Peter Albers

Motion Control in Offshore and Dredging

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About the author

Peter Albers was born on 23rd July 1952. After completing his secondary education in Amsterdam he went on to study mechanical engineering at the Technical University in Delft, The Netherlands. The author completed his masters studies in the Department for the study of Measurement and Control Technology of the faculty of Mechanical Engineering in 1975. During these studies the emphasis already fell on Fluid Power. The first 'trigger' was made by a lector in gears. He attended on very interesting applications with gear pumps and fluid power that could not be realized with other drive systems. In 1985, after working for a number manufacturers and suppliers of both components and complete hydraulic systems, Peter set up 'Ingenieursbureau Albers BV', an independent consultancy service for the fluid power industry.

During its 25 year existence, the company has brought many solutions and completed many designs for machinery manufacturers, larger engineering offices, offshore operators and dredging companies. All these projects have contributed to a deeper understanding of the industry and each project was able to benefit from experience gained in earlier ones. One important philosophy of the company has always been to transfer as much knowledge and experience as possible to the client. The experience and knowledge of 'Ingenieursbureau Albers BV' has been further disseminated through training courses designed by the company.

In 1996 the author was one of the co-founders of the 'Vereniging Platform Hydrauliek' (Fluid Power Engineering Society). This is a group of Dutch experts in the field of fluid power control technology. Since then he has also been its chair for a number of years. One of the aims of the organisation is to spread knowledge. Its bi-annual symposium is one of the ways in which this is achieved. From 2004 till 2007 Peter was visiting lector in fluid power at the Hanze University in Groningen, Netherlands. Since 2008 the author has also been a staff member of the department for Offshore Engineering at the Technical University in Delft, Netherlands. That too has offered a unique opportunity to transfer knowledge to young engineers.

Preface

'Motion Control' is often used as a description in various engineering disciplines. In all these disciplines the reference is to a technological solution that is able to control motion, eg the movement of at least one part relative to another. "Motion Control in the Offshore and Dredging Industries" describes how drives of mechanisms that can be very large are designed and realised.

A distinction is made between rotating and linear drives. In the case of rotating drives, the choice for an electrical drive is becoming more and more prevalent. Linear drives remain important, because of the large forces and the highly dynamic behaviour, in the domain of fluid power drive technology. Both these important technologies are extensively discussed in this book with design rules and the many installation requirements that are useful for practical application.

The book is first and foremost meant for designers of new drive mechanisms. It does however also give a practical explanation of the way in which the different mechanisms described here work.

The author thanks Chris de Haes MSc, who contributed to the correct English translation of this book. Many thanks also go to the technical readers of the draft: Peter Blok, Gerard Elffers and Peter de Vin. With their detailed corrections and suggestions it was possible to improve the quality of this book. Last but not least the author likes to thank Ronald Top who assisted in all technical drawings and Jacques van Schie who designed the beautiful full colour lay-out of the book.

Content


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Chapter 1

Hydraulic energy converters

Motion Control in Offshore and Dredging

Chapter 1

Hydraulic energy converters

To drive systems, mechanical energy must be delivered to the powered system or machinery. This mechanical energy comes from the primary energy source. For drive technology, the primary power sources are usually electrical energy or combustion engines. Other sources of energy are also possible. Energy can for example be obtained from wind, tidal power, waves or fuel cells. In each case the primary power source needs to be transformed from primary drive mechanism into hydraulic energy and subsequently from hydraulic energy into the desired mechanical energy.

In hydraulic drive technology, energy converters can be divided into two groups. In this chapter detailed technical information is being given on the functions of:

- **Conversion of mechanical energy into hydraulic energy, achieved with the use of pumps**
- **Conversion of hydraulic energy into mechanical energy, achieved with so-called actuators.**

For actuators a further distinction is made between longitudinal and rotating energy converters.

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1.1 List of symbols

A	=	area	m^2
A_b	=	bottom area	m^2
A_a	=	annular area	m^2
D	=	outer diameter	m
d	=	rod diameter	m
F	=	force	N
F_{max}	=	allowable axial force	N
I	=	area moment of inertia	m^4
i	=	radius of gyration	m
K	=	correction factor	
L	=	length	m
L_k	=	buckling length	m
m	=	mass	kg
n	=	speed, rotation	rpm
p	=	pressure	N/m^2
ﬁp	=	pressure difference	N/m^2
P_H	=	hydraulic Power	Nm/s
P_{in}	=	input Power	Nm/s
P_M	=	mechanical Power	Nm/s
P_{out}	=	output Power	Nm/s
Q	=	fluid flow	m^3/s
Rm	=	yield strength	N/mm^2
S	=	stroke	m
T	=	torque	Nm
t	=	time	s
V	=	volume	m^3
V_f	=	safety factor	
V_s	=	stroke volume	m^3/rad
v	=	linear speed	m/s
ϕ	=	area ratio	
λ	=	lambda (slenderness ratio)	
$\eta_{m\text{-}MH}$	=	mechanical hydraulic efficiency of motor	
$\eta_{m\text{-}V}$	=	volumetric efficiency of motor	
$\eta_{m\text{-}tot}$	=	total efficiency of motor	
$\eta_{p\text{-}MH}$	=	mechanical hydraulic efficiency of pump	
$\eta_{p\text{-}V}$	=	volumetric efficiency of pump	
$\eta_{p\text{-}tot}$	=	total efficiency of pump	
v	=	kinematic viscosity	$1 mm^2/s$
ω	=	angular speed	1 rad/s

Conversion table:

1	m^3/hr	=	0,0166	lpm
1	bar	=	100	
	kPa	=	10^5	N/m^2
1	rpm	=	0,104	rad/s
1	m^3/rad	=	6.283.184	cc/rev

1.2 Hydraulic pump types, constant output

To be able to use hydraulic power for a drive system a fluid flow needs to be generated by a hydraulic pump to obtain a velocity or speed of the driven equipment. The load of the driven equipment dictates the pressure that is necessary.

