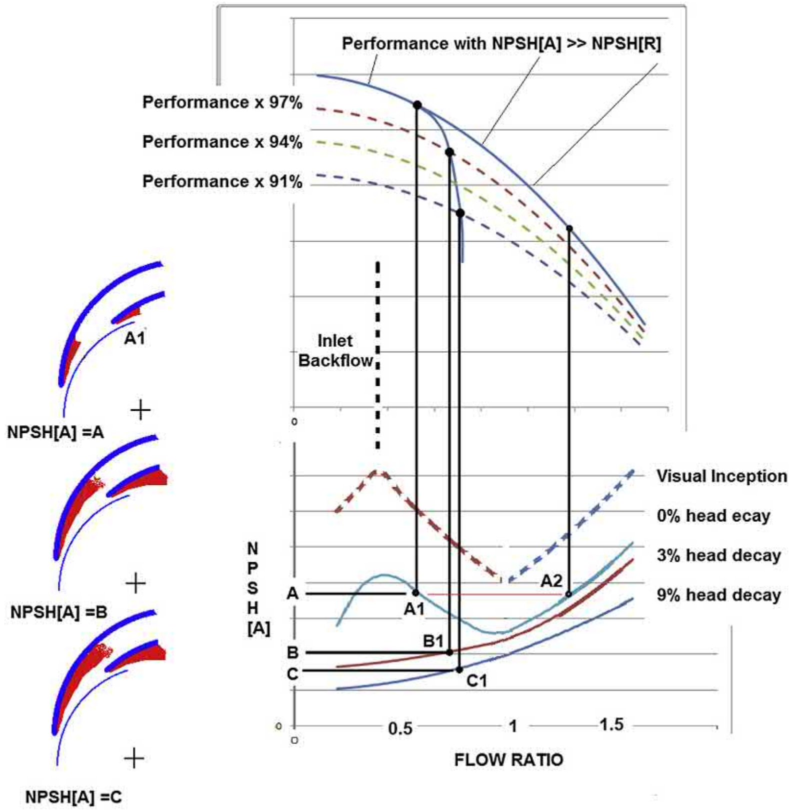



**Fig. E.14** The cavitation length and intensity are related to the relative flow ratio.

- With further reduction in NPSH [A], microscopic cavitation inception beads will become visible on or very near to the blade inlet edge. The cavitation volume is too small [relative to the impeller passage area] to have any perceptible effect on pump performance. In these conditions, the NPSH [R] and NPSH [A] curves cross twice.
  - At flows less than design, this visually incipient cavitation will be most prominent on the concave/visible face of the vane. It will most probably also exist on the less visible convex face but be smaller in size.
  - At flow greater than design, cavitation inception will be most prominent on the convex/less visible surface of the vane. It will still exist on the visible surface but will have become less intense.
  - There will be a cross over point where cavities on convex and concave surfaces are of similar length. This helps define the

optimum inlet flow [sometimes called the 'shockless entry flow']. The angle of the approaching flow will closely agree with the vane inlet setting angle at this point. In many [but not all] case this will correspond with the pump design flow at maximum impeller diameter.

- If we continue to reduce NPSH [A], to a level of a point **A**, a flow is reached during the performance tests, where the decay in performance output becomes JUST detectable. This is the 0% head decay condition. The side-graphic conceptually shows that the cavitation volume inside the impeller is starting to become significant at this point and disrupt the internal flow patterns. There is only one unique point where NPSH [R] curve intersects with NPSH [A]. Though flow can still be increased from here, for the moment, let me just concentrate on what happens up to and around this flow point. Later I will describe what happens over the fuller flow range of the pump,
- If the NPSH [] is now reduced to level **B** then any further increase in flow beyond A1 now will produce a detectable decay in performance — in this case 3% of the non-cavitating head. Again there is only one intersection point for NPSH [A] and NPSH [R]. As the side graphic shows, the cavity volume has now developed to a significant level. Its downstream boundary approximately coincides with the vane-to-vane passage space inlet and it will be causing flow choking/blockage.
  - At lesser flows than this, pump output will not appear to have been affected. When the pump output head has decayed to 97% of its non-cavitating value [Point B1], it is known that cavitation is already now well established within the machine. This condition of 97% head decay is widely used as a definition for cavitation existence. However it is occasionally and mistakenly taken as a definition of cavitation onset or commencement.
- If finally the NPSH [A] is reduced to level **C** flow can still be increased, but with only partial success. The side graphic shows that cavity volume now occupies a large proportion of the passage through flow area. Blockage is severe and the pressure head output will have fallen to [say] 94% of the non-cavitating value.
- While it may still be possible to further increase flow, the head output will fall drastically to levels of 91% or less.

As if this wasn't complicated enough, this picture becomes more complicated at flows much greater or lesser than design flow.

### *Lesser flow*

I described the combined sheet and cloud cavity length as being related to the free stream liquid pressure. But it turns out that it is also related to the flow approach angle, relative to the vane-setting angle — that is the difference between the two [Figs E.4 and E.5]. We saw how the inception cavitation performance took the rough form of an almost symmetric 'Vee' shape. But cavity growth conditions and finite head decay curves are not exactly symmetric about the knee of this incipient vee. The low flow cavity can develop substantially and out into free space until, at about 3% head decay, it invades the vane-to-vane passage space. This behaviour differs from cavity development at high flow.

### *Greater flow*

Here any significant growth in cavity volume almost immediately enters the vane-to-vane passage space and causes a blockage in impeller passage flow. Hence, the superficial symmetry of the incipient cavitation curve, about the best efficiency flow, will not be reflected in the 3% head decay curve.

Significantly, this change in 3% head decay curve slope occurs at or near to the point of minimum inlet loss. However the criteria curve for all three cavitating states form a continuous family, centred on the flow of lowest inlet loss/bep. I have deliberately left the curves incomplete in the region between zero flow and about 40% of the low inlet loss bep. That is because here an entirely different mechanism appears and significantly modifies all three curve shapes. See Appendix D.

- From a troubleshooting perspective, we can already deduce that if an impeller exhibits cavitation damage that is restricted to its concave/visible surface, then it has most probably been operating for some time at a flow less than bep/min loss flow.
- If, on the other hand, damage is largely on the convex/invisible face, then the pump has most probably operating for some time at flows in excess of bep/min loss flow.
- If damage is evident on both faces, then the pump has either;
  - Been operating at near to its bep/low loss flow or
  - Has been operating randomly above and below bep/low loss flow

## Condition 3-heavily developed cavitation — [choking and severe performance breakdown]

If the suction pressure/NPSH [A] is allowed to fall below level C, then the behaviour changes yet again. The combined cavity volume grows even more and causes flow passage choking in either case. The pump is now no longer functions as an efficient machine. This might all seem academic because few machines these days would be allowed to routinely operate with highly developed cavitation. Historically though, pumps in power stations that were responsible for lifting liquid from the condenser up to the deaerator [Condensate Extraction Pumps], did operate with very very high levels of cavitation. In effect, the pumps functioned as a flow control valve: If the condenser level [NPSH [A]] was high, then the pump flow could increase. But doing this helped the condenser level fall, cavitation intensity would increase and pump flow would reduce again. This was called ‘Free level Control’. Such pumps could only continue to function for long periods because the impeller inlet energy, as defined by the eye peripheral velocity, was very low and so the cavitation damage potential was also very low. In order to achieve such low inlet energies this class of machine was typically a very large and slow speed, single or two stage pump. But as power station sizes grew, so did the size and cost of these condensate extraction pumps. Eventually it became more economic to dig a hole in the station floor *[to increase NPSH [A]]* and install higher speed vertical multi-stage bowl pumps operating somewhat above 0% head decay line.

So far, I have only dealt with the shape and appearance of cavitation. I have attempted to relate the level of cavitation presence to its effect on output performance. It would be easy to conclude that if there was no detectable decay in performance, then there will be no detectable cavitation damage either. Sadly, there does not appear to be any simple connection between cavitation presence and the rate of damage to the impeller. All we can say is that if cavitation is entirely avoided, then there will be no damage. But in practice, total cavitation avoidance is a condition far too expensive to achieve in most applications. Moreover, in many cases [Condensate Extraction Pumps for example] some cavitation is actually permissible because the impeller will eventually fail for other reasons [corrosion, erosion] before cavitation damage becomes severe. Even then, the impeller may still be repairable.

## CAVITATION PART 3 IMPELLER DAMAGE

It is known that cavitation damage will generally be more severe if the inlet tip speed of the impeller is increased, or the hardness of the impeller material is reduced. In addition, certain liquids are more susceptible to cause damage than others, while in almost every case, damage rates decrease as temperature increase. Cavitation is also known to imprint mechanical damage, mostly in the impeller inlet region, occasionally to the adjacent casing, and on rare occasions in a zone where, at first sight, the pressure might be so high as to suppress it. I have earlier touched on the noise issue so now let us look at the issues of cavitation damage.

There is still ongoing research as to how cavitation actually causes damage. The most common explanation is that as the cavitation bubbles condense and collapses [in the cloud zone] some do not do so uniformly. At some point, their spherical shape becomes distorted, resulting in formation of micro jets at the last instant (Fig. E.15).

This random micro jet may be directed harmlessly out into the liquid, but noise is still emitted. On the other hand, it may be directed towards the surface of the impeller. In this last case, the jet causes erosion damage at a micro level. Actually, there seems to be an incubation period before damage becomes evident. Then, over an extended period, this micro damage accrues as macro damage.

While cavitation bubble collapse occurs even at the onset of visual inception, there is usually insufficient collapse energy for the micro jets to establish significant damage before the impeller fails for other reasons. But by the time sheet and cloud conditions have become well established, cavitation damage within the lifespan of the impeller is a finite risk in most cases. As yet there does not appear to be any simple exact connection between the effects on performance and the rate of impeller damage. But

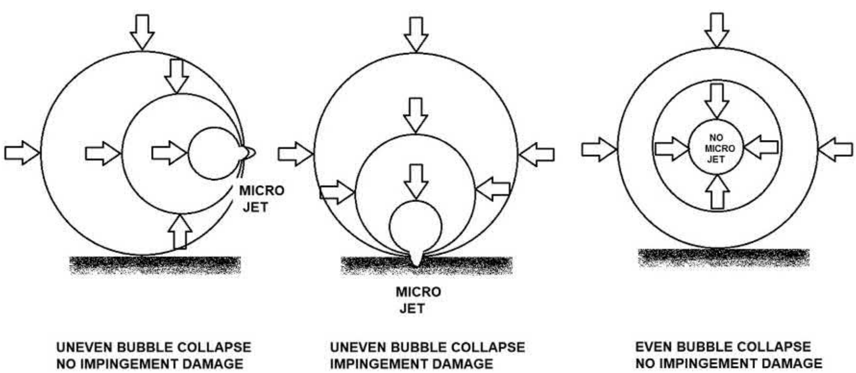



**Fig. E.15** One model of cavitation damage assumes that some vapour bubbles condense and collapse in an uneven manner. This creates a micro jet that is focused in random directions. Those that strike an adjacent surface remove small quantities of material.

some general behaviour trends exist and they are often useful in estimating likely impeller life.

What are the main factors known to influence cavitation damage rates? It is known to be influenced by;

- Liquid approach velocity – a useful analogue of this is impeller inlet tip speed.
- Liquid approach angle, or more generally the ratio of pump flowrate relative to its best efficiency flowrate.
- Liquid characteristics - some promote damage more than others; Cold Water based liquids are more prone than Hot Hydrocarbons, for example.
- Liquid temperature. Higher liquid temperatures tend to reduce cavitation damage intensity.
- Impeller material properties-chiefly hardness.

Let me briefly discuss each of these factors.

## Liquid approach velocity

Precise evaluation of this requires some basic analysis of the flow conditions ahead of the impeller. Fortunately, for the normal range of impeller setting angles, the peripheral velocity of the impeller eye [U1] is a close and suitable analogue.

$$U1 = 3.142 \times \text{Impeller eye diameter [D1, metres]} \times \text{RPM}$$

Experience says that cavitation erosion rates increase in a way related to this velocity. Estimates vary between tip speed to power of three or five. Obviously this is a key factor. If speed is, say, increased by a factor of two, then the damage rate increases by a factor of at least eight,

## Liquid approach angle

As above, knowledge of this also requires basic analysis of the flow conditions as it approaches the impeller. It is actually the DIFFERENCE between the vane-setting angle and the approach flow angle that is important. [Known as the approach flow angle of attack or angle of incidence.] At flows less than design, the liquid approach angle will be smaller than the vane setting angle. At flows greater than design, the approach angle will be greater than the setting angle. For the purpose of this explanation, a much more convenient but less accurate analogue is the ratio of the flow in question to the best efficiency flow [at full/design diameter]. The effect of flow angle difference is described earlier. Generally, the damage rate increase as this ratio increases. For the reasons explained above, incremental flow increases when operating at flows higher than design produce a more rapid increase in damage intensity than when operating at lower flows.

## Liquid characteristics

The liquid properties also influence the damage rates. The rate at which the cavity bubbles grow and collapse govern the micro jet impact energy and hence the rate of material removal. It is known, for example, that in general water-based liquids seem to impart more damage than [say] those based on hydrocarbon products.

## Liquid temperature

Higher temperature seems to reduce the impact severity. This effect [and the preceding one] can be related to the liquid specific volume, that is, the ratio of the bubble vapour volume to collapsed liquid volume. It is higher for water than [say] hydrocarbons.

This is why it is uncommon to see severe cavitation damage in impellers of hot hydrocarbon pumps [above 300 °C say] whereas it is very common to see such damage on cold-water pumps.

## Impeller materials

It is perhaps intuitive to predict that hard materials will be more resistant to [micro jet] damage than softer ones. The **M**ean **D**epth of **P**enetration **R**ate [**MDPR**] has been examined in laboratory rigs. Though this is an artificial condition, it is useful for ranking the resistance of different materials. A broad relationship of the following form appears from such tests (Fig. E.16).

Some materials blur the picture however. Some stainless steel [AISI 316] is known to be an effective solution to many cavitation erosion problems in the field. This despite its relative softness. However, it appear to work harden under the influence of the micro jets and so performs better than expected.

Often cavitation damage is terminal and can mark the end of an impellers life. As a general rule, it would be unwise to reinstall an impeller where the cavitation damage has penetrated and reduced the vane section thickness to below 75% of the original value. But sometimes, such damage and its effects can be effectively repaired.

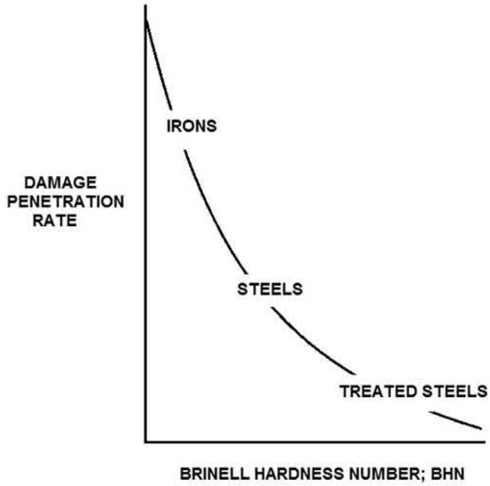



**Fig. E.16** It has been shown that for ferrous materials, the susceptibility to cavitation mica-jet pitting is somewhat related to material hardness. This general rule can have relevance to other materials. On the face of it, Austenitic stainless steel appears to perform better than its hardness would imply. That is because it work-hardens under the micro-jet impacts.

It might be helpful to display some of these influencing factors on a familiar background (Fig. E.17).

This Figure conceptually shows the previously discussed criteria curves for cavitation inception, 0% and 3% head decay. On this same, now familiar format, I show conceptual curves of min NPSH [A] in order to achieve some arbitrary impeller life for two common materials. Soft materials such as aluminium, cast iron, bronzes etc. seem to possess low resistance to cavitation damage and so its intensity must be suppressed by relatively high NPSH [A]. On the other hand, hard materials, such as chrome steel and duplex stainless steel seems to offer more resistance to cavitation damage so less suppression and higher cavitation intensity can be allowed. This translates to a lower NPSH [A] for the same life as the softer materials.

The criteria curves for various degrees of cavitation form a consistent grouping. This ranges from the horse-shoe shape of the inception curves to the almost square law relationship of severe cavitation that is 90–100% head decay. To a first approximation, the corresponding curves for cavitation damage are consistent with this grouping.

Continuous operation of the pump with an NPSH [A] below the relevant line assures that the impeller will not achieve this arbitrary life.

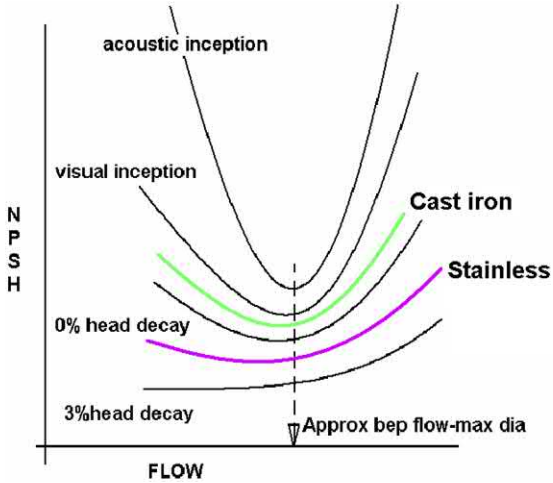



**Fig. E.17** This illustrates the hardness effect shown in Fig. E.16. Under any given NPSH [A] conditions, Cast Iron Impeller will have the same cavity length and damage potential as a Stainless steel example. But to achieve the same life, the Cast Iron impeller will need more site NPSH [A].

Continuously operating the pump with an NPSH [A] greater than the relevant line will assure an impeller cavitation resistance life in excess of some arbitrary level. I say arbitrary because, as yet, there is no internationally recognized definition for accepted impeller cavitation life.

However, a *de facto* standard is emerging, that of 40,000 h continuous operation. This criterion was tabled by Dr E Grist [Ref. 30], when he was responsible for much of the policy behind the design and specification of pumping equipment in the British CEGB [Central Electricity Generating Board]. The basic idea was that the impeller of large powerful boiler feed pumps should remain serviceable between station shutdowns of about 5 years [40,000 h]. At the end of that period, the impeller, though damaged by cavitation, should still be fit for weld repair. The criterion was intended for application to pumps handling water at high temperature.

Later, Vlaming [Ref. 31] adopted this criterion in a study of large powerful brine re-injection pumps in the Saudi Arabian oilfields. Based on his semi-empirical analysis, he evolved a form of acceptance criteria that attempted to factor in all of the earlier mentioned variables. His work differed from that of Grist in that the subject of his analysis were chiefly pumps constructed of stainless steels and handling cool saline water. This notwithstanding, his criteria was later extended to a wider range of liquids and materials of construction. A common denominator in both works was that many of the subject pumps had high impeller inlet tip speeds and represented the state of the art for high inlet energy pumps at that time.

From a pragmatic standpoint, most pump users and many pump designers are less interested in the detailed physics of pump cavitation and are more interested in avoiding the consequential damage!

While there does not seem any simple direct relationship between performance head decay and cavitation damage rates, it can form a useful basis. Applying some 'experience' safety margin to any particular head decay criterion curve turns out to be a largely effective approach, despite their being only a tenuous physical link between the two.

If the subject pump will normally operate over a narrow flow band, then it will require less margin than a pump expected to operate over its maximum operating range – see Fig. E.18.

## CAVITATION NOISE

At various points in the foregoing, cavitation noise has been mentioned, so it might be useful to make a summary here. Cavitation noise does not automatically mean that damage is taking place. The cavity cloud may be collapsing in the inter vane space or in the core of vortices, not adjacent to pump component surfaces.

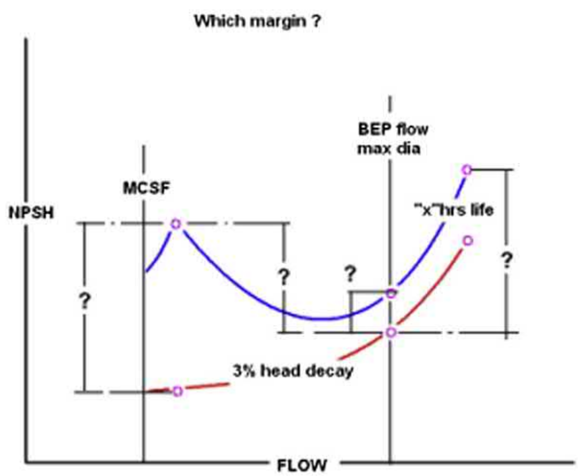



**Fig. E.18** Applying a small NPSH margin will be effective if and only if the pump will operate at or near to the bep [Max impeller Dia]. If it is expected to routinely operate down to MCSF, or run-out flow, then a much larger margin will be required.

It was mentioned that cavitation onset can first be detected by very high frequency noise emission. The suction pressure/NPSH [A] at which this acoustic inception occurs is very high relative to any performance decay criteria, or even visual inception.

As with visual inception, its intensity is flow-related. It also exhibits somewhat of a 'vee' shape with its minimum at or about the flow of minimum inlet loss [zero incidence flow]. As NPSH [A] is reduced, the inception noise intensity increases.

Once the cavitation presence has developed to the point of causing detectable performance decay, this indicates the co-existence of significant sheet/cloud cavitation. The collapsing clouds gives cavitation its traditional 'pumping gravel' noise. Above [say] 20–40% of design flow, cavitation noise is essentially steady in frequency but its volume will be flow-related. . Below this point the noise can take on a rhythmic 'chuffing' or 'surging' characteristic, which is a sure sign that inlet backflow has set in (Fig. E.19).

Centrifugal pumps emit most overall noise in a frequency range of up to 16,000 Hz depending on size and speed. Cavitation noise is much higher frequency than this. It can easily be differentiated in low speed pumps, but can get lost in the overall noise of high speed pumps.

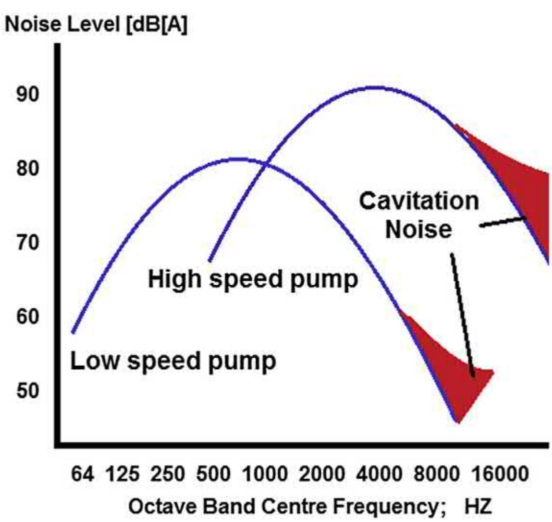



**Fig. E.19** Cavitation noise is limited to high frequencies. In high speed pumps it can be swamped by the general machine noise.

## CAVITATION TESTING AND DEFINITION

By now the significance of a pump's NPSH [R] will be obvious. Within the pump industry, there are a number of ways in which these curves are derived. This is not the place to describe these tests in detail but it is worth outlining their basic concepts. I should add that in the case of multistage pumps we are only interested in determining the behaviour of the first stage. That is because this stage should generate sufficient head – even when partially decayed, to suppress critical cavitation levels in the second and subsequent stages.

The two most popular test techniques involve establishing a particular form of performance curve, from which the traditional 3% departure can be determined.

- In the first case, a series of performance tests are conducted in which flow is varied, under a range of constant inlet conditions NPSH [A].
- In the second, more discrete performance tests are conducted in which flow is held constant while suction conditions are varied.

### Constant NPSHA test

In this approach, a series of normal performance tests are conducted. In between each test, the static suction pressure is reduced-which obviously reduces the NPSHA. Ideally, deaerated water is the test medium. *[This is because unless otherwise deaerated, air dissolved in the water will come out of solution as the suction pressure is reduced. This can create false cavitation effects.]* In difficult cases, the water temperature may also be raised so as to achieve really low levels of NPSH [A].

Initially there will be a large surplus of NPSHA over NPSHR. There may be little if any internal cavitation due to the large margin. In this case, the performance curves extend out the highest flows (Fig. E.20).

As the test continues, the suction pressure-NPSHA is further and eventually a point is reached where combined cavitation sheet/cloud commences BEFORE the highest flow is reached. At this point, the performance curve will drop - imperceptibly at first. But as any increase in test flow is attempted, the curve drops more and more. This 'cut-off' behaviour is central to the test definition.

In subsequent tests, this pattern is repeated and, due to cavitation, the highest attained flow reduces at each step (Fig. E.21).

We can now draw a line parallel to the initial performance curve, but lower in head by 3%. Where this line crosses each 'performance cut-off'

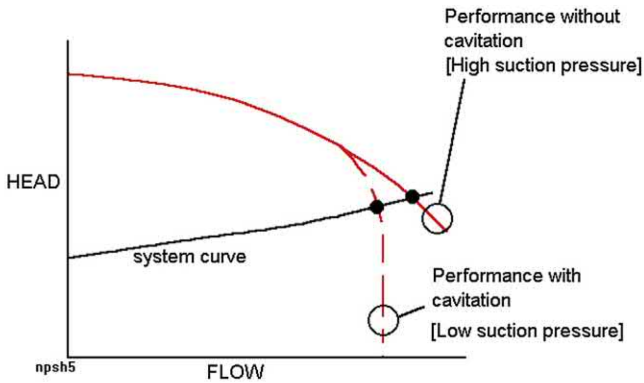



**Fig. E.20** Reducing NPSH [A] will eventually cause the pump maximum output to decay.

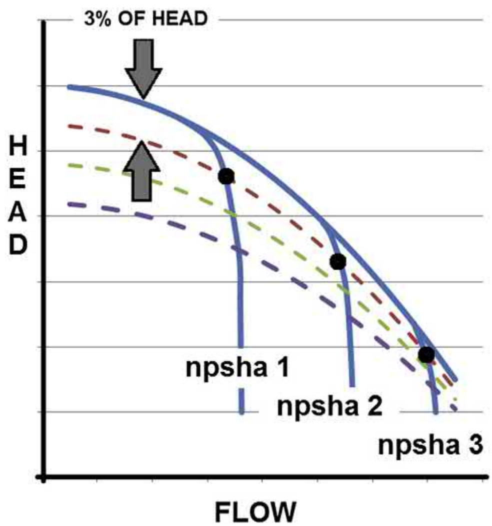



**Fig. E.21** One method of NPSH [R] evaluation runs a series of HQ tests at different but constant levels. The point of departure from the reference curve, taken at relatively high NPSH [A] indicates the presence of significant cavitation within the impeller. A departure of 3% is widely invoked.

establishes the NPSHR at which cavitation is agreed to be established–at each flow.

For many years, this simple test was acceptable. Its chief weakness was in how the suction pressure was reduced in order to reduce NPSHA. Mostly this was accomplished by valve throttling on the suction side. Alternatively, the static suction lift was varied. But both approaches were susceptible to some error, since the liquid might not be sufficiently deaerated. But as margins became tighter and tighter, a more precise and consistent approach took over.

## Constant flow plus NPSH test

In this next approach, the flow is maintained as a constant during each test series, and the NPSHA was varied. Usually this involves a closed loop and a vacuum pump controlling the NPSHA. This is called the vacuum suppression method. It has the added bonus of helping to maintaining deaeration of the test liquid.

Fig. E.22 summarises the results of such a test. Of course, the results from the two approaches are related, with three variables- Head, Flow NPSH. In fact, they represent two different views of a three dimensional relationship between these three parameters [Fig. E.23]. The location of the three cavitation stages are also shown, and a number of important features are depicted

- There is often a large 'Plateau' where both Head/Flow performance and impeller life is unaffected by the NPSHA Similarly there is a 'Cliff Edge' beyond which the performance of the machine has almost completely collapsed. Irrespective of flow, the heavily developed cavitation stage represents the cliff-edge of the performance surface. This line usually falls continuously as flow is reduced.
- The 3% head decay line, which typically represents a combination of sheet/cloud cavitation, is near to the performance 'cliff edge'. It is

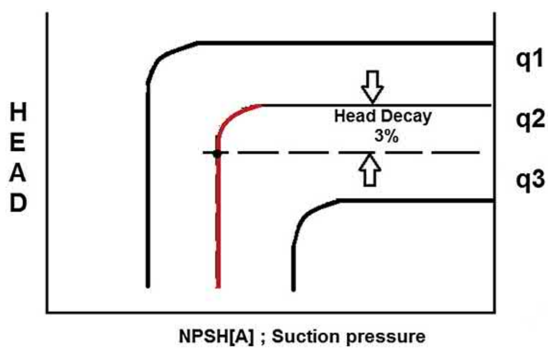



**Fig. E.22** Another method is based on holding the pump flow at a series of constant values [q1, q2, q3] while gradually reducing NPSH [A]. The point where head decays by some percentage defines significant cavitation onset; 3% decay is a widely accepted value. At this level, the cavitation extent will be similar to that shown in Fig. E.8.

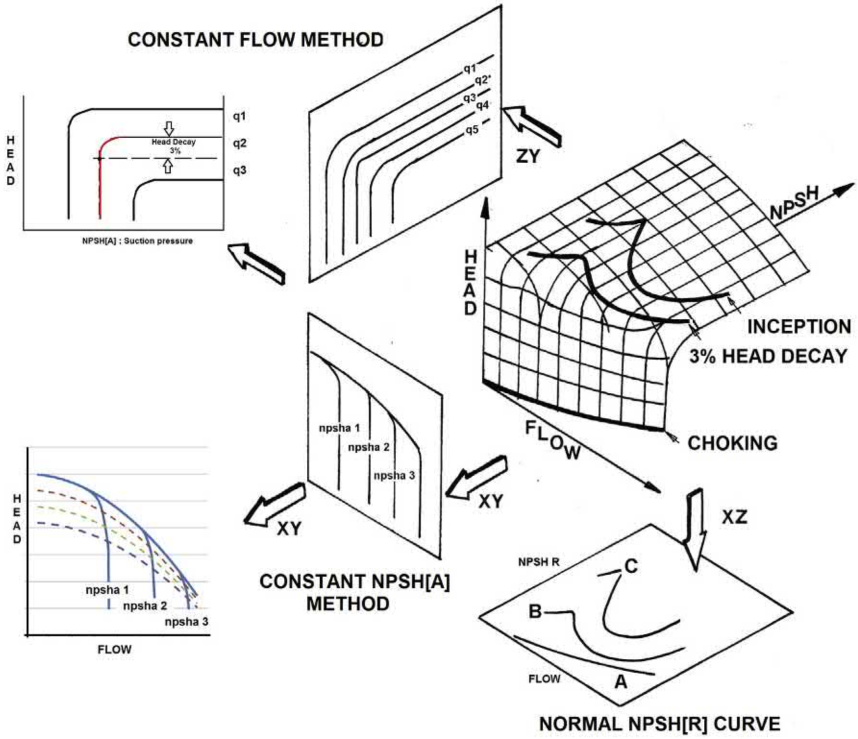



**Fig. E.23** The two testing methods just give different views of the same 3D surface, connecting NPSH, Flow and Head. Either method can lead to the conventional NPSH [R] curves that a manufacturer will published. Referring to Fig. E.13, behaviour on the flat horizontal surface is chiefly related to the impeller inlet edge shape. The vertical 'cliff edge' indicates the points where cavitation chokes the impeller and its performance collapses.

closest to the edge-at or near to the bep of the pump — [full diameter impeller]. It also falls as flow is reduced, however it may rise again to an inflexion point before falling once again. This point is important.

- The cavitation inception line is, in general, much higher on the plateau than the 3% line. But it will generally be closest to the cliff edge at the same or similar flow as the 3% line.

## ADDITIONAL PHENOMENA

### Effect of impeller diameter reduction

A minor weakness in both approaches relates to the choice of a fixed 3% head decay. It is known for example that the NPSHR curve for a pump is almost entirely dictated by the geometry of the impeller inlet annulus and the approach duct shape. So if an impeller has its diameter trimmed, then one might not expect this to have much effect on the NPSHR curve, but it does. If one assumes that the NPSHR curve is completely independent of what is happening at the rim of the impeller, that is its shape is independent of the generated head [and only related to pump flow], then one can see that 3% head decay when the impeller is untrimmed and putting out a lot of head is more than 3% head drop of a trimmed impeller putting out less head. . This is reiterated in Appendix C. This argument can be carried further when multistage pumps are considered. It used to be the practice that such pumps be evaluated in the same way, that is, 3% decay of the TOTAL head. Of course only the first stage is normally subject to cavitation. Most modern NPSH [R] criteria now require the 3% decay to be independently based on the output head of only the first stage (Fig. E.24).

## Cavitating surge

The phenomenon of cavitating surge is mentioned in Appendix D. In the upper left hand corner of the three dimensional surface [Fig. E.23] exists a wedge shaped volume called the surge zone. Operation in this zone will

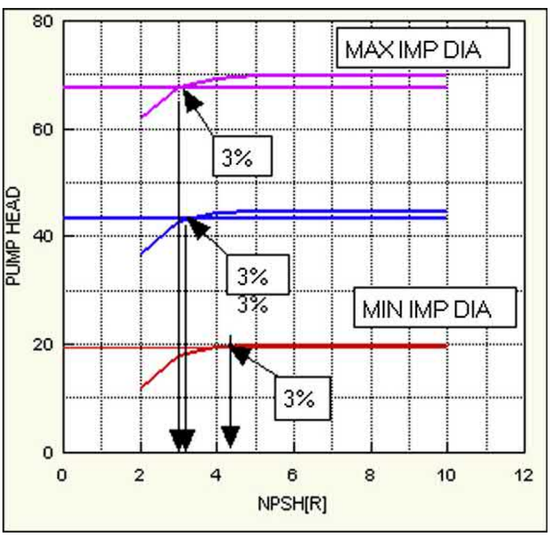



**Fig. E.24** The 3% head decay criterion will yield a lower NPSH [R] for high head impellers than low-all other things being equal. By the same logic, multistage impellers will show even lower numbers if the total head is used. For this reason, all modern specifications require only the first stage head to be measured during a cavitation test.

cause the output pressure to regularly pulse at frequencies of between 0.5 and 3 Hz – depending upon the inlet tip velocity of the impeller. It may also be accompanied by a rhythmic 'chuffing' sound (Fig. E.25).

A fundamental precondition of entry to this zone the existence of inlet backflow at the impeller eye [Appendix D].

A further precondition is that cavitation must also be present. Without these two components, surge will not occur. Since inlet backflow is a precondition, surge is thus normally confined to the low flow part of the pump performance map (Fig. E.26).

There are a number of ways in which the pump can inadvertently be brought into this surge zone. For example, reducing the flow of a pump where the site NPSH does not simultaneously increase very much [condition [A]] will normally induce cavitating surge once the pump enters the backflow regime. Should the site NPSH curve rise more steeply as flow is reduced – due to say a high frictional component in the system makeup [curve B] then the surge zone may be avoided if flow is reduced.

Similarly, reducing the NPSH [A] to a pump that is already operating inside the backflow regime will ultimately produce surge conditions (Fig. E.27).

Unless the inlet tip speed of the impeller exceeds 20–25 m/s, surge is unlikely to be obtrusive, even if the surge sounds are audible. However the

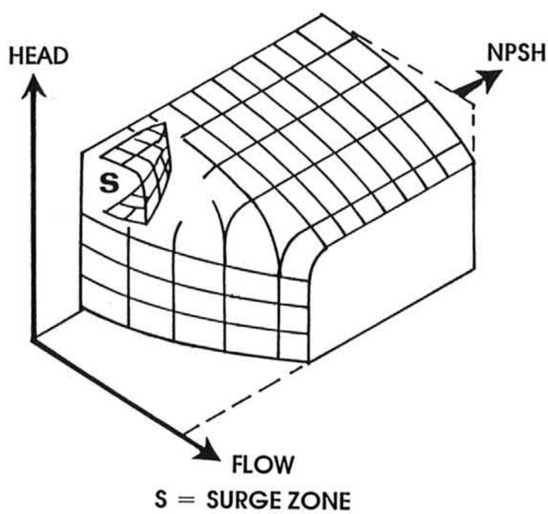



**Fig. E.25** The performance surface described in Fig. E.22 contains a zone where cavitating surge is possible.

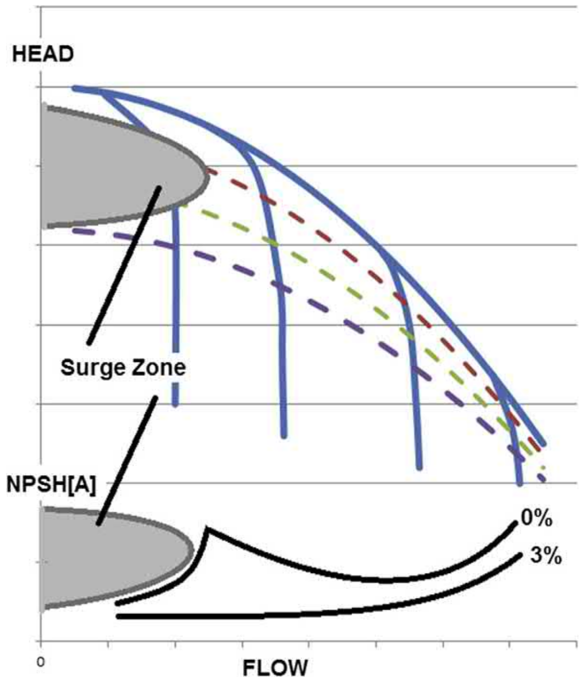



**Fig. E.26** Surge zone relative to NPSH [R] curves. Surge could not normally occur outside of the inlet backflow regime.

effects become more serious, as a function of inlet tip speed. At levels in excess of 35–40 m/surge can be powerful enough to damage pipework and pipe supports

## HOW CAN CAVITATION AND SURGE BE AVOIDED?

There are two routes to avoiding cavitation troubles. Both conditions must be respected.

- The most obvious is to provide adequate NPSH margins
- Less obvious, but just as important is the hydraulic 'fit' to the application

### NPSH margins

As already mentioned, operation right on the 3% decay line will result in significant cavitation. Already there are significant amounts of cavitation. To avoid cavitation damage, NPSHA must exceed this NPSHR line. The amount depends upon the fluid pumped, impeller material, fluid temperature and relative flow rate.

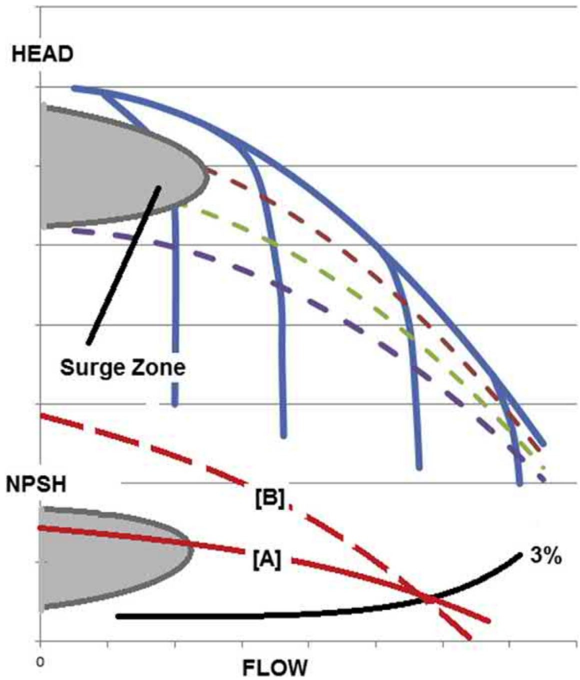



**Fig. E.27** NPSH [A] curves that are steep [B] are not normally preferred, but they are more likely to avoid the low flow surge zone than those with traditional flat curves [A].

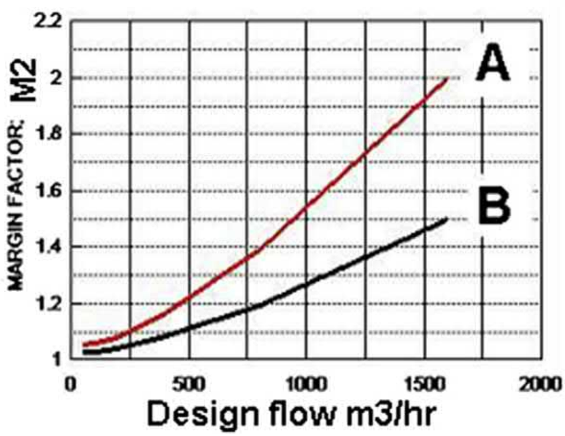



**Fig. E.28** Typical NPSH [A] margin factors [M2] for end suction pumps of 2950 rpm. This chart attempts to exhibit the inlet tip speed effect.

Define this margin as a ratio M1, where:

E1-M1 = NPSH [A]/NPSH [R] = M2 × M3

M1 gives some sense of the margin required to achieve a useful life. Fig. E.28 attempts to illustrate how a component of this margin factor, M2, relates to design flow,[10] materials, and liquids

- [A] – cold water pump with bronze impeller and cast iron casing.
- [B] – hot hydrocarbon pump with 11/13% chrome impeller and carbon steel casing.

In the above example M2 refers to operation at the pumps design flow [bep] at maximum impeller diameter.

For flow other than design, multiply M2 by a further factor M3 (Fig. E.29).

M1 = M2 × M3

Finally, the minimum site NPSHA can be calculated from;

NPSHA > NPSH × M1

## Hydraulic fit – oversize pump

Some Cavitation problems are deeply rooted in the pump selection – in its hydraulic 'fit' to the application. Palgrave's 4th rule of pumping yields

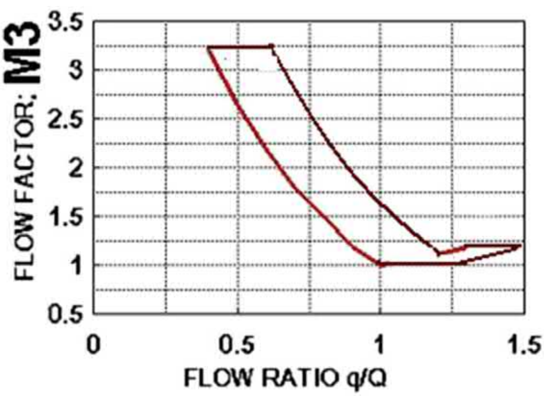



**Fig. E.29** Typical flow related component [M3].

[10] Which relates to inlet tip speed.

pump selections that are too large – by varying degrees. The consequence of this is a heightened risk of 'low flow problems', and inlet backflow was mentioned. Another risk is cavitation damage. The 3% decay line may suggest ample margin relative to the site NPSHA but when M1 [M2 & M3] are factored in the NPSH for acceptable life may exceed that available. This is a very common outcome of oversize pumps.

At low flows, the two curves may differ substantially. Simply providing a standard fixed margin,[11] irrespective of where the pump is running on its curve may end up in trouble.

In pumps where the inlet tip speed [$U_1$] < 20 m/s, there may be cavitation noise, but that does not necessarily mean significant damage is taking place. [Much of the cavitation is collapsing in free space between the impeller blades.] An inducer would also help **if, and only if** it was of a design flow smaller than the pump design flow (Fig. E.30).

In pumps where inlet tip speed >25 m/s cavitation damage is only part of the problem, however. Breakdown of the inlet flow patterns, in concert with cavitation, will generate significant low frequency flow surges *see* Appendix D. It also superimposes pressure pulsation onto the pump discharge pressure. These surges can be very severe and are responsible for

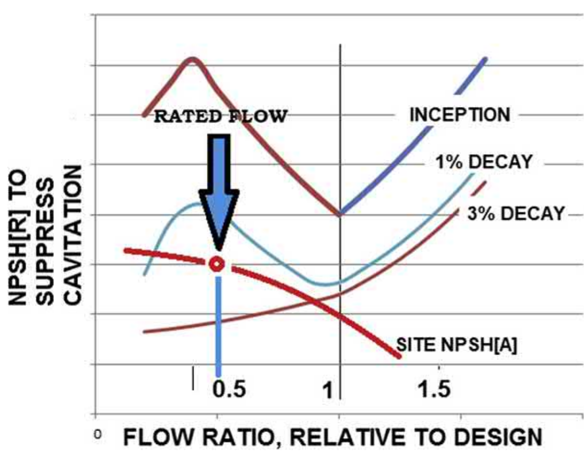



**Fig. E.30** Oversize pumps may result in unexpected cavitation damage and failure to achieve operating life. In these cases, impeller damage will normally occur on the concave, more easily visible face of the vane. In extreme case, damage will also occur on and in the inlet duct.

[11] That is not at all flow related.

significant reductions in seal and bearing life. They can also generate 'organ pipe' resonance in the system pipe work. So simply installing a harder material is not the complete answer.

The only real solution is to obtain an alternative impeller with a lower design inlet flow. This squeezes the surge zone below the operating min flow and out of harm's way. Note that impeller damage in this condition is usually most apparent on the visible [concave] sides of the impeller vane inlet. However, if there is also random cavitation damage on the impeller hub and shroud, perhaps even on the casing, then this is a sure sign of operation at too low a flow.

In the field the only practical alternative is to fit a so-called choke ring. This emulates the eye of a lower capacity impeller design and often proves successful.

## Hydraulic fit – correctly sized pump

If the pump is operating near to its design point[12] then there should be little excuse for ending up with no real margin on site. The same is true for flows above design, say 110%.

In such cases, where the margin *does* turn out to be insufficient, and the installation parameters cannot easily be changed, the only quick solution is to change the impeller material to something that is harder. This simply makes the impeller damage more slowly, but is not an elegant solution. [Though one user reports its cheaper to operate in the cavitation mode and replace impellers every year, rather than elevate the whole plant 2 more metres, in order to get the real margin needed.] Note that impeller damage in this condition is usually apparent on both sides of the impeller vane inlet.

Certain end suction pumps may be available with a helical inducer of course and this is a bolt on solution. Some prejudice exists over inducers, but this is largely based on experience with earlier 'first generation' constant lead inducer designs. Modern 'second generation' variable lead inducers are more flexible and much less prone to bad habits as the earlier types.