The choice of pump will be determined by a number of factors, which will need to be assessed by the designer:

a. System pressure or output
b. Output flow
c. Strokes/Revolutions per minute
d. Fixed or variable stroke volume
e. Input speed
f. Type of hydraulic fluid.
g. Weight and size
h. Suction conditions
i. Sensitivity to dirt
j. Pressure variations and noise
k. Purchase Cost
l. Characteristics of the driving engine
m. Delivery time

Almost all pumps applied in hydraulic technology work on the displacement principle.

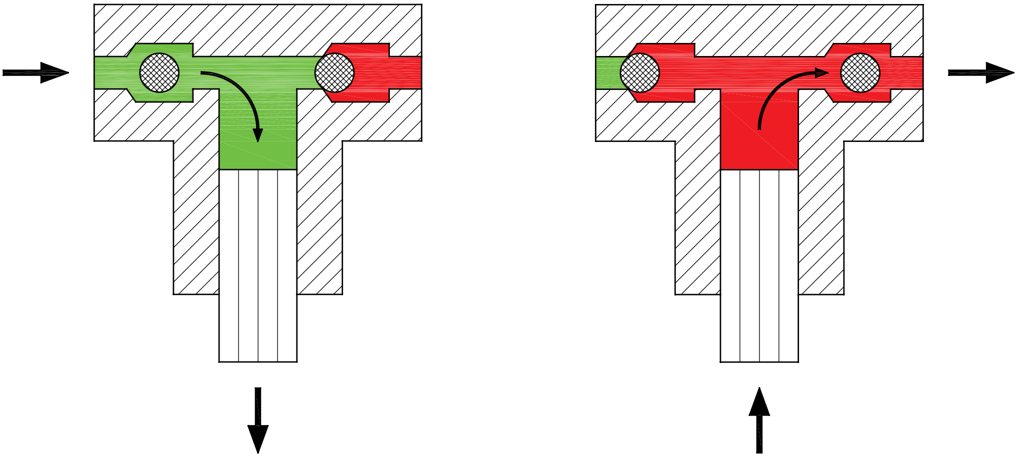


Figure 1.2.A Principle of a positive displacement pump

When the piston extends, a partial under-pressure is created in the cylinder as a result of which, oil will be sucked in through the suction valve from the suction pipe. Because of the pressure in the system, the check valve in the outlet port will remain closed. When the piston is moved inwards, the oil will be pushed into the system via the check valve in the outlet port.

The output from displacement pumps is more or less independent of the pressure in the system and is determined by the capacity per stroke V_s cc/rev and the rotational speed rpm of the pump. In subsequent paragraphs we will describe how various leaks in the pump are of influence on the volumetric efficiency of the pump, thus determining the effective pump output. The rotational speed of the pump drive can be constant when driven by an electric motor, or variable when driven by a combustion engine or variable speed electric motor.

Most of the factors determining the choice of pump, mentioned above, are self evident. However, one of the conditions often overlooked is the suction condition. Because this is a displacement pump, it has its own suction characteristics. If the pump with the suction connection is mounted above an oil reservoir, then the pump must be able to suck the oil against the gravitational forces. Due to the hydro-static height difference, between the level of the inlet side of the pump and the level of the liquid, and also as a result of the pressure losses due to the flow in the suction pipe, a under-pressure will be created at the suction side of the pump. As a result the air dissolved in the liquid will gradually separate out from the liquid in the form of free air bubbles, see also paragraph 1.6.9

When such an air bubble moves to the pressure side of the pump, a new process develops.

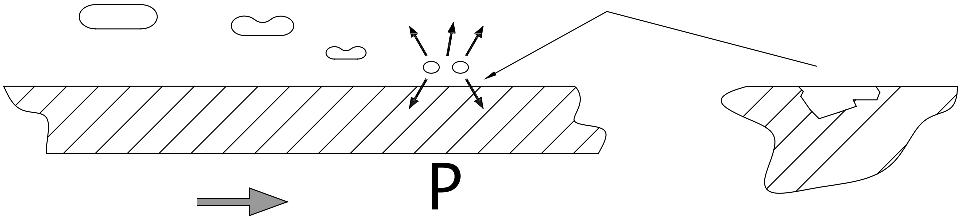


Fig 1.2.B Cavitation in hydraulic pumps

The air bubble becomes smaller due to the higher internal pressure. At some point the bubble will split itself into smaller bubbles. This is known as an implosion. It is coupled with very high local pressures, of up to 1000 bar and local temperatures of more than 1200 °C and oil gets burned. If this happens near a metal surface, then erosion will take place. This combination of processes is known as cavitation. In practice this effect can be easily observed. It will sound as if several marbles have been placed in the suction pipe rather than hydraulic liquid and oil is getting black. Manufacturers will specify the suction conditions, i.e. the minimum allowable inlet pressure at the inlet port, in their technical documentation.

All the most important characteristics of the most common pump types have been summarised in the table at the end of this chapter.