Inducers are really just integral booster pumps. In extreme cases, where an inducer is impractical, a separate booster pump in the suction line may solve the problem satisfactorily.

An old trick that was used to diminish the effects of cavitation was to inject air into the suction pipe in small quantities. Obviously this is safest if

[12] Actually operating there, not just BELIEVED to be operating there.

the pump is not on a suction lift [risk of de-priming]. It appears to work on two counts:

Firstly, it dampens the high frequency noise emitted by cavitation by reducing the speed of sound in the liquid. The objectionable noise appears to be reduced.

Secondly, the air presence appears to slow down the cavitation erosion mechanism in some cases.

## Hydraulic fit – undersized pump

Selecting a pump that is too small – [in order to save first costs] can again lead to unexpected cavitation problems. As before, the NPSHA may slightly exceed NPSHR, but similarly the damage line for acceptable life [Incorporating M2, M3] may lie above the site NPSH. Unexpected damage will result. Note that impeller damage in this condition is usually apparent only on the invisible [convex] surface of the impeller vane inlet.

Sometimes a pump is obliged to operate beyond [say] 110% of its design flow, with an NPSH margin that is too small. This is the most damaging situation [because fluid velocities are the highest] and cavitation damage relates to [at least] 4th power of velocity.

Because the NPSH curve rises steeply above 110%, then NPSHA has to be raised very significantly in order to get sufficient margin and any real increase in run-out capability (Fig. E.31).

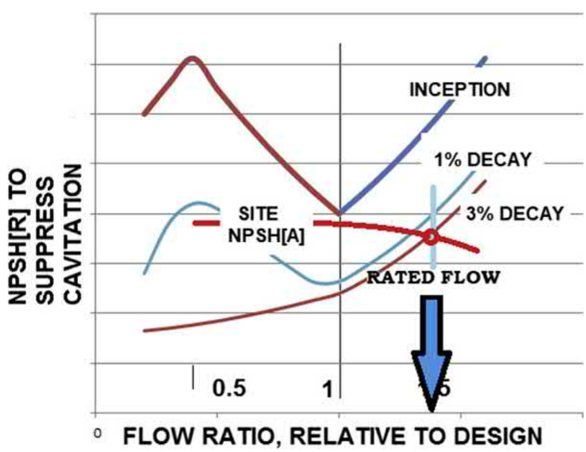



**Fig. E.31** Undersize pumps may also result in unexpected cavitation damage and failure to achieve operating life. In these cases, impeller damage will normally occur on the convex, less visible face of the vane.

Solutions to this problem are limited:

- One approach is to fit an inducer if available.
- Another is to fit a higher capacity impeller design [Manufacturers often have alternatives available]. This will probably operate on a less efficient part of its curve, but at least it will pump!

In the field, impellers can be modified to combat this problem, but it is difficult to accomplish a long term solution to the problem in this way. The modification entails cutting away a portion of the impeller inlet edge thus increasing the passage inlet area. This approach is described in Appendix K.

# APPENDIX F

# Simple demonstration of Lomakin effect

The *Lomakin* effect describes the stabilising forces caused by liquid passing across close fitting clearances such as wear rings and shaft bushes. What might seem at first sight to be an obscure and unimportant phenomenon, can in fact be hugely important in influencing the mechanical behaviour of the rotating masses. But first, a very simple experiment can be conducted that conveys the principle (Fig. F.1).

In a sink or hand-basin which has a chain mounted plug, fill the sink as much as is safe.

Using the chain in a vertical position, lift the plug off its seat by a small amount – say 1/16″.

Water will flow out through the annular gap created.

While liquid is flowing, move the chain from side to side.

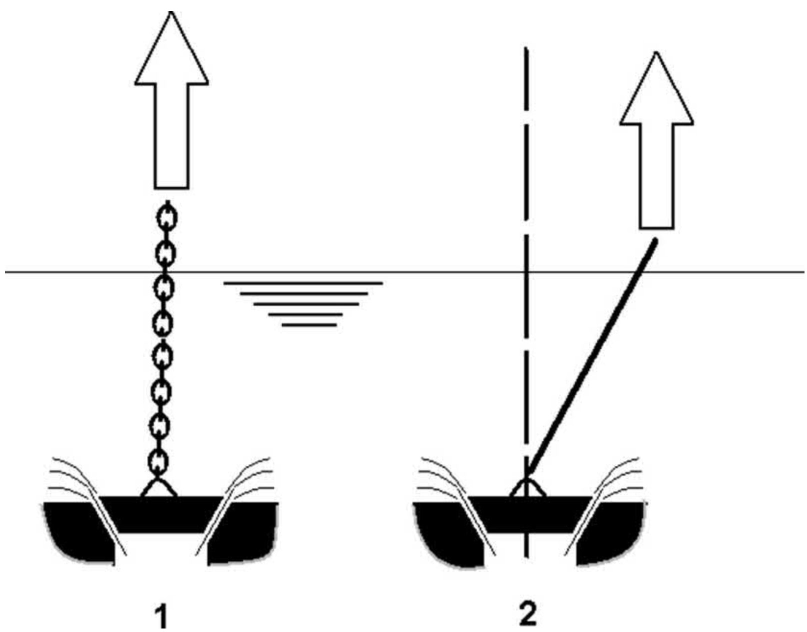



**Fig. F.1** [1] No sideways force on plug; [2] Despite side force on plug, LOMAKIN forces help maintain plug central in drain.

Despite the sideways displacement of the chain, the plug remains central in the seat. Often a substantial offset can be made to the chain before its equilibrium is destroyed.

The water flowing through the annular gap creates a self-centering force. Even with the very low liquid 'head' achieved in a hand basin, these forces are strong enough to overcome the gravity forces wanting to straighten the chain. As the differential pressure/head acting across impeller increases, the so do the restoring forces. This speed—related contribution can begin to influence the shaft "critical speeds".

For example, the shaft "dry" critical speed in air can be related to the static shaft deflection or "sag" resulting from the weights of the various rotating parts. Simplistically, the LOMAKIN restoring forces reduce the shaft deflection and, by implication, raise the apparent "wet" critical speed value. In some pump designs, even the dry critical speed is above the running speed. This is often the case with most [but not all] OH2 type pumps, many BB2 types, but few BB3 types [Refer to Fig. A.1]. But the Lomakin stiffening effect means that for many modern designs, the "wet" critical speed is in excess of the running speed. So the Lomakin contribution is generally more important to BB3 machines than BB2 (Fig. F.2)

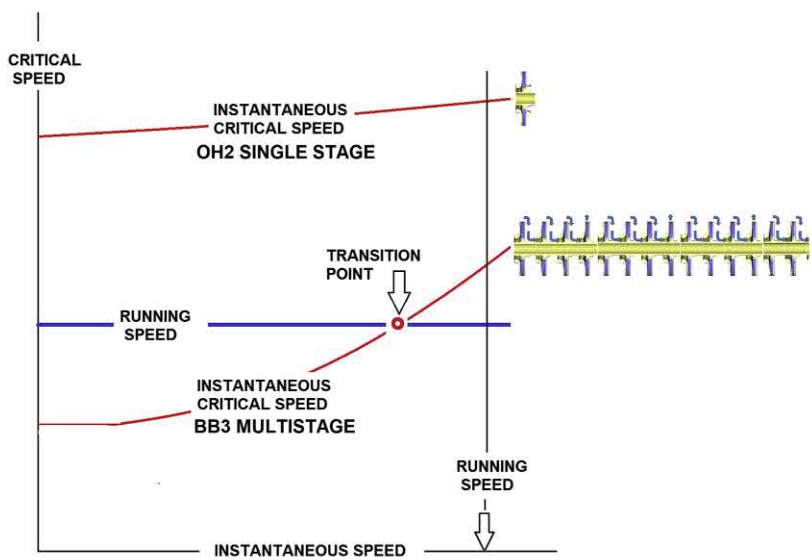



**Fig. F.2** Speed-related Lomakin stiffness effect can influence shaft behaviour, particularly in pumps with long slender shafts. Pumps with stiff shafts are less influenced.

With BB3 machines in particular, the speed-related transition from dry to wet critical speed occurs during the run-up to full speed. Here the machine may briefly display mechanical discomfort before settling down to a steady super-critical behaviour. Obviously Variable Speed machines should never dwell at this transition speed.

## APPENDIX G

# Reverse pumps as turbines [See also Chapter 14]

Externally, the differences between a centrifugal pump and turbine are not great. A non-engineer may not even notice the difference. Technically, the obvious difference is that pumps absorb power and do work on the liquid. Turbines do the reverse; they absorb power from the liquid and deliver it as useful work. In principle, a turbine *is* just a pump in reverse so it is reasonable to query if pumps can be used as makeshift turbines.

Conversely, can turbines operate as pumps? In rare cases dual-purpose pump-turbines are designed but they are beyond the scope of this discussion. It is in the detail that pumps and turbines differ significantly. The vanes on a pump impeller are usually few – somewhere between 5 and 8 in the main. The passage areas between them will change rather gently. A turbine will have many more vanes and the passage areas between them will change more severely. The reason for this crucial difference is simple.

- In a pump impeller, the passage flow is generally decelerating and diffusing. It is difficult to decelerate efficiently and great care is needed. That is why the process is performed very gently, using long lightly loaded passages.
- In a turbine runner, the flow is accelerating which is inherently a more efficient and, importantly, more stable process. This allows more liberties to be taken in terms of passage loading.

These facts help us answer the initial question. A pump impeller can operate effectively as a turbine runner because the internal flow patterns become more favourable. Instead of having to carefully diffuse the flow, it now has to moderately accelerate the liquid and, at least in turbine terms, tolerate only a mild level of passage loading.

On the other hand, using a turbine runner as a pump impeller is not so favourable. Generally, the passage areas increase too rapidly to allow efficiently controlled diffusion. The passages are overloaded.

Briefly, pumps can be run in reverse as passable turbines. Experience says that their efficiency may at worst be only 4 or 5% lower than a purpose made turbine. On the other hand, turbines operated as pumps show poor efficiency. Furthermore, the characteristic curves are not very useful.

Just for clarification; in a pump, the flow is *outwards* through the impeller. Using the same machine as a turbine, the flow is reversed and now passes *inward* across the runner.

The direction of rotation is also reversed. [Just to confuse matters further, a pump can operate as an *outward* flow turbine, though not as effectively. This is called the 'Windmill' mode].

## DELIBERATE OPERATION AS A TURBINE

Deliberate use is generally restricted to situations where the machine is being used chiefly for its quiet head breakdown properties. Any energy recovered in the process is a welcome bonus [If energy recoveries were the only concern, a true turbine might be used to extract perhaps 5% more power and be physically smaller.] Typical head breakdown applications would be where pressure has been bestowed upon liquid due to [say] a chemical reaction. This pressure has to be dissipated to a lower pressure before further processing can take place. Examples would be:

- Hydrocracking of hydrocarbons in refineries
- Rich solutions in Gas sweetening plants [Gas Scrubbers]
- In the Purification of water by the Reverse Osmosis process the high pressure embodied in the membrane reject needs to be dissipated before discharge to waste
- On the downhill leg of pipelines, the static head can be dissipated to protect a downstream pumping station

In every case the reverse running pump is being used in situations where a throttle orifice, or even pressure control valve may have been previously used.

## ACCIDENTAL OPERATION AS A TURBINE

Pumps can accidentally get into the turbine mode if not installed correctly. If a stationary pump can be exposed to a reverse differential pressure, and there is no impediment to reverse flow, then the pump will turbine. Examples would be:

- Pumping against a large static head without a non-return valve, or with a stuck open valve. When the pump stops, the discharge line empties back through the pump. This may happen in mines and quarries.
- In a parallel pump installation, the non-return valve may stay stuck open when one pump is shut down.

In either case, reverse rotation may be a serious problem. When turbines are deliberately installed as part of an energy recovery system, the power is fed back into the motor in order to reduce its power consumption. An electric motor forms an excellent governor and so speed is controlled. In the two examples just described, the electric motor would be de-energised so unable to exert any speed control. In the turbine mode, the shaft speed may significantly exceed the design speed of the motor. This may cause problems.

Another possible risk is present when a pump enters the outward flow turbine mode. This could occur if a booster pump continues to run after the main pump has shut down. In this situation, the main pump may begin to 'windmill' as mentioned earlier. The wind milling speed will be quite low since this is not an effective turbine mode. If the main pump has ring-oiled bearings, then the speed may be insufficient to allow oil pickup. Bearing failure will follow.

In practice it would be very rare for a turbine to be used as a pump, either deliberately or accidentally. On the other hand, pumps can and frequently do function as turbines, both deliberately and accidentally.

When a pump operates as a turbine, its best efficiency flow and head increases by roughly 40% — depending on size.

## APPENDIX H

# Thermal temperature rise

When a pump runs at low flows, only part of the motor input appears as useful work in the form of flow and head. The remainder goes into heating the water in the casing by the churning of the impellers. This is why pump efficiency is low at these flows.

There are two types of temperature rise problems and two relevant equations.

### CLOSED VALVE

When a pump is run at closed valve, the water is continuously churned within the casing getting hotter and hotter. Since the heat cannot escape (except by radiation and this is negligible unless it is an immersed vertical type) the water temperature continues to rise until it boils – this causes vapour locking and probable seizure.

$$\begin{aligned}\text{Temperature increase per minute} &= \text{Delta T/min}\\ &= 42.4 \times \text{BHP } [0] \;/\; W_L \times S_H\end{aligned}$$

*Where:* 42.4 = Constant [BHP to BTU/Min]; BHP [0] = horsepower at zero flow; $W_L$ = Weight of liquid in pump lbs; SH = Specific heat of liquid [1.0 for water]

### LOW FLOW

If the valve on the pump discussed above were opened slightly, the small – flow would enable some of the heat to be carried away and the temperature of the pump casing would drop to some equilibrium value. This stable temperature is given by:

### PUMPS WITHOUT A BALANCE DEVICE LEAK OFF

$$\text{Delta T} = H/778 \times [[1\,/\,E] - 1]/[1\,/\,SH] \qquad \text{H.1}$$

## PUMPS WITH BALANCE DEVICE LEAK OFF

$$\text{Delta T} = H/778 \times [1 / E]/[1 / SH] \qquad \text{H.2}$$

*Where:* H = Total Head in feet; e = efficiency [decimal] at given flow

The application of these equations is best seen in a worked example. Generally questions on temperature rise fall into two types.

- How long can the pump be run at closed valve safely without seizure or boiling?
- What is the minimum continuous flow that will give a permissible rise in casing temperature [This would also allow a leak off orifice, to be designed].

*Pump data is as described:* Po = 103 BHP; SH = 1.0 [for water, 0.5 for crude oil] and $W_L$ = 100 lbs.

## TIME AT CLOSED VALVE

$$\text{Delta T} / \text{min} = 42/4 \times 103/100/1 = 43.6^{\circ}\text{F/min}$$

At an ambient temperature of 50 °F this would cause boiling in:

$$\frac{212 - 50}{43.6} = 3.7 \text{ minutes}$$

With a 1.5 safety factor, one would normally allow operation at closed valve for *no more* than 2.5 minutes.

With a submerged vertical pump, cooling of the surrounding water will be significant and so 4 minutes operation could be allowed.

## MINIMUM THERMAL FLOW

Turning to the minimum continuous flow, we have to produce a graph of temperature rise versus flow. Eqs. (H.1) or (H.2) is solved for a number of flows near to b.e.p and the results plotted against flow (Fig. H.1).

Pump could run at 20 Igpm without excessive temperature rise. Note this is Thermal Minimum Flow NOT the **M**inimum **C**ontinuous **S**afe **F**low (Table H.1).

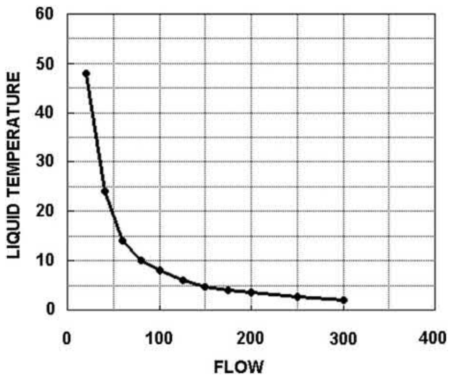



**Fig. H.1** This shows the temperature rise between inlet and outlet as a function of pump flow. There is usually one zone where the temperature change is quite low over a wide range of flows. Then there is a second [low flow] zone where the temperature change increases very significantly for even quite small flow reduction. Operation inside this flow-sensitive region can be dangerous.

**Table H.1** Temperature rise at low flow

| Flow [Igpm] | Head [ft] | Efficiency | Delta T [°F] |
|---|---|---|---|
| 20 | 208.5 | 0.02 | 13 |
| 40 | 208 | 0.04 | 6.4 |
| 60 | 207 | 0.07 | 3.5 |
| 100 | 206 | 0.11 | 2.1 |
| 200 | 204 | 0.20 | 1.05 |
| 300 | 202 | 0.28 | 0.67 |

# APPENDIX I

# Site testing

## SITE TESTING

The following procedure, thanks to Gary Dyson, is intended as a simple guide to allow engineers to determine how their pumps are performing and how to relate this information to the performance curve. In practice it is rare to find all the measurement facilities are available. The basics of flow and head are covered along with rules of thumb and calculation techniques.

### What is the pump flow?

Flow-measuring devices are rarely available, but sometimes the process flow *is* known in terms of mass flow or weight, and that needs conversion.

If no flow-measuring device is available the differential pressure of the pump can be used to determine the pump flow. However this is subject to considerable interpretation error.

Differential pressure is the difference between the suction and discharge pressures. The measured differential pressure must be converted into differential head to compare with the pump curve. The use of head, rather than pressure, makes test data independent of the pumped liquid.

### Procedure

Measure the suction pressure
Take this measurement on the suction side, as close to the suction flange as possible.
Measure the discharge pressure
Take this measurement on the discharge side of the pump, as close to the discharge flange as possible.

### Next

- Determine the Specific Gravity at the pumping temperature
- Calculate the Differential Pressure from;
  Differential Pressure = Discharge Pressure − Suction Pressure

■ Convert the Differential Pressure to a head reading

$$\text{Differential Head (feet)} = \frac{\text{Differential Pressure[psi]} \times 2.31}{\text{Specific Gravity at pumping temperature}}$$

## Performance curve

Now refer to the performance curve provided for the pump (Fig. I.1).

On the performance curve, locate the calculated differential head on the Y axis. Follow the horizontal line till it intersects the head curve. Then follow a vertical line down till it reaches flow readings on the X axis. The intersection is the pump flow − bearing in mind the uncertainties, and assuming that the pump performance has not decayed appreciably.

Ensure the total flow through the pump is being measured. Some pumps may have a low flow protection system, which involves a constantly open bypass line. By this means the low flow temperature rise and flashing is avoided, see Chapter 7. This flow must be included in the total flow. Ignoring it will have the effect of shifting the pump curve to the left by the value of the bypass flow.

The pump flow has been measured using a flow meter, what is the pump head?

If the pump flow has been measured with a flow meter the pump head can be checked (Fig. I.2).

For most practical purposes, we are interested in *changes* in performance so consistency in measurement is more important than accuracy.

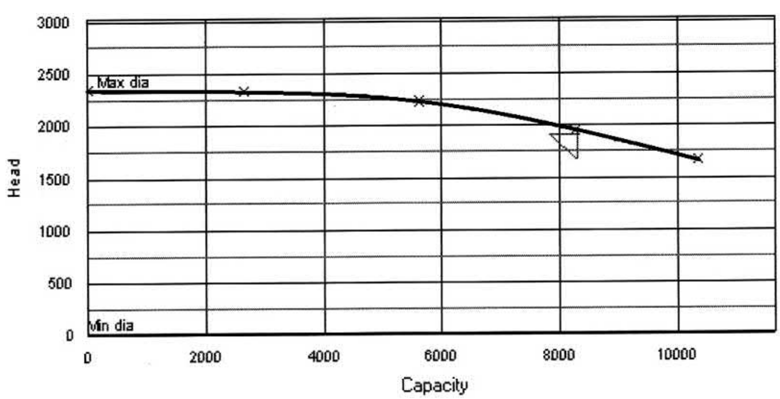



**Fig. I.1** Typical pump performance curve of head vs. flow at maximum impeller diameter. See Appendix C for performance changing by diameter reduction.

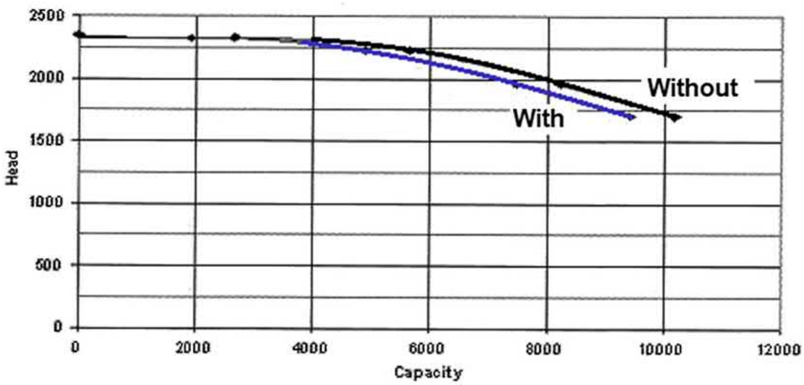



**Fig. I.2** Demonstration of the effect of not including a bypass flow.

## Procedure

Measure the suction pressure

Take this measurement on the suction side as close to the suction flange as possible.

Measure the discharge pressure

Take this measurement on the discharge side of the pump as close to the discharge flange as possible.

## Next

- Determine the Specific Gravity at the pumping temperature
- Calculate the Differential Pressure from:

$$\text{Differential Pressure} = \text{Discharge Pressure} - \text{Suction Pressure}$$

- Convert the Differential Pressure to a head reading

$$\text{Differential Head (feet)} = \frac{\text{Differential Pressure[psi]} \times 2.31}{\text{Specific Gravity at pumping temperature}}$$

On the performance curve provided by the pump manufacturer, locate the flow-rate on the bottom of the curve. Follow the vertical line till it intersects the head curve. Then follow a horizontal line across until it reaches the head readings at the side of the curve. Read the pump head rate off this scale (Fig. I.3).

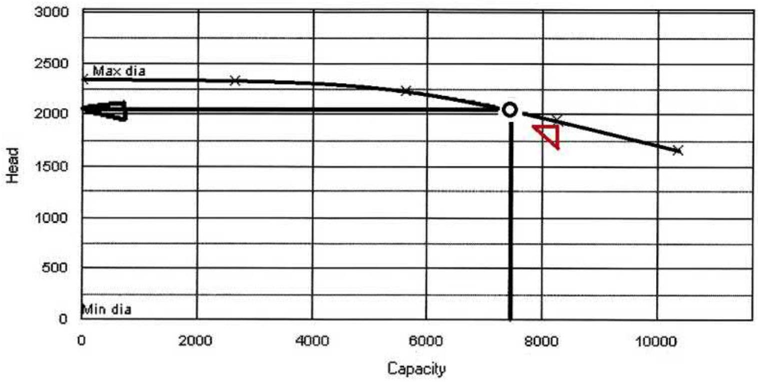



**Fig. I.3** Typical pump performance curve of head vs. flow.

## Absorbed power

In order to calculate the power being used by the pump some information from the motor is required. There are several ways to measure the horsepower.

The least accurate, is to use the ratio of the full load power and the full load amps to the actual amps being drawn. There is considerable room for error using this approach to determine absolute values. But for the purposes of monitoring *changes*, it is often sufficient.

The second involves actual information provided by the motor vendor on the motors efficiency at various loads, the power factor and voltage.

*Motor full load rated horsepower*

Taken from the motor name plate.

*Motor rated amps at full load*

Taken from the motor name plate.

# APPENDIX J

# Internal wear in pumps

Chapter 1 explains that the concept of a centrifugal pump has been in existence for a very long time. In that period, different manufacturers have made many different interpretations of the concept. Consequently it is unrealistic to suppose that the outcome of any of the general advice given in this volume could be guaranteed in all or any circumstances. This advice is given in good faith, but is only intended to illustrate the likely scale of any outcome (Fig. J.1).

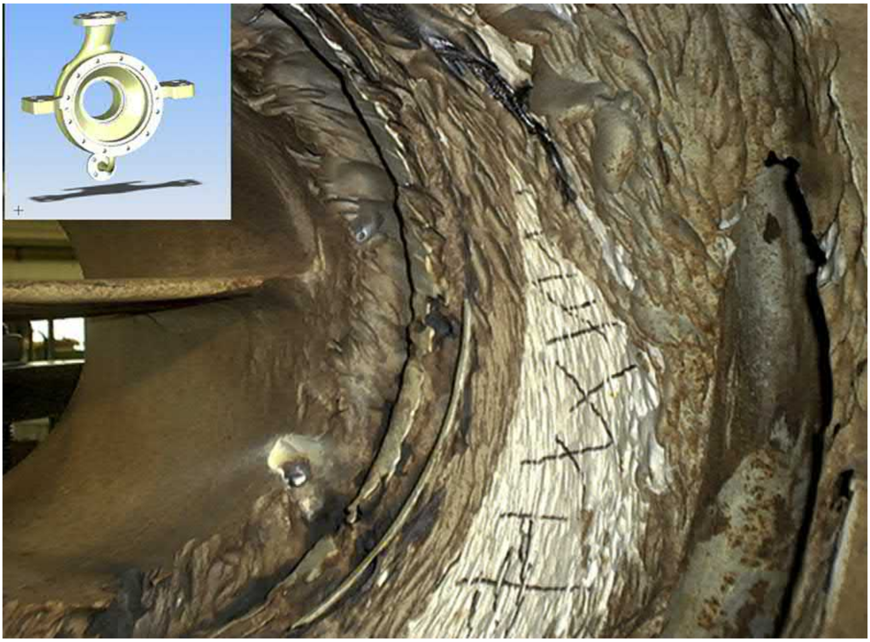


**Fig. J.1** This view is into a very badly eroded process pump casing. As complex as it seems, there are just a few different erosion mechanisms at work here. For reasons of urgency, this pump was subsequently repaired and put back into service. *[David Brown Pumps]*.

In some circumstances, pumps are required to handle liquid which has abrasive particles suspended in it. This may be a planned occurrence or it may occur unexpectedly. If it was planned or foreseen, the pump materials and shaft speed will most probably have been selected to best suit these conditions. However, pumping abrasives is still sometimes an inexact science [Ref. 27]. Furthermore, experience says that, even if the liquid is part of some defined process; it may be difficult to get firm data on the size, shape and concentration of the solids; its granulometry. Often this is not a case of prudence on behalf of the pump user, but simply because he does not know. Even when the liquid is part of a defined process, estimating the likely solids content can be very difficult. However, past knowledge of similar processes will often allow an experienced pump vendor to make an estimate.

On the other hand, handling abrasive solids may never have been envisaged at the pump selection stage. This is most often the case. Conventional pumps designed for high efficiency on clear liquid applications can show limited tolerance to the unscheduled presence of abrasive particles suspended in the liquid.

Confronted with a worn conventional pump, there might not appear to be any logical pattern to the damage. The Figure above illustrates this. All pumps will experience wear in one form or another. There is already an adequate body of literature dealing with wear of external components such as bearings and shaft seals. I will not repeat that here. Instead, I will concentrate on wear of the pump internal wetted components. Note that these comments relate to pumps that have been accidentally subjected to erosive particles in the pumped liquid. Where this situation can be foreseen then special solids, handling pumps are readily available, though there may be upper temperature limits.

The wetted parts of all centrifugal pumps will probably fail from corrosion, abrasion and or erosion. Again, the corrosion aspects are already well covered elsewhere so will not be discussed here. Furthermore, pure abrasion [dry rubbing] is not a sustainable state, so it will also be omitted (Fig. J.2).

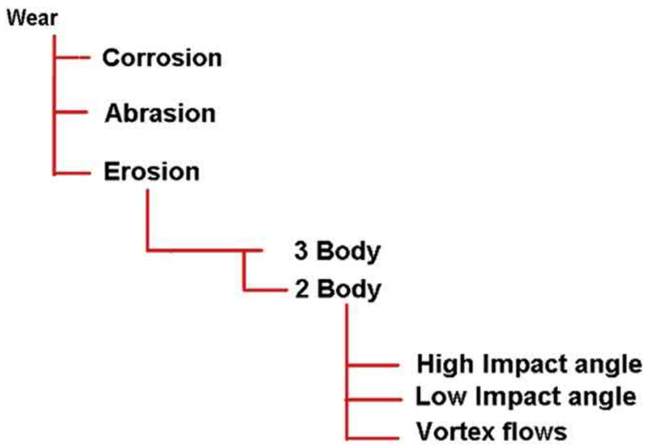



**Fig. J.2** A family tree lists the most common sources of internal wear encountered in centrifugal pumps. Only the Erosion issues will be discussed here.

This section will describe the essentials of erosive damage inside pumps [see also Refs. 18 and 27]. Erosion can conveniently be categorised as three-body or two-body. Three-body wear occurs when three bodies are in contact whereas in two-body wear only two contacts occur (Fig. J.3).

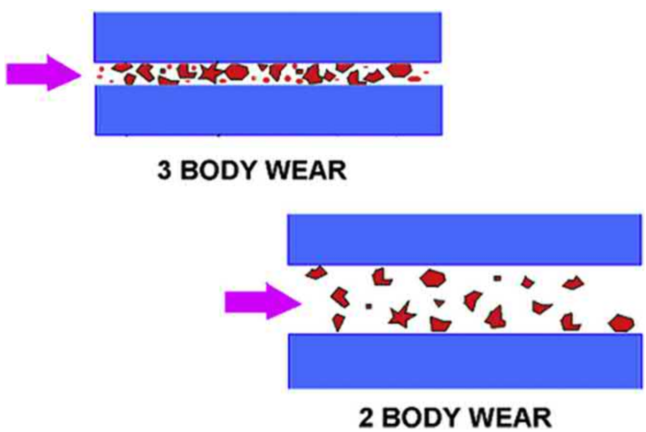



**Fig. J.3** This helps describe the difference between two body wear and three body wear and is largely self-explanatory. In pumps, three body wear occurs when the average particle size is similar to component gaps between moving components – in wear rings for example. Two body wear occurs when the component gap is much larger than the particles – on wetted surfaces for example.

Typical three-body wear may occur at pump wear rings when the pumped liquid contains abrasive particles that are comparable in size to the wear ring gap. That is to say, both wear rings are also on contact with the same abrasive particles. If the particles are significantly harder than the wear rings, they will remove material. Initially, the wear ring gap will relentlessly increase. Eventually, the wear ring gap will increase to be equal to or larger than the particle size. From then on, the wear mechanism will move to the slower two-body wear (Fig. J.4).

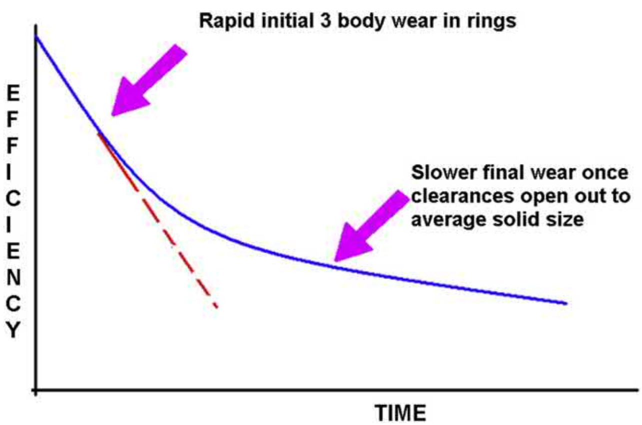



**Fig. J.4** Pumps may initially encounter 3 body wear conditions but will usually move towards two body wear over time. Since two body wear rates are lower, the damaging effects seem to slow down. Three body wear is rarely sustainable for any length of time.

Increasing wear ring clearance will be detected externally as a reduction in pump output. Initially the output will reduce quite quickly. But as the picture moves from three to two-body erosion, the gap increase more slowly. Hence the output now decays more slowly [Fig. 7.4]. Two-body erosion tends to be the most frequent wear class in pumps so it warrants further discussion (Fig. J.5).

In considering two-body erosion, the angle of particle impact seems very important.

- Particles that impinge at a very low angle remove material by gouging and scraping the surface.
- Those attacking at a high angle have their kinetic energy largely dissipated into the surface microstructure and tend to result in micro fracture and shattering.

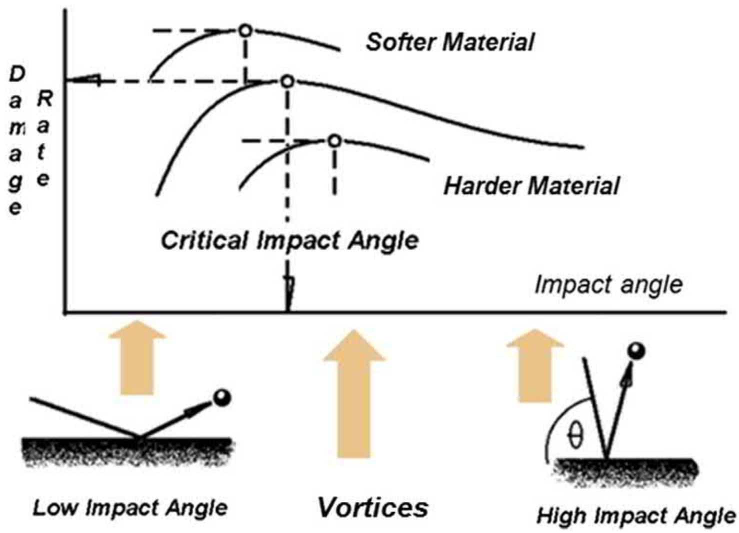



**Fig. J.5** In two-body conditions the wear rates are a strong function of the impact angle and the material hardness – all other things being equal. For each material there exists a critical impact angle where material removal is highest. Vortices hold the greatest risk, because they are most likely to include the critical impact angle as well as possessing heightened particle velocity.

- At some intermediate critical angle, as one mechanism merges into another, we find the maximum removal/damage rate.
- There is an additional trend for softer materials to have a lower critical angle than hard materials.

## LOW IMPACT-ANGLE

Wear predominantly from low angle particle attack is likely to occur all over the wetted surface of the machine. The particles may contact the wetted surface many times, as they pass through the pump. Areas where low angle attack might be expected are casing passage wall, in suction and discharge nozzles and on the shrouds of impellers. In fact, anywhere where the flowpath is generally parallel to the wetted surface. Naturally, this type of damage is likely to be most severe in areas of high liquid velocity such as near the impeller rim and immediate casing (Figs J.6 and J.7).

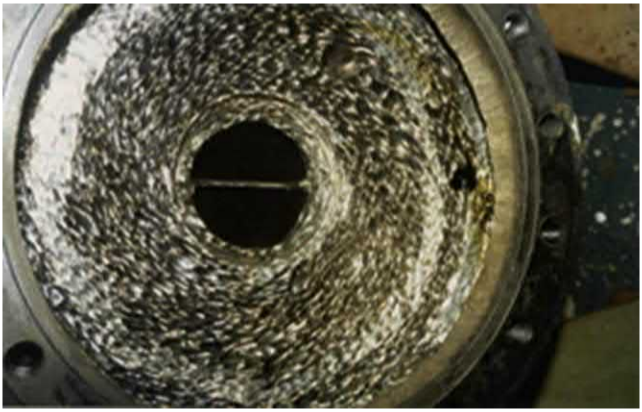


**Fig. J.6** A good example of low impact angle damage on the sidewalls of a pump casing. A somewhat repetitive pattern is evident, that also discloses the spiral inward path of the particles towards the wear ring. *[David Brown Pumps].*

**Fig. J.7** This shows corresponding low impact angle damage to the outer shroud wall of an impeller. Note that the damage rates appear to increase with radius from the shaft axis. This is because the impeller angular velocity also increases with radius.

Low impact-angle damage patterns seem to be linked to the same Von Karman vortex mechanism that is also responsible for making flags regularly flutter in the wind. From some initiating nuclei, maybe cast surface imperfections, the vortex patterns form and gain strength. Eventually the patterns merge to present a continuous appearance. The patterns will have some underlying frequency of replication which gives it a characteristic ‘fish scale’ appearance.

No simple relationship appears to exists between velocity and damage rate, but in pumps it seems to lie in the somewhere in the [very wide] region of $(\text{velocity})^3$ and $(\text{velocity})^5$. Low angle attack also occurs in the annular gap of wear rings and the like-especially those where a high differential pressure exists over them. Experience says that, in general, hard surfaces resist this attack most. In addition, small fine particles seem less damaging than larger ones.

## HIGH IMPACT-ANGLE

Wear from high angle attack tends to occur where vanes ribs or bosses protrude into the flowpath. The particle kinetic energy is dissipated into the component surface which may then micro-fracture. The impacting particles, or what subsequently remains of them, may then make further surface contacts at low impact angles.

For this reason, high impact angle attack tends to occur at fewer sites within the machine, typically at locations where the flowpath splits around some feature. Area most vulnerable include suction anti-swirl vanes, impeller blade entry edges (Fig. J.8) volute cutwater edges, diffuser vane inlet tips, bearing spiders.

**Fig. J.8** This image shows high impact angle damage to the leading edge of an impeller vane. It has lost its streamlined shape and become rather blunt. It has also taken on a typical polished appearance. *[David Brown Pumps].*

## VORTEX FLOWS

It would be wrong to assume that low angle or high attack angles are always mutually exclusive. In fact they can coexist, in which case the highest damage rates might be experienced. The most likely location for this to occur is in the core of any vortices that might exist on the flowpath. Vortex flows are often unstable and turbulent (Fig. J.9).

This means that statistically, there is a high chance of particle attack angles approaching the critical levels somewhere within the unsteady vortex. Furthermore, velocities will be very high in the vortex core.

*But the most important feature is that the particle count inside a vortex will be many times higher than in the same volume of main flow. This is caused by the hydro-cyclone concentration effect.*

Combining all these aspects means that vortex flows possess the highest erosive damage potential along the flowpath.

In normal circumstances, there are a number of classical vortex categories that might be encountered in centrifugal pumps.

- Kamm type
- Taylor type
- Von Karman
- Horse-shoe

**Fig. J.9** View of a volute cutwater [lip] subject to Horse Shoe Vortex [HSV] wear at the junctions with the casing sidewalls. It is clear that the general casing surface has been otherwise largely unaffected by the suspended solids [The original casting surface finish is still visible.]. There is little if any evidence of low impact angle damage on the other parts of the wetted surface.

## Kamm type vortex

This type of vortex forms immediately behind or downstream of any body shape that ends suddenly, and without streamlining. Such vortices tend to be rather stable in form and are sometimes known as ‘standing vortices’. This type of vortex can be seen attached to the rear of bluff-tailed road vehicles being driven in the rain. The spray forms a standing vortex that, moving with the vehicle, draws water up from the road and deposits it on the rear window (Fig. J.10).

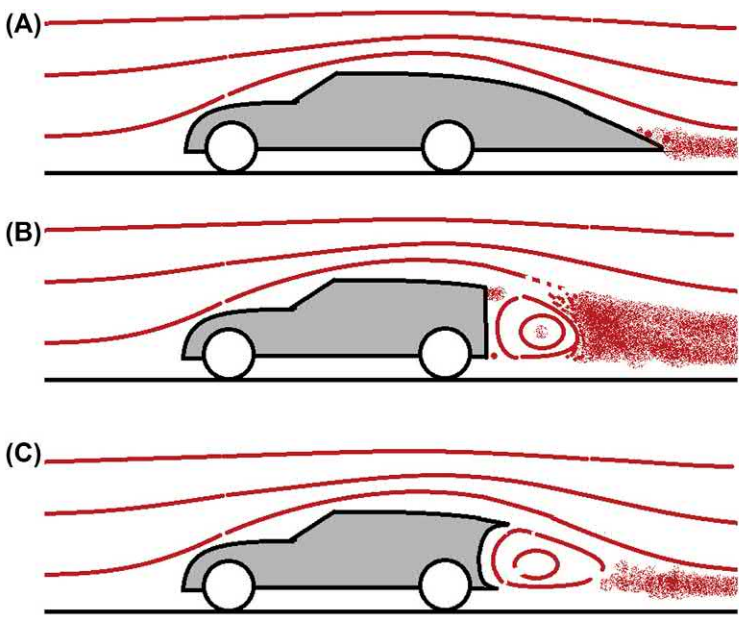



**Fig. J.10** (A)Streamlining the rear of a vehicle best manages the airflow. It helps supress flow detachment and drag inducing vortices. But its physical size makes it often impractical. (B) A vehicle without streamlining will naturally develop a standing vortex at its rear. This helps to creases the drag resistance on the vehicle. The vortex has the side effect of entraining and accumulating road dirt, some of which may be deposited onto the rear surface of the vehicle – as here. (C) Prof Kamm found that by subtle re-shaping of a vehicle rear, this vortex shape could be managed to the point of almost emulating a streamlined body shape. The subsequent airflow pattern became similar to that of a streamlined vehicle. The drag levels were also much reduced compared to an unstreamlined vehicle.

One objective of streamlining is to avoid such a large standing vortex forming. This is because they induce drag forces. Normally, the standing vortex tries to self-design a streamline form and somewhat smooths the main flow leaving the vehicle. In about 1938, Prof W Kamm found this effect so powerful that, if correctly designed, a vehicle with a bluff un-streamlined tail produced almost as little drag as a longer one with full cumbersome streamlining. In pumps, this self-streamlining effect can also be useful; however, the solids concentration effect of this, as with all vortices, is usually very unwelcome.

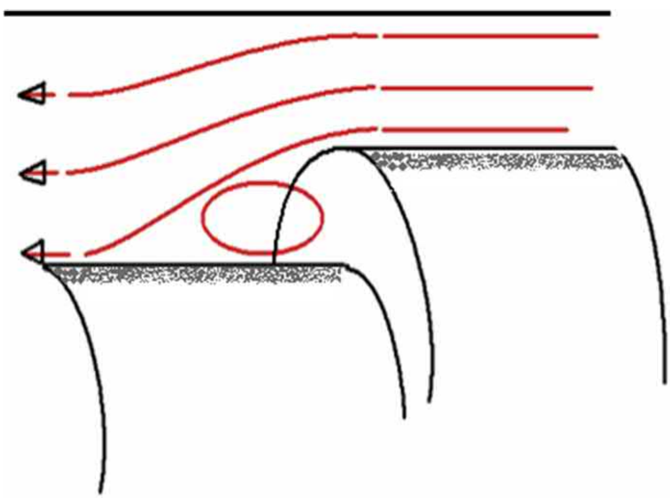


**Fig. J.11** Flow leaving out of an annular passage suddenly expands and is unable to follow such an abrupt change in profile. Instead it forms a standing corner vortex, which helps smooth the flowpath. This is very similar to the 'smoke rings' that some smokers can create. It is also the circular equivalent to a Kamm−type vortex.

## Taylor type vortex

In certain circumstances, the flow through a long annular gap can create a series of ring type vortices. They have much in common with the 'smoke rings' that can be generated by some smokers (Fig. J.11).

Such Taylor type vortices occur through annular gaps but can occur at the efflux point or where there is a sudden and significant change of section area. In this case, they can be thought of as a coaxial version of the Kamm vortex. Again the flowpath is improved but with the unwelcome creation

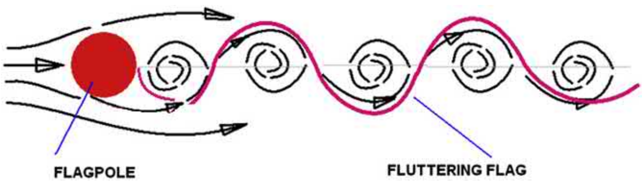



**Fig. J.12** Von Karman vortices are alternately shed from the sides of a bluff body positioned in a crossflow. These alternating vortices are chiefly responsible for making flags flutter in the wind. They are also responsible for the whistle or hum from overhead cables in the wind.

of a solids-entraining vortex. Taylor vortices are most commonly seen in seal chambers and the exit flowpath of wear ring gaps, breakdown bushes.

## Von Karman vortices

This type of vortex creates the regular fluttering of flags on the wind. It can be imagined as a series of Kamm type vortices that alternately form on one side of the flagpole, and then the other as the wind blows. This flip-flop characteristic imparts the fluttering corrugations onto the flag surface (Fig. J.12).

Such vortices are also responsible for the Strouhal effect that relates to both telephone and power lines bouncing in the wind.

This type of vortex is most likely to occur downstream of very short ribs or tubes crossing the flowpath. They can also be generated along the interface between a Kamm type vortex and the main bulk flow.

Each travelling vortex 'packet' can carry with it core of concentrated solids. This may travel some distance from the source and along down the flowpath before the solids encounter a solid surface.

## Horseshoe vortex

This is a rather complex, but common, vortex variety. There is a risk of this flow pattern forming wherever a vane or rib crosses the flowpath (Fig. J.13).

The risk is restricted to the region where the rib intersects with passage walls that are more or less parallel with the main bulk flow path. It is most severe where and when this intersection is completely orthogonal. In pumps the area most susceptible are the casing cutwater/volute lip and the entry edge of impellers and also multi-vaned diffusers. This is a three dimensional flow effect but I will attempt to explain its development using two dimensional descriptions.

Consider flow with only mild levels of suspended solids approaching [say] a volute cutwater/lip at some point midway between the adjacent

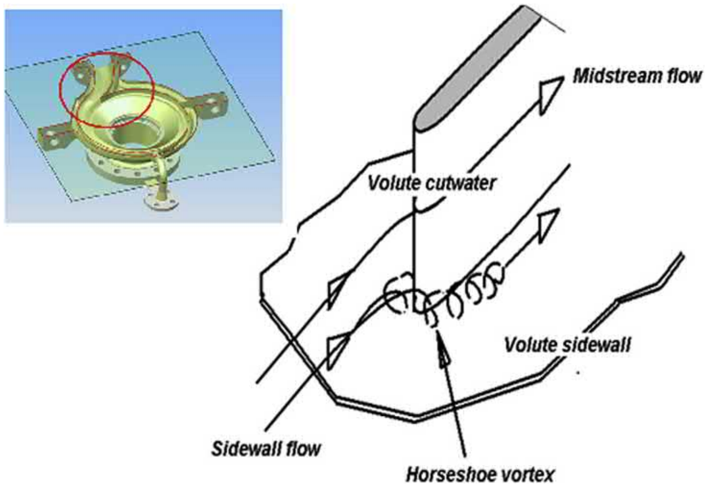



**Fig. J.13** Horse Shoe Vortices [HSV] can form at the junction between sidewalls and ribs or vanes positioned at/near to right-angle with the wall. This is a simplified picture, but shows the basic mechanism.

casings walls. The casing walls will generally be too far away from this point to influence the flowpath. The approaching flow simply splits and passes either side of the cutwater edge. High impact angle damage may occur, but since few or no vortices are created at this point, the erosive damage rate will be relatively low. The flow then continues with a flowpath parallel to the adjacent case walls.

Approaching flow that is near to the casing walls experiences a different set of conditions. Here the frictional effects of the boundary layer reduce the flowpath velocity close to the walls. The closer to the wall, the greater the flow retardation effect. Unlike particles at midstream, this introduces an additional tendency for the flow to veer slightly towards the wall and to become entrained into the boundary layer. Upon then encountering an orthogonal rib, the flow not only splits but also does so with component at right angles to the flow.

The net result is a vortex that spreads either side of the orthogonal juncture. The vortex shape is adequately described by the term 'horseshoe vortex'.

Again, such vortices can entrain a cargo of erosive particles. However, in this case the prime damage direction will be chiefly perpendicular to the main flow and hence through the adjacent casing wall. In many designs, the casing wall at this point is also part of the pressure boundary. This might raise the potential risk of dangerous liquid leaking out to atmosphere.

## IDENTIFYING THE CAUSE OF COMMON DAMAGE PATTERNS FOUND ON THE INTERNAL PARTS OF CENTRIFUGAL PUMPS

The general internal damage causes have already been outlined earlier. The most common of these experienced by pumps is that of corrosion. In most case, the evidence for this is so clear-cut that little help is needed in identifying the root cause. A severe example is shown here. Frequently the solution can simply be the choice of better materials-if available (Fig. J.14).

The remaining most common damage causes are erosion and cavitation.

From the foregoing it should have become clear that erosive damage in pumps can have several root causes in co-existence. Consequently, there is seldom one single global solution where conventional pumps are concerned. However, by correctly identifying the several root causes and applying relevant solutions in turn, then significant pump life improvements are possible. I should stress that cavitation damage can also be experienced in some of the same sites as erosive damage. However, it would be very unusual, but still possible, to witness cavitation damage AND abrasive damage simultaneously at the same damage sites.

In this next section, I attempt to show images of some typical damage patterns and give some explanations and solutions. I should mention that in each case the patterns just show a generic trend. Specific mechanisms can produce a wide range of visual symptoms that might at first depart from the examples shown here. But they should show some similarities.

**Fig. J.14** This bronze impeller has failed largely due to corrosion. It was successfully replaced by an impeller made of Duplex Stainless Steel. There is little, if any, evidence of particle erosion. *[Weir Pumps]*.

# 1. IMPELLERS

## SINGLE STAGE

### Single or double suction: fully shrouded

This Figure highlights the most common areas of damage found on these types of impellers. The ballooned numbers refer to the subsections in which cause and effects are discussed in more detail.

**1]** Wetted surface area in and around the impeller hub disk-to-shaft interface.

**2]** Vane surface area local to intersection with shroud, or [less frequently] hub disk inner.

**3]** In-board area at junction of wear ring with shroud disk outer.

**4]** Shroud disk outer surface.

**5]** Peripheral surface termination of vane and intersection with shroud disk.

**7]** Along the vane entry edge (Fig. J.15).

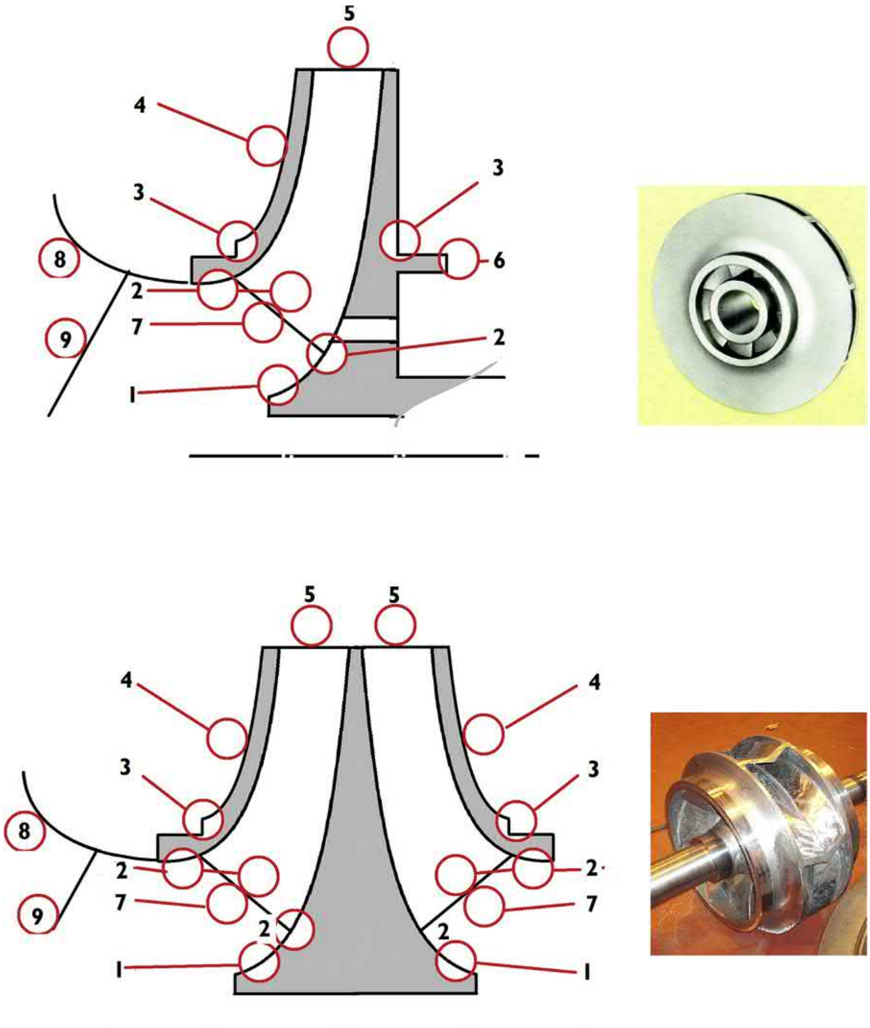



**Fig. J.15** These are radial sections through typical centrifugal pump impellers [Single stage-single/double suction — fully closed/shrouded.]. They highlight the most common areas of damage.

## *Zone [1]: hydraulic hub damage*

### Cause

This type of damage is most often caused by pumping very significant quantities of suspended particles having a noticeably higher density [SG] then the pumped liquid.

Inertial effects oblige these heavier particles to continue in an axial direction rather than turn and follow the flowpath into an outward radial direction. The solids impinge on the hub causing mainly low impact angle damage in the form of gouging and grinding. The damage surface generally looks rough and grooved.

### Solution

Hard surface coatings applied to affected zone. In extreme cases, provide a renewable protecive sleeve (Fig. J.16).

*NOTES*; A note of caution, damage in this area can also very occasionally be caused by cavitation, as a result of poor hydraulic profiling in the area surrounding the impeller lock nut. However cavitation damage has a very different appearance, consisting of many surface pits rather than surface grooves. Fig. J.18 is a picture of such cavitation in a sewage pump. Sewage pump impellers very often have to be compromised from optimum design in order to bestow the pump with some capacity to handle large solids. A part solution here is better streamlining to the locknut.

**Fig. J.16** Here is shown surface damage localised to the hub profile of an impeller. This can be typical in pumps handling relatively large and dense particles. *[Weir Engineering Services]*.

**Fig. J.17** This cut away compressor impeller shows similar damage pattern on the hub surface [horseshoe vortex damage is evident at the blade root, and is discussed later — see Fig. J.23]. *[Barry Hall]*.

**Fig. J.18** In this Sewage pump impeller, the damage is not caused by particle erosion, but by cavitation. Sewage pump impeller profiles are often severely compromised from the ideal in order to better handle a very wide range of suspended items. This can result in poor cavitation resistance.

### *Zone [2]: grooving damage at intersection of blade and shroud at impeller inlet tip*

#### Cause

Horse-shoe vortices are likely to form at the blade to shroud intersection. Abrasive particle caught and concentrated in the vortex create grooves, cutting into the shroud. The damage surface will generally be smooth, even polished in some cases.

#### Solution

Inhibit the development of Horse-shoe vortices by creating sweepback where the vane intersects the shroud. The intersection should be less than 15 degrees, instead of the more normal 90 degrees [see IMPROVEMENT sketch], this approach will largely avoid the formation of horseshoe vortices in this region

NOTES; Again care must be taken to differentiate erosive particle wear from cavitation, which can also afflict this region. Cavitation damage will usually be in the form of small discrete matt pits, unlike horseshoe vortex wear which will be large, polished or show linear scratches. When it occurs, severe cavitation will usually appear concentrated on either the concave or the convex surface in this region of the blade [in flowpath direction] (Figs J.19–J.21).

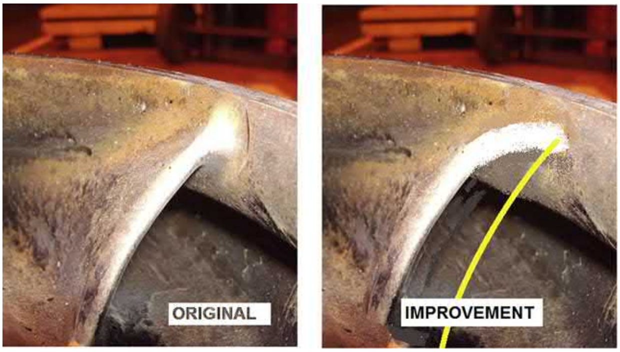



**Fig. J.19** The left hand image shows that a Horse-Shoe Vortex has developed where the impeller vane leading edge intersects with the impeller outer shroud. The right hand image has been enhanced to show how such erosion can be supressed or even eliminated. Introducing some 'sweep-back' to each vane edge over a short length — say 10% of the edge length, or more, helps destroy the underlying vortex mechanism. The light line represents the original vane edge shape before cutting back to form a sweep-back.

**Fig. J.20** Another example of significant horseshoe vortex damage attacking the vane inlet tip junction.


**Fig. J.21** Another example of improved resistance to Horseshoe Vortex. Note how the intersection angle between vane leading edge and both hub/shroud disk is now very much less than 90 degrees. In most cases the intersection with the shroud disk is more important than with the hub, because the liquid velocities are higher there. *[Weir Engineering Services]*.

### *Zone [2]: grooving damage at intersection of blade and hub at impeller inlet*

#### Cause

The first pictures shows damage to a centrifugal pump impeller. The remainder are of a centrifugal compressor.

In both cases, and as in the previous section, such damage is caused by Horseshoe Vortices generated in the flowpath as it encounters the blade. The entrained solids concentration has caused accelerated erosion at the junction.

#### Solution

As before, the solution involves reprofiling the inlet edge in order to disrupt the Horseshoe vortex mechanism (Fig. J.22).

**Fig. J.22** Horseshoe Vortex at junction of vane leading edge with impeller hub. There is little evidence of any other significant erosion from other causes. Even the cast passage surface appears largely intact. This illustrates the damage potential of vortices.

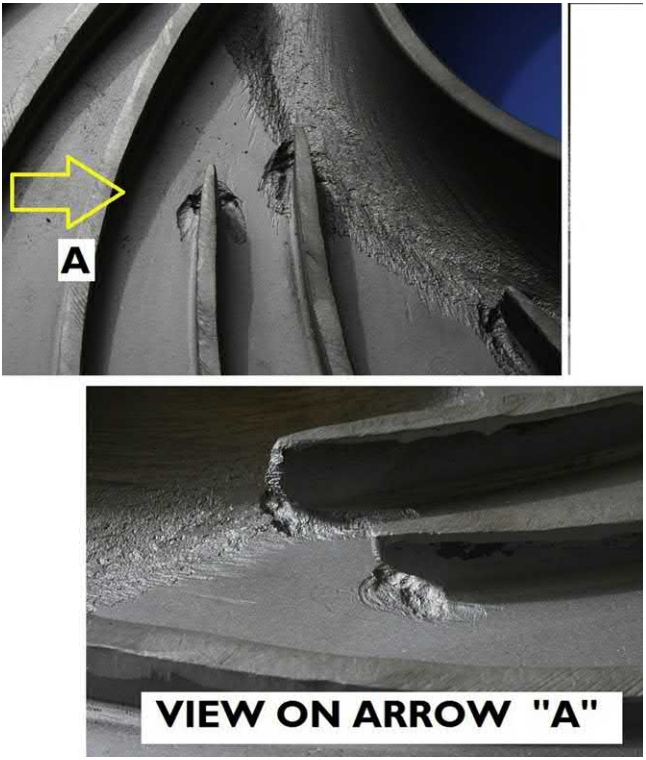



**Fig. J.23** Here is a compressor impeller inlet [see also Fig. J.17] that shows well developed HSV damage at the vane-to -hub intersection. *[Barry Hall].*

### *Zone [2]: damage present on the flowpath before entering the vanes*

Sometimes pitting damage can be identified that is on the flowpath but upstream, of the impeller blade. At first sight, this is counter intuitive. The explanation is as follows;

#### Cause

In most pumps that operate at less than approximately 30–40% of their inlet design flow [which might not be outlet design], the flow picture going into the impeller breaks down and a partial zone of backflow appears in an annulus around the impeller entry [see Appendix D]. This 'backflow' liquid can escape upstream for some considerable distance before being re-entrained back into the main flow. Vortices are created both within the backflow volume as well as at the interface between inflow and outgoing backflow. Cavitation can of course form in all or any these vortex cores. Those cavities forming in the backflow volume are most likely to collapse on the adjacent walls of the inlet duct. Those forming at the core of interface vortices will be swept along by the inflow and collapse at random and unexpected points around the inlet annulus.

**Fig. J.24** Cavitation damage ahead of the impeller vane leading edge is usually evidence that the impeller has been operating inside the Backflow Regime. Cavitation can form in the filamentary vortex cores created along the interface between the incoming and outgoing liquid masses. It is then swept out ahead of the impeller before collapsing. *[See also Fig. D.9]*.

## *Zone [2]: pitting damage on face of blade at inlet*

### Cause

In most cases, pump cavitation will occur first and most intensely at or near the maximum diameter of the impeller inlet [the so-called impeller eye tip]. Here liquid velocities will be high and static pressure low [Appendix E].

### Effect

As the margin between NPSH [R] and NPSH [A] diminishes, the boundary of cavitation [and any damage] will expand both at right angles to the flowpath and down the blade inlet edge and towards the impeller hub. It will also expand in the direction of flow. Cavitation damage most commonly occurs on the concave, and most easily visible, surface of the blade. This is because most pumps are oversized for their duty flow. As a result, the vane setting angle is larger than that of the approaching flow.

Occasionally, damage occurs on the less easily visible convex surface. In such cases, the pump flow is probably undersize relative to the actual duty. Hence the approaching flow angle exceeds that of the vane setting angle.

Equal damage amounts on both faces would normally point to operation at or near to the design inlet flow.

### Solution

Prevention requires new or alternative impellers to give a better hydraulic fit, if the NPSH margin cannot be increased. Otherwise harder materials help slow down the damage rate.

### Attack initiated on front [visible and concave] blade surface and proceeding to back

The appearance of cavitation damage can often help to yield more information on the way the pup has been operated.

As mentioned above, damage on the visible concave face is indicative of operation at flows less than design. This conclusion would be reinforced if at or near the centre of the visible damage patch the cavitation actually penetrates through the vane thickness. Therefore, there will appear to be an intense zone of penetration surrounded by a lesser zone of cloud cavitation damage (Figs J.25–J.30).

**Fig. J.25** Light cavitation marks on concave [visible] vane surface indicate classic off-design flow damage. This image shows how, in a pump operating near to its design flow, cavitation damage decreases with radius from the shaft. It is disclosed by the wedge-shaped pattern. This is principally because the peripheral velocity also decreases with radius.

**Fig. J.26** Intense low flow cavitation damage on concave [visible] face, plus damage in area of passage inlet throat. This pump has probably been operating over a range of flow conditions in order to experience these two different damage patterns.

**Fig. J.27** Heavy centralised cavitation damage implies that pump has been operating at significantly off-design at low flow.

**Fig. J.28** Heavy cavitation damage at passage inlet throat. Again implies high cavitation intensity.

**Fig. J.29** Light 'off-design' −low flow damage. Note damage intensity is related to radius from shaft i.e., peripheral velocity.

**Fig. J.30** Heavy off design flow damage.

### Attack initiated on back [less visible and convex] surface and proceeding to front

If there is visible penetration through the blade, but little or no cloud damage around it on the concave surface, then most probably the pump has been operating at a flow rather larger than design. Inspection should show that most of the cloud cavitation damage, leading up to penetration, will be on the invisible convex surface. It can be examined by use of mirrors or touch.

Unless the impeller material is weld repairable, then severe penetration damage would normally signify that the impeller is well beyond use. But if penetration has not occurred, then how much damage is acceptable? One popular criteria considers the impeller beyond use if the damage has penetrated the vane thickness, normal to flow, to the extent of 75%. Of course, impellers of weld-able materials may have this thickness reinstated by weld deposit if and only if the blade profile and thickness distribution [in the direction of flow] can be reinstated, along with the inlet edge shape.

There are cases where patches or inserts have been welded in to damaged blades. This can be intricate, requiring considerable effort. Normally this approach would only be justified in cases of dire emergency — where replacements are not readily available and following a risk assessment (Figs J.32–J.35).

**Fig. J.31** Damage on rear convex less visible vane surface indicates operation at high flow [see also Fig. J.36].

**Fig. J.32** Example of cavitation on the rear/convex less visible surface eventually penetrating through to concave/front face.

**Fig. J.33** Here the cavitation damage on the front/concave visible surface has begun to penetrate through the vane thickness.

**Fig. J.34** There is a lack of pitting around the damage area and this suggests that the penetration is taking place from the rear convex less visible vane surface — hence high flow operation. *[A J Semple]*.

**Fig. J.35** As before, little pitting on the visible face. Great amount of pitting on the reverse side, ultimately penetrating through to the front.

**Fig. J.36** Damage on the reverse vane side and also the hub disk [see also Fig. J.31].

## *Zone [2]: alternate vane cavitation*

### Cause

Generally when cavitation damage occurs its intensity is more or less the same or similar on each blade. However, very occasionally, cases arise where only some of the blades are damaged. This can happen when the impeller has an even number of vanes. The cause is analogous to rotating stall in other turbomachines. As the pump through-flow is reduced, the flow angle approaching the impeller differs more and more from the vane-setting angle. At a certain point, the flow cannot smoothly adapt to this change. Instead a stall condition begins to arise, causing some flow blockage effects inside each passage. This condition will occur in impellers with either odd or even vane number

### Effect

It seems that initially, in impellers of even vane count, the equal flow blockage picture can flip to a point where one passage unstalls a little, while the next one stalls a little more and so on. The nett flow stays the same. As flow is further reduced, this difference increases until alternate passages are either stalled or unstalled. This leads to the circumstance of alternate vane cavitation. Such a condition is widely reported in helical flow inducers, particularly those of constant lead angle. However, it appears much less common in centrifugal pump impellers

### Solution

The scale of the problem can obviously be reduced by fitting an impeller having an odd vane count number (Fig. J.37).

**Fig. J.37** Only Alternate vanes show cavitation damage. This is quite unusual. Obviously it will only occur in impellers with even vane number.

### *Zone [2]: microcracking*

Cracks are sometimes discovered in cast impellers of the fully shrouded type. These occur at the point where the vane inlet edge joins either the outer shroud, or the central hub. Usually these appear after the machine has been in service for some time. They are most likely [but not exclusively] in pumps that have been operating at low flows. That is because the inlet backflow produces fluctuating or fluttering forces on the impeller vane inlet region. These forces induce bending effects.

#### Cause

Centrifugal pump impellers are often produced by the casting process. Fully shrouded types can represent a challenge due to their complex shape, which will include cored out liquid passage shapes. From a hydraulic point of view, the blades will ideally be rather thin, particularly in the passage inlet region. On the other hand, adjoining mechanical features such as the central hub and or the outer shroud will often be much thicker. This combination creates the possibility of micro cracking at the intersection point.

#### Effect

After molten metal has been poured into the impeller mould, the thin sections such as the blade, particularly near the inlet, will cool and freeze first. The thicker adjoining sections cool more slowly and in the process, they shrink slightly in size. The simultaneous shrinking of hub and shroud induce tensile loads into the blade inlet edge. Such loads are roughly normal to the flowpath direction. The material microstructure in the intersection region can be complex due to the different freezing rates of the two components. This discontinuity can leave the casting with a latent stress raiser. The fluctuating bend stresses of backflow operation [Appendix D] may intensify the cracks over time. Such cracks can also appear in bowl type casings.

#### Solution

Experienced foundries will be aware of this risk and take steps to try to equalise the freezing rates in the intersection region. The designer can help by providing large fillet radii at the juncture (Figs J.38–J.40).

**Fig. J.38** Micro cracking at the vane intersection with the hub disk is here revealed by dye penetrant. *[C K Verma]*.

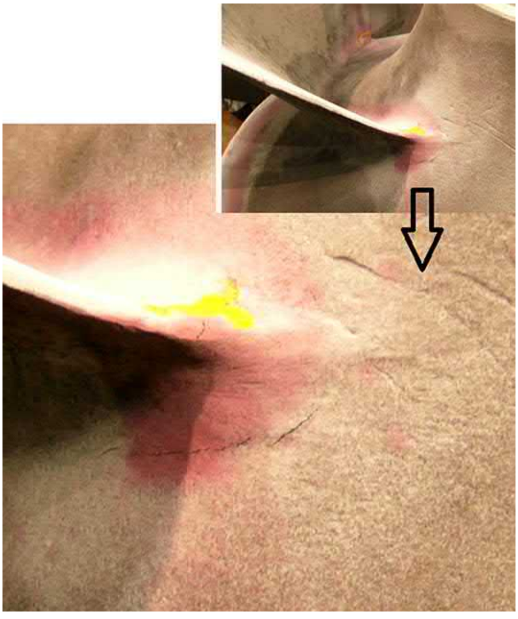


**Fig. J.39** Another clear example of micro cracking at the vane to hub intersection.

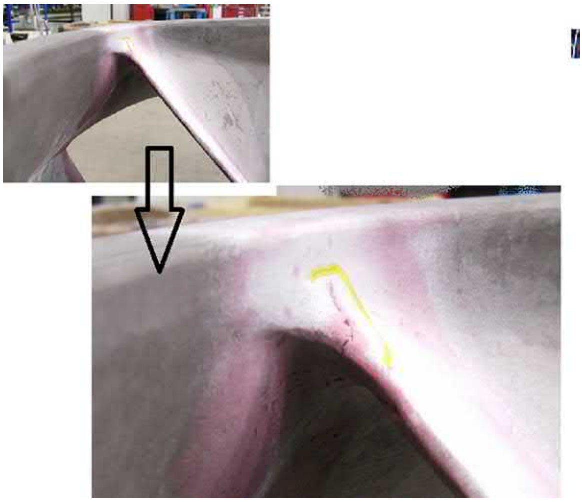


**Fig. J.40** Micro cracking at the intersection with the impeller outer shroud.

**Fig. J.41** Tensile shrinkage forces in the cooling casting can induce micro cracks on the vane edge. These can be the centre, as here, for high impact angle erosion damage. *[Andrew Percy].*

### *Zone [2]: localised damage at intersection of blade and hub*

#### Cause 1

Most often this is caused by cavitation damage.

#### Effect

Cavitation damage is always most likely to occur near the largest diameter of the inlet annulus [eye]. That is because liquid velocities will be highest here. However, in some severe cases, cavitation damage can be evident near the hub as well. In fact certain rare cases exist where the hydraulic design philosophy results in the hub being the most likely area.

As described earlier, damage exclusively to the visible concave face is symptomatic of prolonged operation at flows less than design flow and *vice versa*.

#### Solution

The pump design flow needs to be brought much closer to the true operating flow, or vice versa. Alternatively, the margin of NPSH [A] over NPSH [R] needs to be substantially increased.

#### Cause 2

A less common cause relates to micro cracking − see earlier. Here the damage will be very much less widespread (Figs J.42−J.44).

**Fig. J.42** Normally, Cavitation damage is most intense at the vane intersection with the outer shroud disk. But certain hydraulic designs [as here] operating at low flow, will show more long term damage at the hub intersection.

**Fig. J.43** This design exhibits more damage at the hub than at the shroud intersection. This counter intuitive behaviour is a feature of certain hydraulic designs and can be associated with low flow operation.

**Fig. J.44** The 'nick' in the vane edge at its intersection with the hub is unlikely to be cavitation and more likely to be erosion at the site of a micro crack. The root cause is identical to Fig. J.41.

### *Zone [2]: localised erosion damage to vane concave surface*

Damage marks to the concave surface sometimes occur. These do not have the same finely pitted appearance as cavitation. Instead there is distinct coarse grooving pattern-more or less at right angles to the imagined surface flow direction

#### Cause

Inlet backflow [described in Appendix D] occurs if the pump flow is throttled to [say] less than 30%–40% of its design value. Initially, the phenomenon is contained within and rotates with the impeller. As flow through the pump is further throttled it becomes more intense, eventually escaping out into the inlet duct.

Within the initial backflow structure, vortices occur and bring with them their associated erosion potential. This type of damage is restricted to the first part of the visible [concave] surface of the vane. The surface pitting will appear coarser and less well defined than cloud cavitation damage.

#### Solution

Elimination or reduction of inlet backflow. This is described in Appendix D (Fig. J.45).


**Fig. J.45** This erosive wear pattern discloses that a backflow vortex was formed on the concave/visible face of the vane.

### *Zone [3]: inboard/upstream end of wear ring*

A deep, usually polished shroud groove adjacent to the inboard face of the front wear ring face is seen.

#### Cause

Many wear rings are a tight [interference] fit on the impeller. It is common practice to make wear ring larger in diameter than the location seating on the impeller shroud. The reason for this is to leave a rebate or step behind the ring so that, when in due course of wear and tear, they are to be removed, there is an adequate shoulder to allow grip of a puller or lever.

#### Effect

An unfortunate consequence of this is that a sharp step and corner is created in the flowpath. Flow entering the annular wear ring gap thus has to accelerate and can form a Taylor type standing vortex in that corner. As noted previously, the presence of a vortex will significantly increase the local solids count and encourage erosion.

If the corresponding case wear ring face is not flush with the [ideally machined] wetted inner face of the casing, then there is also a risk of similar damage in this area- [see Collector – Zone 19] (Figs J.46–J.48).

**Fig. J.46** The entry vortex, resulting from a vena contracta inflicts substantial damage on the early section of the flow path. Once the flow pattern stabilise, the damage reverts to mild two body wear. The helical pattern is a result of the axial motion of the particles and peripheral motion of the wear ring surface. See also Fig. J.129.

**Fig. J.47** Another example of vena contracta vortex wears. Note similar helical wear pattern.

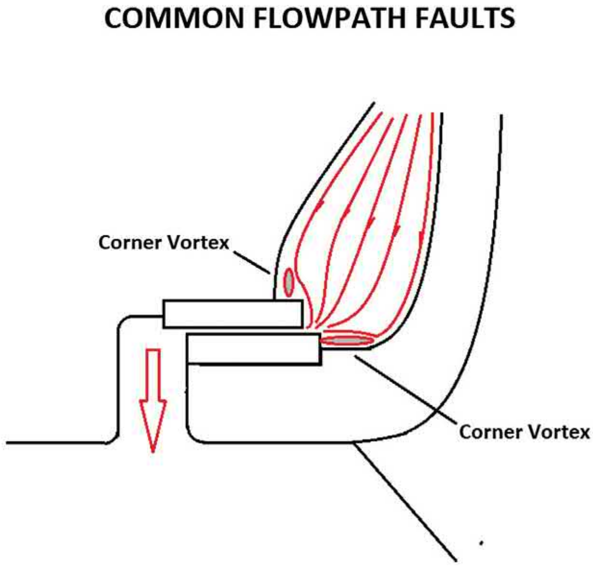
**Fig. J.48** Less severe vena contracta vortex wear. Little sign of helical wear means that the particles have been small relative to the flowpath gap and so only two-body wear has occurred.

### Solution

Reduction [Ideally elimination] of the step will avoid this issue. There are several different methods of impeller wear ring retention in use. So this figure only gives guides as to the elements of an ideal flowpath. Each actual case would need to be reviewed to establish how many, if any, of the features could be assimilated. These features include;

- Gradual change in the clearance between impeller sidewall and adjacent casing wall.
- Smooth transition in outer shroud profile to annular clearance gap.
- If hard coatings are applied, they should 'fade out' at some radial distance up the shroud and not stop abruptly.
- Smooth surface to impeller shroud and casing sidewall.

Note that these comments only apply to managing the flow conditions leading up to the wear ring annulus. However, the flowpath features leaving the annulus is also important, and are covered separately in Collector – Zone 16.



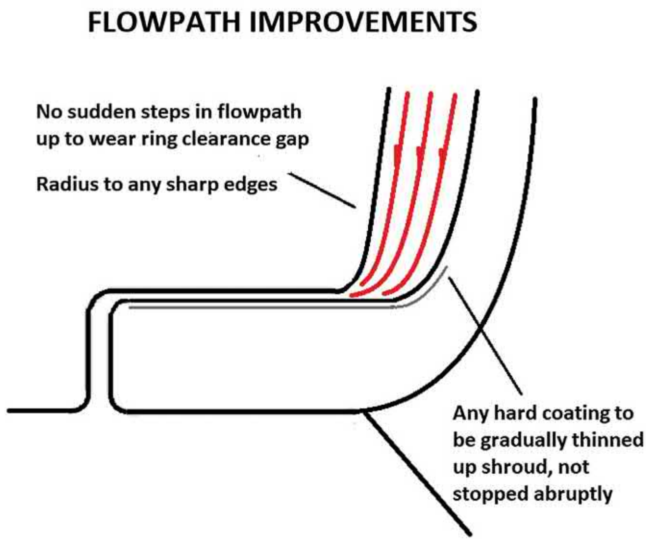




**Fig. J.49** Upper — Most wear rings are mounted in this way so as to assist subsequent removal. But it allows a corner 'Taylor type' vortices to form, in addition to a vena contracta just inside the annular gap. Lower — This sketch shows conceptually the elements of an ideal wear ring mount. There are no sharp edges or turns in the flow. The flowpath area changes progressively and avoids stall cells forming. It is often possible to incorporate some of these features, but not all.

### *Zone [3]: wear ring surfaces*

On pumps where some suspended solids have been foreseen, it has been fashionable to provide clean liquid flushing directly to the wear ring clearance-annulus. The idea here was twofold;

- Firstly to try and create some backpressure and attempt to slow the flowrate across the annular clearance.
- Secondly to try and dilute the solids concentration passing through the annular gap.

#### Causes

Success has been mixed; there are some good examples in existence. However, it seems in most cases that the introduction of a discontinuity in the gap flowpath has produced damaging vortices. Wear ring surfaces are most prone to low impact angle damage. The first picture opposite shows a wear ring from a pump handling small amounts of river silt. Although the conditions largely equate to two-body wear [xref] there is some evidence of three-body wear due to larger particles. The wear patterns also disclose the underlying helical absolute particle motion.

The second picture shows typical flushed wear ring damage. Vortices forming around the liquid injection points cause accelerated wear. For this reason, flushed rings are unpopular nowadays. Most anecdotal experience is that unflushed wear rings are likely to last longer than those that are flushed.

#### Solution

Remove the flush holes. Most abatement success seems to be found with hard coatings or hard materials. The latter group demand great attention to their mounting and retention arrangements (Figs J.50 and J.51).

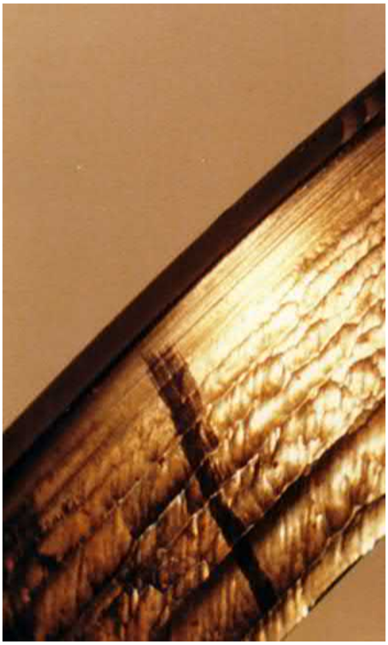


**Fig. J.50** An unflushed wear ring exhibiting a combination of two and three body wear.

**Fig. J.51** This flushed wear ring shows how the injection disturbance is profound and is likely to create many cases of high impact angle contact within the flowpath gap.

### *Zone [4]: pitting damage to impeller outer shrouds*

The impeller shrouds exhibit a pitted appearance that is not characteristic of corrosion.

#### Cause

The spinning impeller shroud disks experience a drag force known as 'disk friction'. Pump designers traditionally treat this as unproductive loss and a great deal of effort has been expended in researching this 'efficiency—reducing' phenomenon. In fact, useful pressure head generation can be a by-product of this phenomenon, viz. the Tesla type of pump. Liquid very close to the impeller shroud outer wall will experience drag forces tending to centrifuge them out towards the rim. The energy transfer process from disk surface to adjacent liquid is complex but seems to consist of closely spaced repeating vortex cells. Of course, the vortex cells create the possibility of solids concentration and enhanced erosion rates.

#### Effect

This mechanism produces low impact angle damage to the shroud surface. The scale of the pitting depends on the hardness of the impeller materials and particles, as well as the surface velocity of the impeller and its surface finish. In the first image, damage intensity can be seen to increase with distance from the shaft centre, i.e., with increasing peripheral velocity. The second image shows a coarser repetitive pattern on both the inner and outer shroud walls.

#### Solution

The process can be slowed by smooth machining of the impeller shrouds. Hard surface coatings resistant to low impact angle attack are also possible (Figs J.52 and J.53).

**Fig. J.52** Low impact angle attack on the bronze impeller shrouded a pump handling very fine river water silt. The damage rate increase with distance from the shaft – in conjunction with increased velocity.

**Fig. J.53** This process pump impeller has suffered low impact angle attack from suspended fine coke particles. *[David Brown Pumps]*.

### *Zone [4]: isolated damage on outer shroud surface*

#### Cause

Isolated areas of damage are quite common and are usually associated with some sort of significant discontinuity on the shroud surface.

#### *Effect*

One of the main messages behind this whole Appendix is that vortices will spell trouble if suspended solids are also present. Here are two examples that illustrate their damaging effect.

This first shows how the discontinuity formed by cast letters on an impeller shroud will generate vortices-probably of the Kamm type. Local penetration of the shrouds can be seen to already be significant, whereas the low impact angle damage suffered by the rest of the shroud surface is very mild if any. Indeed the cast surface finish is still visible.

In the second case, an extremely small discontinuity formed by a surface defect in the impeller casting [a blowhole] has rapidly escalated. Again the low impact angle damage to the rest of the surface is negligible. Even machining tool marks are still evident.

Note that in both these cases, the general direction of the damage marks reveals the outward motion of liquid that is close to the shroud surface.

#### Solution

As with the previous category, smooth shrouds, along with hard coatings are possible solutions (Figs J.54 and J.55).

**Fig. J.54** Discontinuities on an impeller shroud surface can create Kamm type vortices. Here cast identification lettering has generated substantial damage. Elsewhere the original cast surface shows little if any damage [see also Fig. J.130].

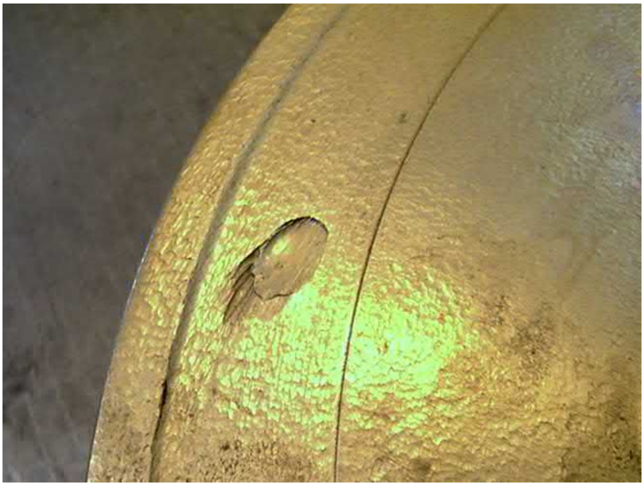


**Fig. J.55** Flow erosion marks on this impeller shroud casting defect disclose outward flow of shroud boundary layer.

### *Zone [5]: groove damage in region of blade outlet edge*

Pump impellers sometimes exhibit marks/stains on the peripheral part of the vane exit tip. In some case, where the suspended solids content is very small, this might only be slight, even after many thousands of running hours. In other cases, where the content is high, this develops into deep grooves after only a few thousand hours.

#### Cause

The flow picture that creates this is difficult to portray in a two dimensional image. As an analogy, imagine one body of flow passing over another. If the flow intersects at a small angle, then the weaker one becomes entrained in the other. However, if they cross at rather a large angle, then a region of shear forms between the two flows. The flow in this transition zone rolls up into filamentary vortices all along the interface between the two bodies. The two main flow bodies tend to continue as separate items, unless obliged otherwise. Damage intensity seems chiefly related to impeller tip speed [pump pressure head] and levels of solids in suspension.

#### Effect

In this instance, the main flow exiting the impeller passage crosses over flow pumped up the shroud walls [and creating disk friction]. Sometime this mechanism is insufficient to cause erosion, but is enough to cause staining or marking of the impeller blade edge. There seems little difference between the damage patterns of single or double suction impellers.

#### Solution

No simple effective solution seems to exist for existing impellers (Figs J.57–J.60).

**Fig. J.56** Stains on this large cooling water impeller indicate early stage of twin vortex erosion.

**Fig. J.57** More developed twin vortex damage on this double entry impeller river water pump. Elsewhere little damage is visible. This indicates the high damage potential of vortices in the flow.

**Fig. J.58** Advance twin vortex damage to this single suction API 610 Process pump handling Catalyst fines. *[Union Pumps]*.

**Fig. J.59** Advance twin vortex damage to this single suction API 610 Process pump handling Catalyst fines.

**Fig. J.60** Early twin vortex damage patterns on this double suction booster pump in offshore oil installation.

### *Zone [5]: pitting damage to vane tip*

Cavitation-like damage is sometimes detected on the downstream peripheral part of the vane exit tip.

#### Cause

At first sight, it might seem that cavitation damage in the impeller rim region is not likely. Cavitation is normally associated with regions of low pressure, whereas the impeller rim will generally be a region of high-pressure head. While this assumption is correct, it is slightly simplistic. The way in which an impeller blade works creates peripheral variations in the pressure distribution around the impeller rim. The reasons for this are touched upon later, but the outcome is a jagged, almost saw-toothed pressure distribution, scattered about some mean value.

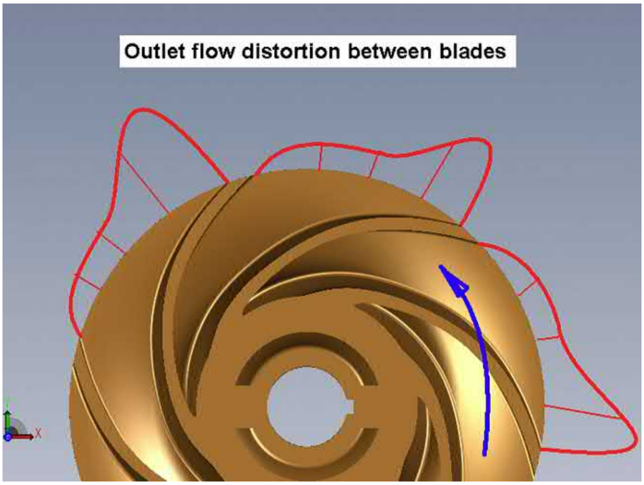



**Fig. J.61** Flow and pressure field surrounding impeller rim is quite abrupt, resulting in severe swings as each blade passes fixed points on the collector.

The natural pressure differences on the convex and concave side of the blade mix as the flow exits the vane edge. However, this flow is squeezed as it passes through the gap between the rotating and stationary collector vanes [s]. This 'squeeze – and – sudden release' mechanism creates pressure pulsations that are radiated out through the pumped liquid.

The low point of this pressure cycle can allow cavitation to occur locally with the associated pitting shown. Obviously gassy liquids will be most prone to gas release at this point. Certain liquids such as Benfield's solution [used in gas stripping applications], can then release acid gas.

One traditional approach to reducing pressure pulsations is to arrange the vane exit edge to be 'skewed' as shown. This non-orthogonal feature means the impeller edge, and its flowpath, will progressively pass any orthogonal collector vane inlet tips. This can result in a much less abrupt encounter than between two orthogonal edges. But it might not be enough to avoid exit tip attack. A more effective approach is to increase the radial gap [B gap] between the impeller vane exit tip and the stationary casing tip. This damage intensity seems so strongly dependent on impeller materials and liquid properties that no firm preventative advice is available (Figs J.62 and J.64).

**Fig. J.62** Cavitation-like markings on outlet edge of impeller vane of this single suction booster pump.

**Fig. J.63** Similar cavitation-like markings to this impeller vane tip.

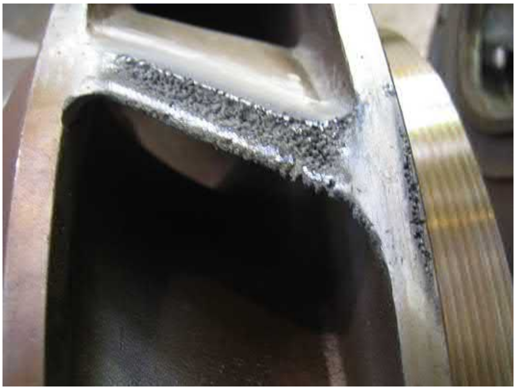


**Fig. J.64** Again cavitation-like markings to the vane outlet tip. Unusually, this also exhibits damage to the rear [hub] shroud of this single entry impeller. The pressure distribution in this zone is quite complex.

### *Zone [5]: impeller shrouds bulged or broken*

Occasionally, impellers may experience 'bulging' of the shrouds, or even breakages.

#### Cause

The way in which a centrifugal pump impeller functions results in higher pressure on its convex surface and lower pressure on its concave surface [see Fig. J.61]. Just before the flowpath enters the impeller, there will be little if any difference, so long as the impeller is outside of the so-called backflow regime. Once the flow leaves the impeller, these two flow stream, of different pressure 'mix out' back again to a uniform pressure field. This mixing process takes a finite distance to take place. At say 120% of impeller diameter, then the two flows will have effectively mixed out. At distances of only few percent, then the mixing process has hardly started and the pressure difference might amount to say 50% of the impeller pressure head. An observer positioned on the casing, and very close to the impeller, could see a very rapid 50% local pressure spike as each impeller blade tip passes by. This mechanism is a large contributor to pump noise as well as vibration. It is also chiefly responsible for the form of failures shown here.

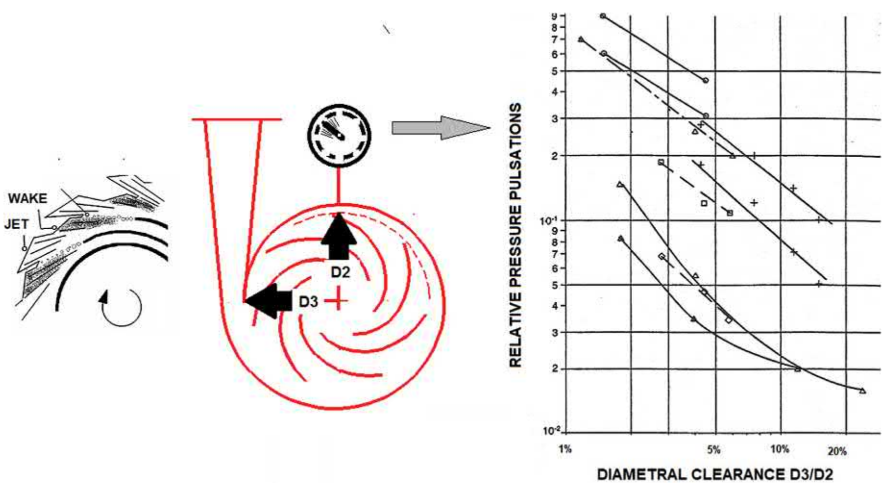



**Fig. J.65** Experience says that the ratio of D3/D2 should equal or exceed 1.10 for volute collectors and 1.03 for multilane diffuser types. This constraint can be violated if the pump is small [Impeller diameter less than say 8 inch and shaft speeds less than 3000 rpm]. *[Sulzer Pumps].*

The rapid rise and fall of pressure with each vane –to-casing encounter causes both of the impeller shrouds to 'breathe' or 'pant'. The gap between the two briefly grows and contracts. At one extreme, this can cause permanent bulging of the impeller shrouds – as shown. At the other extreme shroud failure can occur as a result of high cycle fatigue. The magnitude of such pressure pulsations are, all other things being equal, dependent on how close the impeller is to the collector tips.

### Solution

As a general rule, the distance from the shaft centreline to the collector tip *[volute or diffuser]* should be more than 103% *[D3/D2>1.03]* of the impeller vane tip radius. This should suppress the pulsation levels to manageable proportions. But of course the shrouds themselves must also be adequately proportioned in the first place (Figs J.66 and J.68).