1.2.1 External gear pumps

In many situations where pressures of up to 250 bar apply, pumps with so-called external gearing are used. The important reasons for this are: their simple operation, low costs, good self suction operation, low sensitivity to dirt and relatively low weight. Vane pumps and or internal gear pumps are usually applied if the applications requires a low noise level.

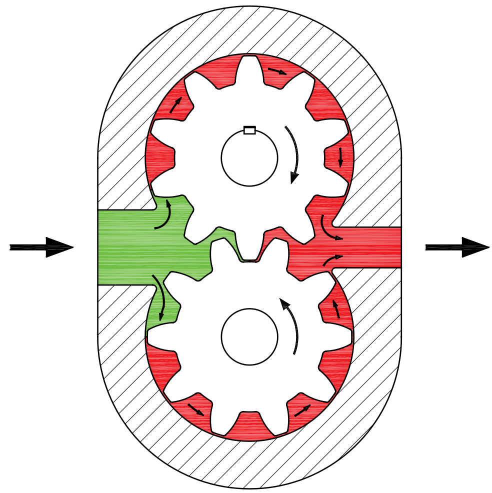


Fig 1.2.1.A Principle of an external gear pump

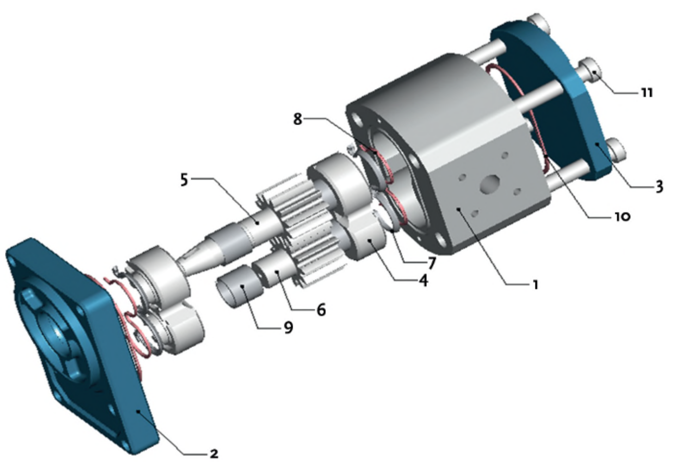


Fig 1.2.1.B Exploded view of an external gear pump (Courtesy of Bosch-Rexroth)

In essence, this type of pump consists of two shaft-mounted gear wheels. The shafts are mounted on bearings, in a tight-fitting housing. One of the gear shafts is driven. When this gearwheel rotates, the 2nd gear will also rotate due to the meshing gears. Oil is thus transferred from the suction connection of the pump to the pressure connection of the pump along the outer circumference of the gears.

The drive shaft needs to be driven with a torque proportional to the pressure at the exit port. A small proportion of the pumped oil flows back to the suction port due to internal leakage because of: the clearance between the teeth of the gear wheels and the housing, the clearance of the bearings on the gear wheel shafts and the clearance between the sides of the gears and the pump housing. Most gearwheel pumps are therefore fitted with compensated side plates to keep these leakages to a minimum.

Standard gearwheel pumps operate at 1000 to 3000 rpm at working pressures up to 250 bar. Higher rotation speeds and pressures are available. Power varies from 1 to 100 kW.

1.2.2 Internal gear pumps

Often, when noise levels are important, gear wheel pumps with internal gears are applied.

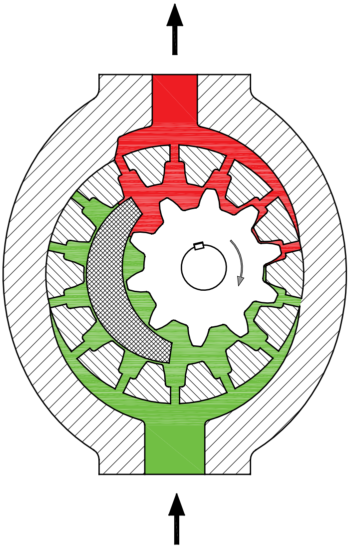


Fig 1.2.2.A Principle of an internal gear pump

Fig 1.2.2.B Section view of an internal gear pump (Courtesy of Bosch Rexroth)

This pump has a gearwheel with external teething, driven by the drive shaft and a gearwheel with internal teeth, which rotates in the pump housing and is driven by the teeth on the circumference of the external gearwheel. Because of the separating segment, both gearwheels can move oil from the suction side to the pressure side of the pump. Compared to external gearwheel pumps, these pumps produce less noise. Their available stroke volumes and maximum working pressures are comparable to those of external gearwheel pumps.

For a particular size of gearwheel, with a fixed diameter and a fixed number of teeth, stroke volumes can be adapted by increasing or decreasing the width of the teeth.

1.2.3 Vane pumps

The vane pump consists of a rotor on which a number of vanes are mounted that can be moved in a radial direction is mounted. The rotor is driven by the drive shaft. The outline of the housing is double-elliptical. This allows the oil to be moved from the suction side to the pressure side. In the section view of figure 1.2.3 two inlet and two outlet ports are used.

The blades move outwards due to the centrifugal forces. To reduce leakage at the outer circumference further, the blades are also pushed outwards by the pressure in the pump. In some types, a small spring is added so that a good seal can be achieved, even at low rotational speeds.

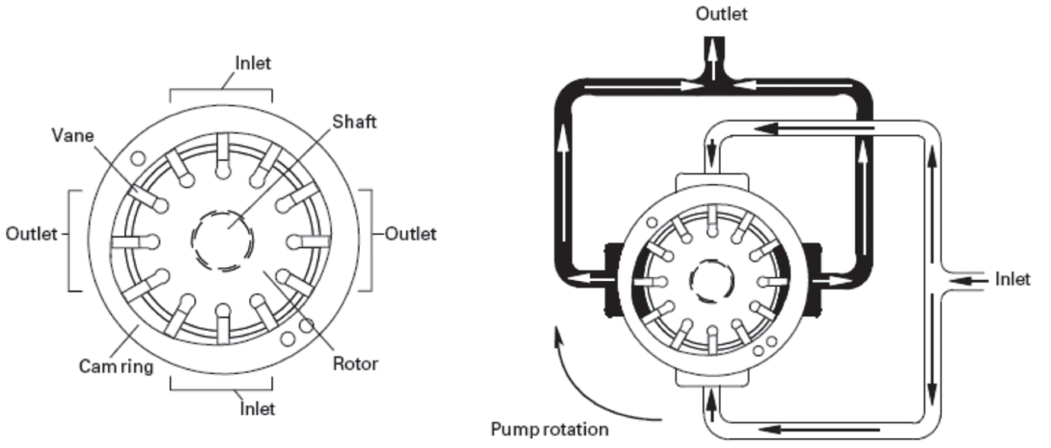
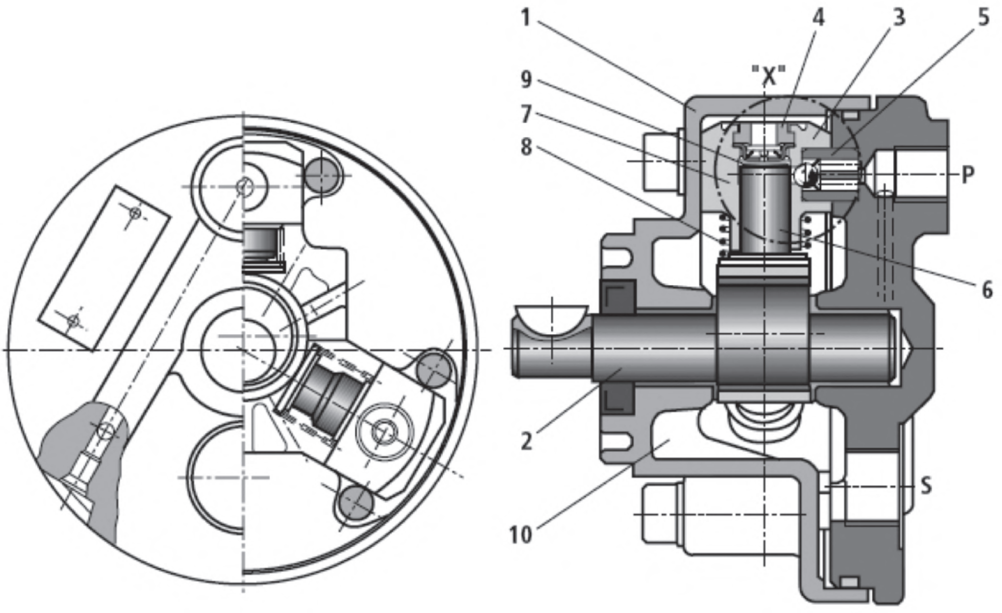


Fig 1.2.3 Principle of a vane pump (Courtesy of Eaton Vickers)

Vane pumps are selected in applications where cold start conditions and consequently high viscosities occur. As they have a fixed stroke volume they are often used in boost circuits or fluid conditioning circuits.

1.2.4 Radial piston pumps

Radial piston pumps consist of a number of cylinders which are positioned in circle and which work on a simplified version of the displacement pumps of figure 1.2.A. The eccentric shaft drives the pistons (6) in and out the cylinder blocks (7).



1	=	pump housing	5	=	pressure valve	9	=	suction valve
2	=	eccentric shaft	6	=	piston	10	=	suction chamber
3	=	pump elements	7	=	cylinder block	P	=	pressure port
4	=	suction valve	8	=	sliding shoes	S	=	suction port

Fig 1.2.4 Principle of a radial piston pump (Courtesy of Bosch Rexroth)

The pistons (6) move within the cylinder blocks (7). These pistons are driven by the drive shaft (2). The piston rods ends have been fitted with sliding shoes (8), which are forced to follow the stroke ring. The radial piston stroke is created because the stroke ring is placed eccentrically to the drive shaft. Build in check valves (4 and 5) ensure that each cylinder is connected with either the suction (S) or the pressure port (P).

Pumps of this type are in general suitable for high operating pressures of up to 1000 bar and are much less sensitive to dirt than the axial piston pump, which will be discussed later.

1.3 Hydraulic pump types, variable output

The output of a pump with constant stroke volume is determined by the stroke volume and the rotational speed of the driveshaft. If we assume a more or less constant rotational speed then the output will also be more or less constant. The drive-side torque of the pump with fixed output is directly proportional to the pressure on the pressure side. This means that the energy required for the drive shaft by a static rotational speed is directly proportional to the pressure on the pressure side. Later we will show that the working pressure is constant for many drive systems. This means that when a pump with fixed stroke volume is used, high, constant input power will be required to maintain the pressure in the system, although the hydraulic power is not used all the time.

In order that hydraulic pumps can be more widely used, pumps with a variable and adjustable stroke volume have been developed.