**Fig. J.66** These shows show plastic deformation under the cyclic 'panting' pressure pulsations. This occurs as each vane tip passes the closest collector component – see Fig. J.67.

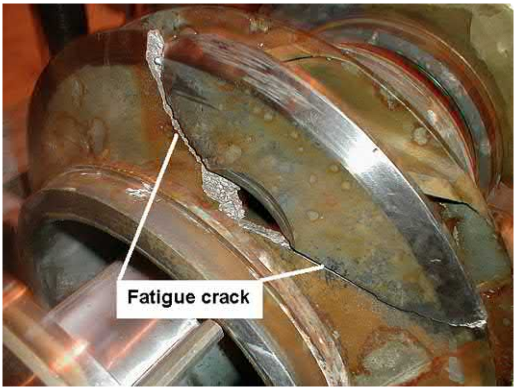




**Fig. J.67** If subject to pressure pulsations for long enough, then the impeller shrouds can fail in high cycle fatigue. This is atypical fatigue pattern and extends from one vane tip to the next-across an unsupported span.

**Fig. J.68** Another example of high cycle fatigue failure caused by impeller pressure pulsations.

### *Zone [7]: general damage to inlet edge*

New 'clear-liquid' pump impeller vane inlet edges are normally smooth and rounded. This creates little disturbance or resistance to the incoming flow. After some service, the edges sometimes appear polished and/or flattened. A ridge line appears along most of its length.

#### Cause

This is usually evidence of high impact angle damage from suspended solids.

#### Solution

Realistically, the inlet edge has 'self-designed' itself to the optimum shape in these particular circumstances. Attempts to re-shape it to the original smooth taper will just result the process repeating in further service. In practical terms, the 'self-designed' ridge profile will usually be acceptable —except if the impeller simultaneously suffers cavitation damage. Such occasions will be very rare, but in any event the cavitation issue will most likely be more serious.

Local coating is a possibility, but the material used must be specifically resistant to high impact angle attack, not low angle (Figs J.69—J.72).

**Fig. J.69** An example where the inlet edge has lost its smooth profile due to high impact angle erosion.

**Fig. J.70** Further example of high impact angle damage. The sharp inlet edge shape has been largely destroyed.

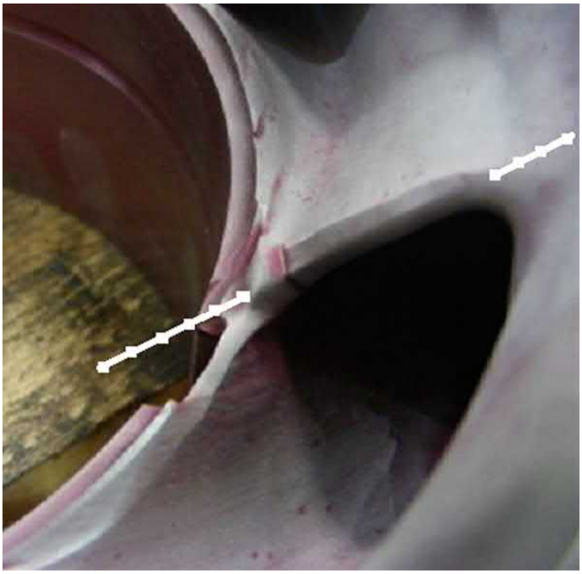


**Fig. J.71** On this radial flow impeller, the edge has not been eroded to a flat shape. Instead, a wedge profile has developed. The ridge of the wedge is highlighted by the dotted line.

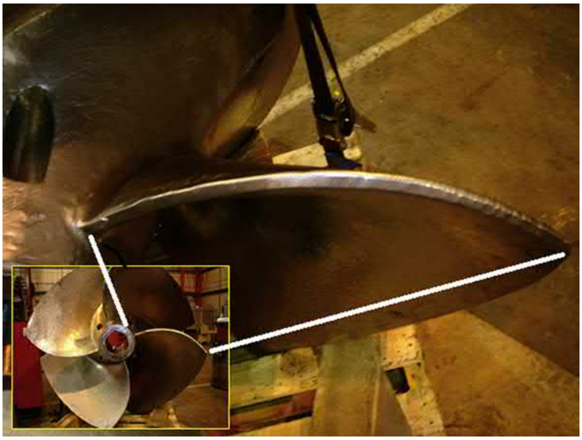


**Fig. J.72** On this example, the edge has not been eroded to a flat shape. Instead, a wedge profile has developed on this axial flow impeller inlet edge. As before the ridge is evident. *[Sulzer Pumps].*

### *Zone [7]: corrugated gouging to vane surface*

The vane surface of most impellers will only suffer from low attack angle erosion. Generally there will be little underlying pattern to this. Occasionally this is replaced by a strong repetitive pattern. This erosion pattern tends to be associated with machines of high inlet tip speed – say over 80 m/s or thereabouts.

#### Cause

The previous section described how impeller vane inlet edges can be 'blunted' as a result of high impact angle damage. A ridge line forms along the inlet edge. The 'as new' vane edge shape will slice into the oncoming liquid with low disturbance to the flowpath. Even when the ridge line develops, the flow disturbance may not be great.

However, under certain speed and granulometry conditions the included angle of the ridge becomes small and almost flattened. When the edge becomes flattened, then the flow disturbance can become significant. In the extreme, Von Karman –type vortices can be alternately shed from each side of inlet edge [see Appendix J] and then swept along the vane surface. Each vortex is of course capable of entraining and concentrating a substantial quantity of abrasive solids. This discloses the vortex pattern, as in this example here.

#### Solution

The most obvious solution is coating with a material resistant to high impact angle attack – as with the previous category. Maintaining the 'as new' profile will suppress the tendency for this damage pattern to appear (Figs J.73 and J.74).

**Fig. J.73** Vortices originating from the vane inlet edge [that has lost its profile due to high impact angle damage] corrugate the downstream vane surface. This constitutes low impact angle damage.

**Fig. J.74** This re-touched image shows a similar pattern at lower liquid velocity. This is more typical low impact angle damage. *[Sulzer Pumps]*.

## Single or double suction semi open and open

Semi Open and Open type impellers are susceptible to the same erosive and cavitation damage patterns as the shrouded impellers. However, as mentioned earlier, such impellers are generally equipped with back 'pump-out' vanes. They have a dual purpose of reducing pressure in the seal chamber area as well as offsetting and reducing unbalanced hydraulic axial thrust.

Occasionally, shrouded impellers are also furnished with back pump-out vanes, chiefly for reasons of axial thrust reduction, but occasionally to help expel suspended solids from the region of the seal chamber. Very occasionally, shrouded impellers are also furnished with pump-out vanes to the front shroud. Here their purpose is to help expel suspended solids from the wear ring annulus region.

Semi open and open impellers are prone to suffer most of the wear symptoms experienced by the single entry shrouded designs previously described. In addition to these wear zones highlighted, additional sites need to be illustrated. These result from the presence of vanes in or near the main flowpath.

**10]** Rear 'pump-out vane' tips, near shroud rim

**11]** Inner diameter of pump-out vanes

**12]** Vane tip surface forming clearance gap with casing wall. This damage zone is often associated with casing damage Zone [22] (Figs J.75–J.77).

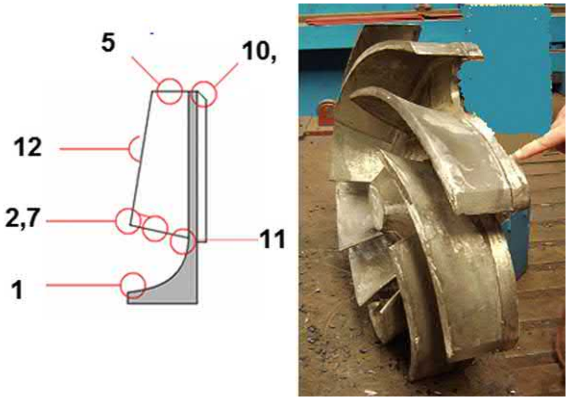



**Fig. J.75** Common areas of damage in semi open and open impellers.

FRONT VANES BACK PUMP-OUT VANES

**Fig. J.76** View of Front and Back vanes of Radial flow semi open and open impellers. The back vanes shown here more or less mirror the shape of the front vane spiral and extend out to the same maximum diameter. In some case, the back vanes are simply straight and radial. Occasionally, vanes of either type do not extend right to the same maximum diameter.

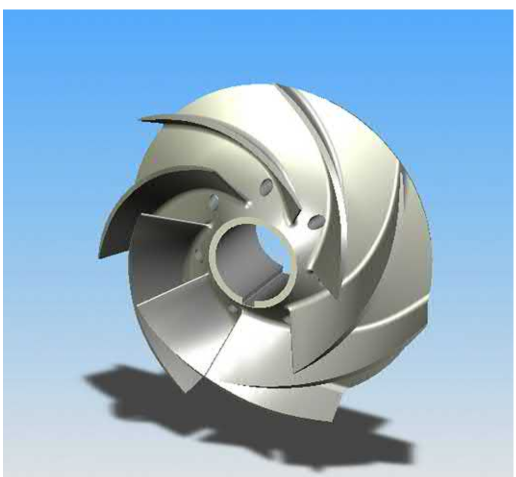


**Fig. J.77** Front view of mixed flow open impeller. Such impeller may or may not be furnished with back pump out vanes.

### *Zone [10]: damage in region of back 'pump-out vane' tips*

#### Cause

As previously described, the purpose of the back vanes is chiefly to modify the pressure distribution acting on the back shroud in such a way that the nett axial thrust is reduced. In addition, the static pressure in and around the seal chamber is reduced. Finally, the vanes can tend to centrifuge solid particles away from the seal chamber area and further improve its environment. The ribs work by creating a modification to the natural swirl pattern that exists in the gap between the rotating impeller and the static casing.

#### Effect

In doing so, Kamm type vortices are formed downstream of the vane. These have the ability to entrain and concentrate suspended solids and this causes the erosion patterns shown. Furthermore, a form of von Karman vortex is formed, which sends packets of concentrated solids out into the mainstream and causing damage some way away from the source.

#### Solution

Both the Kamm vortex and the von Karman vortices can be almost eliminated by streamlining the downstream side of the radial rib. In this way, the responsible driving force is substantially weakened. There is still the probability of diminished damage around the vane tips – as shown in the second image opposite.

It is a mistake to truncate the vanes to a smaller diameter than the shroud rim, because this then leaves a large area for the von Karman fed vortex packets to erode – third image opposite. For this reason it is also a mistake to try and clean up damaged vanes by machining, since it leaves a vulnerable surface (Figs J.78–J.80).

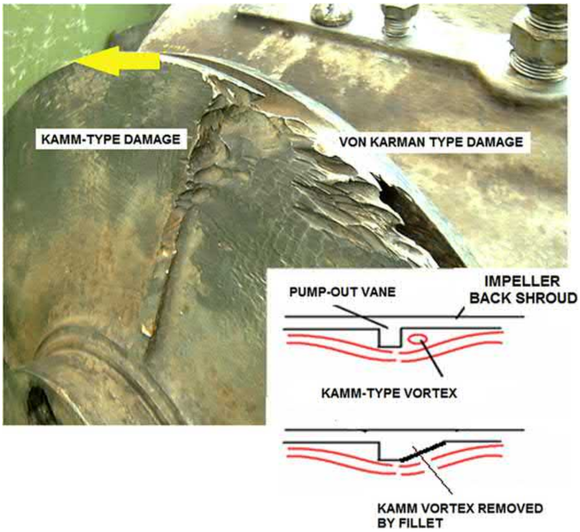



**Fig. J.78** Impeller erosion damage resulting from two vortex sources. *[David Brown Pumps].*

**Fig. J.79** Even if the downstream shape of the back pump out vanes is streamlined in an attempt to supress Kamm type vortices [as here], a form of Von Karman type vortex can form at the very tip. If these vanes do not extend to the maximum, then this can cause significant damage — see next image.

**Fig. J.80** This shows the damage that Von Karman type vortices can cause if the back pump out vanes does not extend to the maximum diameter.

### *Zone [11]: damage in region of 'pump-out' vane inner diameter*

#### Cause

On occasions cavitation damage is seen at the inner diameter of back pump out vanes. These vanes have a dual purpose of reducing pressure in the seal chamber. They have the added benefit of helping to offset and reducing unbalanced hydraulic axial thrust.

Should the NPSH [A] be low enough, then the reduced pressure in the seal chamber area can go below the vapour pressure and cause local cavitation in and around the vane tips. This damage may extend to the adjacent case walls, as seen here.

#### Solution

The principal solution would be to reduce the vortex strength, by, for example increasing the gap between the impeller back vanes and the adjacent casing sidewall. Since this will also increase the nett axial load on the thrust bearing, this step should not be taken without consultation with the manufacturer. A further risk is introduction of tip leakage cavitation [as with ZONE 12].

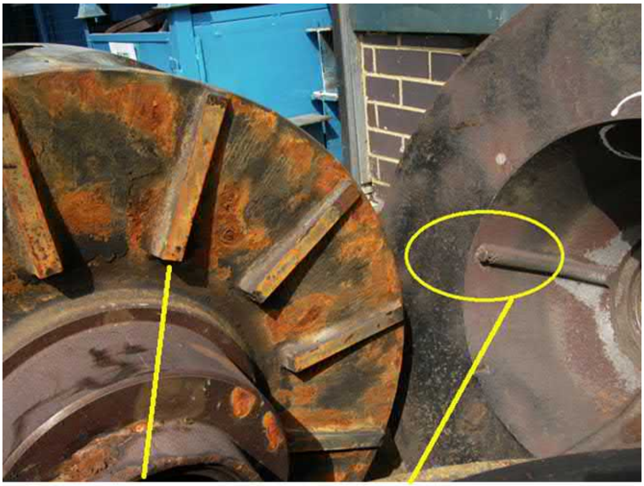



**Fig. J.81** Certain conditions can allow cavitation damage at the inlet tip of the pump-out vanes and corresponding damage to the casing.

### *Zone [12]: pitting damage to impeller vane near midpoint*

The front edge of semi-open impeller vanes are most often machined, so as to allow precise setting of the clearance to the casing wall. Cavitation pitting is sometimes seen on this machined surface – mainly part way along the vane length. There may be damage to the casing at adjacent radii [see also Fig. J.126].

#### Cause

In order to function, the impeller vanes must create a pressure difference between the convex and concave surfaces. There must also be a small but finite gap between the rotating impeller and the casing. Hence a small flow will continuously pass over the front edge. In passing through this gap and then suddenly expanding, the liquid will experience an equivalent pressure drop, similar to the orifice analogy [Appendix E]. Cavitation growth and collapse is thus possible, with associated collapse on the casing wall. In most designs, the maximum pressure difference over the two surfaces builds up to a point near the vane mid length and then declines. So damage pitting is most likely at or near this point. Casing damage will be at a similar radius.

#### Solution

This damage form seems most likely if or when the clearance gap is too large –either due to wear or assembly setting (Figs J.82–J.84).

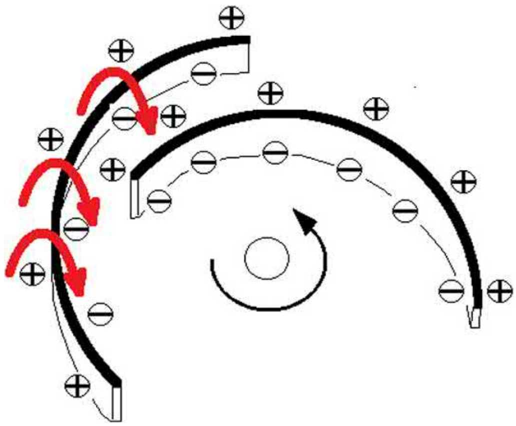


**Fig. J.82** In order for an impeller to function, it must create a pressure difference between the concave and convex surface. With open and semi-open impellers, a gap will exist between the rotating impeller and the stationary casing. The pressure difference will create flow across this gap.

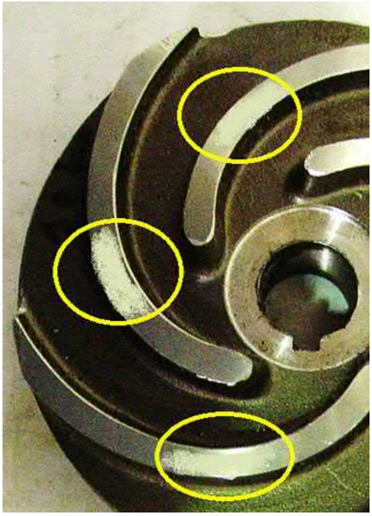


**Fig. J.83** In some conditions, this gap flow will result in cavitation. Localised damage will be evident on the machined face of the impeller vanes. The cavitation cell rotates with the impeller and cause corresponding damage over 360 degrees of the casing. See Fig. J.129.

**Fig. J.84** Equivalent gap flow damage on a mixed flow impeller.

# MULTISTAGE

## Multistage: single entry, fully shrouded

Multistage impellers are vulnerable to the same damage patterns as single stages. However, there are some patterns almost exclusive to multistage impellers. This is a consequent of leakage flow paths which are generally different and unique. Small multistage pumps under [say] 150 kW will usually be furnished with impellers having back and front wear rings. The earlier remarks relating to single stage impellers all apply. Larger and more powerful machines have a different leakage flowpath. They are usually furnished with large front wear rings [typically 15–20% larger than the inlet eye diameter], and back wear rings only a little larger than the shaft diameter.

As in single stage machines, flow at the impeller rim is still obliged to pass **down** the front shroud and through the wear rings en route to the lower pressure zone. But not down the back shroud. Instead, flow that has passed out through the collector, en route to the next stage has gained some static pressure head after leaving the impeller rim. So it will tend to flow back to the nearest zone of lower pressure head – that will be at the impeller rim. This is where multistage marines can differ. Such a route obliges such flow to pass **UP** the back shroud *[except last stage where it flows to low pressure via some sort of thrust balance compensator]* (Fig. J.85).

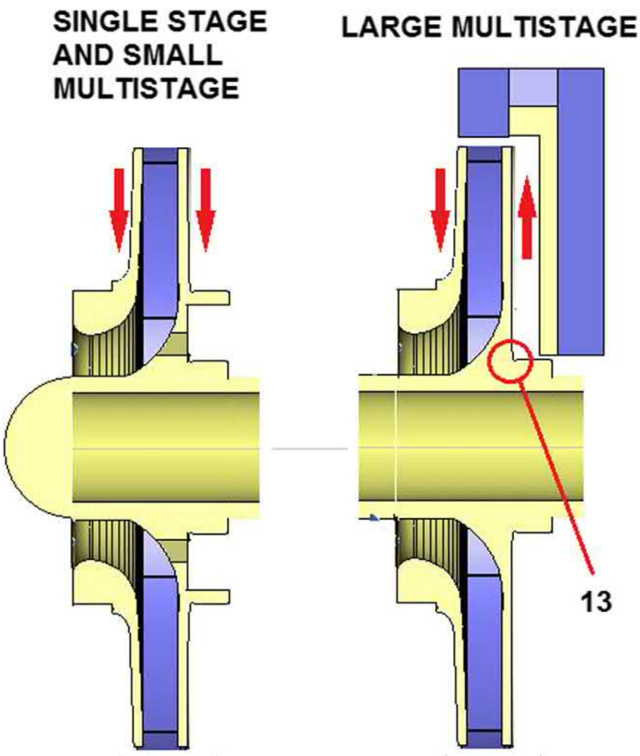



**Fig. J.85** The leakage flowpath of a simple single stage or small multistage impeller differs from that of larger multistage machines. This results in a small difference between the pressure profiles on the back shroud. In general the average back shroud pressure will be relatively higher in the case of larger multistage layout compared to an equivalent single stage layout. Though this difference is small, it acts over a large 'piston' area to result in a significant increase in axial load.

## *Zone [13]: inboard area of multistage back hub*

Multistage machines may exhibit erosion in or around the inboard edge of the rear hub.

### Cause

The sudden expansion of the outward flowpath can generate standing corner vortices. These are susceptible to entraining suspended solids. Thus the corner formed by the back hub/ring and the back shroud outer surface can show various degrees of Taylor vortex type erosion. This first Figure shows damage at an early stage *[Note that the cast in lettering is still visible!]*. If allowed to get much worse, then it will ultimately pierce this shroud, as shown in the second image. In passing through the annular back ring gap, the liquid picks up some angular velocity and so leaves with some swirl velocity. This helical motion is clearly disclosed (Fig. J.86).

### Solution

Careful management of the flowpath exiting the fine clearance can help a great deal. Replacing sharp corners with matching and blended radii helps to reduce or even eliminate the culprit vortex. Hard faced components have also been proven successful in severe applications (Fig. J.87).

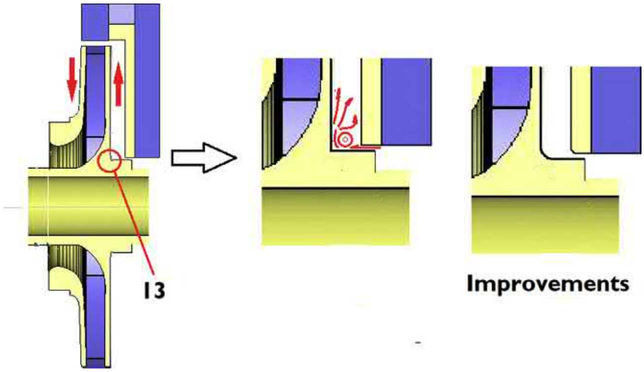



**Fig. J.86**

**Fig. J.87** Flow exiting from the back bush clearance of larger multistage pumps can generate a stable corner vortex and hence an erosion potential.

## Multistage: single entry semi open

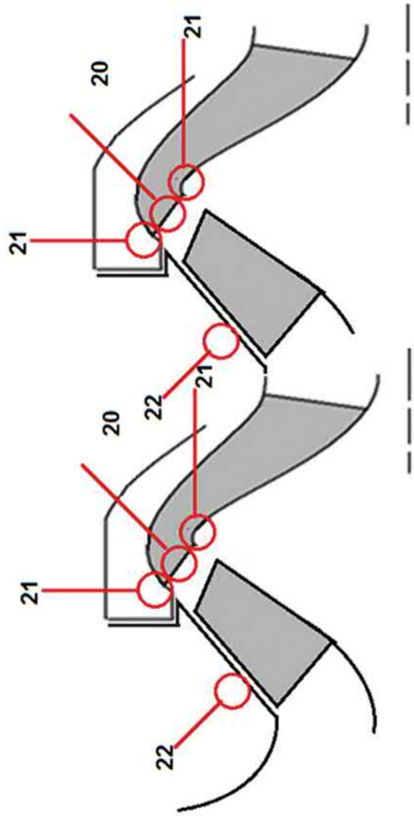
Multistage versions of these Low or Medium specific speed hydraulic designs are almost always only seen in a vertical shaft configuration. The benefits of vertical shaft pump in application where NPSH is low have already been discussed earlier. Historically, these comments chiefly applied to fully shrouded designs and, by implication, tend to be clear liquid applications.

One benefit of open and semi open designs has already been discussed which is that efficiency of a worn pump can be somewhat retrieved. This is accomplished by adjusting the whole rotating element towards the casing and hence regaining the running clearance with the casing/casing liner. Vertical open or semi open multistage designs often find favour where the liquid is not quite clear and or some fine silt may be expected. Being able to regain efficiency in such applications is thus very attractive.

Typical wear points are:

**20** Midspan of collector vanes.
**21** Connection area of collector vanes and casing walls.
**22** Clearance gap between impeller and casing.

These are discussed later in the respective Zone sections (Figs J.88 and J.89).




Mixed Flow - Multistage Vertical

Fig. J.88

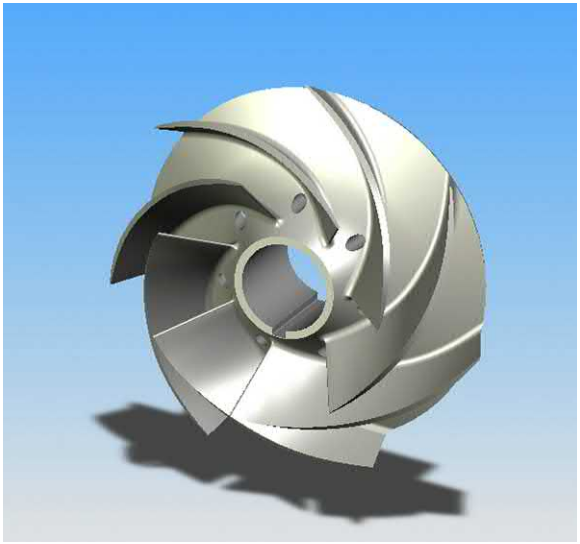


Fig. J.89

## 2. CASING/COLLECTORS

### COLLECTOR: VOLUTE-SINGLE STAGE AND MULTISTAGE

The majority of single stage pumps are furnished with a spiral/volute collection chamber surrounding the impeller. Many high energy multistage pumps are also built this way. As explained earlier, in low specific speed designs [see Ref. 18] the collector casing is likely to suffer more damage than the impeller *[This is due to the relatively higher velocities.]*. Areas likely to show damage are;

**8]** Casing wall upstream of impeller
**9]** Ribs or vanes upstream of impeller
**14]** Holes or tap-off points connecting into the main flowpath
**15]** Casing wall adjacent to impeller back and front shroud
**16]** Casing in vicinity of wear ring annular gap exit.
**17** Region in and around casing cutwater lip[s]
**18]** Side wall maximum diameter, adjoining volute passages
**19]** Pocket in sidewall gap [Where present]
**22]** Casing sidewall or wear plate; Semi open and open impellers only (Fig. J.90).

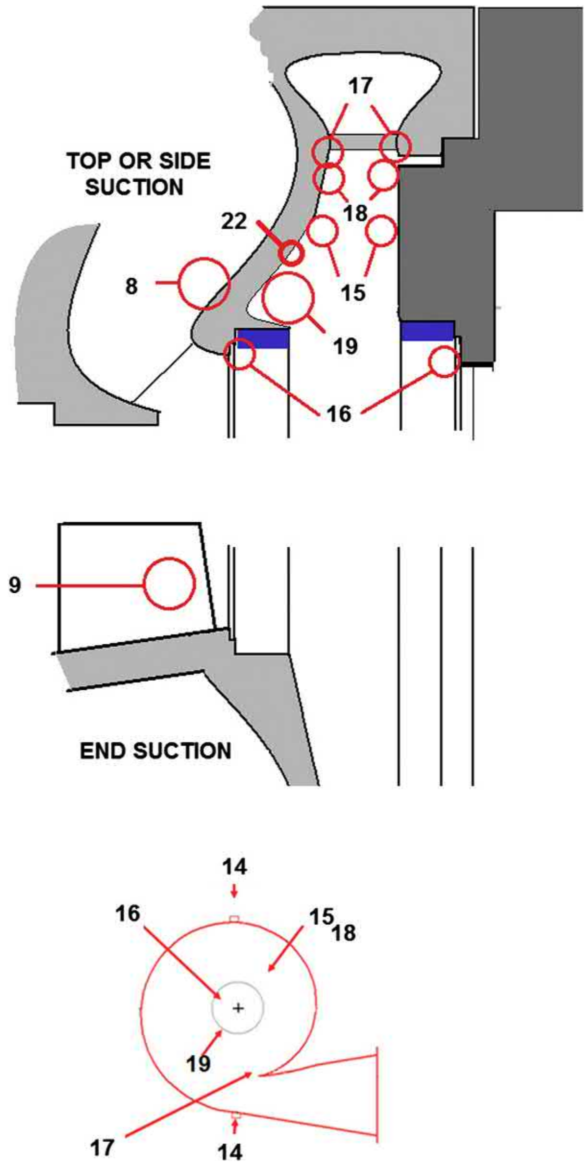



**Fig. J.90** These are the most common damage sites in spiral Volute type collectors.

## Zone [8]: pitting to casing wall upstream of impeller

### *Cause*

Conventional wisdom describes cavitation as originating inside the impeller. However, if the pump is operated outside of its COMFORT ZONE and inside the backflow regime [Appendix D], then this may not be true. In such conditions, cloud and filamentary cavitation may be created in the rejected flow and so upstream of the impeller. They may subsequently collapse on the inlet duct casing walls. This form of damage is particularly common in pumps handling gas cleaning solutions [Benfield's etc.] since acid gas also comes out of solution to enhance the pitting rate. While cavity collapse damage will appear on the upstream casing, it can also carry over and partly into the impeller. Here the damage can paradoxically appear upstream of the impeller inlet edge *[see corresponding ZONE 2-Damage present on the flowpath before entering vanes].*

### *Solution*

Backflow avoidance is the surest way of avoiding this issue. Where such damage is present, the pump is oversize for the duty. Impeller needs rerating to a lower flow and in doing so shift the onset of backflow to below the minimum routine process flow. Increasing the NPSH [A] significantly is another option but this is rarely practical. This condition can also arise if two pumps are operating in parallel when in fact one pump could mostly handle the designated flow (Figs J.91 and J.92).

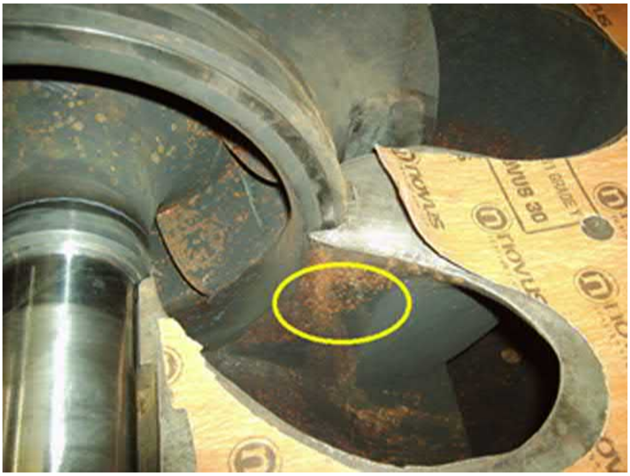


**Fig. J.91** Backflow emerging from a pump that is operating at low flow, can produce cavitation damage on the upstream surface of the flow path [see also Fig. J.24].


**Fig. J.92** Further example of backflow induced cavitation damage on the casing wall upstream of the impeller [see also Fig. J.24].

## Zone [8]: gouging damage to casing wall upstream of impeller

Circumferential gouging marks are sometimes seen in this region of pumps handling abrasive material. One might expect that any that gouge marks would be axial and along the duct wall, reflecting main inflow direction. Why is the damage circumferential?

### *Cause*

If a pump is operated outside of its 'COMFORT ZONE' and inside the backflow regime [Appendix D], then this damage may appear. It is associated with inlet backflow escaping from the impeller and spiralling back along the pipe wall. To do so, the backflow has to leave at a velocity in excess of the impeller inlet tip speed. This often involves high velocities, and if erosive particles are entrained, this results in high erosion potential. Compare with the damage pattern in the next section. In that example, the anti-swirl vanes largely destroyed the spiralling backflow.

### *Solution*

If this damage form is present, the pump is oversize for the duty. Impeller needs rerating to a lower flow and in doing so shift the onset of backflow to below the minimum routine process flow. This condition can also arise if two pumps are operating in parallel when in fact one pump could handle the designated flow at that time (Fig. J.93).

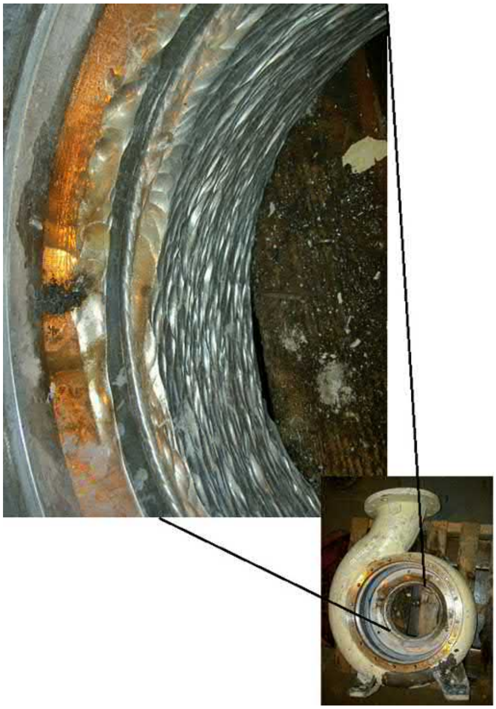


**Fig. J.93** Erosive solids inside the Swirling backflow emerging from the impeller of this pump has caused circumferential gouging of the casing inlet duct.

## Zone [9]: damage to inlet flow straightener vanes

### *Cause*

When pumps are operated at relatively low flows the backflow phenomena [Appendix D] sets in. This may extend a significant distance into the casing inlet duct and against the main inflow.

### *Effect*

This picture looks back upstream from the impeller and shows erosive damage to only one side of the inlet straightener vane. Backflow is ejected from the impeller at roughly the vane-setting angle and with the same sense of rotation. So only one side of the vane is damaged. In this case, the spiralling flow has also etched a pattern of low impact angle attack onto the outer wall of the inlet duct passage. Contrast with the damage shown in the previous image. There the spiralling backflow was allowed to develop without restriction.

### *Solution*

The solution is to replace the impeller with a design flow more suited to the application. In both the above examples, the pump impellers were rated for flow much higher than actually required. In normal operation they were thus operating at partial capacity and hence the risk of backflow occurred.

Sometimes two pumps are operated in parallel when the process needs could actually be satisfied with only one. Each pump thus becomes oversize and this sort of risk arises. There is usually good and valid reasons for running two pumps in such cases, but the hidden risks should be appreciated (Fig. J.94).

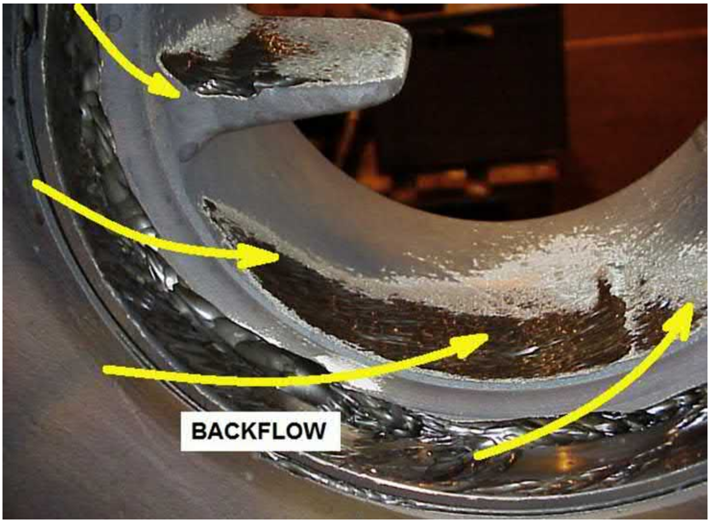



**Fig. J.94** Another example of backflow induced erosion damage upstream of the impeller. Here the anti-swirl vane has helped minimise the extent of damage, but has suffered in the process. *[David Brown Pumps].*

## Zone [14]: damage where vent and drain connections penetrate hydraulic profile

### *Cause*

An earlier section described that standing vortices [probably of the Kamm type] will form downstream of any sudden discontinuity in the liquid flowpath. Drilled 'in' or 'out' holes are the most common candidates. But they are not the only source. Cast identification lettering is another common cause *[see ZONE 22 Damage to casing sidewalls]*.

### *Effect*

Solids concentration naturally follows any vortex formation, erosive damage can be expected.

### *Solutions*

Improvement of the flowpath by removing discontinuity is the general approach. With casing connections, this is not so easy. Casing connection bore sizes are most often selected to provide a more robust external mechanical design, resistant to abuse. One solution to persistent/significant vortex erosion is to reduce the hydraulic discontinuity, but retain the robustness by inserting a smaller bore plug. The smaller bore produces a less energetic vortex and reduces the damage rate. While effective, this approach needs great care. Apart from the obvious increase in vent and drain times, there is a blockage risk with waxy or coking liquids. The pros and cons should be carefully considered (Figs J.95 and J.96).

**Fig. J.95** The flow disturbance by drilled tapping can be quite significant and lead to vortex flow damage.

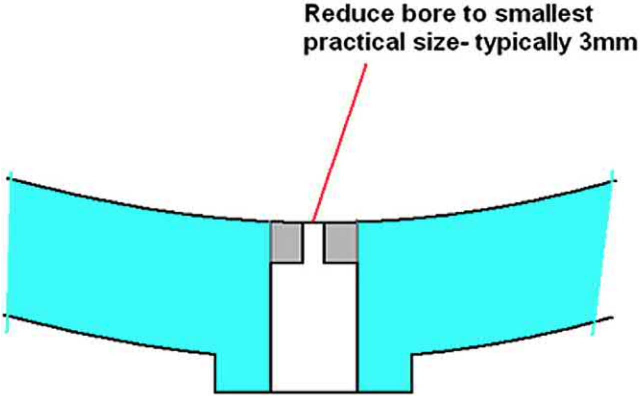



**Fig. J.96** The sketch below shows how this risk can be reduced. Such a solution is impractical if the liquid is viscous/waxy or contains significant solids.

## Zone [15]: damaged casing wall adjacent to impeller back and front shroud

The entire case sidewall exhibits gouging marks in a pattern loosely spiralling inwards. Typically the gouge marks show a repetitive pattern of constant frequency.

### *Cause*

Liquid in the sidewall space between the casing and the shrouds of closed impellers tends to follow a spiralling inward path. If abrasive solids are present, the conditions within the sidewall space are usually conducive to low-impact angle attack. There will be a tendency for the erosion to become less intense as the particles get closer to the shaft. This is because the particle linear velocity will decrease with radius. However, in the wear ring area, the intensity may increase if there are other erosive mechanisms present such as Taylor type vortices [see Zone 16, 19].

### *Solution*

The most obvious solution is to hard coat the shroud and casing sidewalls. The coating must be particularly resistant to low particle attack angle. This will only solve the low impact angle problem. If Taylor vortices also exist then their effects may only be partly fought by coating (Figs J.97 and J.98).

**Fig. J.97** This is a classic example of low impact angle damage. *[David Brown Pumps].*

**Fig. J.98** Further example of low impact angle damage. *[David Brown Pumps].*

## Zone [16]: damage in region of front wear ring flow-exit

### *Cause*

Common misconception is that this sort of damage is simply caused by liquid directly 'jetting' out of the wear ring and impacting the casing adjacent to it. Actually, this is an oversimplification of the cause. Normally the flow leaving a wear ring annular gap also has a strong helical motion imparted to it by viscous drag effects. It then expands into a cavity space that is much larger. However, it is unable to diffuse suddenly enough to follow the new flowpath boundary. Consequently, a number of standing/Taylor type ring vortices will be obliged to form, particularly in the corners of the cavity. Obviously, any solids-enriched vortices have a heightened potential for erosion damage. Were the damage due to a once-only 'jetting' impact, it would already be significant. But the abrasive particles are instead trapped and accumulate in the vortex core that spirals around the periphery of the cavity *[Driven by the helical flow pattern emerging from the clearance annulus.]*. Thus, the particles impact many times before escaping on down into the flowpath. This greatly increases their damage intensity.

### *Solution*

The usual reason for providing such a larger space is to allowing tooling access behind the rings. This is necessary when the time comes to remove and replace them. However, this space is often very generous because there are seldom strong reasons to do otherwise. But in the case where suspended erosive solids are unavoidable, then this space must be better managed.

- Sharp corners must be avoided so as to discourage corner vortex formation.
- The flowpath should avoid sudden area changes since they also encourage vortex formation (Figs J.99–J.102).

**Fig. J.99** Classic Taylor—type ring vortex damage patterns related to front wear ring. Note also Horse-Shoe Vortex damage around casing cutwater/lip. The impeller has been operating on the boundary of its Backflow regime. Consequently, particles emerging from the impeller wear ring have been carried upstream a little. Finally, see that there is corner vortex damage to the casing at its interface with its wear ring outer diameter. This is because the wear ring was not flush with the casing wall [see also Fig. J.49].

**Fig. J.100** Further example of front wearing gap ring type vortex damage. Also corner vortex erosive wear at interface of casing and wear ring.

**Fig. J.101** Another example of damage in the front wear ring zone. First there is the Taylor type vortex damage as flow leaves the wear ring annulus. Then is highlighted the corner vortex formed by protruding case wear ring.

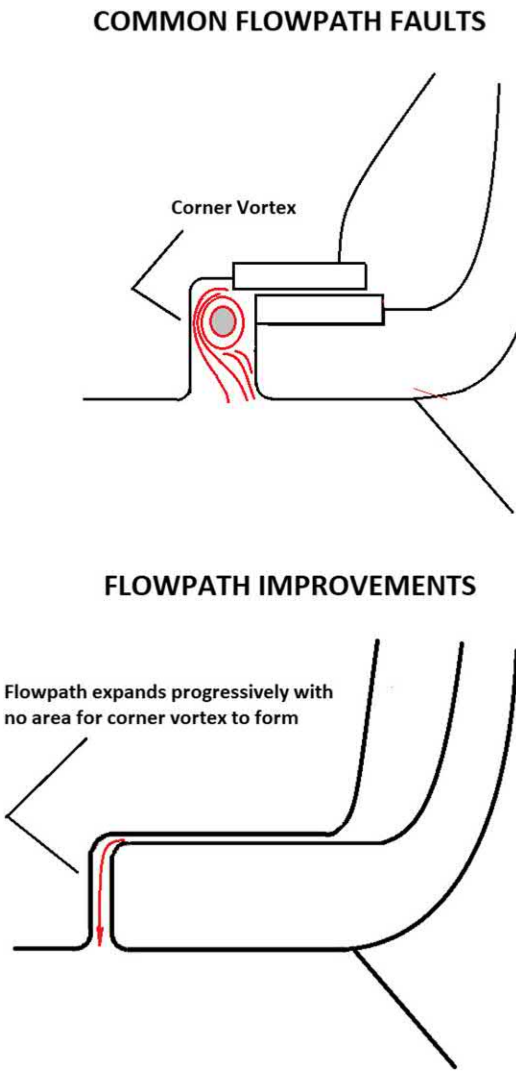



**Fig. J.102** Flow abruptly emerging from the front wear ring clearance annulus has a helical velocity component and helps form a Taylor–type vortex in the cavity corner. Ideally, the flow path exit should be less abrupt and the cavity space restricted. Neither may be practical in some cases.

## Zone [16]: damage to back wear ring flow exit area

### *Cause*

A very common problem in overhung impeller pumps is of severe wear on the area adjacent to the back wear rings. The root cause mechanism is the same as with the front wear ring zone, described in the preceding section *[solids become entrained and concentrated in a 'Taylor type' ring vortex]*. But there is a subtle difference and that results in the damage intensity usually being more severe.

### *Effect*

In most cases there will be a significant difference between the maximum and minimum inscribed radius to the flowpath, when compared to the front wear ring flow field. This gives and larger volume and strength to the vortex formed in the impeller—to-casing sidewall space. This results in a higher proportion of suspended solids being recirculated round in the sidewall vortex. So any solids that do escape inwards from the Taylor type corner vortex, stand a good chance of being transported back again in the sidewall vortex (Fig. J.103).

Even though the back and front wear rings might be of identical or similar diameter, more intense damage will be experienced in the wear ring flow exit area of the back wear ring.

### *Solution*

It can be very difficult to discourage or prevent such Taylor type ring vortices forming in pumps of traditional design.

Installation of anti-swirl ribs in the impeller —casing space for example simply moves and concentrates the problem to their erosion.

Careful management of the flowpath exiting the wear ring clearance annulus is also important. Sudden flowpath area expansions should be avoided, along with sharp corner radii. But at best this will only help to dilute the problem.

The erosion intensity at this location can be very severe. One installation, known to the writer, used an ultrasonic probe to help assess the cover material thickness and hence deduce the erosion rates. Another approach is to add solid wear resistant inserts-either separate or integral with the case wear ring (Fig. J.105).

A proven abatement is to connect the ring vortex area directly to a lower pressure zone [pump suction for example] and to continuously bleed away the ring vortex core contents *[The connection* ***must*** *be tangential to the*

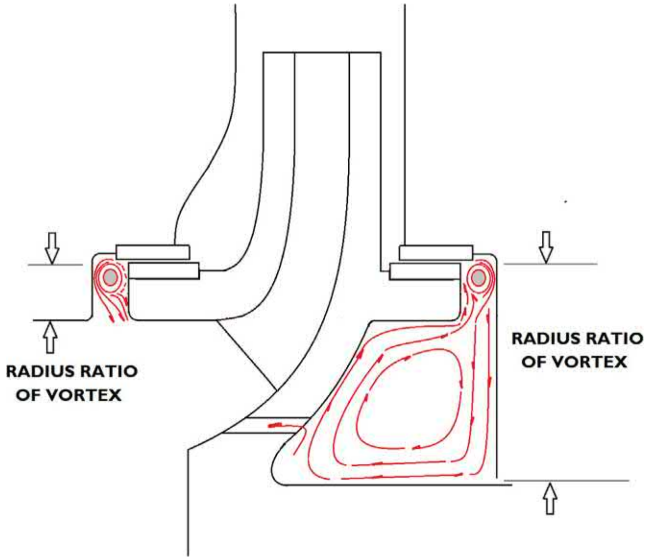



**Fig. J.103** The vortex that forms in the rear sidewall space has a large difference in inner-to-outer radius, relative to the shaft axis. This helps centrifuge more solids into the wear ring exit zone than at the front. Consequently, erosive damage is usually more severe than for the front wear ring, even though both rings they may often be the same or similar diameters.

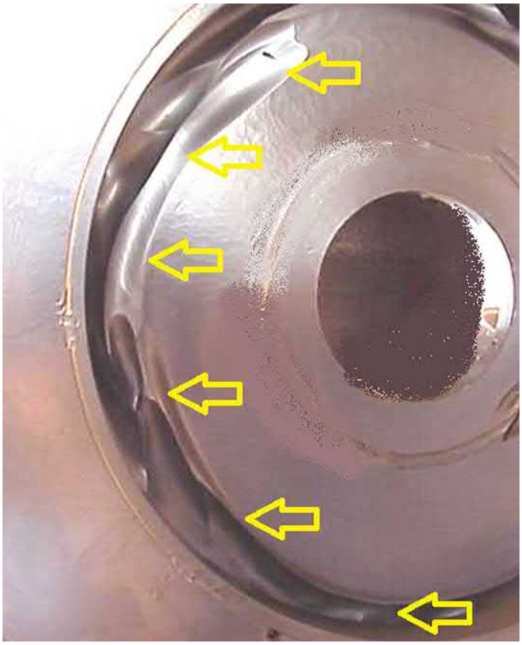


**Fig. J.104** Severe Ring-type vortex damage related to back wear ring gap exit flow and casing cover of API 610 process type pump [note this picture has been re-touched for clarity]. See Fig. J.131 for unretouched image. *[Union Pumps].*

*cavity so as avoid sudden turns in the flowpath. Sudden turns would induce secondary vortices.]*. This largely negates the build-up of solids concentration and makes the vortex much more benign in terms of erosion [Fig. J.106].

This is an appropriate point to mention the grub screws that are frequently used to retain case wear rings [see Fig. J.131].

**Fig. J.105**

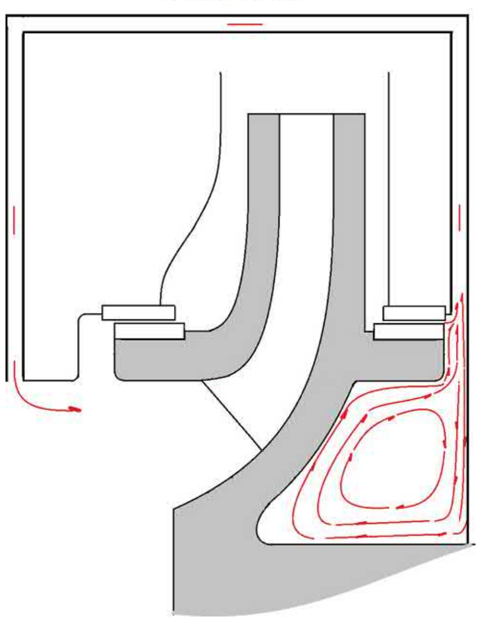



**Fig. J.106** Tangentially connecting the zone that is prone to ring-type vortex damage directly to the suction weakens the vortex formation. It also helps avoid solids accumulation in the vortex.

## Zone [17]: cutting erosion to junction of casing wall and cutwater/tongue

### *Cause*

This Appendix earlier described the mechanism of horseshoe vortex formation. One of the most likely areas for this to occur is where the volute casing cutwater or lip/tongue intersects with the collector passage sidewall.

### *Effect*

The consequent erosion damage often proceeds at roughly right angles to the flowpath and hence through the wall of the casing. Since this can be quite a dangerous form of damage, I include several pictures so as to help its identification.

### *Solution*

The most direct solution to this problem is to ensure that in the immediate vicinity of where the cutwater joins with the passage walls; the intersection sweepback angle is less than, says, 15 degrees. At some distance from the walls, the 15-degree criterion can be relaxed. From a practical point of view, it is often easiest to describe the cutwater as a semi-circle rather than an orthogonal line. This is easily complied with in pumps of low specific speed, where the volute passage width is small relative to the impeller diameter. In the case of rather wide passages, where the ratio of 'b2/D2' exceeds say 0.20 to 0.25, a hybrid of orthogonal edge and 15 degree sweepback is prudent.

The resulting edge shape looks very much like that of a well-worn pump. In that respect, the above advice just results in pre-wearing the pump to a point where subsequent wear rate curve would flatten off (Figs J.107, J.108, J.110 and J.111).

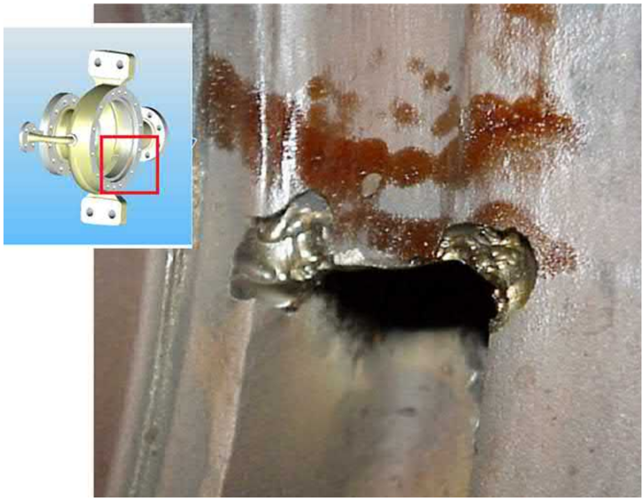


**Fig. J.107** View into the casing of an end suction hydrocarbon processing pump. Horseshoe vortex wear at the junction between volute cutwater lip and casing side-wall. Note erosion is proceeding chiefly at right angles to flow. Also note that the original cast surface is still visible and seems little damage from low impact angle particles. This image emphasises the solids concentrating power of vortices.

**Fig. J.108** View of the volute casing cutwater lip on an axially split case pump. Again notice the horse shoe vortex damage near the junction point.

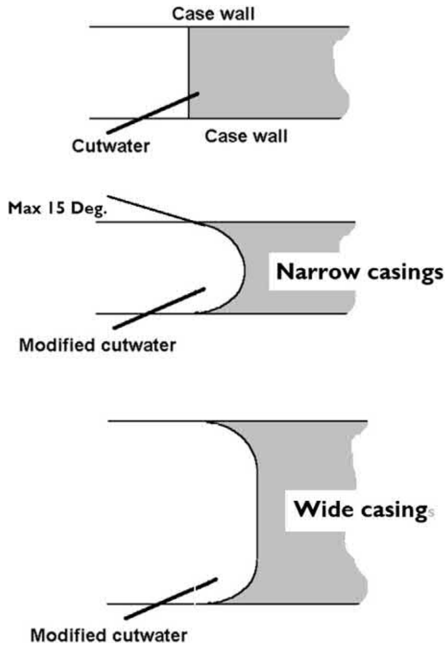



**Fig. J.109** This shows a simple measure to help reduce horse-shoe vortex damage, in this case at a volute casing cutwater. The main objective is to lower the angle at which the vane intersects the wall from 90 degrees to something less than 15 degrees close to the casing wall. The intent is to provide some degree of local 'sweep-back'. It is not necessary to carry the sweep back along the whole of the edge. Sometimes it is most convenient to achieve this by removing a semicircle from the edge [which might already be showing signs of earlier damage] In other case, where the space between the walls is large, this might be unwise-since it could significantly increase the casing throat are and consequently the best efficiency flow and power. In this case each sweep back needs only extend to around 10% of the total cutwater length [normal to flow].

**Fig. J.110** This hydrocarbon process pump shows HSV damage near the cutwater lip, as well as ring—type vortex attack.

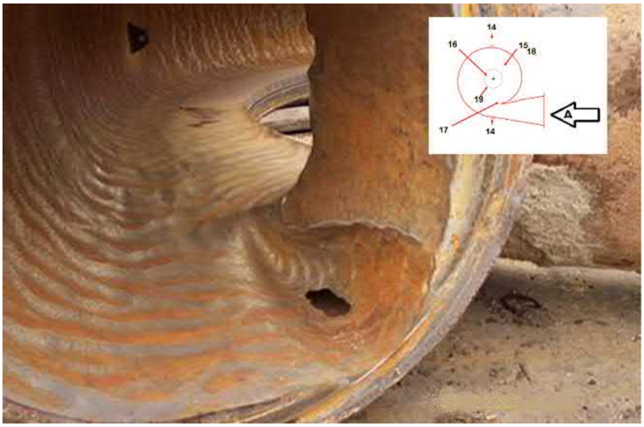



**Fig. J.111** This is a view, in the reverse direction to flow [in direction of arrow A], looking back down the casing liner of a solids handling/dredge pump. It shows the downstream damage of a horseshoe vortex that formed at the volute lip/cutwater. The damage has eaten through the wall of the casing liner. *[J F Gulich]*.

## Zone [17]: cracks to volute cutwater near centre-line

### *Cause*

Earlier in Impeller-Zone 5 the flow picture leaving an impeller was described [conceptually] as having a saw tooth pattern. As each rotating impeller blade passes a stationary collector blade the, liquid is squeezed, then released, as it passes through the radial gap.

### *Effect*

This periodic interaction is one of the chief sources of hydraulically excited vibration in pumps. As each impeller blade approaches the collector vane, the local pressure begins to rise and then fall as the impeller blade passes. This also implies that the whole casing sequentially grows and contracts with each impeller to casing encounter. At the same time, this alternating pressure field exerts bend stresses on the collector vanes. In a volute collector, the maximum bending will normally occur at mid-span. The same applies to fully shrouded diffuser collectors [i.e. those with two integral side passage walls.] A similar situation can also exist with semi-open diffuser vanes except here the maximum bending moment occurs at the vane to hub juncture. In the either case, fatigue cracking of the collector vane can occur at the point of highest reverse bending (see also Fig. J.123).

### *Solution*

It is known that the severity of pressure pulsations is related, among other things, to the diameter ratio of impeller vane outlet diameter to collector vane inlet diameter [This is alluded to in Fig. J.65.]. As this is often the root cause, impressive improvements can result from increasing this ratio. As a secondary step, the encounter between vanes can be softened by skewing one relative to the other. It is usually most convenient to achieve this by shaping of the cutwater. The impeller to casing interaction can then be likened to a progressive and gradual 'scissors' action, in contrast to the sudden 'guillotine' action of more orthogonal blade edges (Figs J.112 and J.113).

**Fig. J.112** This centreline crack on the cutwater of a volute pump is typical of high cycle fatigue impose by sustained low flow operation. The cyclic pressure/flow pulsations applied by each vane pass encounter apply maximum bending moment at the centreline, which may already contain micro cracks.

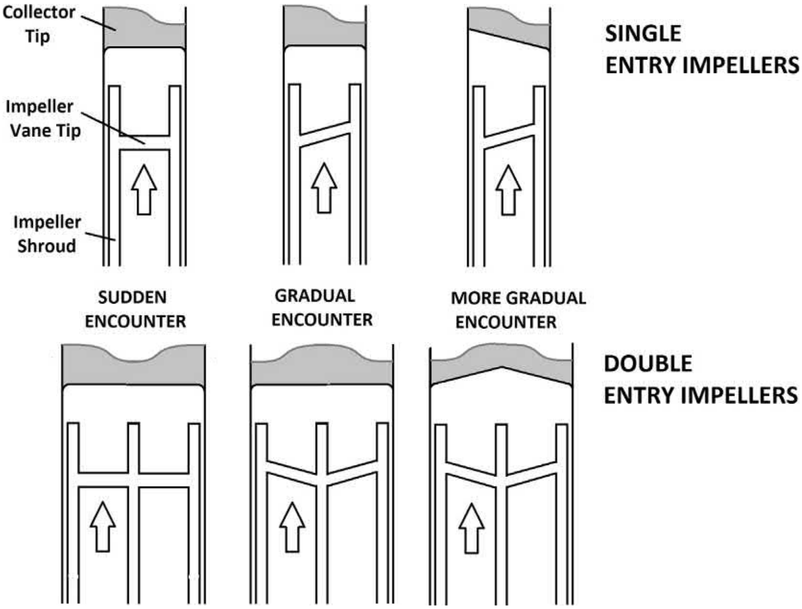



**Fig. J.113** When both the impeller outlet edge and the collector inlet edges are orthogonal in shape, the two encounter each other rather suddenly − resulting in a pressure spike. By skewing the impeller outlet edge, the intensity of encounter can be reduced somewhat. If both the impeller and collector have skewed edges, the pressure spike on encounter is even more reduced. The shape that results from skewing the inlet edge of a double entry casing is sometimes described as 'Birds-mouthing' [Lower right].

## Zone [18]: volute casing sidewall corner, near to impeller rim

Gouging erosion, with repetitive pattern, on casing near maximum impeller diameter.

### *Cause*

Certain casing corner profiles encourage formation of Taylor type corner cortices. Such vortices are capable of entrain large concentrations of any erosive particles that mat be present. Some particles can impact the casing wall at or near to the critical angle causing significant damage. This form of damage will be very localised. Zones of the sidewall at smaller radii, and closer to the shaft might be so unmarked as to still show machine marks or casting surface.

### *Solution*

If possible remove severity of corner, by [say] machining or grinding. An alternative is to locally hard coat, though this might be impractical (Fig. J.114).

**Fig. J.114** This process pump exhibits corner vortex/damage right around, near to the rim of the impeller. This damage pattern has been locally overridden at the cutwater by a horseshoe vortex pattern.

## Zone [19]: casing sidewall near to wear ring

Some casings exhibit high levels of low impact angle damage in and around the area of larger diameter than the wear ring.

### *Cause*

This damage is most often seen in casings which have a large pocketed area near to the case wear ring location bore. Such areas are an unfortunate consequence of good casting design which dictates that the wall thickness should be relatively uniform. Eliminating this pocket would result in heavy casting sections and possible shrinkage defects. Unfortunately, unless discouraged, a Taylor-type corner vortex can form in this pocket.

### *Solution*

There are too many variations of this fault to describe one solution. But the general approach is to, in some way; pad out the hydraulic profile so as to eliminate the corner vortex (Figs J.115–J.117).

**Fig. J.115** This 'between bearings' double suction pump suffered localised damage to both the casing and its cover. The rest of the casing has only suffered slight low impact angle damage.

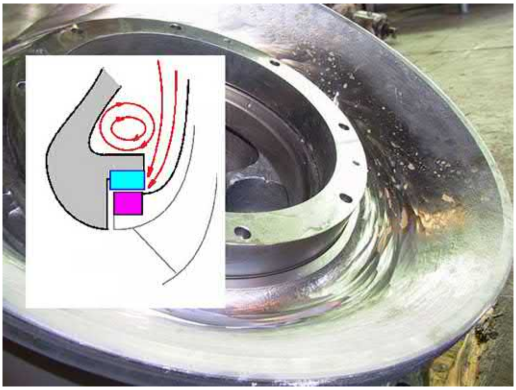


**Fig. J.116** Some casing casting designs encourage the formation of standing vortices — an unexpected consequence of incorporating good casting design features. In this case, severe vortex induced damage was experienced in the relieved pocket of the casing and also its cover, near to the wear rings.

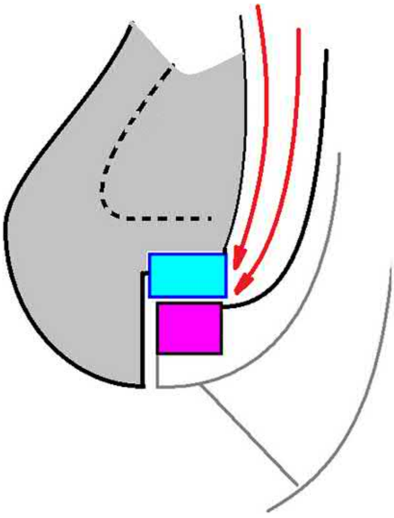


**Fig. J.117** This unfortunate feature can be overcome by filling in the dead space. The precise method of execution will depend on the pump design as well as its serve and materials of construction.

## COLLECTOR: MULTI-VANE DIFFUSER-SINGLE AND MULTISTAGE

Pumps of low to medium Specific Speed [Section 4] are commonly furnished with either spiral volute collectors or with multi-vane diffusers. Pumps of higher Specific Speed tend to only be furnished with multi-vane diffusers, for reasons beyond the scope of this text. The flowpath through low to medium specific speed diffusers tends to be radial or near radial. In contrast, the flowpath through diffusers of high specific speed machines is conical or near axial. Despite this they share many of the same damage patterns.

Radial diffusers are most often produced to a 'semi —open' configuration. This means only one sidewall is cast integrally with the vanes. The fourth wall of the passage is formed by a separate 'closing plate'. This allows a simpler casting with the advantage that the flowpath surface is wholly accessible for manual upgrading [For efficiency improvement.]. It also lends itself to machining-from-solid with the benefit of high accuracy and good surface finish. As a rule—of-thumb, pumps with an impeller outlet tip speed in excess of 100metres/sec tend to have fully shrouded diffusers. Experience says that semi-open diffuser tips are susceptible to fatigue cracking where they join the back shroud [see earlier section Collector — 17.]. This is results from the repeated bending forces exerted at each impeller-diffuser vane encounter *[Unless the diametral gap between the two is comparatively large.]*.

On the other hand, mixed flow diffusers are almost always of closed-shrouded design.

Additional zone of attack for both class of collector include;

**20]** Diffuser vane tip damage

**21]** Juncture of vane with supporting shroud.

**22]** Cavitation damage on casing sidewall or wear plate. Often associated with damage Zone [12] on Semi open and open impellers only (Figs J.118 and J.119).

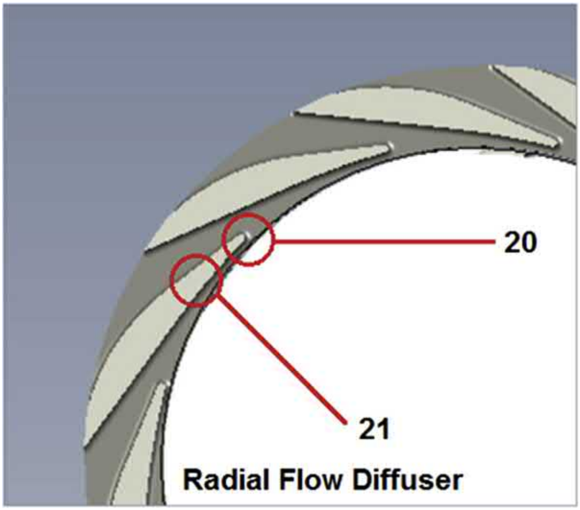



**Fig. J.118** Areas of multiblade diffuser that is particularly susceptible to erosive damage. This shows a diffuser from a low flow-radial flow machine.

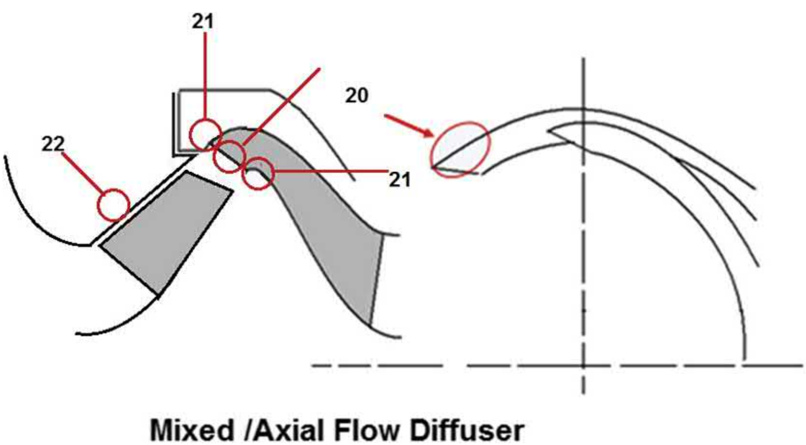



**Fig. J.119** High flow diffuser pumps generally are built around mixed/diagonal flowpath or even axial.

## Zone [20]: thinning and shaping of diffuser inlet edge

### *Cause*

In pumps having relatively narrow impellers [$b2/D2 < 0.1$], compared to their diameter, the liquid velocity leaving the impeller rim is probably the highest in the machine.

### *Effect*

The diffuser tips experience very high approach velocities. When such pumps handle liquid with very fine suspended solids, their diffuser tips experience high impact angle damage and low impact angle damage to their general surface. The velocity of liquid leaving an impeller is not uniform across its width. This is most true in impellers that that are particularly narrow. Generally the velocity mid-passage shows a much higher peak than elsewhere. Consequently, the diffuser inlet tips show more damage mid-passage then elsewhere.

### *Solution*

The damage concentration mid-passage can be reduced by increasing the radial gap between impeller and diffuser [Fig. J.67]. This allows the low and high liquid velocities to 'mix-out' to an average velocity, which is less damaging. This gap increase can be accomplished by machining out the diffuser D3 bore. A widely quoted industry-standard defines a minimum for the ratio of D3/D2 as 1.03 for diffuser pumps. There will be some important side effects though. The head produced by the impeller will reduce by a few percent, for no reduction in power. In fact, there may be a slight flow increase, with a corresponding power increase. Furthermore, the curve shape will be affected – particularly at flows less than 20% of bep. The curve shape will tend to flatten and the head a zero flow reduce slightly. On the other hand, the pressure pulsation at each impeller–diffuser vane encounter will be reduced and this will be evident by reduced vibration levels.

An alternative is to weld overlay the diffuser tips with harder material. This will help reduce both the high and low impact angle damage. This approach is most practical with 'semi open' diffuser. Closed diffusers might only be partially overlaid (Fig. J.120).

**Fig. J.120** Classic High impact angle damage along the edge of a radial flow diffuser vane. In radial flow machines, flow velocity leaving the impeller is usually highest at mid passage and this is reflected in the higher erosion rates at diffuser mid-passage.

## Zone [20]: undercutting at diffuser vane inlet tip

### *Cause*

Whenever a vane edge is simultaneously positioned at right angle to both flow and an adjoining wall, horseshoe vortices can form.

### *Effect*

As frequently mentioned earlier, these have a very high potential for erosive damage abrasive particles are present. The first picture shows typical horseshoe vortex damage to the diffuser inlet tip of a radial flow pump. This pump has a semi-open diffuser, so the corresponding casing wall or closing plate, will be similarly damaged.

But even pumps with wide impellers, at the opposite end of the spectrum, are still vulnerable to such damage. This is illustrated in the second picture which shows the diffuser vanes of a mixed flow bowl type pump.

### *Solution*

To destroy the liquid mechanism that makes horseshoe vortices possible it is necessary to destroy the orthogonal property of the edge, by sweep back or radius as with the impeller [see previous section] (Figs J.121 and J.122).

**Fig. J.121** Classic horseshoe vortex wear and undercutting at junction of vane and hub disk.

**Fig. J.122** Similar Horse Shoe Vortex damage; this time in a mixed/axial flow diffuser intersection with hub.

## Zone [20]: inlet vane tip broken

Each time a rotating impeller vane encounters and passes a stationary casing vane [volute or multi-vaned diffuser] a pressure pulsation is created. This effect is particularly severe at relatively low flows; say less than 30% of bep. These cyclic pressure pulsations can induce fatigue stresses in the diffuser tips. The pulsation levels are mostly influenced by the radial gap between impeller and diffuser tip [The so-called 'B' gap, see, earlier section.]. The root cause mechanism is similar to Zone 17 in volute pumps

### *Solution*

In order to control such pressure pulsations, good general practice limits the radial gap [B gap] such that D3/D2 exceeds 1.03 – more in the case of high energy machines. This also helps controls the corresponding fatigue bend stresses. Vane tip thickness at inlet should normally exceed 10% of the diffuser bore diameter.

The stress levels are particularly high in semi open diffusers since the vane is cantilevered out from the supporting shroud. The vane inlet tips of closed diffusers are obviously less stressed and more secure. With some high-energy machines, there is no option but to include closed type diffusers in the design. Typically this occurs if the impeller peripheral tip speed exceeds 100 m/s (Fig. J.124).

**Fig. J.123** Flow and pressure pulsation caused by the typically saw-tooth exit distribution [see Fig. J.63] induce fluctuating forces on the collector vanes. The intensity of these forces increase as the flow departs from its design condition. The forces apply a bending moment, which is a maximum at the hub-disk intersection. This is where fatigue failure most often occurs, resulting in this sort of breakage.

**Fig. J.124** Bend fatigue failure induced, by low flow pressure pulsations in this process pump.

## Zone [21]: mixed flow diffuser bowl

### *Cause*

An earlier section described the conditions leading to micro cracking at the point where impellers vanes interface with the front and or back shrouds. These conditions also exist in the vanes of bowl type casing, sometimes even more severe.

### *Effect*

Cracks propagate along the intersection between vane and casing walls, in the general direction of flow. The nascent crack site on the vane inlet edge often later becomes the focus for high impact angle erosive damage. First signs are 'nicks' in the inlet edge close to the fillet between vane and wall.

### *Solution*

The root cause of this problem chiefly lies in the initial design. Whenever thin vanes merge into very thick wall sections, this problem can occur as the casting cools in the foundry. Experienced foundries will attempt to slow the more rapid cooling of the vanes. More customised heat treatment of the finished casting may be required (Fig. J.125).

**Fig. J.125** Crack propagation at the vane to hub junction Thus was caused by a combination of micro crack initiation and fluctuating forces from low flow pressure/ flow pulsations [see also Fig. J.63].

## Zone [22]: cavitation pitting to casing sidewall or wear-plate

### *Cause*

Fig. J.81 illustrates the phenomenon of cavitation occurring in the clearance space of open and semi open impellers. Resulting damage is typically located in the first half of the vane length and can have corresponding damage on the casing. The cavitation cloud that initiates this damage rotates with the impeller. Whereas the impeller damage is localised, the casing damage will be spread round 360 degrees. If less than 360 degrees, as shown here, then the impeller is probably slightly eccentric in the liner and this causes the clearance gap to vary with each shaft revolution.

### *Solution*

Options include;

- Increasing NPSH [A] or [submergence of vertical pumps].
- Resetting the impeller to casing clearance gap.
- As last resort, hard coating the sidewall.

**Fig. J.126** Casing damage from tip leakage cavitation of mixed flow semi-open impeller. *[Weir Engineering Services].*

## Zone [22]: damage to casing sidewalls

### *Cause*

The vortex damage initiated by raised lettering on impeller shrouds has already been illustrated [Zone 4; Damage to impeller shrouds.]. Here we have an example of the same sort of damage exhibited on a volute casing cover sidewall. Again, the otherwise normal cast surface of the cover is virtually untouched by the low impact-angle attack. However, The Kamm type vortices caused by the letters and their discontinuity, create areas of accelerated erosion.

### *Solution*

It is good practice to machine away such lettering, in order to remove the potential for this class of damage. It will also reduce the pump power consumption very slightly (Fig. J.127).

**Fig. J.127** Kamm type vortex damage induced by raised cast lettering in sidewall space, between impeller and casing [see also Fig. J.56]. *[Sigmund Pulsometer Pumps].*

## 3. MISCELLANEOUS WEAR PATTERNS

### THROTTLE BUSH

Taylor type vortices were described earlier in this section. True Taylor vortices relate to flow in long annular clearances. One such site is the throttle bush of a typical multistage pump. Two examples are shown here. In the first, the solid particles were extremely small and only two-body contact ever existed in the annular clearance. The helical path of the abrasive particles is disclosed. The bush is about 4 in. [100 mm] in diameter and the pumps were running at 5500 rpm.

In the second case, the particles were larger and three-body contact probably existed. This would explain the less well defined pattern. No helical component is visible (Fig. J.128).

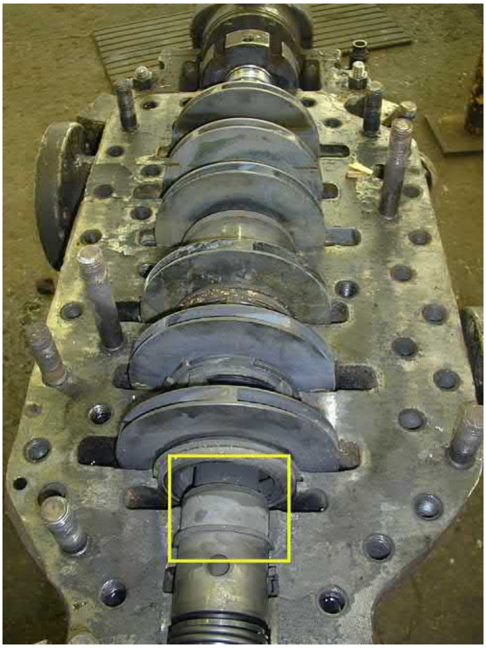


**Fig. J.128** Location of throttle bush.

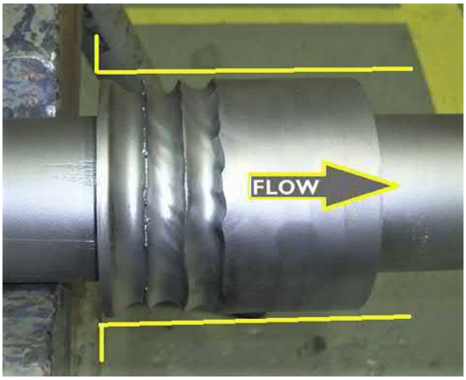



**Fig. J.129** Throttle bush subject to two-body Taylor type vortex wear.

**Fig. J.130** Similar throttle bush, exhibiting mixture of two body and three body wear.

## PROTRUSIONS-GRUB SCREWS

### Cause

In the introduction to this Chapter it was mentioned that Kamm –type vortices form downstream of any item which fails to streamline the flow. One seemingly innocent component would be protruding grub screws. It is common practice in many cases to retain wear rings and the like with grubs crews drilled and tapped along the interface between the ring and carrier component. In most applications this poses little if any risk. However, if abrasive particles are entrained in the liquid, then the risks increase significantly.

The liquid in the impeller-casing space is dragged round at roughly half the impeller angular velocity. Even the presence of flush grub screw creates a disturbance in the flowpath. Particles will collect in the consequent downstream vortex and this will result in 'comet-tail' erosion patterns.

Note that the machining marks are still visible on the cover surface! The low impact angle damage has been too small to completely remove them. Yet the damage from the Kamm type vortex has eaten through the cover wall.

### Solution

Even flush grubs crews will still initiate a slight flow disturbance so erosive damage might still occur. But if the grub screws are NOT flush, then the erosion rates will be greatly increased.


**Fig. J.131** This is a back cover plate of an API610 process pump. Kamm type vortices can form downstream of bluff protrusions in the flowpath. The Grub-screws here only protruded 2mm initially, but caused very substantial Kamm vortex type damage downstream. See also Fig. J.104, which highlights the region of Taylor − type vortex damage. *[Union Pumps].*

## Other factors influencing wear rates

I opened this Chapter, creating the impression that erosion in centrifugal pumps is an inexact science. However, specialist manufacturers seem to have advanced the state-of-the-art to a very high level. Presumably, this reflects a substantial understanding of all the influencing factors. The following table summarises my general experience in what part different factors play in the overall outcome (Table J.1).

**Table J.1** Abrasive wear factors in pumps.

| Factor | Comments |
|---|---|
| Particle size | If particles are expected, they should be made as small as possible. |
| Particle quantity | Damage is roughly proportional to quantity of solid |
| Particle hardness | No simple rule, but at least proportional to hardness |
| Material hardness | No simple rule, but at least proportional to hardness |
| Head per stage [AU3] | No simple rule but at least proportional to $(\text{head})^3$ |
| Viscosity of carrier liquid | The higher the better. Appears to help reduce particle damage rate. |
| Density difference between carrier liquid and solid particles | Small differences seem better than large ones. |

Looking at some of these factors in more detail.

## Product properties

This Figure gives some sense of how pump life is influenced by the particle size and concentration (Fig. J.131).

This is a simplified view, but allows some important conclusions to be drawn. For example there is a region where the pump life is predicted to reduce significantly if the particle size is increased. Beyond this size limit, life is less affected if size increases, but the overall life is now already low anyway. It seems that if abrasive solids have to be pumped, it is best to make them as small as possible, so as to get the least pump wear.

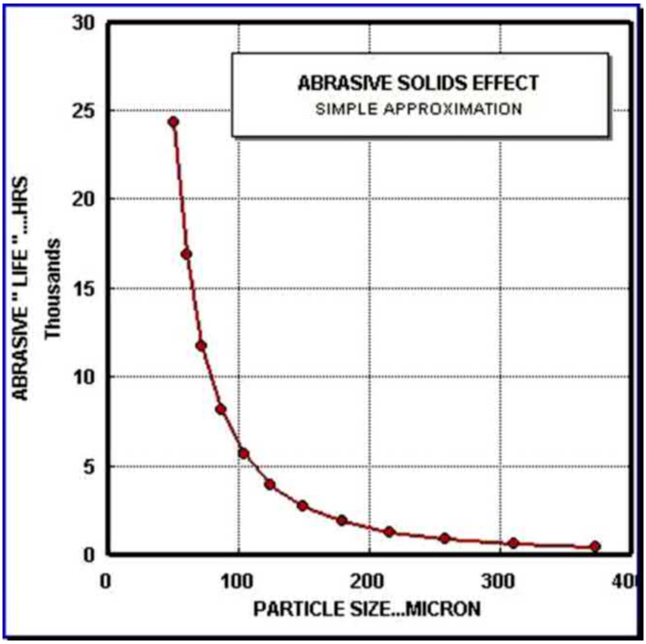



**Fig. J.132** This illustrates a typical relationship between solids size and arbitrary pump life.

## Hydraulic design factors

In pumps where the impeller outlet width is around 10% of the diameter, or less, then the casing is most likely to receive the most wear. In other words, pump of relatively high head, at any given flow, are likely to experience the highest casing wear, all other things being equal.

Where the impeller width is more than 15% of diameter, then substantial impeller wear is likely to be an additional factor. Both these conclusions can be foreseen from a study of average velocities along the internal liquid flowpath [Ref. 18].

## Material surface hardness-coatings

In certain circumstances, hard surface coatings can be a successful wear abatement measure. However, these conditions are quite well defined and coating may not be a universal solution. There has sometimes been a culture of 'one type suits all' in terms of coating selection. This may have helped generate some cynicism towards hard coatings.

Hard wear-resistant coatings seem best suited to low impact angle wear. If the pump is failing due to high impact angle attack, then thin hard coatings are less likely to be suitable long term. So for example, if the pump is suffering high impact angle damage to the casing or diffuser cutwater lip, then hard coating alone will not be successful. A more useful approach is to shape the leading edge in such a way to reduce the high impact angle attack and swing the balance in favour of low impact angle attack, which hard coatings are better suited to. This is the logic behind the cutwater reshaping shown in Fig. J.109.

Coated wear rings, in the other hand, have proven very successful. This is because; the flow field in the gap is chiefly comprised of low impact angle attack. Hard coatings are best suited to these conditions. **However**, careful shaping of the flowpath entering the wear ring annulus is still critical if Taylor vortex type damage is to be avoided [including flushed wear rings].

### Slope of pump and system curves

It is worth reiterating an earlier point, that attention to curve shape is particularly important when pumping dirty liquids.

## 4. EFFECTS OF PUMP AND SYSTEM CURVE SLOPES/ SHAPES

The most outwardly apparent effect of internal wear will usually be a reduction in pump output. It was mentioned earlier that most centrifugal pumps are procured in order to generate flow in a liquid transport system, as opposed to creating hydrostatic pressure. So it follows that any decay in performance will, in normal circumstances, be the first signs of internal problems.

But pump wear might not be the only cause of flow reductions. Many times in this volume it is mentioned that the system curve shape is at least as important as that of the pump when total system flow is concerned. Here is a good example; consider two cases.

In the first, the pump has suffered from running for long periods with unacceptably high suspensions of abrasive particles in the liquid. Predictably this causes a reduction in pump output and a new lower intersection with the appropriate system resistance curve (Fig. J.133).

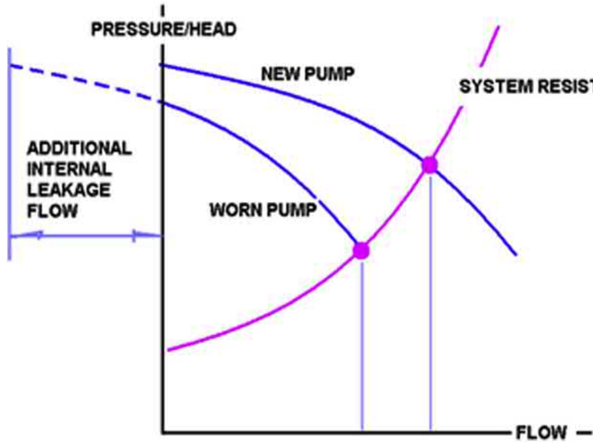



**Fig. J.133** Pump flow performance will decay as a result of increased wear and consequent leakage increase. The pump curve intersection with the system resistance curve will change to a lower head value.

In the second case, the pumped liquid contains no abrasive solids, but instead has a chemical composition that leads to fouling and or deposition on the wetted surface of the pipe system. This has little or no effect on the pump performance curve, but of course increases the flow resistance of the piping system. The system curve then becomes a little steeper (Fig. J.134).

Again the pump curve intersects the system curve at a new lower flow. The outward effect is the same as for the case of a worn pump. Is there any way to separate out the two causes? Well the pressure generated by the pump will also change, but such measurement is not often reliable [Fig. 11.16].

Knowledge of the pump power consumption can often help resolve this issue. For the case of internal pump wear, the absorbed power curve will be shifted to the left – by an amount equal to the increased internal leakage flow [Fig 11.21]. So the reduction in flow will be accompanied by an increase in absorbed power/amps (Fig. J.135).

In the case of system fouling, the response will be different. The power curve will not shift. But the effect is analogous to throttling the pump output; the absorbed power will appear to reduce along with the flow. So the trends in power consumption can complement any pressure head data. Table J.2 attempts to summarise this behaviour.

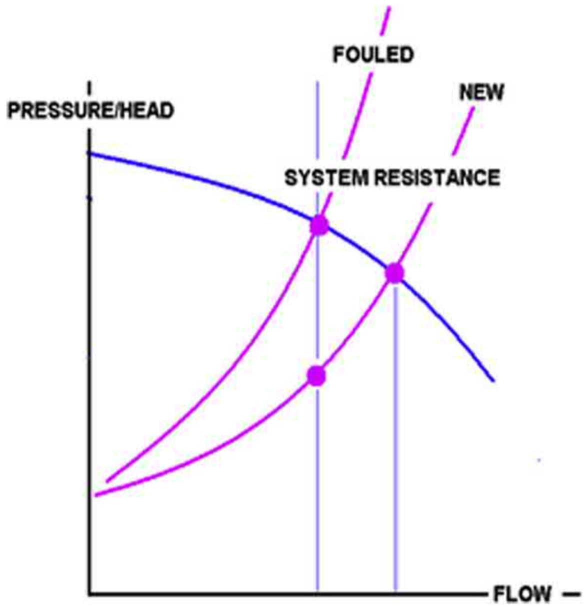



**Fig. J.134** Pump flow performance will also decay as a result of [say] system fouling, or maybe some other root cause. This time the pump curve will intersect with the system resistance curve at a higher head high value.

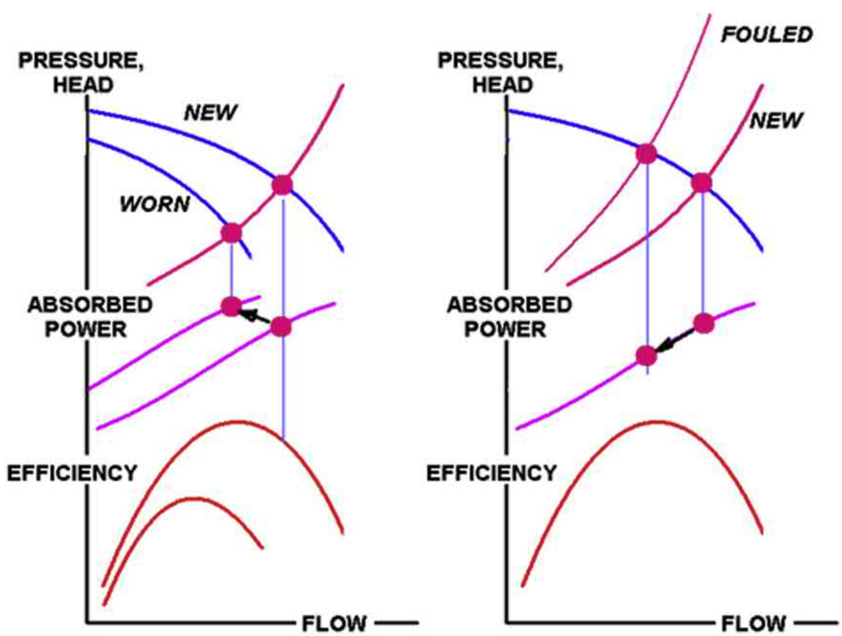



**Fig. J.135** In this hypothetical case, the system flow has decreased by exactly the same amount, even though the root cause is different. One way of identifying the root cause is to monitor the pressure head generated by the pump. Another complimentary measurement is that of absorbed power.

**Table J.2** Knowledge of the way in which pump performance appears to decay, can help to point to the source of the issue.

| | **Pump flow appears to?** | **Pump differential pressure/head** | **Pump absorbed power?** |
|---|---|---|---|
| Pump internal wear | Reduce | Tends to reduce | Tends to increase |
| System fouling | Reduce | Tends to increase | Tends to reduce |

Can a pump be made more immune to this cause of performance decay? Sections 3 and 4 discussed how the different curve 'shapes' evolved, as a function of the so-called specific speed, and by the choice of impeller geometry (Fig. J.136).

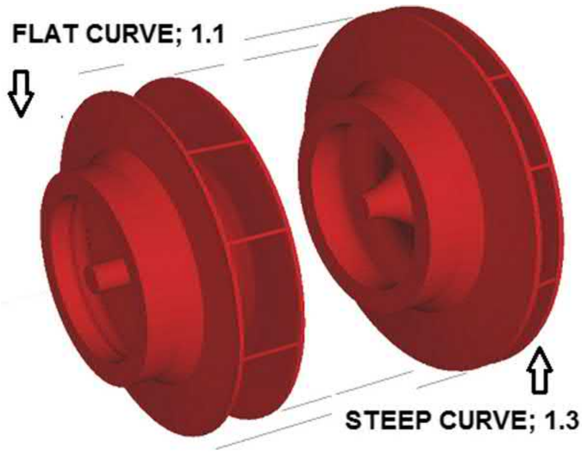



**Fig. J.136** These two impellers will both generate the same flow and pressure head at their design point. However, the impeller on the right will generate a higher pressure head at or near to zero pump flow — maybe 30% more. The impeller on the left will generate much less head — at zero flow-maybe only 10% more. It is likely to be slightly more efficient and will usually cost less to manufacture. But in service, it will usually appear to wear out more quickly than the impeller on the right — as explained in the text.

Curve shape is often defined, within the pump community, as the ratio between the pressure head that the pump generates at its design flow to the pressure head at zero flow *[defined as **H**ead **R**ise **T**o **S**hut **O**ff; **HRTSO**]*. It turns out that pumps with relatively 'steep' curves, were this ratio leans towards 1.5 or even 2, have impellers that are relatively large and narrow. It also turns out that the pump weight and, to an extent its manufacturing cost, is roughly proportional to 'Impeller Outer Diameter 2.6 to 3.0'. In other words, pumps with steep curves tend to be more expensive than those with 'flat' curves. Pumps designed and produced for the general market place and with no particular application in mind will tend towards flat curves so as to make them more competitive. But it turns out that pumps with flat curves do not APPEAR to last so long under abrasive wear conditions. The explanation lies again in the relationship between pump and system curve.

Consider the same pump installed in two very different systems. In the first case the system resistance curve consists mainly of static lift head, so is quite 'flat'. The second system has very little static lift but considerably more frictional resistance. By a convenient coincidence, in this instance, both these different systems yield the same resistance at the same design flow (Fig. J.137).

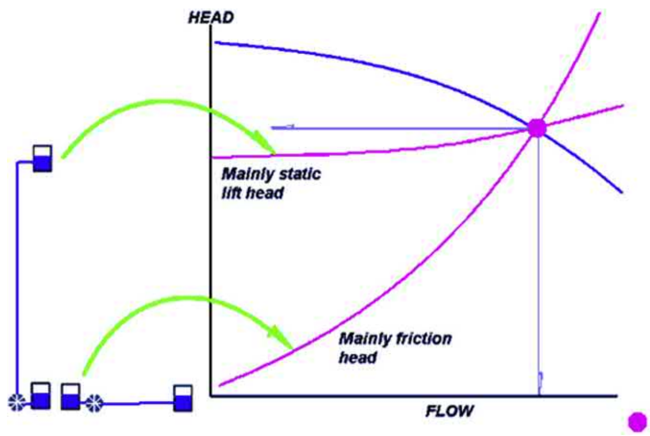



**Fig. J.137** This shows resistance corves for two different systems but which result in exactly the same resistance at a certain rated flow. In one case the pump has mainly to lift the liquid from one level to another, so the system is comprised of mainly static head. In the other case, the pump has little if any lift work to do but the system contains elements which, in total, contribute a great deal of friction to the resistance curve. Identical pups would probably be chosen in both cases.

Now consider this pump to suffer performance decay due to internal wear. We can examine how it would behave in these two different systems. When operating against the first 'flat' system resistance characteristic, the two curves intersect at a much lower flow than with a 'steep' system (Fig. J.138).

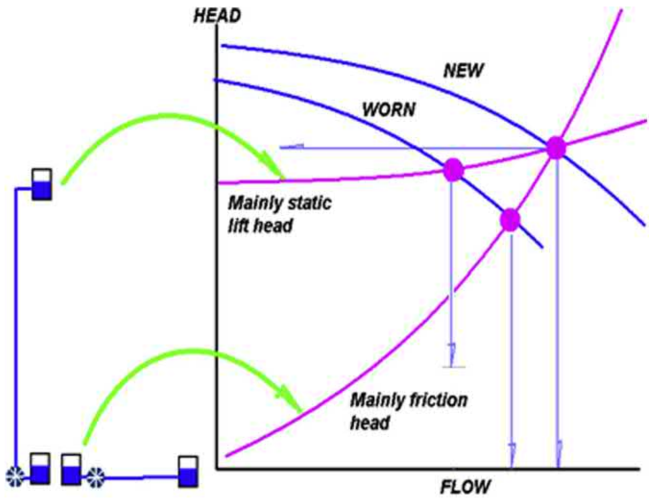



**Fig. J.138** Here the effect of internal wear is considered on these two identical pumps. Evidently the intersection of pump and system curve [the equilibrium flow] has a lower value when the pump operates against a system resistance comprised mainly of static lift. The pump will appear to have worn out faster than had it been operating on a system comprised mainly of friction resistance.