1.3 1 Swash plate pumps

In one of the possible designs of this type of pump, the input shaft drives a cylinder block (1). A number of pistons (2) (always an odd number: 9, 11 or 13) are positioned within the cylinder block, parallel to the input shaft. The reason for applying an odd number of pistons is that the flow fluctuations and consequently the pressure fluctuations are less than with an even number of pistons. Sliding shoes (5) which run against a swash-plate (4) are fitted to the rod-ends of the pistons. The rotational movement of the cylinder block is converted into a linear movement of the pistons.

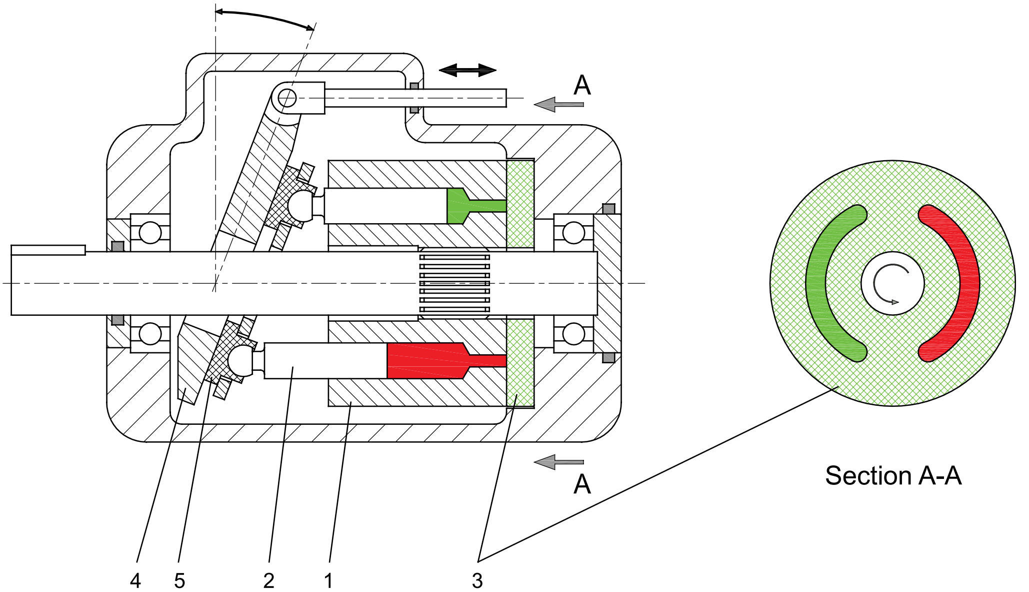


Fig 1.3.1.A Operating principle of an axial piston pump

The cylinder block runs against a so called port plate (3). In this disk plate are two semi circular (kidney shaped) grooves, the suction and pressure ports of the pump.

The null position of the swash plate is when it is placed perpendicular to the pump shaft. In this position the pistons do not move linearly and the pump displacement is zero, despite the fact that the pump shaft is rotating. By making the position of the swash-plate angle variable, it is possible to continuously regulate the stroke volume.

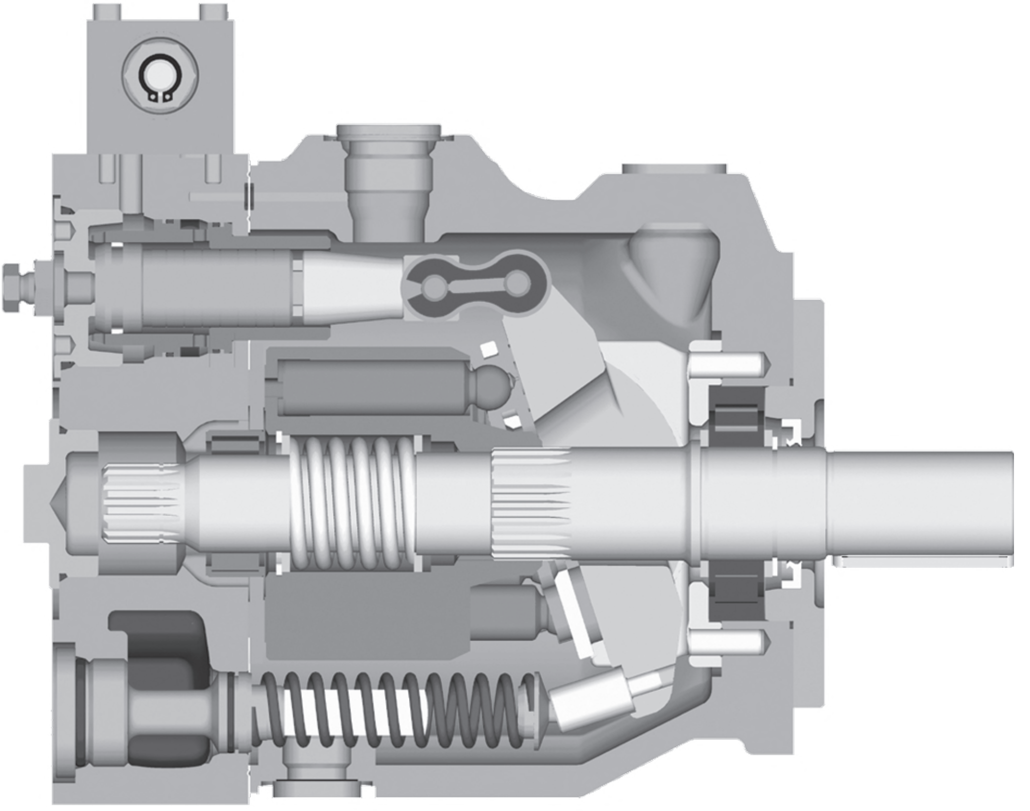


Fig 1.3.1.B Section view of an axial piston pump (Courtesy of Parker Hydraulics)

There are also models where the swash-plate can be moved beyond the null position shown in the illustration. In those cases, it is possible to adjust the angle of the swash-plate in both directions. If the rotational direction of the input shaft remains the same, it will be possible to continuously adjust the flow in two directions. The suction and pressure port of the pump swap function when the swash-plate passes through the null position.