- This leads to the first important conclusion that, using rated flow as the criterion, the SAME pump would appear to wear out more quickly if operating in the system with a 'flat' resistance curve.

One can ask the complimentary question of how, as they wear, do two pumps with different curve slopes 'shapes' react on the same system curve (Fig. J.139)?

- This leads to the second important conclusion that, using rated flow as the criterion, a pump having a steep curve, besides being more expensive to manufacture, will also appear to wear more slowly than on with a flat curve.
- Finally, pumps with steep performance curves, pumping against steep system resistance curves will appear to wear out much more slowly than pumps with a flat curve shapes operating on a flat system resistance curves.
- Maximising the angle between pump and system curve minimises the apparent wear-out rate.

This leads neatly onto one criterion for pump maintenance intervals. The decision to intervene will be driven by at least two considerations; either,

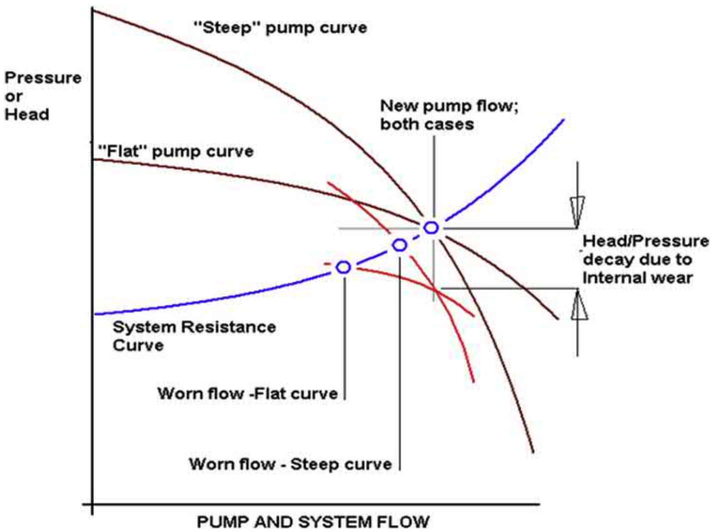



**Fig. J.139** This summary illustrates that a pump having a 'steep' performance curve and operating in a system the consists mainly of frictional resistance will appear to wear more slowly than a pump having a 'flat' curve sited in a system comprised chiefly of static lift.

- Failing to accomplish process flow, and/or
- A worry that prolonging intervention will introduce unacceptably high maintenance costs.

A rough rule for **performance-based intervention** is as follows (Fig. J.140).

Consider intervention when:

$$dQ^2 + dH^2 > 25$$

where;

dQ is % flow deficit from new

dH is % diff. head deficit from new

This attempts to factor in the effects of different pump and system curve slopes/shapes. The intervention factor of 25, can, if necessary, be re-calibrated to suit specific circumstances.

**Example 1:**

dQ = 5% dH = 3%

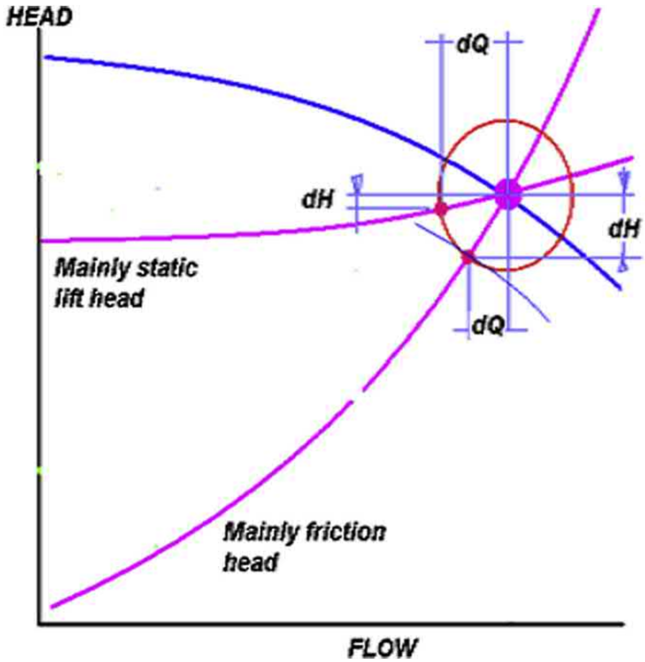



**Fig. J.140** This illustrates a general rule for helping to estimate when a pump should be withdrawn for routine maintenance. The text explains how this general rule can be tailored to individual/critical circumstance.

- Sum = 34, thus intervene

**Example 2:**
dQ = 3% dH = 3%

- Sum =18, thus do not intervene, **unless** process obligations are not being met.

# APPENDIX K

# Hydraulic modifications

NOTE; Chapter 1 explains that the concept of a centrifugal pump has been in existence for a very long time. In that period, different manufacturers have made many different interpretations of the concept. Consequently it is unrealistic to suppose that the outcome of any of the general advice given in this volume could be guaranteed in all or any circumstances. This advice is given in good faith, but is only intended to illustrate the likely scale of any outcome.

## INTRODUCTION

With time, pumps may become mismatched to their service. While sourcing replacement equipment is an obvious solution, it is often the case that existing machines can be modified to suit. This section outlines some of the measures that can be formally adopted. It also describes some less formal methods that can provide short term/emergency solutions.

Ideally, most troubleshooting exercises would commence with an open mind *[immediately violating Palgrave's Third Rule]*. They will consider the root cause might lie in either the pump or the system. Eventually, it might be concluded that the pump, as it stands, is not suitable for the application. We know from experience that pumps are almost always over specified, at the outset [Chapter 2] so it might never have been the right one for the job. On the other hand, the needs of many pumping systems change with time. So a pump that was more or less correctly matched at the outset becomes mismatched over a period of time. For example, the pipe system might experience progressive corrosion or fouling to the wetted pipe surface. It might be impractical or even uneconomic to correct this. On the other hand, the plant in which the pump system is installed may have had its planned output increased or decreased. Either way, the pump performance output may now be unsuitable. Elsewhere is stressed the importance of achieving a good hydraulic 'fit' of pump to system.

Does that mean that, in such circumstances, a new pump has to be purchased and installed? Not necessarily! Experience says that the cost to remove a pump set and install it with a different one can amount to between two and four times the capital cost of the original. Therefore, there is quite an incentive to try to persuade the existing pump to deliver the new performance requirements. Added to which, this approach largely avoids the need for any retraining relative to the new machine. And many pre-existing spare parts may still applicable.

Most centrifugal pumps are available with a range of impeller outer diameters. In mass-produced pumps, these are only available in discrete steps, whereas with engineered pumps this diameter is, within limits, infinitely variable. So, correct or better matching of system needs can sometimes be met by simply selecting a larger or smaller impeller diameter. This is the process that a manufacturer will have followed when making the original selection. The chief obstacle may occur on the occasion where when more output is demanded from the pump. In such cases, the driver originally provided may be incapable of providing sufficient power. Even so, it may or may not prove practical to install a more powerful driver.

Beyond the quite significant range of performance adjustment is the possibility of changing the basic hydraulic design. Manufacturers often have hydraulic libraries from which alternative impellers may be selected (Fig. K.1).

Typically the design flows of these alternatives will extend in steps that provide only small efficiency gaps [<3% points] in the transition from one pump frame size to the adjacent. Low flow specific speed pumps generally have a more rounded/flatter efficiency curve so larger flow steps can be

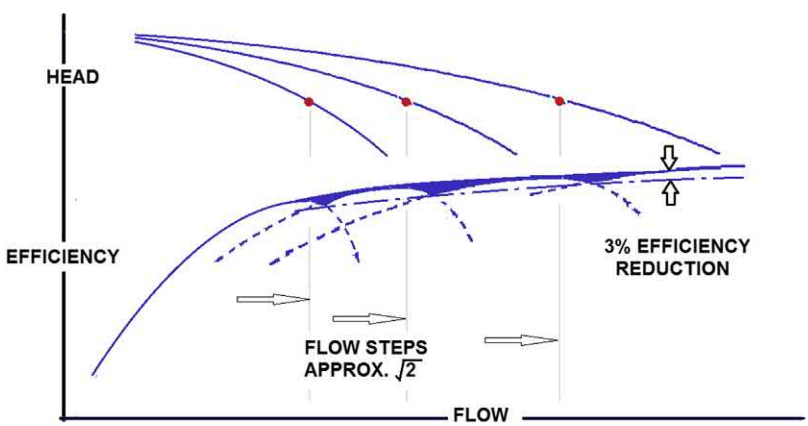



**Fig. K.1** Manufacturers will plan their ranges in such a way that there are no 'holes' in the efficiency coverage. Generally this is accomplished by having two or more alternative impellers available inside the same casing. This can be a great help when it comes to troubleshooting hydraulic misfits.

chosen before the 3% efficiency reduction occurs. On the contrary, in higher specific speed pumps, the efficiency curve is generally more ‘peaked’ and so smaller flow steps may be chosen. From such hydraulic libraries, suitable alternatives might already exist.

## MODIFICATIONS IN THE FIELD

But what can be done within the constraints of existing pump components? Can its output be modified in any significant way, either on a stop-gap basis, or even permanently?

In fact, the performance of an existing machine can often be adjusted or modified to quite a degree. The way in which a pump apparently fails to meet current needs can generally be categorised as follows;

**1)** Pump pressure head does not appear to match EXPECTED new system pressure head. — *Turn to the first section*
**2)** Pump can be modified so that its pressure head meets the new system pressure head needs, but does so at a flow far removed from its optimum efficiency flow, maybe even outside of its COMFORT ZONE [Appendix D]. — *Turn to the second section*
**3)** Shape of the current pump performance curve is unsuitable for mutual co-operation with other pumps in the system. — *Turn to the third section*
**4)** The NPSH [A] at the new system flow requirements does not exceed the current pump NPSH [R] at that flow by sufficient margin to assure adequate resistance to long-term cavitation damage. — *Turn to the fourth section*

But no modifications should be contemplated without some clear knowledge of the internal details of the subject pump. Existing pumps may well have already been modified in the past, without all or any details being recorded. It is in the nature of many day to day operations.

So there are no substitutes for having the responsible trouble-shooter make the actual inspection. In any event, he may detect nuances that seem unimportant to others (Fig. K.1.1).

### Pump pressure head does not match the expected new system requirements at the new flow

#### *Pump pressure head fails to meet new system pressure head at the new flow. Under-pumping results*

In most installations, it is much more common to [locally] measure pump inlet and outlet pressures than it is to measure true flow. Low pressure head


**Fig. K.1.1** When modifications are envisaged, ideally the actual trouble-shooter confirms the exact hydraulic details before evolving a solution. There may be geographic constraints to this or, like here, access difficulties. Here the author is examining a large water pump installed in about 1923. This is ample time to absorb several modifications and changes, not all of which might be documented [G Walker].

is instinctively assumed to indicate under-pumping. However, Fig. 11.6 illustrates how a low differential pressure might result from over specifying the requirements at the time the pump was bought. Today, the machine might actually be over-pumping! Fig. 11.16 helps explain how pressure readings alone are really insufficient data for diagnosis. So since the prime purpose of most centrifugal pumps is to stimulate system flow, then it is worth making efforts to more directly assess whether this is really being achieved. Otherwise, instigating any of the following steps to increase output pressure head will simply exacerbate the over-pumping problem and increase the risk of driver overload.

To increase output, a number of modifications can be made to the impeller and collector either individually, or in combination. In third section, I explained how an impeller potentially displaces a flow that is loosely related to the swept area of its vanes. Increasing the swept area is one way of achieving more flow and vice versa. Simplistically, this may be accomplished by increasing the impeller passage width at outlet [Fig. 4.18]. A similar effect is achieved by increasing the number of impeller vanes. However in most cases, this last effect shows significantly diminishing returns once the vane count exceeds 7.

But the collector also has a part to play. The total cross sectional area of the collector passages, at their smallest point *[the so-called collector throat]*, constricts this flow potential and helps define the point of highest efficiency. For a given impeller, increasing this throat area A3 moves the point of highest efficiency to a higher flow but a lower pressure head- and vice versa.

The foregoing gives some clues as to how the performance of an existing pump might be modified. Maximum increase occurs when both impeller and collector are modified. Worster in Ref. [10] gives a good theoretical reasoning to the relationship between impeller and collector hydraulic characteristics. One can condense the Worster approach and define two theoretical performance lines; one for the impeller and one for the collector *[A guide to roughly estimating these two lines is given later in this section.]* (Fig. K.2).

**Firstly, the theoretical Impeller line:** This falls away from a maximum at zero flow. This theoretical maximum is almost entirely dependent on impeller diameter and shaft speed. The rate at which it falls, relates to the vane count, the vane setting angle, and the swept area as characterised by the meridional outlet width *[I should mention here that width, b2, is in turn, just an approximate analogue of the real parameter which is the*

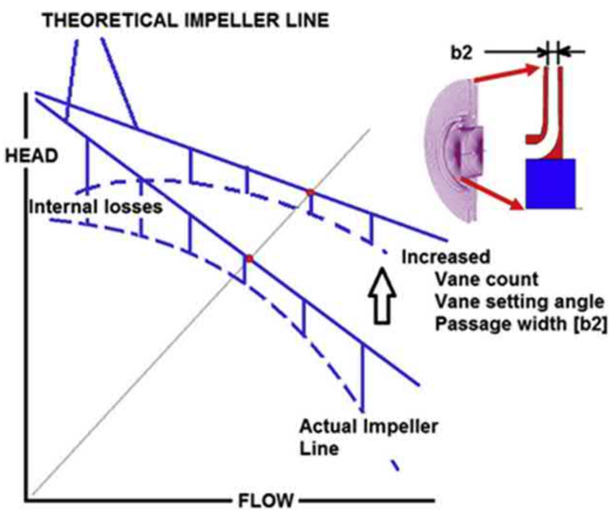



**Fig. K.2** The real impeller output differs from theoretical predictions as a result of internal hydraulic losses. Increasing the passage outlet width [b2] has the effect of reducing the slope of this impeller line, and consequently increasing real performance.

*passage outlet area.]*. Increasing all or any of these will tend to reduce the rate [slope] at which the line falls away *[i.e. Doubling the impeller outlet width and hence the passage outlet area halves the slope at which the line falls away — see later]*.

The 'Actual' impeller line is closely related to the 'Theoretical' line and follows broadly similar trends. The point where the Actual performance curve comes closest to the Theoretical line is termed the Best Efficiency Point [Bep] of the machine. The casing/collector line [see next] somewhat influences this BEP position. The ratio of actual head to theoretical head at this point is called the Hydraulic Efficiency (Fig. K.3).

**Secondly, the collector line**: This is the loci of design head and flow points for impellers having the same spiral angle of development. This line rises from zero, at a rate which is chiefly dependent on the total collector throat area [see Fig. 4.05]. Increasing this area will tend to reduce the rate [slope] at which this line rises from zero.

In principle, the point where the Impeller line crosses the Casing line will help define the flow point of Best Efficiency [bep]. These are theoretical curves based on frictionless liquid. Real liquids will induce losses in

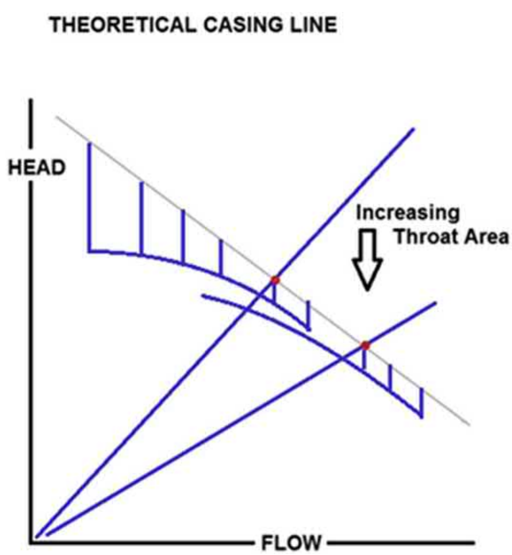



**Fig. K.3** A theoretical collector line can also be constructed by connecting [and extending] the origin point with the point of pump best efficiency [bep]. Increasing the area of the collector throat will reduce the slope of this line. Consequently the performance will increase, as shown.

both Impeller and casing. But where the real curves cross still helps to identify the real best efficiency point.

From the foregoing, it will be obvious that Impeller/Collector changes, either in combination or in isolation, give some control as to how the output of an existing machine may be modified. In particular, the best efficiency point can be also moved. On the basis that the pump best efficiency/design point is very closely related to the intersection of the Impeller and Collector lines, one can summarise as follows.

**1.1.** Reducing only the slope of the Impeller line will result in their intersection at a larger pump output flow. This will occur at a higher head than before.

**1.2.** If only the collector slope is reduced, the intersection will be at a lower head and higher flow than before.

**1.3.** Increasing the slope of the Impeller line will result in their intersection at a reduced pump output flow. This will occur at a lower head than before.

**1.4.** If only the collector slope is increased, the intersection will be at a higher head and lower flow than before.

Let's look at how these modifications might be achieved in practice.

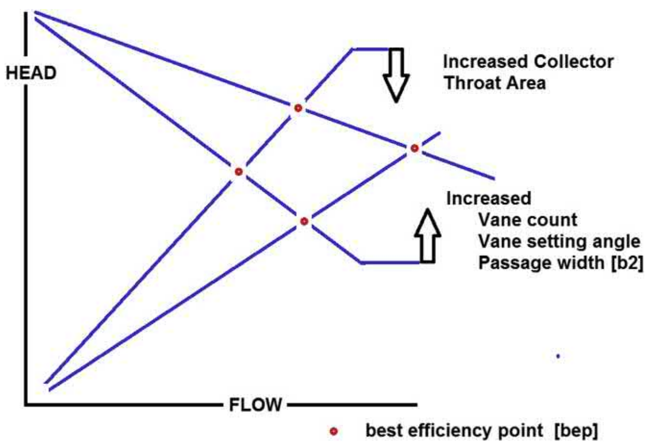



**Fig. K.4** This shows the general effects of modifications. Increasing the passage outlet width [b2] will push flow out to a high level and at a higher head. Increasing vane count and or setting angle has a similar but limited effect. Increasing the collector throat area will push the flow out to a higher level at a lower head. It is necessary to increase both impeller and collector areas if more flow is required at a similar head to the original.

## Impeller modification

Earlier, I indicated that the impeller displacement flow is loosely related to the swept area of the blades. By implication, it is closely related to the vane-to-vane passage area at outlet *[of which the width b2 is also an analogue]* (Fig. K.5).

This influences the so-called 'Euler velocity triangles'. These vector triangles are used by designers to help describe the average flow conditions of liquid particles at the passage exit. They formed part of Worster's analysis, previously referenced. They are designer's tools and their background is inappropriate here. As already described, they predict that increasing the passage flow area will result in increased best efficiency flow and reduced pressure head.

One widely used method of increasing the local passage flow area at outlet is most commonly known as '**Under-filing**'. This self-descriptive process involves removal of material from the concave surface of the vane tips (Fig. K.6).

Of course metal can similarly be removed from the convex surface and is known as '**Over-filing'.** This has no significant effect on Head/Flow performance but can have some beneficial influences on noise and vibration behaviour.

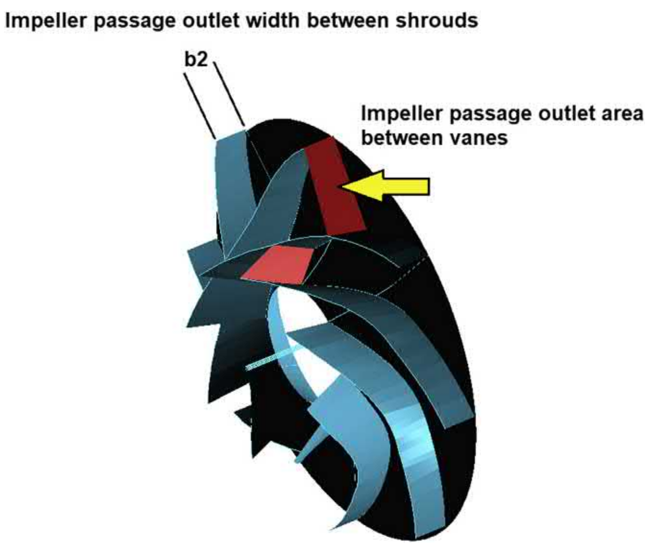



**Fig. K.5** The impeller passage width at outlet [b2] is a more convenient analogue of the blade swept area and/or the Outlet Area Between Vanes. This area is normal to the general flow direction. Proportional changes to b2, are reflected in the other two, all other things being equal.

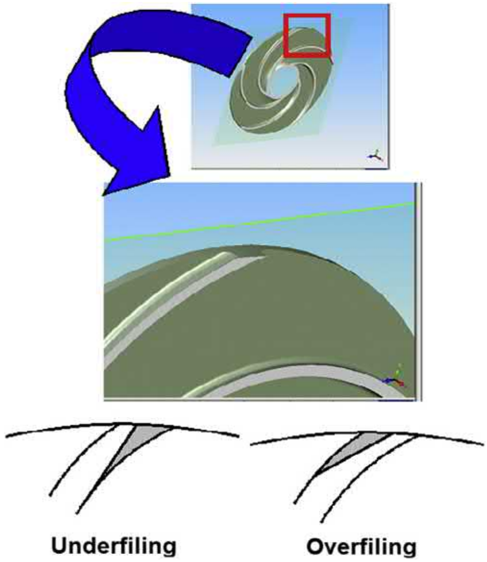



**Fig. K.6** Adjustments to the impeller blade outlet edge are the most common form of modifications. The performance effects of under filing are more profound than those of over filing. Over filing is chiefly considered as a means of reducing hydraulically excited vibration.

## Underfiling

Under-filing locally increases the vane-to-vane area at impeller passage exit. This local modification has a similar effect to the passage width [b2] being widened. In fact, the local vane setting angle is also increased slightly as well (Fig. K.7).

Because this is a combined effect, it cannot be well defined by one simple rule that suits all impellers – given the wide range of vane counts and vane setting angles that exist. Under-filing mainly influences the vane head contribution; which is higher in relatively wide impellers than narrow ones. So the effect of under file tends to be greater with relatively wide impellers (Fig. K.8).

If under filing is contemplated, it is a good idea to have the Manufacturer involved. But where he has gone out of business and/or no technical support is available, then under file should not be undertaken without qualified advice if ANY of the following conditions apply (Fig. K.9)

- Impeller vane count is less than 5
- Impeller diameter is greater than 380 mm [15 in.] and Shaft speed exceeds 3650 rpm [hence Impeller tip speed exceeds 70 m/s]

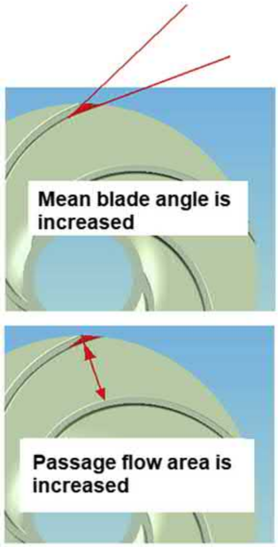



**Fig. K.7** Under filing has two effects: Firstly the local outlet vane angle is increased — this tends to increase the pump head by reducing the Impeller line slope see FIG K2. Secondly, the passage area between vanes at outlet is increased. From FIG K5, this is analogous to an increase in width, b2.

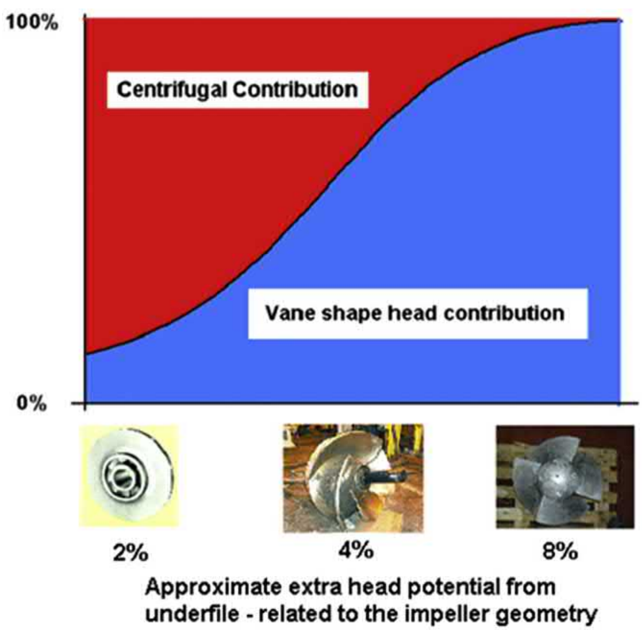



**Fig. K.8** Under filing can be conducted on all impeller, but it has more effect on impellers that are relatively wide compared to their diameter. These are just indicative numbers.

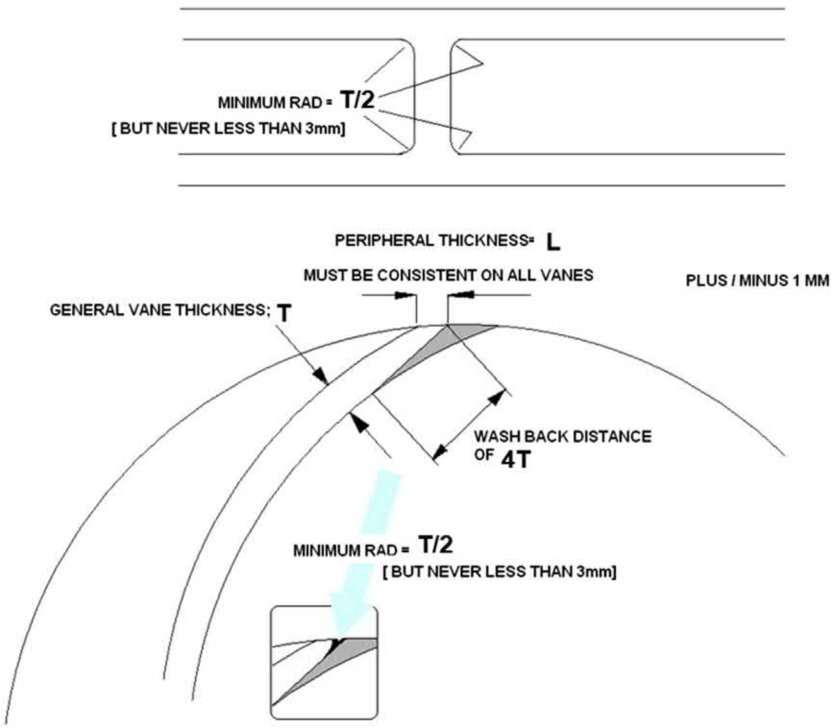



**Fig. K.9** These are very general guidelines for under filing. Attention must be paid to all the fillet radii between vane and shroud. This must always exceed 3 mm, so as to help avoid fatigue crack initiation under pressure pulsation conditions.

- Vane Thickness at outlet tip [T] is already less than 5 mm [0.2 in.]
- Impeller material is not ductile

This Figure gives some general guidance on under filing of impeller vane tips, where impeller peripheral tips speeds do not exceed the implied limit. One can expect under filing to 'pivot' the performance curve about the zero flow point and to progressively increase head and flow (Fig. K.10).

## Overfiling

In contrast to under-filing, over-filing has little or no effect in terms of performance output change. This is chiefly because the process does not change the passage outlet flow area between vanes. What it does do is locally reduce the vane camber angle so it will reduce the theoretical pump head. But this is not borne out much in practice. That is because the simple

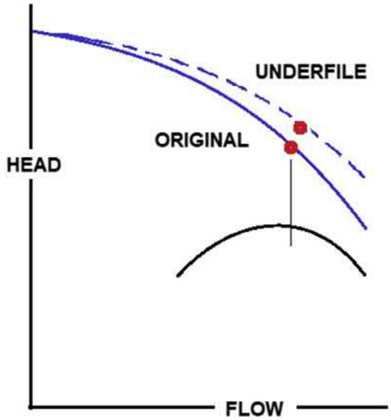



**Fig. K.10** For practical purpose, one can assume that under filing does not change the value of head at zero flow.

theoretical head calculation assumes constant velocities across the impeller passage whereas in fact there are significant departures from this assumption and the small change of local vane camber/setting angle is hardly influential.

I have mentioned that overfilling is sometimes employed in cases where vibrations at vane pass frequency are an issue. That is because the flow leaving the otherwise blunt trailing edge of the impeller vanes can create a significant turbulent wake interaction with any stationary collector vanes [Fig. 10.12]. Overfilling helps reduces the width of its turbulent wake and hence the intensity of hydraulic stator-rotor interaction.

## Impeller passage width increase

As an alternative to under filing, the passage width between shrouds [b2] can be increased. Typically this means grinding away metal to thin the shrouds. The impeller shroud thickness is an extremely critical dimension in high-energy machines, so this approach always needs approval from the OEM, or a suitably qualified agent. In smaller/low energy machines, the minimum mechanical design shroud thickness will almost always be less than the foundries can cast to so there is more scope here. Because of these concerns, and because the volume of material removal is large, this second approach is very rarely recommended.

### Collector modifications

The foregoing describes how increasing the impeller passage flow area can increase the performance output of the pump. Earlier I described how the best efficiency flow point was partly defined by the impeller geometry and partly by that of the collector. This is reiterated earlier in this Chapter by reference to the analysis of Worster [Ref. 25]. Therefore, it should come as no surprise that the pump performance output can also respond to changes in the collector geometry. If the minimum flow area of the collector can be increased, then the pump flow will also increase, and *vice versa*.

As a very rough approximation, one can say that for area increase ratios of less than 1.3, the flow will roughly increase by the square root of the change, and vice versa. This rough rule can help in initial feasibility thoughts.

The most common approach to increasing the area is to expand the so-called base circle diameter [Fig. J.66]. This can be conducted on both volute and multivaned diffusers. In some cases the material [or most of it] is removed by machining, but in others it has to be manually ground back in its entirety (Fig. K.11).

Attempts should always be made to regain the original tip thickness – even though the thickness will then increases at a greater rate in the flow wise direction. Blunt collector tips will increase hydraulically induced vibration levels. But the tips should not be too thin either, in order to help avoid cracking caused by pressure pulsations [see Figs J113 and J125].

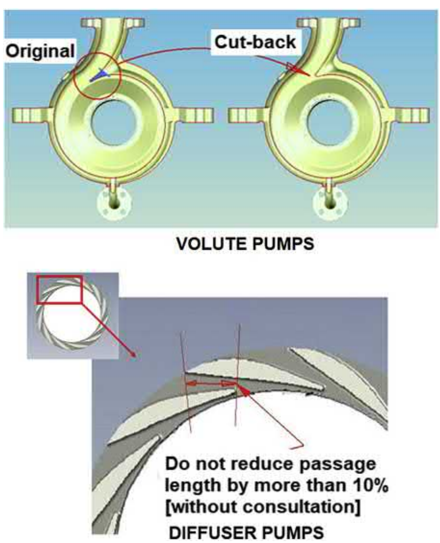



**Fig. K.11** Area increase of collector throats can be achieved by cutting back the collector tips. There are restrictions to this – particularly with diffuser machines.

Thickness values of less than 1.5% of the base circle diameter should be viewed with caution.

Since the passage areas after the so-called 'throat area' will diverge, this can help accomplish the necessary area increase. There will be a need to finesse the machining by hand grinding as well. In some rare cases, hand grinding alone can – if the cutwater tip is thick enough, give the required change. Occasionally, often for reasons of in-field convenience, the same effect is achieved solely by manually cutting and grinding.

The operation will of course increase the radial gap between impeller and collector vane [s]. This allows the distorted flow leaving the impeller to 'mix out' into a more uniform profile. Hence, the pump vibration levels will reduce slightly. However, the mixing process also incurs an energy loss, so the pump efficiency will most probably reduce slightly as well.

Collector area increase might be the unexpected by-product of modifications to help reduce horse-shoe vortex formation [Figs J108–J111] in pumps that are unexpectedly handling solids. Normally in such cases, this modification will result in less than 5% consequent increase. Hence, invoking the rough rule mentioned above, the flow will increase by less than 3%. This will rarely be problematic.

In Multiblade diffuser design, the passage length is often optimised with reference to experimental data in the public domain. This commonly refers to rectilinear passages being presented with uniform inflow. In actual pump diffusers, which are seldom rectilinear, the inlet flow is far from uniform – varying significantly over each passing impeller vane pitch. This may result in the passage flow picture cyclically fluctuating from stalled to unstalled, as each impeller vane passes. It may therefore be questionable to make much connection between the behaviour of the two. Nevertheless, this would imply that the diffuser path length should not be reduced by more than, say, 10%.

### Combined impeller and collector modifications

Of course, it is quite possible to invoke modifications to both impeller and collector, in which case the maximum effect will result. This was depicted in Fig. K.4. Fig. K.12 expands on this somewhat.

## *Pump pressure head exceeds the new system pressure head at the new flow and over-pumping results*

Sometimes production profiles dictate a reduction in pump output as time passes. In other cases the routine process requirements have changed. The

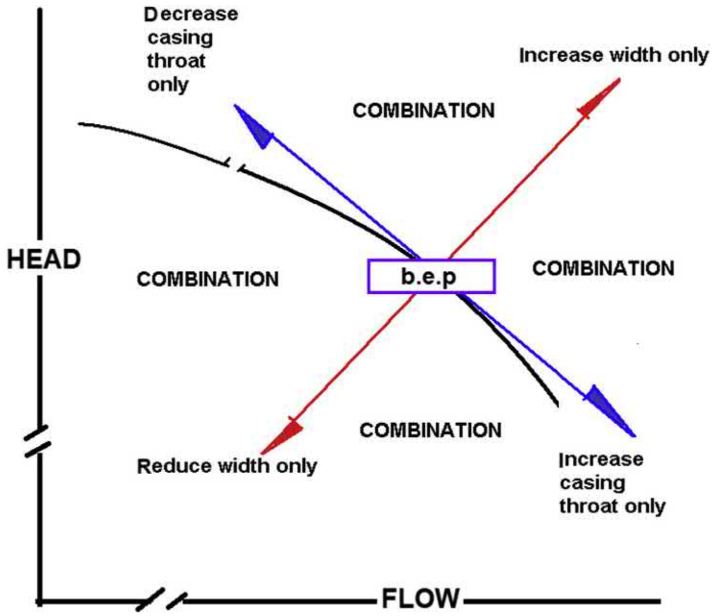



**Fig. K.12** An illustration of which changes are required in order to achieve a particular performance change objective.

most obvious ways of producing a corresponding reduction in pump output are as follows.

The least elegant approach to performance reduction is to simply dissipate the surplus head energy at the pump discharge [throttling] either by a control valve or by pressure head-reducing orifice plate. This acts as if the flow-related internal losses have been increased and the effect follows a square law.

However, since the impeller still does the same input work, the power absorbed remains the same. This is not, therefore, an efficient process for other than temporary or very short term changes. However, it has the attraction of being easily reversible. Where the orifice loss is exceptionally high, one can expect an increase in 'white' noise [Fig. 10.13] (Fig. K.13).

### Flow bypass or recycle

A more sympathetic approach than throttling is to bypass some discharge flow back to pump suction [Fig. 10.26].

This has some appeal in that the flow handled by the pump changes little. If the pump is already operating inside its COMFORT ZONE, this will not change. However, the liquid returning to the pump suction will

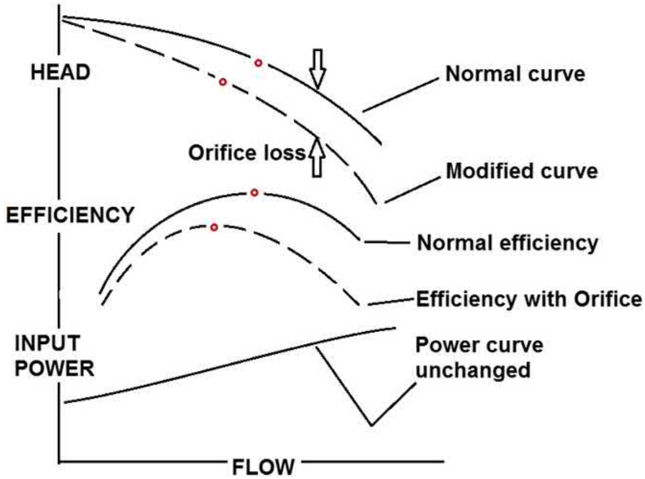



**Fig. K.13** Application of a pressure head reducing orifice to the pump discharge is a common way of temporarily derating pump output, in order to better fit circumstances. This is an inefficient process and should not be considered a long term solution. The orifice loss is generally assumed to be proportional to the square of flow. Not shown is the NPSHR curve which, like the absorbed power curve, remains unchanged.

always be at a higher temperature, due to the pump's inefficiency. So it might not be a viable solution in some extreme cases, such as very high power pumps, liquids with a high vapour pressure and or low site NPSH [A].

### Simple reduction in impeller diameter

This effect can be well predicted by the so −called 'Affinity Laws' [Appendix C]. The locus of best efficiency head and flow essentially follows a parabolic line, which in many cases is helpful. However, it is possible that the pump will be taken outside of its COMFORT ZONE by such a process [Fig. K.23].

### Reduction in impeller outlet area

Just as an impeller responds with more flow when its flow passages area is increase, the reverse is true when the areas are reduced. The slope of the theoretical impeller line and associated real performance line is increased. Area reduction is a much less convenient process, since it involves adding material to the existing pump component in some way. Consequently, this is only carried out in the field in very unusual circumstances. Reducing the

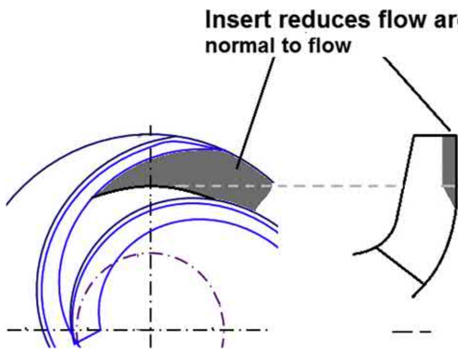



**Fig. K.14** Occasionally loose inserts are added in order to constrict the impeller passage are and consequently flow. This is only effective with relatively wide impellers and, in any event, is mechanically awkward.

impeller flow area will almost always involve a considerable amount of time in cutting and attaching area chokes in some way (Fig. K.14).

Even so, the physics of pump hydraulic design means that substantial changes are only effective when the width ratio, b2/D2, exceeds [say] 0.15. At values much lower, then the effect is rarely worthwhile. Various methods of attachment have been used and the selection depends upon materials and impeller speed. The size of such inserts means that the initial dynamic balance will almost always be compromised and rebalancing will be required. All this means that this approach is very much a last resort, whose chief benefit is a reduction in end-of-curve power absorption *[By increasing the slope of the theoretical and actual pump curve, the flow and power at flow greater than the [new] best efficiency flow are reduced. It may even become non-overloading* [Fig. 8.15].

## Reduction in casing throat area

A corresponding option is to reduce the throat area of the collector. With a volute casing design, this can be achieved by welding *[where possible]* extensions to the cutwater (Fig. K.15).

This is most readily carried out if the extension simply follows the casing base circle diameter. This causes a compromise in hydraulic design terms but is useful in emergencies. Roughly speaking, the passage areas of a volute casing are proportional to angular distance from the collector cutwater, so [say] at 270 degrees from the cutwater, in the direction of flow, the passage area will be roughly [270/360] of the final throat area. So if the throat area needs to be reduced by 10%, then the welded cutwater needs to extend approximately 36 degrees further back around the base circle.

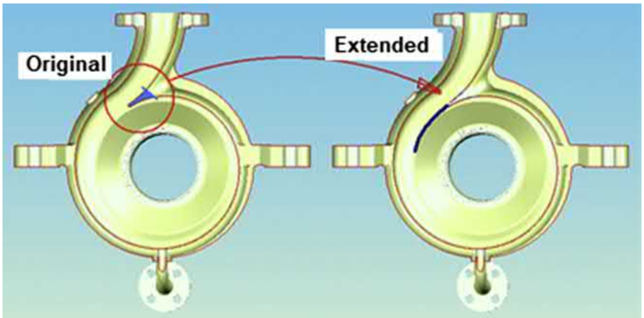



**Fig. K.15** One simple way of reducing collector throat area is to weld extensions to the volute cutwater. Although this is a slight hydraulic compromise, it has proven practical and effective.

Where welded extensions are not possible, then cast and shaped attachments have been bolted to the cutwater to accomplish a similar area reduction. One such item is the so-called 'slipper' inserts; a term which adequately describes their appearance. These 'slipper' attachments have the advantage of being easily removable and hence reversible (Fig. K.16).

In another approach, the weld material is built up opposite the cutwater and suitably shaped. This is quite labour intensive and not easily reversed, so again it is only considered as a last resort (Fig. K.17).

Multivaned diffusers have less scope for similar such area reduction and in general are less suitable for modification. If the collector has an even vane count, then alternate passage can be blocked off with inserts, but this would


**Fig. K.16** Attaching pre-formed insert to the volute casing is an efficient and effective approach to area reduction.

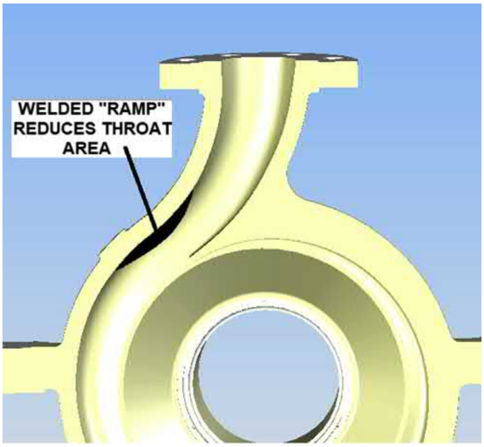



**Fig. K.17** It is possible, with weldable materials, to create a build-up that can be ground and dressed to shape. This is effective, but may be costly.

only be justified in a real emergency *[to help in short term reduction in end of curve flow and power, say]* and has very little to otherwise recommend it.

At this point it might be worth summarising the conceptual effects that these different modifications can make, either independently or collectively (Fig. K.18).

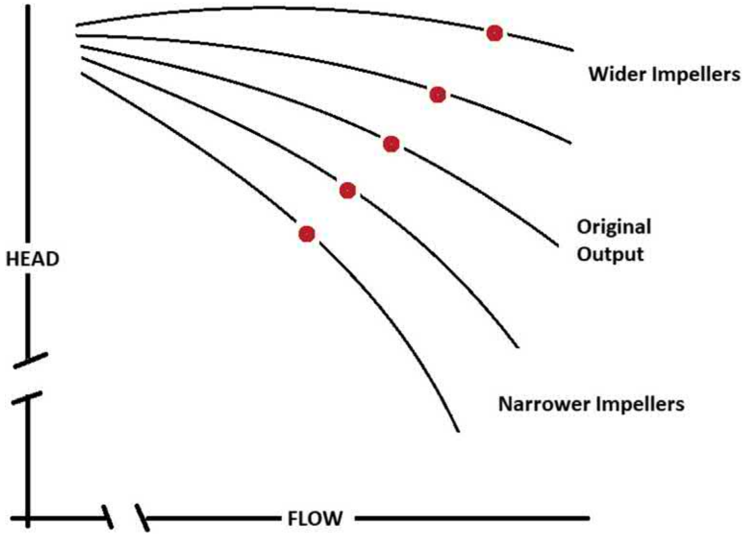



**Fig. K.18** This illustrates the general effect of impeller passage width increase. As the impeller gets relatively wide, there is actually a tendency for the head at zero flow to reduce a little.

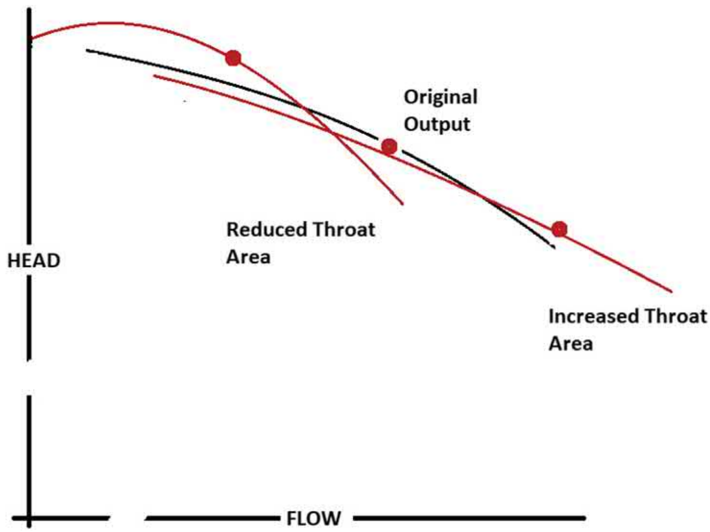



**Fig. K.19** With change of throat area, the point of best efficiency more or less slides up or down the original curve.

Increasing or reducing ONLY the impeller outlet width [and hence the passage outlet area between vanes] has the effect of NOMINALLY pivoting the performance curve about the zero flow point (Fig. K.19).

Increasing or reducing the collector total throat area by less than [say] has the effect of NOMINALLY sliding the best efficiency point along the original performance curve.

Increasing or reducing both the outlet width [passage outlet area] in the same ratio has the NOMNAL effect of sliding bep to greater or less flow at the same or similar pressure head.

## Pump can be modified to meet the new system requirements, but now does so at a flow far removed from its point of optimum efficiency, maybe even outside its comfort zone

Modifications aimed at increasing or decreasing the output of existing components have centred on the impeller rim outlet and the collector inlet. Very little can similarly be achieved solely by modifications to the impeller inlet *[Unless it was basically deficient in the first place].* Actually, in terms of behaviour, the zones of impeller inlet and impeller outlet can usually be considered in isolation and not linked. Semi-Open and Open impellers can easily be machined in such a way that their inlet and outlet geometry can be

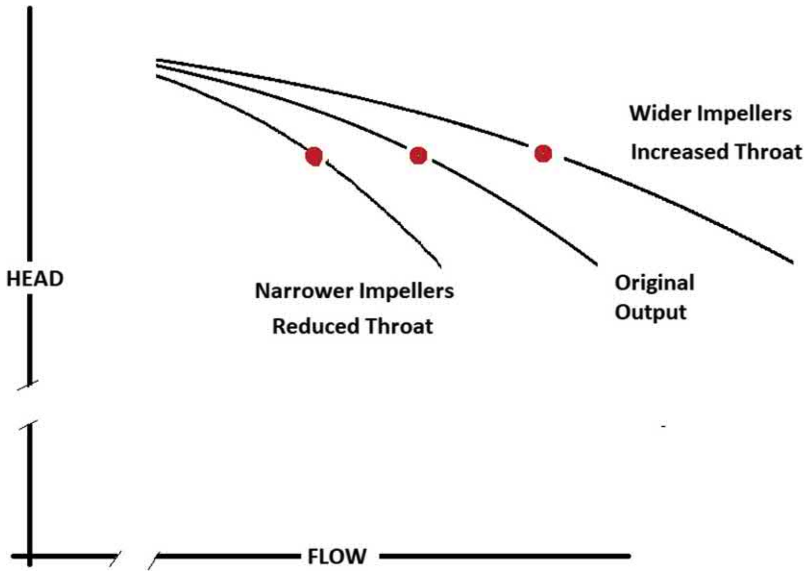



**Fig. K.20** If the impeller and collector areas are changed in the same proportion, then the output flow maintains almost the same pressure head.

adjusted simultaneously. For example reducing the outlet width b2 of such impellers will reduce the best efficiency flow point. Without any other concurrent adjustments, the optimum inlet flow will remain the same and obviously mismatched. But by suitable machining, the optimum impeller inlet flow and outlet flow can be brought into line at a stroke. But two separate adjustments would be needed in order to achieve the same conditions with shrouded impellers (Fig. K.21).

As an illustration, an open or semi-open impeller could have its outlet width reduced from A to B. This would reduce the flow in general and the best efficiency point in particular. However, if the inlet width remains unchanged at 1, then the range of optimum inlet flow does not change either. If, on the other hand, the inlet width is also reduced to [say] 3, then the optimum inlet flow range would move in step with the optimum outlet flow. Reducing the inlet width to 2 would not move the optimum inlet flow as much.

We are familiar with the classic NPSH [R] curve which rises more or less continuously as flow increases from a Minimum Flow out to Run-out. It will be obvious that increasing the pump output by modifications in the impeller outlet region alone does almost nothing to change this NPSHR curve. So with a nominally fixed site NPSH [A], the NPSH margin will

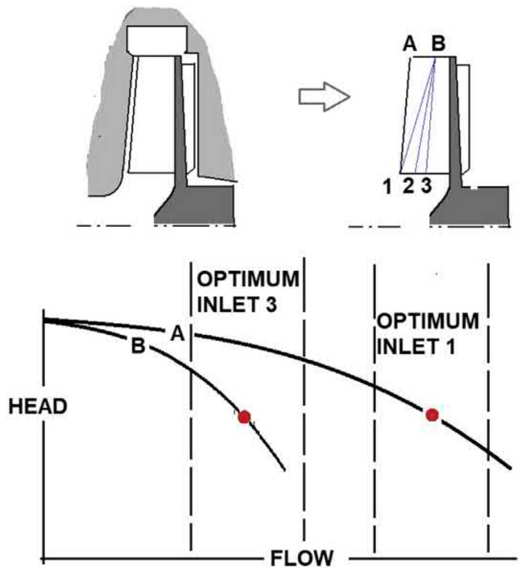



**Fig. K.21** If this semi-open impeller is machined to follow line B1, the outlet flow will shift to a lower flow but the optimum inlet flow and NPSHR curve will be essentially unchanged. However, by machining instead to Line B3, the optimum inlet can be shifted in step with the outlet. Machining to B2 will achieve a partial shift. It is more difficult to accomplish this simultaneous shift when modifying shrouded impellers.

clearly be reduced *[But in fact, the NPSH [A] will also most probably reduce as flow increases, so unfortunately the nett margin will decrease a little bit more.].*

But it can come as a surprise that reducing the design point flow can also reduce the margins. At various point in this book I have shown that even though the classic 3% NPSHR curve continues to reduce as flow is reduced, at least up to the point of inlet backflow onset, this is not always true of other aspects of hydraulic behaviour.

Even though the 3% head decay NPSHR is reducing, the NPSHR to prevent cavitation damage will actually rise as flow reduces below design flow. So while the low flow NPSH margin might appear to improve from a performance decay perspective, it can actually be reducing in terms of cavitation damage prevention. Generally it will continue to do so until flow has been reduced to the inlet backflow onset point, after which it is most likely to reverse and fall again. And this behaviour will be mirrored in the noise and vibration performance.

For example,

Consider an impeller originally designed for a flow of 100 flow units. If the impeller diameter were to be reduced to [say] 70% of its design

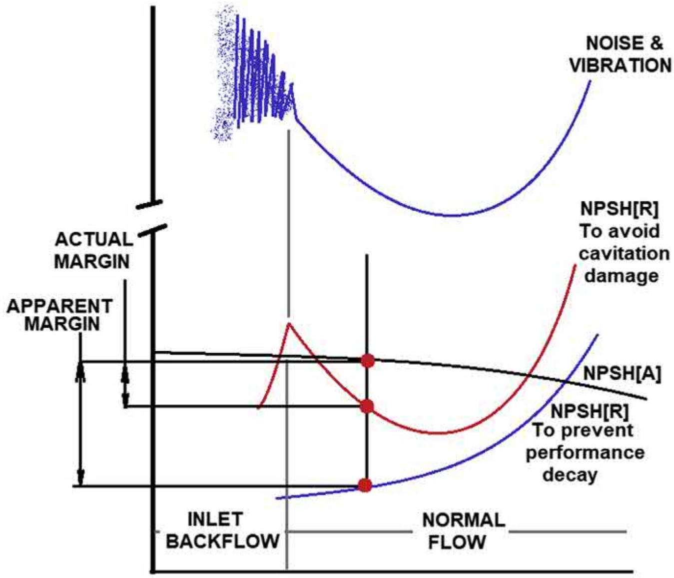



**Fig. K.22** Characteristics such as Noise, Vibration and Cavitation damage rates exhibit minimum levels at or near to the pump optimum inlet flow. The line defining 3% decay in head, due to NPSHA deficiency, departs from this pattern and through most of its practical range shows no minimum. This later curve is more widely known and can create the incorrect illusion that low flow operation generates an improved safety margin against cavitation.

diameter, then the new pump best efficiency flow will nominally be 70% of the original-say 70 flow units [In accordance with the Affinity Laws.]. However, the inlet design flow remains at 100 units. Imagine now that the collector is modified by reducing its throat area. The combined effect is to reduce the best efficiency flow of the reduced diameter impeller to let us say 40 units. On the face of it, this new configuration of impeller and collector might provide a better hydraulic 'fit' to the new reduced system flow requirements. But the inlet is still designed for 100 units so in fact, due to the combination of impeller trim and throat area reduction it is now forced to operate at 40% of this. This may well take the pump into its 'backflow' regime [below its COMFORT ZONE] and make it vulnerable to associated operational problems (Fig. K.23).

It is now widely appreciated that once a pump's flow is reduced to between 25% and 40% *[Depending on the impeller inlet tip speed*] the flow picture at impeller inlet breaks down [The *flow picture collapse is described in*

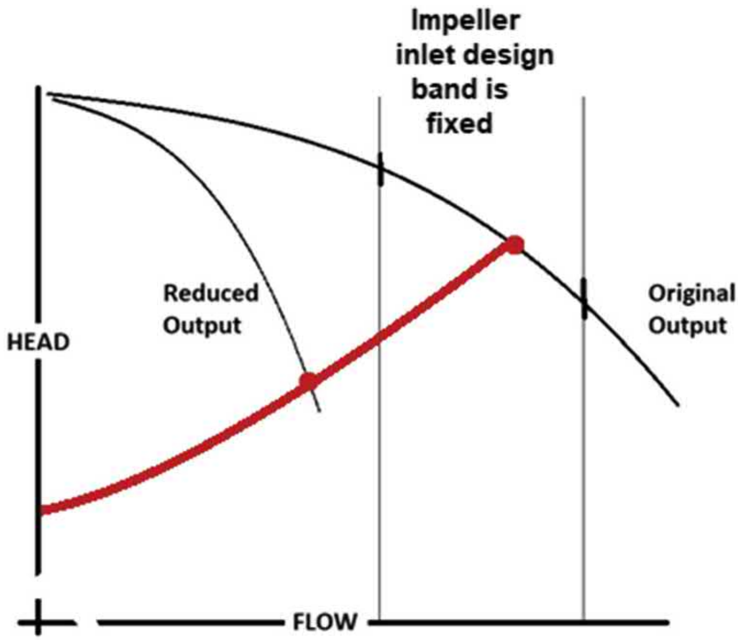



**Fig. K.23** Fig. K.20 illustrates how semi open impellers can simultaneously have their optimum inlet and outlet flow adjust together. This figure shows how, if a shrouded impeller has its performance decreased by the various means described, the inlet performance remains unchanged unless otherwise addressed. A similar event occurs, to a lesser degree, as and when an impeller has its outer diameter trimmed to reduce its output.

*Section D, which also outlines a method of estimating the onset.*]. This condition can trigger several undesirable issues and so is best avoided.

As another example,

It is possible in some circumstances to substantially increase pump flow by a combination of, say, impeller under filing and collector throat area increase. Perhaps an increase of 15% was achieved. If the pump was already operating at 5% beyond it bep flow, these modifications would result in the impeller inlet now operating at more than 120% of its design flow. Again, this might now take the pump beyond its COMFORT ZONE and into the region of increased cavitation damage risk. Modifications to existing impellers do not generally have the potential to increase flow by much over 15%, though isolated exceptions do exist. Therefore, while the high NPSH margin issue cannot be disregarded in such cases, it does not always become a constraint.

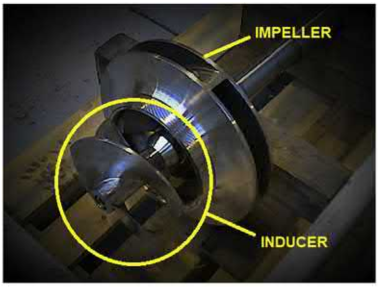
These two examples are of course contrived, but they show how modifications to the impeller outlet and or the collector inlet can inadvertently put the pump at risk of cavitation damage. This arises because the impeller inlet geometry will not have been modified in step with such changes. In fact only limited scope normally exists to make any concurrent adjustments to the impeller inlet geometry-particularly with shrouded impellers. These modifications will be described later. But if it happens that the system flow is being increased or decreased significantly, the most elegant option is to install an alternative design of higher or lower flow impeller, as described earlier. So, in brief;

- Substantial Design point Flow reductions achieved by combined modifications to impeller outlet and casing inlet **alone**, can also increase the risk of generic 'low flow' problems if not identified and addressed at the outset. This might include;
  - Cavitation damage to;
    - Impeller vanes – most probably on the concave visible surface.
    - Inlet duct surfaces near to impeller.
  - Noise increases: see [Fig 10.10].
  - Increase in vibration levels.
  - Introduction of low frequency noise and flow surges.
- Substantial Design point Flow increase achieved by combined modifications to impeller outlet and casing inlet **alone** may reduce the NPSH margins to an unacceptable level. Unless correctly identified and addressed at the outset, this raises the risk of;
  - Cavitation damage-most probably on the convex invisible vane surface.
  - Increased noise and vibration.

Generally when reworking an existing impeller to deliver more or less flow the inadvertent risk of increased long term cavitation damage is the issue of most concern. So, in the absence of an alternative impeller, I will outline some methods of somewhat reducing that risk with the existing item.

### *Cavitation damage intensity has increased because pump has modified so as to shift design flow/bep [at max diameter] to a higher/greater value*

It is known that the most obvious outward signs of cavitation, such as performance drop-off NPSH [R] become more pronounced as pump flow increases. The less obvious signs such as noise, vibration and cavitation erosion damage also increase once the zone of pump best efficiency flow has been much exceeded. Therefore, any modification that results in significantly increased pump flow [say more than 10%] might run the risk of inducing problems from all or any of these sources. In practice, the risk mounts quite quickly once bep flow zone has been exceeded [Fig. K.22]. This often limits the options to any increase in the flow of an existing impeller by modification, unless the NPSH [A] can simultaneously be increased substantially. One way of achieving this increase would be to install a separate booster pump ahead of the main pump and in series with it. This pump has a lower NPSH [R] [*possibly because it runs at low speed and/or configured with a double suction impeller*] and so can operate safely, while boosting the NPSH [A] to the main pump and restoring its margin. This is very occasionally done, but is not a widespread practice. In a limited number of cases, this booster is known as an 'inducer' and is integral with the main pump. These inducer devices are most often available in, 'end suction/overhang impeller' pumps. They are much less common in 'between-bearings' machines (Fig. K.24).



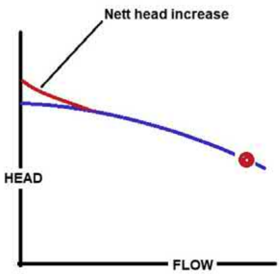




**Fig. K.24** Inducers can operate while cavitating themselves, yet still generate enough head to avoid cavitation in the centrifugal impeller. The assembly operates as two impellers in series. The inducer will be susceptible to inlet backflow of course, but its presence in the inlet duct inhibits inlet backflow in the main impeller. This and the head from the inducer in series will tend to increase the low flow head output of the combination [Pompes Guinnard].

Substantial flow increases are really best addressed by alternative higher capacity impellers, as alluded to in the above section. By definition, these impellers will also have a bep flow zone that is displaced to a higher flow. Most probably, the NPSH margin will be more favourable than with a modified impeller, but it is still not guaranteed to be acceptable. It might still be prudent to conduct a detailed review.

Repositioning the NPSHR curve of an existing impeller to a higher flow requires some basic hydraulic understanding and ability to assess the impeller geometry. However, in an emergency any short-term corrective measures might be useful.

As a generalisation, the high flow part of the curves for both the high and low head decay conditions, follow each other very closely. Actually, the high flow part of the NPSH curve is largely controlled by the minimum impeller passage area [normal to flow]. This will occur at or very near to the impeller passage inlet.

In Section E, I describe that in general, the early stages of performance head decay [i.e. low NPSH margin] are chiefly governed by the shape and sharpness of the impeller leading edge.

Further reducing the NPSH margin, leads to more advanced performance head decay. These later stages of decay are more influenced by the minimum flowpath passage area [normal to flow] (Fig. K.25).

Therefore, the potential for significant high flow NPSHR improvement by local sharpening or re-profiling the impeller leading edge alone is not great. Although still limited, much more can be accomplished by actually

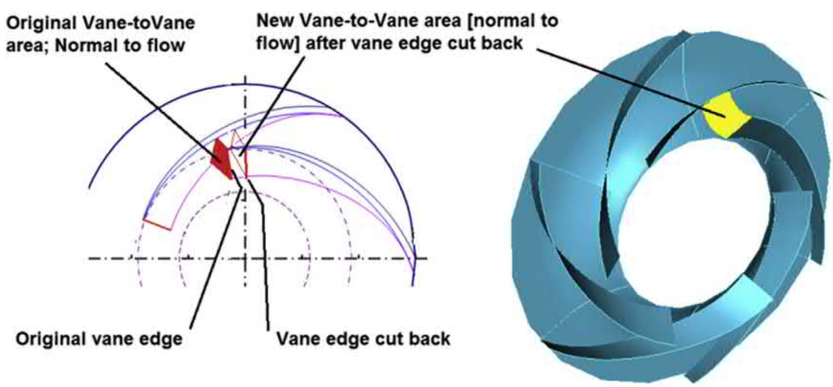



**Fig. K.25** The impeller passage inlet area between vanes is a major factor in helping to determine the HPSHR [3% head decay]. Removing small sections of the vane [cutting back] will increase the area and can help to reduce the NPSHR to some extent.

cutting back the impeller leading edge. This increases the inlet vane-to-vane passage area. Given that these are likely to be desperate 'in field' measures, only general guidelines are given here.

Firstly, the inlet vane-to-vane area needs to be assessed. If the subject pump has more than one stage, only the first active stage needs to be considered. The most practical method of in-field assessment is to create a cardboard [or similar] template that will JUST pass through the minimum point in the impeller flow path. This can often be an iterative process involving several failed attempts. Since this a field measurement, great precision can rarely be achieved. Nevertheless some care should be taken at this stage. In some cases, it might be very difficult or even impossible to make a satisfactory template. In that case, the only [reluctant] alternative is to estimate the inlet passage area height [front-to-back shroud gap] and width [vane leading edge to nearest point on the adjacent vane] by [say] internal callipers. This involves some judgement as to where the mean line lies between each pair of surfaces. This is quite a compromise, but is better than nothing.

The next step is to estimate what flow shift is required in the NPSHR curve. The higher flows part of the NPSH curve can be shifted approximately in proportion to the amount that the inlet area between vanes [A1BB] normal to flow can be increased (Fig. K.26).

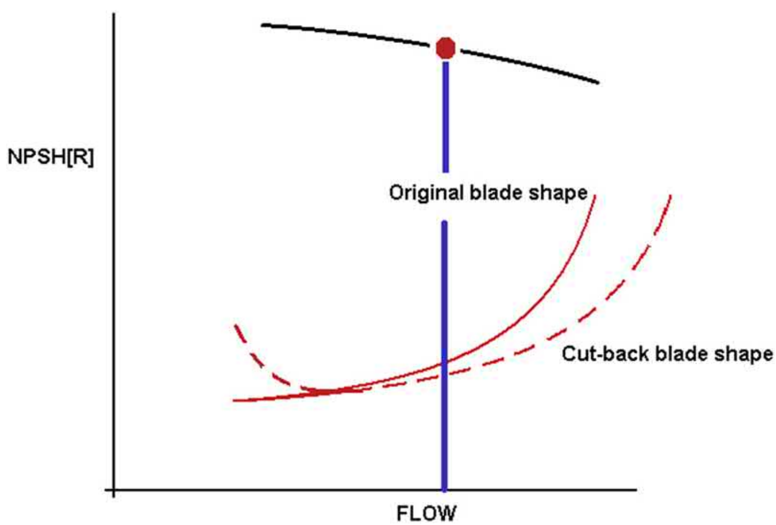



**Fig. K.26** Increasing the vane to vane area at passage inlet will chiefly influence the high flow part of the NPSHR curve. The low flow zone will also be affected if the cutback process is carried to excess.

In other words, if the total area [normal to flow] can be increased by 10% consequent to cutting back the leading edge, then that high flow part of the NPSH curve will be shifted by no more than 10% towards a higher flow *[This presumes all other parameters such as vane edge sharpness and profile can be regained.]*.

Since in the field this entire process is imprecise, it will be prudent to factor in a contingency margin of ignorance. This margin cannot have a fixed value, given the different range of applications, levels of criticality and diversity of designs. But it should probably never be less than 10%. The inlet passage area measured by template in the first step should then be increased by the flow increase ratio and then by the contingency margin of at least 10%. If only passage height and width measurements by calliper were possible, this minimum margin should be increased by [say] 15% or more.

Area increase ratio = New design flow/original design flow × contingency margin

A new template is now created, based on the profile of the original. To this original are added incremental strips equivalent to the desired area increase. This increment must be shared between the two vane surface sides of the vane-to-vane template (Fig. K.27).

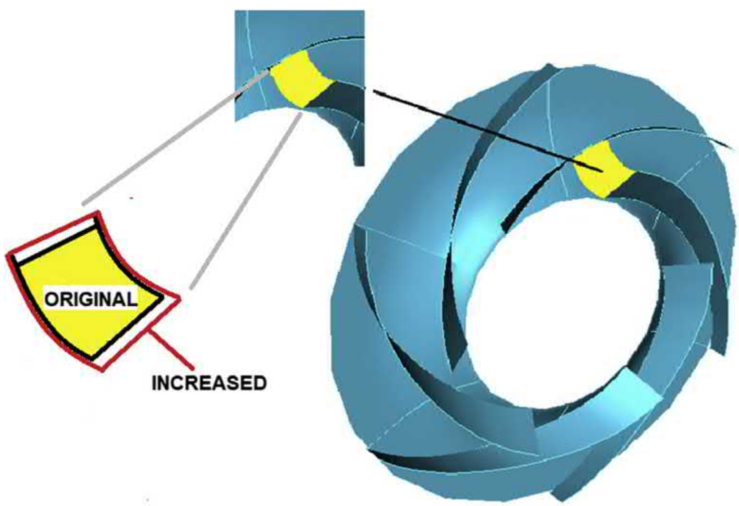



**Fig. K.27** The incremental area increase should be shared by both sides of the template that touch the vane. The remaining two sides of the template should not normally be touched and the corresponding hub and shroud profiles only cleaned up after vane cutback.

Material is removed from each blade inlet edge such that the new template can now pass through the minimum area point. The new edge should be more or less parallel to the original. It is a fact that most impeller passage inlets will diverge appreciably in the vane-to-vane direction, while converging very much less in the shroud-to-shroud direction. In removing material only from the vane/blade, the divergence vastly outweighs the convergence effect.

- Expect that this new template will be tight in the shroud-to-shroud direction; hence some further iteration will be called for.
- As a general constraint, the vane length, as measured in the direction of flow along a mean line, should not be cut back more than 10%. More is possible but expert guidance should be sought. The chief risk here is that the pressure head at zero flow will reduce and produce a performance curve with an unsatisfactory and flatter shape-perhaps even unstable. Vibration levels may increase. Both these potential effects result from increased flow separation inside the impeller passage, since the shorter blade is more heavily loaded.
- In order to avoid other complications, neither the hub nor shroud profile should be significantly modified in the process *[Other than cosmetic dressing to tidy up where after the vane edge has been cut.]*.

In the cut back process, it will usually be the case that the vane thickness [normal to flow] increases a little. The vane leading edge profile MUST now be restored to at least the original thickness. This is to ensure that the incremental area increase is achieved. It also helps to ensure that the flow resistance created by the vane itself is not increased. This involves removing material which ideally should be taken from both convex and concave faces of the blade in equal amounts. In practice it is sometimes very difficult to remove significant amounts of material from the convex [less visible] face. There is then a temptation to achieve the thickness reduction by removing material only from the concave [visible] face. This is counter-productive, because it results in the local inlet vane setting angle being now reduced whereas, optimum flow physics requires the angle to be increased, or at least maintained.

### *Cavitation damage intensity increased if pump modified so as to shift design flow/bep at max diameter to a lower/smaller value*

Contrary to the case with high flow cavitation, the most obvious outward signs, such as performance drop-off NPSH [R] can become less pronounced as pump flow decreases. It is widely assumed therefore that the less obvious outward signs such as noise, vibration and cavitation erosion damage also decrease. Consequently, the assumption arises that;

- The NPSH margins will have increased.
- On the whole, the picture is improved in this respect.

Section E outlines that this is not the case and in fact all three phenomena increase with progressive flow reduction below the bep flow zone. Note that flow reductions resulting from modifications to the casing alone, could be substantial, yet the original design flow zone for the impeller inlet would not have changed, so all three of the less obvious distress signs will be unchanged [Fig. K.22].

Once again, substantial flow decreases are best addressed by lower capacity impeller designs *[unless flow bypass methods are employed. Such approaches tend to keep the pump in its COMFORT ZONE-if it was in the first place.]* Low capacity designs will also have a correspondingly lower COMFORT ZONE (Fig. K.28).

Regarding NPSH margin, the same caveats apply, as with High capacity designs and a detailed review will be prudent.

Earlier I outlined emergency measures to help relocate the high flow section of the NPSH curve to a higher flow. This was accomplished by increasing the minimum cross sectional area on the flowpath. Similarly, the low flow part of curve can be displaced to a lower flow by reduction of the flowpath areas. This is more difficult given an existing component.

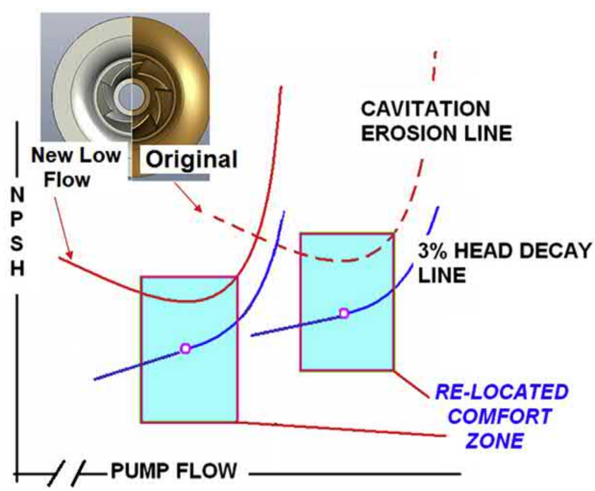



**Fig. K.28** Shows how low flow designs will relocate the pump comfort zone.

Increasing area was achieved by removing material whereas reducing area requires adding material.

It is true that the inlet passage area normal to flow A1BB could be restricted with, say, welded inserts. However, a crude but more practical approach utilises what is various known as a choke ring or bull ring. This is described in Fig. K.29 and Appendix K.

The overall aim is reduce the area of the inlet annulus in proportion to the outlet flow reduction accomplished by any of the other means described earlier. As before, these are likely to be emergency measures, where no expert guidance is available. Therefore, only general principles are described.

So if the equilibrium flow has been reduced to 70%, of its original bep flow then the area of the inlet annulus, normal to flow, needs a similar reduction. In most cases, either a shaft [between bearings machines] or an impeller retainer [overhung impeller machines will pierce the inlet area]. So this needs to be factored into the calculation *[In fact, the NPSHR curve for an impeller depends not only on the axial flow velocity, but also the peripheral tip speed. So while the choke ring can help reduce the axial velocity proportionally, the tip speed does not reduce in step, if a shaft/nut restriction exists.]*.

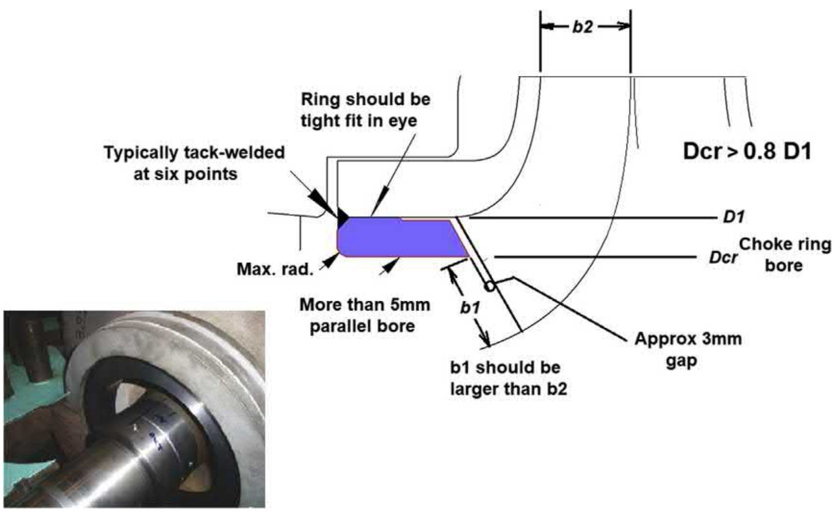



**Fig. K.29** Choke rings have a long history of helping to solve inlet backflow related problems. The reduction in eye diameter [D1] is rarely more than 80% for pumps in the field. Larger reductions usually require shop test qualification because the NPSH [R] will increase.

Fig. 10.44 shows that the while the 3% NPSHR curve is scaled to lower flows, it unfortunately experiences a small increase on the Y axis. This mostly results from the abrupt step in the flowpath caused by the ring. However, this will be more than offset by the benefits of inlet backflow suppression [where it exists of course] and relocation of the pump COMFORT ZONE. In any event, long term cavitation damage rates should be significantly reduced.

Obviously the sizing of a choke ring is best accomplished on the test bed, or in direct consultation with the OEM. However this is not always practical – the OEM may not even still be in business! But the need to do something positive still exists-even in the short term. In such an emergency, only very limited hydraulic information is likely to be available. But some reasonable assumptions can be made that wil help towards an initial choke ring design.

Generally a simple spreadsheet can be constructed to help these calculations. I list here the steps involved.

- Firstly an ideal [but simplified] model of the impeller inlet annulus is generated.
- Then a performance prediction is made for this annulus. This is compared with the real performance. This idealised annulus design, will not include all the nuances of the real design, so there will be differences.
- By adjusting the assumptions, the idealised model is crudely calibrated to the real performance. If these adjust are too severe, it is not recommended to proceed. It means the assumptions are inadequate.
- Finally, the eye diameter of the calibrated model is adjusted, within constraints to suggest how muchthe inlet backflow point might be adjusted and the consequent increase in NPSH [R].

The first information needed is the Impeller eye diameter; D1 and the size of the hydraulic hub D0. If the pump is as shown in upper Fig. K.29.1, this data is unambiguous and can be easily measured.

However other configurations such as lower Fig. K.29.1 make this measurement more arbitrary. In view of all the possibilities, I can only advise that what seems a sensible decision is made.

The main purpose here is to try and determine the free eye are, normal to flow.

$$\text{Amo} = [\text{pi} \times \text{D1}] - [\text{pi} \times \text{D0}]\ \text{metre}^2$$

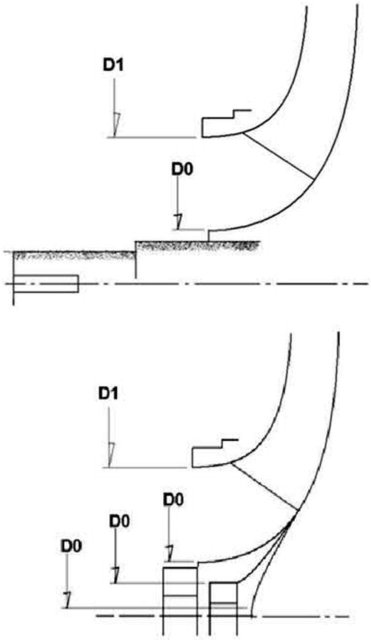



**Fig. K.29.1** Determining the precise of the entry orifice, the 'eye' may be straightforward as in the upper example, or sometimes judgement is required, as in the lower designs.

From a test curve – corrected to maximum impeller diameter – as outlined in Fig. K.39, the flow at best efficiency point is determined.

The average axial flow velocity into the impeller eye follows from (Fig. K.29.2):

$$\mathrm{Cm0} = \mathrm{Qbep/Amo/Number\ of\ Impeller\ eyes}$$

Generally, the velocity entering the vane array will be slightly higher than this; an acceleration of 5% is typical. So $\mathrm{Cm1} = 1.05 \times \mathrm{Cm0}$. Furthermore, the impeller will be continuous handling any leakage flow, as defined by the volumetric efficiency. So for practical purpose, we can take this to be 5% of Qbep.

This leads to an **average** value of $\mathrm{Cm1} = [1.05/.95] \times \mathrm{Cm0}$.

However, due to the streamline curvature in the radial plane, the liquid velocity near the front shroud wall will be higher, while at the rear hub wall it will be lower. The scalar values of 1.2 and 0.8 are only typical of many radial flow machines and some variation may be expected (Fig. K.29.3).

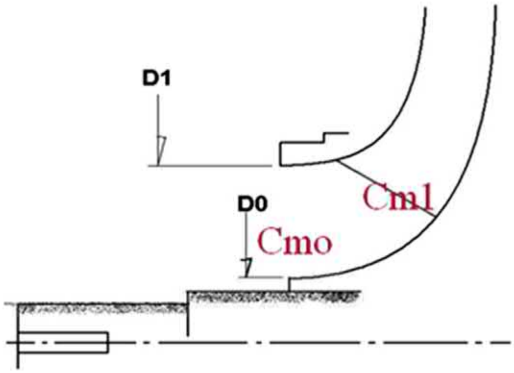



**Fig. K.29.2** Designers will normally accelerate the liquid from its entry value, Cmo, to a higher value just as it enters the vane array, Cm1. This helps to stabilise the liquid flowpath.

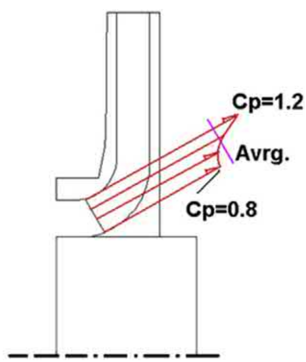



**Fig. K.29.3** The liquid velocity varies across the inlet annulus, due to the streamline curvature. Viscous friction will somewhat reduce the values of 1.2 and 0.8 but this is a secondary effect.

Finally we can approximate the liquid velocities in the radial plane as;

$$\text{Cm1-shroud} = [1.05 \times \text{Cm0}/0.95] \times 1.2\ \text{m/s}$$
$$\text{Cm1-hub} = [1.05 \times \text{Cm0}/.95] \times 0.8\ \text{m/s}$$

This calculation is applicable to pumps where the inlet duct is axisymmetric. Where the flow approaches in a direction perpendicular to the shaft, these values could be multiplied by a further 1.05 to acknowledge velocity variations in the duct [see Chapter 3].

We can also estimate the peripheral velocity [or inlet tip speed] at both D1 & D0:

$$U1shroud = pi \times D1 \times RPM/60 \text{ m/s}$$
$$U1\ hub = pi \times DO \times RPM/60 \text{ m/s}$$

Simple fluid mechanics [*Velocity Triangles]* then tells us that the fluid approaches the impeller blade at an angle such that,

Tan [shroud approach angle] = Cm1-shroud/U1-shroud − degrees
Tan [hub approach angle] = Cm1-hub/U1 hub − degrees

In many or most cases, the vane setting angle will equal these angles or exceed them by [say] 2 degrees, depending on the shape of NPSH [R] curve required. The following process has to assume this to be true.

Calculation so far has provided us with some further information; some indication of what the vane setting angles are likely to be. In some case, the impeller will be available and the setting angle may actually be measured directly. Figs D.18 and D.19 outline ways in which this might be accomplished. But it is worth conducting this calculation anyway, as a reality check.

If using a spreadsheet [*strongly recommended]* it should be laid out in such a way that D1 and Q bep can be variable input.

We can now take advantage of the equations outlined in [Ref. 25] to attempt to model the NPSH [R] behaviour of the machine.

These equations can be simplified to;

NPSH [R] = [[1 + Cb] × Cm1-shroud$^2$/2g] + [Cb × U1-shroud$^2$/2g] − metres

Experience says that the blade cavitation coefficient, Cb can be approximated by,

$$Cb = [[\text{TanBlade setting angle at shroud}]^2] + 0.04$$

Assuming-

- Blade setting angle corresponds to the approach angle determined above Plus or Minus 2 degrees.
- Vane thickness is less than 0.02 × D1. Where Impellers fail this criterion, the constant 0.04 will need increase − but I have no simple guide for this.

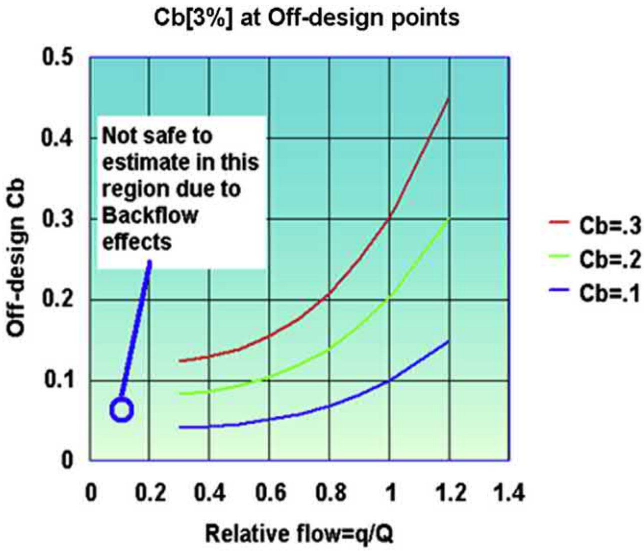



**Fig. K.29.4** This shows pseudo-cavitation coefficients derived from statistical analysis of tests and following the principles of Gongwer [Ref. 25]. Impellers with few vanes will exhibit flatter curves, while those with many vanes will be vice versa.

This allows the NPSH [R] to be estimated at the nominal zero incidence condition [q/Q = 1].

Fig. K.29.4 helps to estimate how Cb varies with change in flow. Entering this chart at q/Q = 1, with Cb value developed above, allows Cb for a range of [*Relative*] flows to be estimated.

If the calculations so far have been conducted on a spreadsheet a more complete NPSH [R] prediction can now be produced.

At this point the known NPSH [R] curve for the pump should be introduced to the sheet, for graphical comparison purpose [Fig. K.29.5, Chart 1].

Precise agreement cannot be expected — given the many assumptions made — some of them explicit and some implicit. But there should be some broad agreement. If not, then the sheet value for Q bep can be increased or decreased [Fig. K.29.5, Chart 2]. *[This is quite justified, because not all designers prescribe the zero incidence flow to be the same as bep flow.* That would be the case, for example, where unusual NPSH [R] behaviour was desired]. In extreme, the value of the constant in the Cb prediction used 0.04 can be increased.

If no reasonable agreement still results, then **STOP!** The pump configuration lies out-with the assumptions made here.

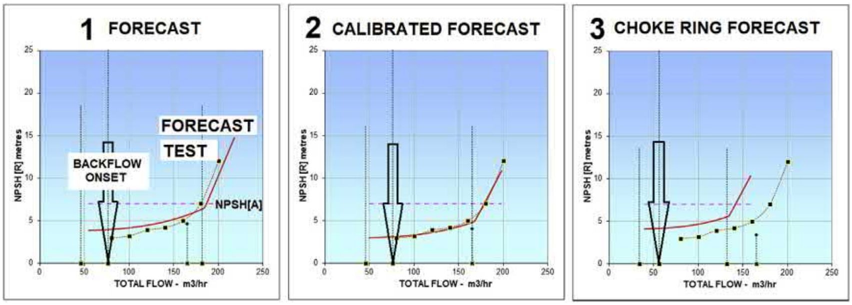



**Fig. K.29.5** NPSH [R] forecast curves will probably differ from the real test curves. By some adjustment to assumed zero incidence flow and Cb, the curves might be brought into agreement. The calibrated spreadsheet can then be used to help forecast the effect of changing the Impeller eye diameter using a choke ring.

If reasonable agreement still persists, then, in the spreadsheet reduce eye diameter D1 and observe both the shift in backflow onset, zero incidence and increase in NPSH [R] [Fig. K.29.5, Chart 3].

As D1 is reduced, the vane setting angle also changes, and accordingly the zero incidence flow. There is no simple way to factor this in, which is one reason why the results of the process detailed here can only be approximate. But it will help suggest the likely scope of changes possible. It is not recommended to reduce the eye to less than 85 to 80% of the original, without confirmatory testing. That's because there are too main assumptions involved.

## Shape of the pump performance curve now becomes unsuitable for mutual co-operation with other pumps in the system

Any of the foregoing modifications might be successful in increasing or decreasing the performance of a specific pump. But they may also have the unexpected consequence of making the pump performance curve less suitable for co-operative operation.

On the other hand, the curve shape, even for a single pump, might be already be unsuitable from a control standpoint *[Generally pumps having curves in which the pressure head changes continuously and significantly with flow are easier to control with precision.]* and so again some further modifications might be beneficial.

But if the pump is expected to operate simultaneously in co-operation with others, then there is one more matter to consider. Chapter 5 describes how the overall performance curve can be developed when several pumps are operated together.

- If the pumps are operating in parallel, the individual flows at common pressure head values are added, in order to derive the total flow of the combination.
- If the pumps are operated in series, then the individual pressure heads at common flow values are added in order to derive the total head of the combination.

If or when pumps having similar performance characteristics are operated in combination, then the pumping load will be more or less shared between them. Even so, at very low flows, the individual curves are likely to differ slightly [due to differences in manufacture, or wear rates]. And because the curve slope is likely to also be flat here, then substantial load sharing differences might arise.

However when the performance characteristics distinctly and definitely differ, load sharing difficulties are much more likely to arise. In Chapter 5, I touched upon issues relating to multiple pump operation when there are noticeable differences in the curve shape of each pump. Differences might occur when, for instance, pumps of different manufacturers are asked to operate together in a plant expansion. Or if, in an emergency, pumps with rather different ratings are pressed into combined service. Clearly, there will be some incentive to modify one or more pump curve shapes to ensure better load sharing in the combined performance.

The most common issue is that the low flow curve shape of at least one of the pump combination is too flat. In other words it does not initially fall away with enough slope from the zero flow point (Fig. K.30).

The self-evident solution is to locally increase the head at or close to zero flow. However, that is usually one of the most difficult things to achieve from an existing impeller. At zero flow, the pump pressure head is chiefly dependent on the impeller outer diameter, with other aspects of the pump geometry playing a relatively minor role. Increasing the zero flow head without increasing impeller diameter always involves complex modifications with limited scope for improvement. Increasing the impeller diameter will of course simply increase the pressure head all along the curve.

So, though little scope exists to locally increase/pressure head close to zero flow alone, some minor possibilities can be considered.

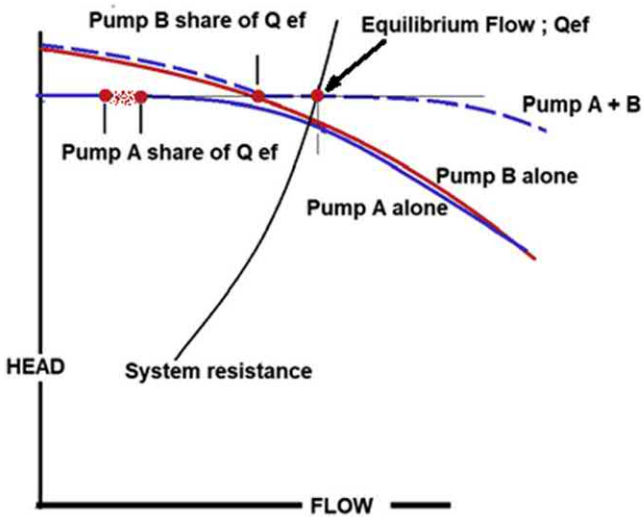



**Fig. K.30** Illustrates how poor low flow load sharing is likely if one of the parallel pump pair has a very flat characteristic curve. Not only will this pump fail to share its load, but its contribution will also be erratic due to the shallow intersecting angle with the Equilibrium head line.

## Increasing head at/near to zero flow

### *Inlet backflow effects*

Appendix D described the inlet backflow effect. Among its many negative consequences is a reduction in apparent pump pressure head. This effect is particularly apparent in some older designs that exhibit a very pronounced 'flattening' of the curve below, say 40% of bep flow (Fig. K.31).

This effect increases as flow is reduced below the backflow onset point. Anything that limits inlet backflow will therefore help increase pump head at that point. That is because the backflow mechanism wastefully absorbs some input power which would otherwise generate useful pressure head.

Pumps that have a significant shaft obstruction [between-bearings] are already penalised because the impeller eye diameter will have been enlarged by the designer, in order to compensate. This results in an otherwise higher inlet tip speed and, from Appendix D will cause a higher backflow onset value. So such pumps will be more difficult to correct than, say, overhung impeller pumps.

Measures known to inhibit backflow onset to a lower through-flow value include;

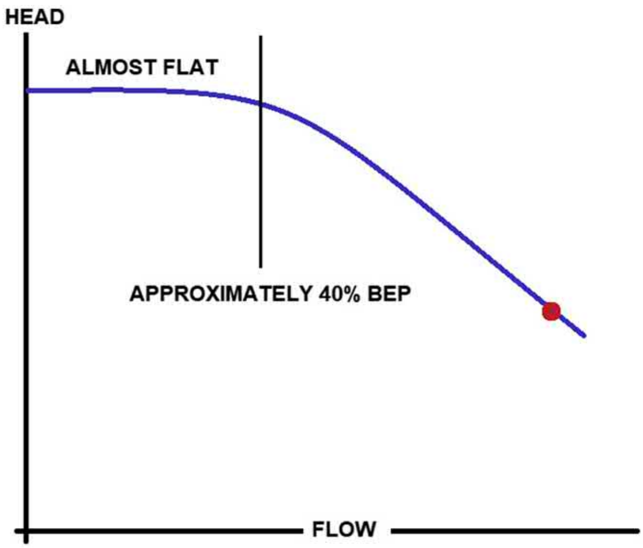



**Fig. K.31** Many older designs exhibited a very flat curve at flow less than 30–40%. This was inherent in their hydraulic geometry. Pumps with such curve shapes will prove troublesome when operated together at low flows. Although manufacturers would discourage such practice and instead encourage operation of just one pump, owners sometimes prefer the security of an already running standby – in case of emergency.

### *Backflow catchers*

These special devices attempt to directly tackle the impeller backflow, and then make it as benign as possible. This is accomplished by de-swirling the liquid. They are described in Appendix D.

### *Reduction in flowpath entry area, normal to flow*

- Reducing the impeller inlet annulus – adding choke rings see earlier [Fig. K.29].
- Blocking alternate vane passages.

Note these are severe measures that may have flow-related effects. While successful at low flow, it will have corresponding negative effects at higher flow. So it should not be considered without review, if the pump is still expected to cycle through a wide range of flows.

### *Installation of helical cavitating inducer*

These devices are intended to improve the NPSH margins by performing as an 'in-series' booster. They have the ability to add sufficient suction pressure to the main stage inlet to suppress cavitation, even though they are cavitating themselves. A secondary benefit is their series-pump head addition effect is flow-related and improves curve slope (Fig. K.24).