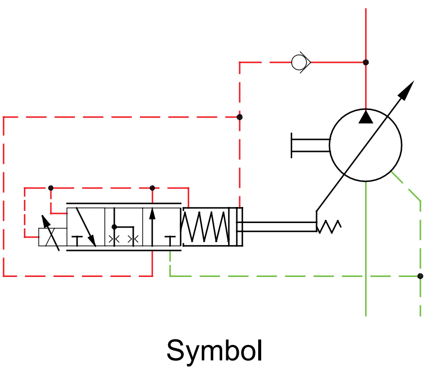
Because of the clearance that is present between the pistons and the bores in the cylinder drum and between the cylinder drum and the port disk, internal leaking takes places from the high pressure side to the pump housing. The leakage is removed through a separate oil drain connection, which is connected to the highest point of the pump housing. The highest point is chosen to prevent the pump housing from emptying when the pump is not driven. Because high temperatures can develop where the leaks occur, extra flushing oil is often pumped through the pump casing to provide extra cooling. For this purpose always a second drain port is available.

Adjustment of the swash-plate can take place in several different ways. The way this is done is among other things dependent on the function that the pump performs in the hydraulic system.

1.3.2 Proportional flow control

The stroke volume of a variable pump can be varied between 0 – 100% through the use of a pneumatic, hydraulic or electrical signal. For a pump built into a closed loop system this can be between -100 % and + 100%. If an electronic signal is used, extra pilot pressure is necessary to convert the electrical signal into a required control pressure.

The accuracy in which a particular output can be adjusted is highly dependent on the quality of the proportional valve that is being used to regulate the system. The highest level of accuracy will be achieved if the position of the adjustment mechanism is fed back to the control mechanism electronically.



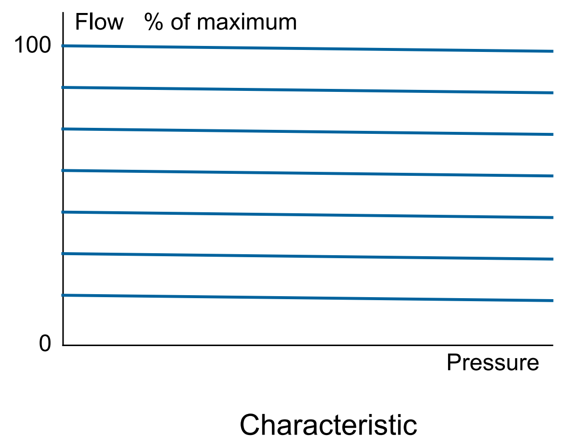
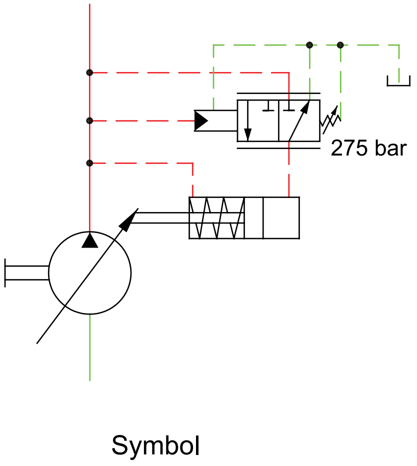


Fig 1.3.2 Proportional flow controlled pump (Courtesy of VPH)

This pump control mechanism may also be called the primary control as the output flow of the pump and thus the speed of an actuator can be varied from 0-100% or from -100 to +100%. The output pressure of the pump is determined by the induced pressure, this is the pressure that is generated in an actuator by external load. If the induced pressure is low, then the pump pressure can be low too.
The mechanical power to drive the pump shaft is proportional with the stroke volume, the drive speed and the pump pressure

1.3.3 Pressure control

With a pressure control function the system pressure of the pump is compared with the mechanical preset pressure in an adjustable spring. In this example the preset pressure is 275 bar.
The swash plate is controlled by a small hydraulic cylinder with large spring to its maximum angle, controlling the pump to its maximum stroke volume. The output flow from the pump creates a system pressure. This system pressure is compared with the preset value. A control pressure is send to the small control cylinder that keeps the swash plate angle at such a value that the system pressure equals the preset pressure.
If the measured system pressure exceeds the preset pressure in the spring, the pressure control valve shifts to the position of the adjustable spring and allows a small fluid to the control cylinder. The stroke volume gets reduced until the system pressure stabilises at the set value. When the necessary flow in the system increases, the system pressure will drop a fraction, which means that the stroke volume will be increased automatically so that the system pressure is returned to the desired level.
The main advantage of this type of pump control is that the pump output flow equals he amount of flow that is necessary in the system.



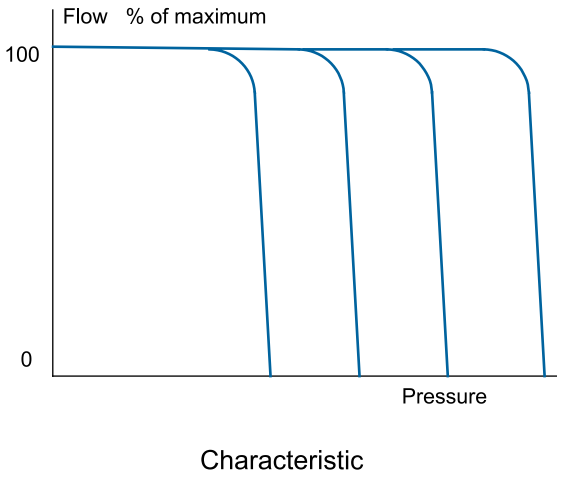


Fig 1.3.3 Pressure controlled pump (Courtesy of VPH)

This type of pump regulator is applied to achieve a so-called 'constant pressure' system. The pressure in the outlet port will be almost constant, equivalent to a constant pressure network for a pneumatic system or even equivalent to the voltage of an electrical circuit in domestic premises. It is possible to connect more than one user to the network. The maximum pump output must then be equal to the total required flow of all users that could possibly be driven at the same time. The distances between the users and the pump unit in large industrial installations and onboard ships are often large, sometimes as much as hundreds of meters. By installing a ring main rather than individual pipes to each user, large savings can be achieved on the pipe-work.

The main disadvantage of this type of pump regulator is the energy losses that occurs if the operating pressure of one of the users is significantly lower than that of the constant pressure system. This pressure difference causes loss of power, which will be converted into heat.

The mechanical power to drive the pump shaft is proportional with the stroke volume and the pump shaft speed.

1.3.4 Load sense control

If a load sensing regulator is applied, we also talk about a central pressure pipe for all individual users, the same as in the 'pressure control' application. A hydraulic 'loadsense' line is built between each user and the pump. The pressure regulator of the pump is now governed by the highest load pressure occurring amongst the connected users. The important difference with a standard pressure regulator is that the output pressure from the pump is 20-22 bar higher than that of the highest load pressure. This pressure difference is needed to make sure that the load sensor control mechanism can function properly and stable.

The advantage of a Load Sensor Control system is that the pump pressure will automatically adjust to the highest system pressure demanded by the actuators.

Because of the need for individual 'loadsense' line to the pump, this type of system is less suitable for systems involving long distances, for example on board ships.

Another disadvantage is that the stability of the regulator is very dependent on the distance to the user and the viscosity of the oil.

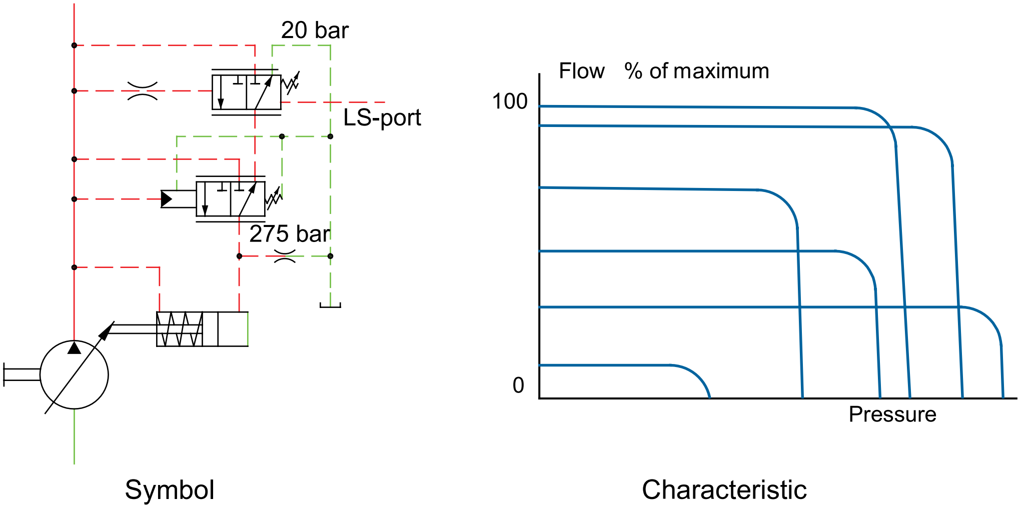


Fig 1.3.4 Variable pump with loadsense control (Courtesy of VPH)

1.4 Summary of pump characteristics

Type	Speed Max	Flow at 1500 rpm	Max Operat-ing pressure	Stroke volume	Temperature range	Viscosity range	Noise	Optimum efficiency
	n_{max}	Q	p	V_s		$\nu_{min}\ldots\nu_{max}$	L_{wa}	η_{max}
	min^{-1}	lpm	bar	cm^3	°C	mm^2/s	dB	
External Gear Pump with pressure compen-sated side plates	800-3000	1,5-24	175	1-16	-15-+80	42-90	68-85	0,89
Internal Gear Pump	1200-5000	5,6-576	63-250	5,1-500	-20-+80	20-100	73-83	0,87
Inernal gear Pump, pressure compensated	2500-4500	5,4-370	175-300	3,6-250	-20-+80	28-100	69-79	0,93
Vane Pump, fixed displacement	900-5000	12,8-188	70-290	8,5-270	-10-+80	16-100	75-82	0,80
Screw Pump	4500	13,5-2625	80-160	9-1750	-10-+90	2-100	62-80	0,90
Axial piston Pump with swashplate	500-2000	8,6-74	200-420	6-1000	-15-+80	15-100	83-87	0,90
Axial piston Pump, fixed dsiplacement, bent axis type	950-6000	71-3584	350-400	94-1000	-25-+80	10-100	85-98	0,90
Radial piston Pump	300-2000	1,7-198	210-700	1,2-133	-10-+70	15-100	69-83	0,90