This is effect is enhanced by the constriction they offer to any backflow emanating from the main stage impeller. But the possible restriction on run-out flow still needs to be borne in mind.

### *Flow straighteners in inlet duct*

These most often take the form of axial ribs aligned with the normal flow direction. Normally these do little if anything, except correct residual levels of swirl in the inflow. As pump flow reduces to and below the backflow onset point, the suddenly outgoing backflow swirl is dissipated against these ribs [Fig. J.95] (Fig. K.32).

Without any such restriction, backflow would develop to a very significant volume and absorb work that would otherwise appear as output head *[Unless otherwise dissipated, this backflow may extend 10 pipe diameters or*

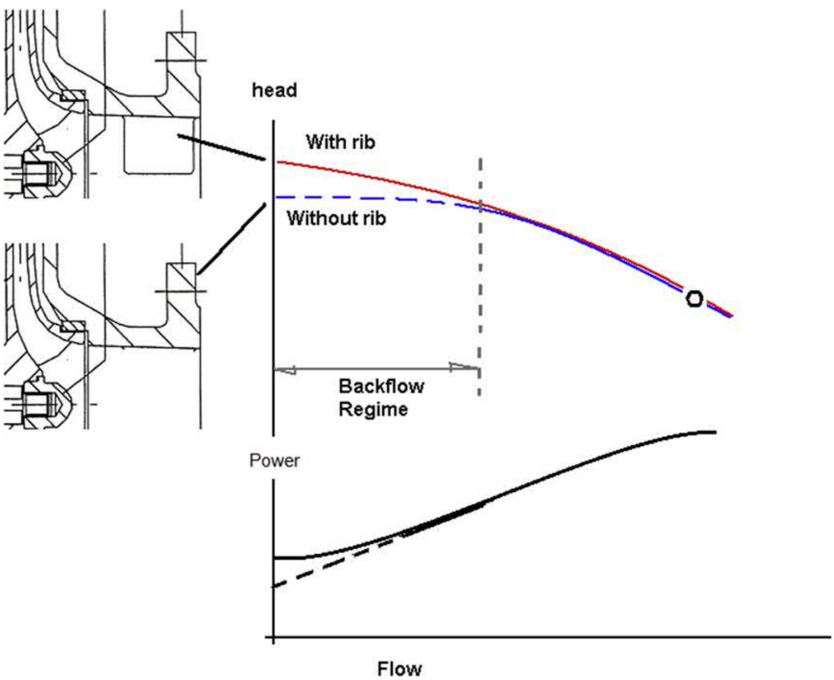



**Fig. K.32** Presence of axial straightening rib in inlet duct will inhibit inlet backflow and hence increase the low flow head. However, this will absorb a little more power and, if suspended solids are present, the rib will be exposed to erosion Fig. J.95.

*more back upstream. – see* Appendix D*]*. While successful, this approach can have a downside. Interaction between the discrete backflow 'cells' emerging from each rotating vane passage, and the fixed straightener vanes creates a periodic torque 'ratchet' reaction that materialises in a low frequency vibration signature *[Remember that in order to escape from the impeller the liquid cells must be travelling at a velocity somewhat greater than the eye inlet tip speed!]*. This effect diminishes as the gap between impeller vanes and straightener vanes increase. Unfortunately, the head improving benefit of the ribs also decreases with increased gap as well!

### *By modifying/constricting the casing inlet duct*

Normally, good piping practice dictates that the inlet duct design obliges the flow to constantly accelerate towards the inlet branch connection and then on to the impeller inlet annulus. This implies that the flow cross sectional area constantly reduces. The underlying objective here is that accelerating flow helps to discourage the formation of localised unstable stall cells. However, if pumps are continuously operating close to the edge of the backflow regime, it is known that some degree of flow constriction can be helpful in delaying the onset of inlet backflow. So for example, replacing accelerating taper nozzles with constant diameter pipe may help to supress backflow and force a commensurate increase in head. But this is only a short term gain, because with further flow reduction, backflow will eventually set in. So while this approach may defer or dilute the backflow effects such as surging and vibration, it helps little when it comes to load sharing of co-operating pumps.

### *Imposing swirl onto the inlet flow*

The above paragraph described how attempting to manage the incoming flow has little effect at flow greater than the backflow onset point. Paradoxically, managing the flow to instead possess significant swirl can steepen the entire curve shape *[Because the subtractive term in Euler equation now becomes non-zero.]*. Of course this will consequently reduce pump head as well, but this may often be overcome by simultaneously increasing the impeller diameter. The nett effect of these two measures can be a performance curve that tends to fall more continuously from the zero flow point. The pump designer always has this easy solution available at the outset. But he will usually be reluctant to adopt it because the resulting impeller diameter will be larger than an equivalent having nominally zero inlet swirl. From second section, it is known that pump weight can be proportional to almost the cube of impeller diameter. This impacts significantly on pump cost-more particularly where expensive alloys are concerned. So the designer will leant wards strategies that do not rely on inlet swirl. But that does not rule it out as a valid in-field modification.

There are a range of strategies available to help achieve inlet swirl. Pumps with axial inlet ducts and single stages can have an array of guide vanes installed that deliberately impart swirl either with or against the direction of rotation. . Before the days of variable speed drives, this approach was chiefly employed to provided pump output variation. Elementary fluid mechanics shows that these are most effect in low head [High Specific Speed] pumps.

Multistage pumps, of radial flow configuration, also commonly deploy straightener vanes between stages. In the same way, they may be arranged to provide a little swirl with or against impeller rotation. This is a widespread design practice, principally aimed at curve shape manipulation.

Double Entry pumps with radial entry ducts are commonly arranged to either deliberately impart swirl or not to [see Chapter 3]. In this case the motive is more aligned to NPSH [R] reduction and less to curve shape manipulation.

### *Increased internal leakage*

Palgrave's 2nd Rule says that that when pump performance decays, the slope of the characteristic always becomes steeper. Figs 11.20 and 11.21 attempt to explain how pump performance is shifted as a result of internal leakage. So in problem cases where pump load-sharing has become acute and a very short term solution is sought, then this might be an option to consider.

It will involve deliberately increasing some or all of the running clearances between rotor and stator-in effect pre-wearing out the pump. Naturally the pump pressure head will appear to reduce, but where maximum diameter impellers are available as spares, this can be overcome by trimming them to a larger diameter than previously (K.34).

Of course, efficiency will be reduced compared to a specific low flow alternative design, but in an emergency it might be worth considering this approach. Introducing controlled recirculation line connecting pump discharge to pump suction would have a similar effect [Fig. 10.26]. In this case the recirculation can be controlled somewhat compared to the fixed value associated with wear ring clearance increase. The connection point back into the suction line needs careful consideration, because the injected

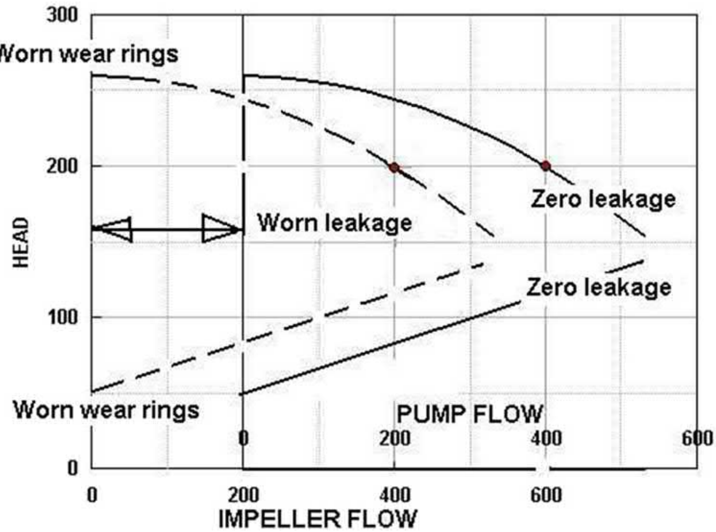



**Fig. K.33** A crude method of increasing curve slope is to deliberately enhance the pump internal leakage, as illustrated here. This is very inefficient and would be a last resort.

liquid will be warmer than the suction flow. This might induce NPSH problems – particularly if injection is quite close to the impeller inlet. Injection several pipe diameters upstream would be recommended in this crude but effective short term solution.

### *Conic impeller trim/cut*

A big drawback of this [last] solution is that the pump overall efficiency is substantially reduced. The pump will consume more power that previously. However, if spare impellers are readily available one should consider an alternative which is, simply trimming them to a conic angle, rather than cylindrically. It is known that this can also increase the pump head at flows below the backflow point. It does so by reducing the inequalities in the low-flow impeller passage picture [?] and this helps diminishes both the backflow tendency and intensity. Little if any efficiency penalty results (Fig. K.34).

Cone angles up to 50 degrees have been employed in the past. But 30 degrees would be a more usual limit. For the most part, the impeller will still perform in accordance with the Affinity Laws based on an impeller diameter equivalent to the RMS diameter. However, at flows less than the backflow onset point, the behaviour will now result in higher head than predicted by the Affinity laws (Fig. K.35).

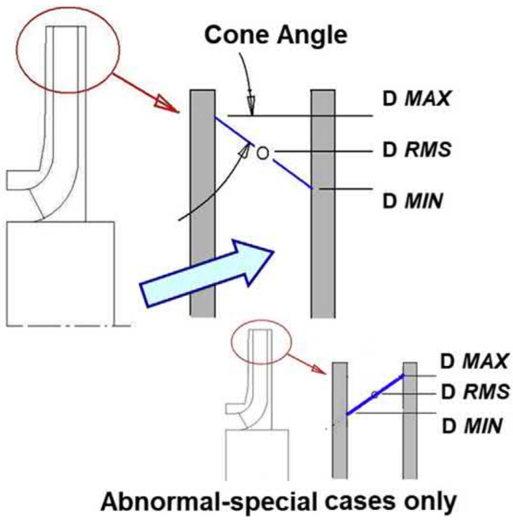



**Fig. K.34** Trimming the impeller outlet diameter to a conical profile, as opposed to cylindrical, is known to bestow more slope to the reduced output curve.

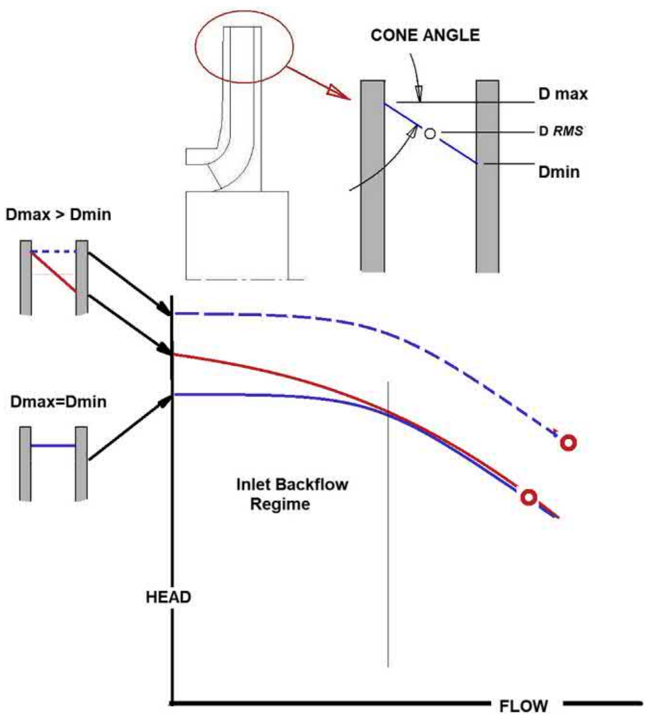



**Fig. K.35** Conic diameter reduction is known to increase head and curve slope inside the backflow regime. Outside of his regime, the normal Affinity Turn-down laws apply. Both examples have the same RMS impeller diameter.

This may result in a more favourable performance curve shape. The amount by which the low flow section of the curve exceeds the Affinity Law prediction depends on the cone angle. Obviously a zero cone angle conforms to a normal cylindrical impeller turndown and close compliance with the Affinity Laws would be expected. With a large cone angle, say 30–40 degrees, pressure heads in excess of the Affinity Law prediction would result along the low flow section of the curve.

There does not appear to be any simple method of predicting the intensity of this departure with respect to cone angle. It seems that by applying a conic turndown, the length of the streamlines on the hub and shroud are made more equal and this helps which lead to a more uniform work distribution inside the impeller. Consequently the flow picture leaving the impeller is more in balance and so backflow tendency less likely. As a rough rule, the conic angle should not be so great as to result in the true/developed hub streamline length being less than that at the shroud. Experience says that the conic angle solution seems more effective when the impeller is relatively wide, compared to its diameter. Generally such impellers will inherently have larger natural differences in streamline lengths.

This chart gives a VERY APPROXIMATE guide as to what change is and is not possible from a replacement impeller. It is intended for simply for troubleshooting purposes and to eliminate or include this approach in a list of solutions. It should not be used on its own and/or without expert guidance. It suggests – for example that if the pump with a normal cylindrical impeller trim currently has a Head Rise To Shut-Off [slope] of 1.15, then it is most unlikely that this could be increased to 1.4- even if the replacement impeller is trimmed to a cone angle limit of 30 degrees. On the other hand, increasing HRTSO to 1.2 might be possible. But I stress again, this is just a very coarse guide – because many other factors contribute to this behaviour (Fig. K.36).

A note of caution chiefly related to double suction impellers: Figs J114 and J66 illustrate possible outcome from the interaction between impeller and collector flow fields. Conic trims result in the blade outlets taking on a 'chevron' shape and this will help soften the impeller/collector interaction. Figs J67–J69 illustrates the effect that high levels of pressure pulsations may have. Although reduced by a conic trim, the pulsation levels still exist and can still initiate fatigue crack propagation. Consequently, the 'Vee trim' angle formed at the D [min] diameter **must** be well rounded. It should not

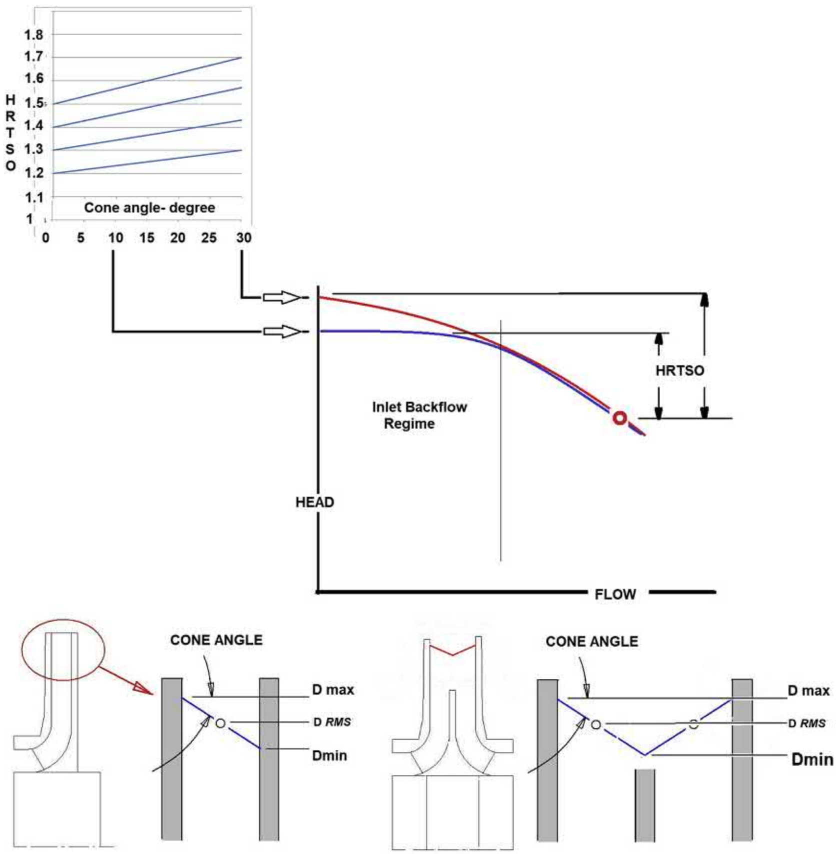



**Fig. K.36** This figure gives a coarse guide as to what is and is not possible from using conic impeller trims. As the main text explains, this is just to show if this is a solution worth pursuing when wanting to change the curve slope of an existing impeller.

be sharp and well defined; else stress concentrations may be formed. Fig. K.37 shows the possible consequences.

### *Alternate vane trim*

If the impeller is furnished with an equal number of vanes, then another method of curve shape adjustment exists. It is known that if every other vane is equally reduced in length [from outlet], then the slope of the now reduced pump curve increase (Fig. K.38).

It seems that there are two effects:

- Firstly, the zero flow head is reduced broadly in accordance with the Affinity Laws applied to an RMS diameter reduction.

**Fig. K.37** The notch formed by a conic trim on a double suction impeller can be the source of fatigue cracks. The pressure pulsations resulting from impeller −collector flow interactions will propagate these cracks.

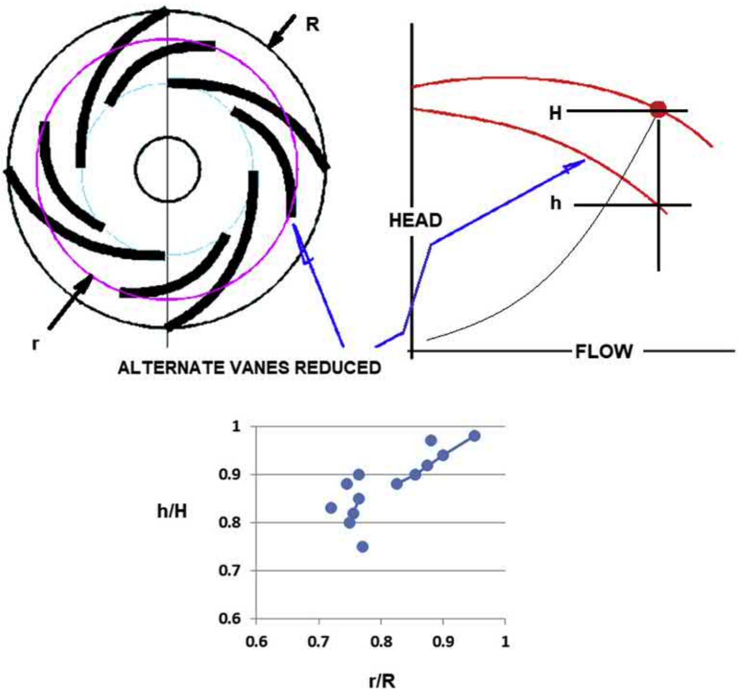



**Fig. K.38** Trimming alternate vanes is one way of increasing the slope of a pump curve and was sometimes used on older pumps in order to produce a stable curve. As the main text explains the nett effect is similar to that of an internal 'flow squared' loss. Consequently, the pump efficiency will also reduce.

- Secondly there are flow-mixing losses between the flow streams issuing from adjacent passages. This component behaves in a similar manner to a discharge orifice. In other words it introduces a flow-related pressure head loss, over and above that forecast by the Affinity Laws. The loss effect increase with flow. Because this is in effect an additional hydraulic loss, the pump overall efficiency will clearly suffer.

This approach is not common these days and was more often applied to older designs having relatively large vane setting angles at outlet. The included graph simply indicates trends-derived from historical data. It only indicates what might be possible if using this approach to try and produce a stable curve of constantly falling slope as a function of flow.

## HOW TO ESTIMATE IMPELLER AND CASING LINES WHEN NO HYDRAULIC DATA AVAILABLE

Often the hydraulic data needed to conduct a detailed calculation of the theoretical impeller and casing line is not readily available. This might suggest that the earlier discussion on modifications to these items is of little use. However, these two lines can be estimated with sufficient accuracy to help indicate the scope of modifications needed and to suggest actual dimensions. From troubleshooting perspectives, this will help evaluate whether modifications to an existing machine would likely be sufficient, or whether an alternative solution needs be sought.

It should be mentioned that the simplified approach described here is applicable to pumps in which the flow enters the impeller with little or no spin or swirl. This would include most single entry horizontal shaft pumps and more modern double entry designs. Many, but certainly not all multistage pumps would qualify, so care is needed her [Some designs deliberately induce pre-swirl from their stage return guide vanes] Vertical shaft pumps operating suspended on open sumps might not be included.

**[1]** The first requirement is a performance chart for the pump in question. This needs to represent the performance at the normal; maximum diameter [not oversize-[sometimes possible]. Ideally a works test performance curve will be available. This needs correction to the maximum diameter performance by using the Affinity Laws-[Described later].

This predicted curve becomes the basis for the modification assessment. In some cases, the performance at maximum diameter might already be known, in which case, this step can be eliminated (K.41).

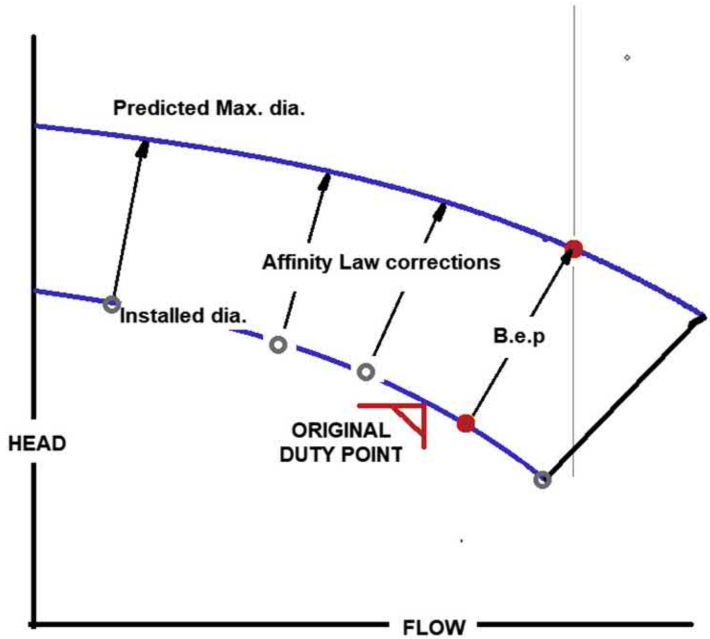



**Fig. K.39** Generally the installed impeller diameter will have been trimmed by the manufacturer to fine tune the pump performance to the best 'fit' to the duty. In order to assess the likely scale of modification it is first necessary to gain some idea of the performance at Maximum impeller diameter. Sometimes, this is already known, but where it is not, the Affinity laws can be used to make a sufficiently accurate forecast.

**[2]** The next step is to estimate the hydraulic efficiency of the pump. We have the actual performance, but need the hydraulic efficiency in order to estimate the theoretical performance (Fig. K.40).

Designers go to a great deal of trouble to try and accurately predict the hydraulic efficiency. For purposes of troubleshooting, and infield rectification, it is inappropriate to cover in the same level of detail. In view of the approximate nature of the calculation mentioned here, it is also appropriate to include an approximate estimate of hydraulic efficiency. For the purpose of this calculation, the hydraulic efficiency can be taken as the square root of the overall pump efficiency expressed as a decimal – not a percentage [and **not** the wire to water value]. This is just what it says, an approximation. Greater accuracy should not be inferred.

At the best efficiency point of the machine, the pressure head should be divided by the hydraulic efficiency in order to help determine the theoretical head at this flow. This gives us one significant point on the Euler line.

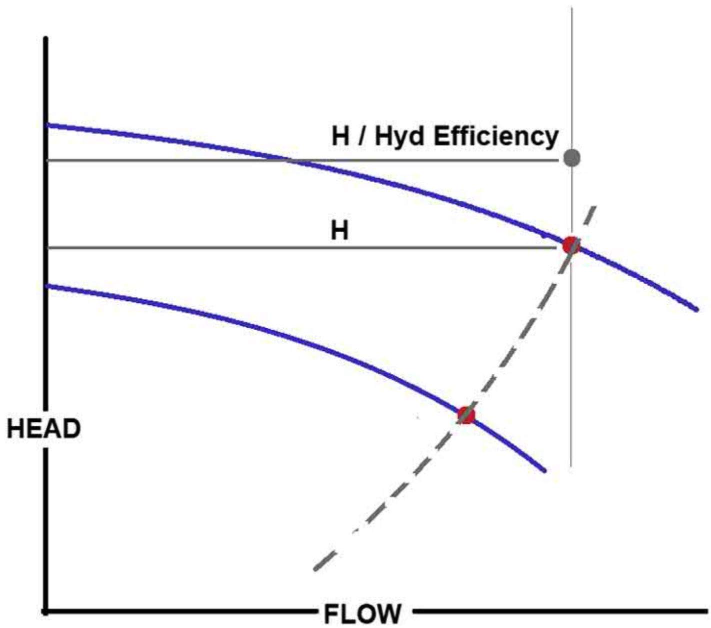



**Fig. K.40** The theoretical head can be calculated if the actual head at best efficiency is divided by the hydraulic efficiency. Hydraulic Efficiency cannot be simply calculated but, a way out of this dilemma is to take the square root of the overall efficiency - expressed as a decimal. This is sufficiently accurate for most troubleshooting purposes. But it is just an approximation. This calculation establishes one point on the theoretical head curve.

**[3]** It would be useful if a second point on this line could be estimated. The most convenient point would be at zero flow. Calculating the theoretical zero flow head requires information not readily available outside of the deign community. The only readily available data is the previously mentioned maximum impeller diameter. However, this information can still be used to form a useful crude estimate of the theoretical head at zero flow.

$$\text{H@0} = \text{U2} \times (\text{K} \times \text{U2})/9.81 - \text{metres}$$

where U2 = Impeller outlet tip speed in m/s
and K = [approximately];
0.75 for Impellers with 2,3 or 4 blades
0.77 for Impellers with 5,6 or 7 blades
0.8 for impellers with 8,9 or 10 blades

I again stress that this is for impellers without pre-swirl in the flow immediately approaching the impeller.

Real head will; of course depart from these values, due to hydraulic losses as well as inlet backflow effects. Furthermore blade outlet setting angle is assumed to lie within the common range of 18–23 degrees. Fortunately, most impellers will lie within this range (Fig. K.41).

Connecting these two points forms an estimate of the theoretical impeller input head line. The line should be extended down towards higher flow/lower pressure head. Then, as a check, a line parallel to this should tangentially touch the original corrected performance curve in the region of pump best efficiency flow. If there is a significant difference between the bep flow and the tangential contact point flow, then any further decisions based on this process will be questionable. Given the approximate nature of the calculations, differences greater than 20% should be questioned.

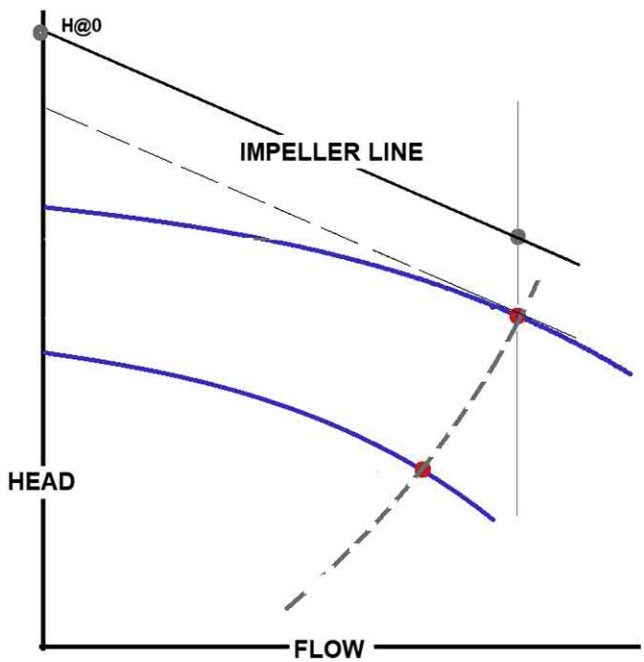



**Fig. K.41** The value of theoretical head at zero flow can be estimated from the equation in the main text. Connecting the two points establishes the full line. As a reality check, a second line drawn parallel should touch the maximum diameter curve within plus/minus 20 % of bep – given all the approximations used.

**[4]** The equivalent casing line can be derived by connecting the first derived point with the graph origin [and perhaps extending it a little] (Fig. K.42).

As approximate as they are, these Impeller and Casing lines can help develop a picture of the pumps' hydraulic personality.

Experience will show that low flow/high pressure head pumps [Low Specific Speed] have an impeller line that is relatively 'flat.' The zero flow pressure head might be no more than 115% greater than that at bep. The corresponding casing line can be termed as 'steep'. In such designs, the bep flow, as indicated by the point of intersection, is not as sensitive to deviations in impeller line slope as in casing line slope. The performance of this type of hydraulic design is known as 'Casing controlled'.

This suggests that actual bep flow is less influenced by discrepancies in impeller flowpath design or manufacture than that of the casing and particularly its throat [s] (Fig. K.43).

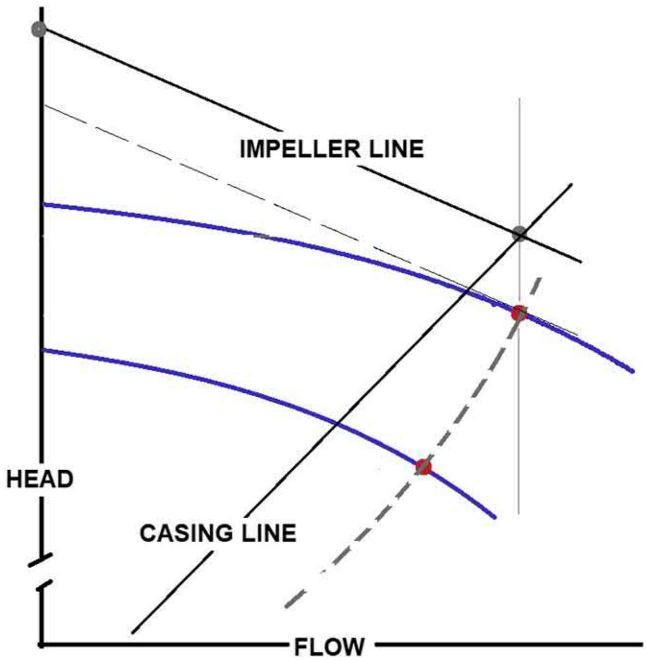



**Fig. K.42** Connecting the bep point to the graph origin establishes the casing/collector line. Knowing these two lines allows a lot of judgements to be made regarding performance modifications.

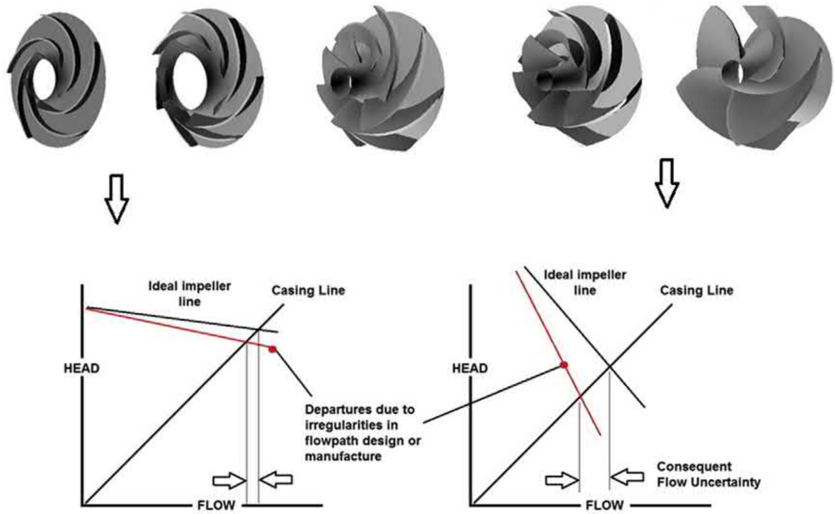



**Fig. K.43** The characteristics of these two lines changes with impeller geometry. Low specific Speed pumps, typified on the left, have their best efficiency point largely dictated by the collector area. High specific speed pumps, on the other hand, or more controlled by the impeller geometry – see main text [Sulzer Pumps].

On the other hand, in pumps of high flow and low pressure head [High Specific Speed] the impeller line will be relatively 'Steep' —the zero flow head being maybe more than two times higher than at design flow. Meanwhile, the corresponding casing line will be relatively 'flat'. In this case, the bep flow, as indicated by the point of intersection, is not as sensitive to deviations in casing line slope as in impeller line slope. The performance of this type of hydraulic design is known as 'Impeller controlled'.

This suggests that in such pumps, the actual bep flow is less influenced by discrepancies in casing flowpath design or manufacture than that of the impeller and particularly its vane-to-vane passages. Furthermore, in such high specific speed designs, the generated pressure head is achieved not only by centrifugal action, but also by the 'lift' head of the vane effect. As a result, decisions based the impeller/casing line intersection alone become questionable. This will apply to any pump having a Specific Speed in excess of much in excess of 4000 in Usgpm-rpm units.

Knowing whether a pump is Casing or Impeller controlled shows where to best direct effort when performance modifications are being considered.

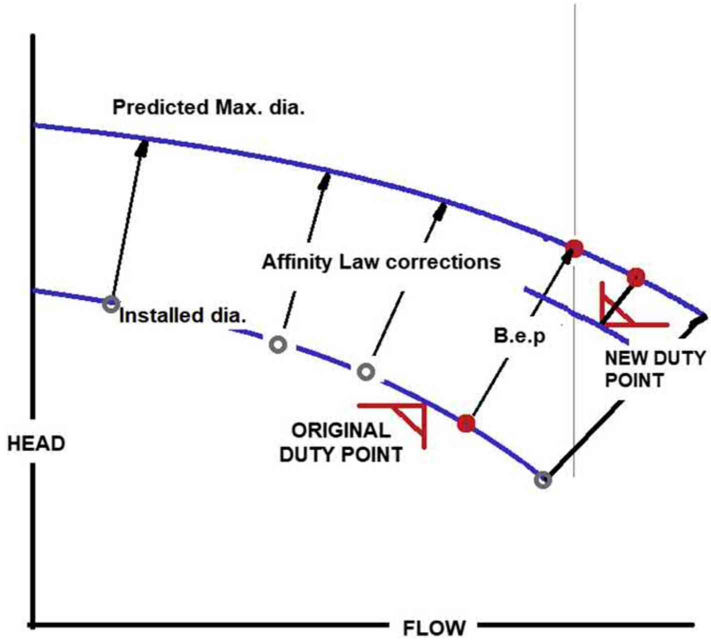



**Fig. K.44** This example is contrived to illustrate how hydraulic modifications might bring about beneficial performance changes.

**[5]** Now imagine that this pump is required to satisfy a new set of duty conditions, as shown (Fig. K.44).

Of course this could be achieved by fitting a new impeller of larger diameter than originally installed. But imagine that it was required that at the new duty point, the pump was operating at its best efficiency. Clearly that would not be the case with a new larger trimmed impeller. And in fact the efficiency differences in this case between a trimmed impeller and a modified pump might be acceptable. But for the purposes of illustration I will presume they are not.

The first step is to correct the new duty point to its equivalent at maximum impeller. Again the Affinity Laws are employed (Fig. K.45).

The Theoretical Impeller and Casing lines are already known for the unmodified pump. On the basis that the hydraulic efficiency remains unchanged we can deduce a new intersection point. Unchanged hydraulic efficiency is justified at this level of approximation (Fig. K.46).

This shows that the collector throat area needs to be increased in the ratio of A/B. The Impeller outlet area would need a very small reduction in

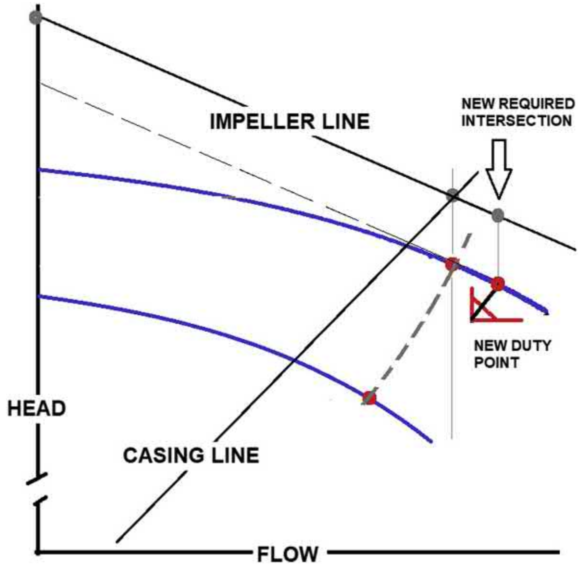



**Fig. K.45** Using the Affinity Laws, the new duty point can be extended to meet the [predicted] maximum diameter curve. To place the bep at these new conditions, the Impeller and Casing lines must now intersect at this new flow. This will, in turn, become the bep when the impeller diameter is trimmed to the new duty point.

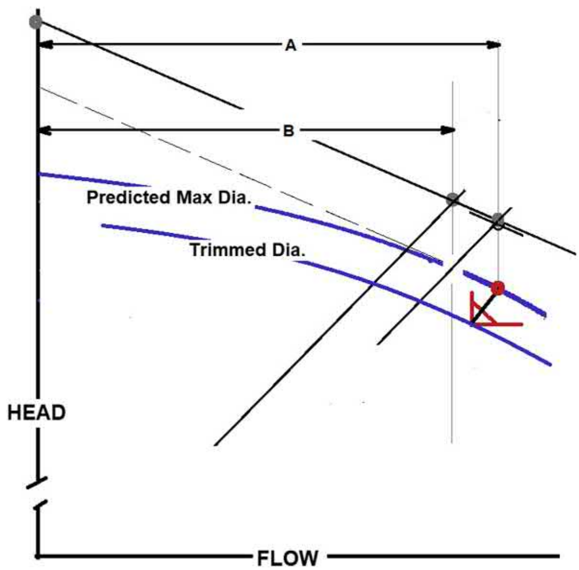



**Fig. K.46** The total collector throat area must be increased in the ratio of A/B. Required impeller modifications seem trivial in this case.

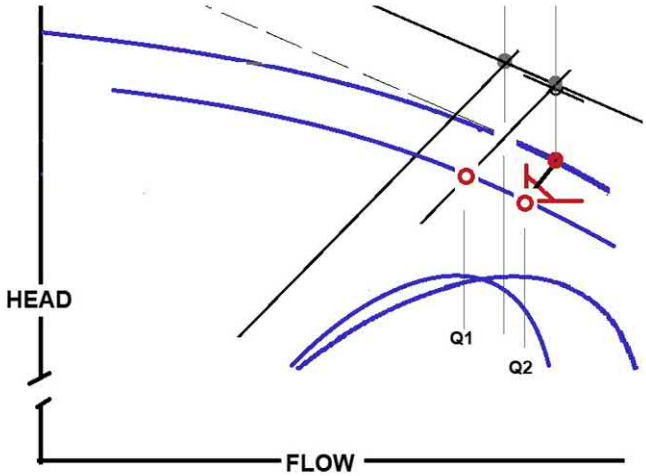



**Fig. K.47** If a new larger impeller is simply trimmed in diameter, the bep will be located at Q1. This does not coincide with the new duty. But if the casing throat is simultaneously enlarged, then bep will move to Q2 and will now coincide.

order to fully comply. Realistically, this looks to be too trivial to further consider. So, by

- Installing a new impeller trimmed to a larger diameter than before.
- ore and
- increasing the collector throat area,

The pump best efficiency point will now coincide with the new duty point [Q2]. If the impeller had been only trimmed, then it would be operating at a flow somewhat larger than its corresponding best efficiency [Q1] and consequently absorbing more power (Fig. K.47).

# References

[1] R. Palgrave, Some Critical Issues in Process Pump Design and Operation, Process Pumps; the State of the Art, Institute of Mechanical Engineers, London, June 25, 1991.
[2] A. Buseman, Das Forderhohenverhaltnis radialer Kreiselpumpen mit logarithmisch – spiraligen Schaufeln, Zeitschrift für Angewandte Mathematik und Mechanik 8 (1928) 372.
[3] J.F. Gulich, Centrifugal Pumps, Springer, 2010.
[4] M. Ryall, et al., TBA – Weir Cone Pump.
[5] U.M. Barske, Design of Open Impeller Centrifugal Pumps, Technical Note No. RPD 77, Royal Aircraft Establishment, Farnborough, January 1953.
[6] W.H. Fraser, Recirculation in Centrifugal Pumps, Materials of Construction of Fluid Machinery and Their Relationship to Design and Performance, ASME Book No H00208, 1981.
[7] J.L. Hallam, Centrifugal Pumps: Which Suction Specific Speeds Are Acceptable? Hydrocarbon Processing 61 (1982).
[8] H.H. Anderson, Hydraulic design of centrifugal pumps and water turbines, ASME 61WA 320 (1961).
[9] API 610, A Standard for Pumps in the Petroleum, Heavy Duty Chemical and Gas Services Industries.
[10] R. Palgrave, Using Noise Data to Analyse Hydraulic Problems on Site, ASME Pumping Machinery Symposium, San Diego, July 9-12, 1989, pp. 117–123.
[11] M.J. Prosser, The Hydraulic Design of Pump Sups and Intakes, BHRA Fluid Engineering, 1997.
[12] Hydraulic Institute/Europump, Life Cycle Costing.
[13] Variable Speed Driven Pumps, BPMA – Gambica-Action Energy, 2002.
[14] Hydraulic Institute/ANSI Standards for Centrifugal Pumps.
[15] C.C. Heald, D.G. Penry, Design and operation of pumps for hot standby service, in: 5$^{th}$ International Pumping Symposium, Texas A&M, 1988.
[16] A.J. Stepanoff, Centrifugal & Axial Flow Pumps, second ed., John Wiley & Sons Inc, 1957.
[17] D.F. Denny, C.J. Young, Prevention of Vortices and Swirl at Intakes, BHRA Fluid Engineering, sp583.
[18] R. Palgrave, Process Pump Design for Hot Abrasive Service, in: Institute of Mechanical Engineers, London, December 1989.
[19] R. Palgrave, P. Cooper, Visual studies of cavitation in pumping machinery, in: 3rd International Pumping Symposium, Texas A&M, May 1986.
[20] R. Palgrave, Diagnosing pump problems from their noise signature, in: 11th BPMA Technical Conference, Cambridge University, April 18-20, 1989, pp. 9–28.
[21] K.R. Blockwell, D. Kleinbub, Performance Characteristic of the Oil Ring Lubricator – An Experimental Study.
[22] R. Palgrave, Operation of centrifugal pumps at partial capacity, in: 9th BPMA Technical Conference, Warwick University, England, 1985, pp. 57–70.
[23] R. Palgrave, Avoid low flow pump problems, in: 12 BPMA Technical Conference, Regents College, London, April 17-19, 1991, pp. 124–139.

[24] R. Palgrave, Hydraulic power recovery using reverse running pumps, in: 10th BPMA Technical Conference, Churchill College, Cambridge, March 24–25, 1987.
[25] C.A. Gongwer, Theory of cavitation flow in centrifugal pump impellers, TRans ASME 63 29-40.
[26] I.C. Massey, R. Palgrave, The suction instability problem in rotodynamic pumps, in: NEL Conf, Pumps and Turbines, vol. 1, September 1976. Paper 4-1.
[27] R. Palgrave, Mechanics of wear and its effect on the hydraulic performance of conventional centrifugal pump, Institute of Mechanical Engineers, 9th European Fluid Machinery Congress, The Hague, April 23–26, 2006.
[28] Prof. W. Kamm, Research Institute of Automotive Engineering and Vehicle Engines Located Near Stuttgart.
[29] R.C. Worster, The flow in volutes and its effect on centrifugal pump performance, Proceedings of the Institute of Mechanical Engineers 177 (1) (1963).
[30] E. Grist, Cavitation and the Centrifugal Pump – A Guide for the Pump Users, Taylor and Francis, 1998.
[31] D.J. Vlaming, A Method for Establishing the Net Positive Suction Head Required by Centrifugal Pumps, ASME 81-WA/FE-32, November 1981.
[32] R. Palgrave, Pump pressure pulsation reduced in oil pipeline stations, Institute of Mechanical Engineers, 11th European Fluid Machinery Congress, Edinburgh, Scotland, September 12–15, 2010.
[33] G.F. Wislicenus, Fluid Mechanics of Turbomachinery, vols. 1 &2, Dover Publications, New York, 1965.

# Index

'*Note*: Page numbers followed by "f" indicate figures, "t" indicates tables.'

## D



## E



## F

## L



## M



## N



## O



## P

## W



## Z

Second Edition

# Troubleshooting Centrifugal Pumps and their Systems

**Ron Palgrave**
Pump Hydraulic Specialist, Gateshead, UK

Those who operate centrifugal pumping machinery sometimes have to deal with the situation where a pumping system fails to perform. The easiest, but most inconvenient and costly solution is to replace the machine. The new edition of ***Troubleshooting Centrifugal Pumps and their Systems*** helps analyze the source of the problem which sometimes lies in the system itself and not the pump. It also shows that the pump characteristics can be reconfigured to suit changing conditions. The book provides guidance to those faced with deciding when to withdraw a pump from service for repair and how it should be subsequently treated as well as assist those who feel ill-equipped to try to analyze unsatisfactory pump system behavior.

This book is for those responsible for the selection or daily operation of centrifugal pumping machinery and who might be faced with a situation where a pumping system fails to perform.

## Key Features

- Provides more information on the pump hydraulic design aspects in separate appendices
- Describes the basic mechanisms of abrasive wear in centrifugal pumps including the different wear patterns and their causes
- Discusses what performance improvements and approved repair techniques can be applied to extend a pump's life cycle

## About the Author

Ron Palgrave is a retired pump designer and specialist in the hydraulic design and performance of centrifugal pumps. He has nearly 60 years' experience with 5 manufacturers. He has written several technical papers, magazine articles, and is the author of the 1st edition of *Troubleshooting Centrifugal Pumps and their Systems*. He has been granted several patents relating to pump technology. Mr. Palgrave currently practices part-time and is one of the most experienced hydraulic engineers in the industry.

ISBN 978-0-08-102503-1

9 780081 025031

ELSEVIER
Butterworth-Heinemann
An imprint of Elsevier
elsevier.com/books-and-journals