Fig 1.4 Hydraulic pump set with three variable axial piston pumps, running in parallel at 300 bar system pressure (Courtesy of Huisman)

1.5 Formulas

The most important formulae for hydraulic pumps are:

$$T = p \cdot V_s \tag{1.1}$$

$$P_{in} = T \cdot \varphi = T \cdot \frac{2\pi}{60} \cdot n = P_M \tag{1.2}$$

$$P_{out} = p \cdot Q = P_H \tag{1.3}$$

$$\eta_{p-tot} = \eta_{p-MH} \cdot \eta_{p-V} = \frac{P_{out}}{P_{in}} \tag{1.4}$$

$$\eta_{p-MH} = \frac{p \cdot V_s}{T} \tag{1.5}$$

$$\eta_{p-V} = \frac{Q}{\varphi \cdot V_s} \tag{1.6}$$

n	=	speed, rotation	rpm	V_s	=	stroke volume	m^3/rad
p	=	pressure	Pa	$\eta_{p\text{-}MH}$	=	mechanical efficiency of the pump	
P_H	=	hydraulic Power	W	$\eta_{p\text{-}V}$	=	volumetric efficiency of pump	
P_{in}	=	input Power	W	$\eta_{p\text{-}tot}$	=	total efficiency of pump	
P_M	=	mechanical Power	W	ϕ	=	angular speed	rad/s
P_{out}	=	output Power	W				
Q	=	fluid flow	m^3/s				
T	=	torque	Nm				

A formula often used, together with its units of measure is:

$$Q = \frac{\eta_{p-V} \cdot n \cdot V_s}{1000} \tag{1.7}$$

where:

Q	=	fluid flow	lpm	Vs	=	stroke volume	cc/rev

The following formula is used to quickly calculate the hydraulic power and from that the net power input required:

$$P = \frac{p.Q}{600} kW \tag{1.8}$$

P	=	power	kW	Q	=	fluid flow	lpm
p	=	hydraulic pressure	bar				

1.5.1 Volumetric efficiency / mechanical efficiency

The volumetric and mechanical efficiency vary between about ,85 and ,98 and are often a function of the rotational speed of the pump, the working pressure and the viscosity of the fluid. An example of the curve for the volumetric, mechanical and total efficiency of a pump is given in figure 1.5.1.A.

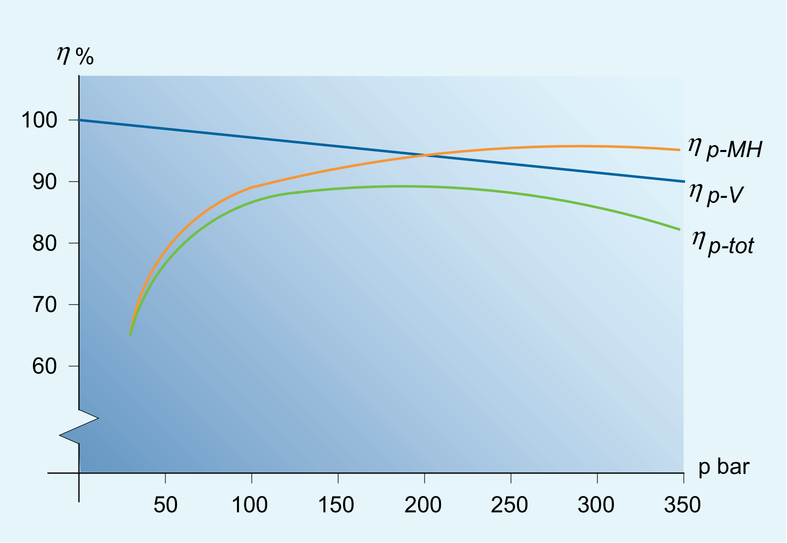


Fig. 1.5.1 Volumetric ($\eta_{p\text{-}V}$), Mechanical ($\eta_{p\text{-}MH}$) and total efficiency ($\eta_{p\text{-}tot}$) of a hydraulic pump. (Courtesy of VPH)

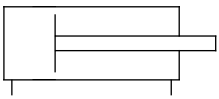
1.6 Actuators: Cylinders

Cylinders are the most often used actuators in hydraulic control technology. They can be used to apply large forces, whilst the movement of the driven machinery can be controlled accurately with relatively high dynamic characteristics. Of the types available as standard, listed below, the double acting differential cylinder is the most commonly applied. For this type of arrangement, both the outward and inward movements of the cylinder produce volumetric output at the annular side and the bottom side of this type of cylinder.

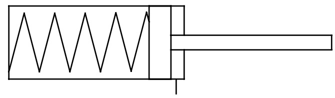
1.6.1 Different types of cylinders

Single Action Cylinders

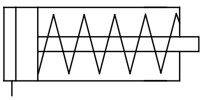
Piston/Rod cylinder



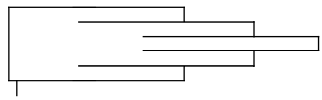
Pulling cylinder



Pushing cylinder

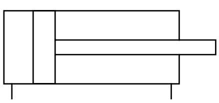


Telescopic cylinder

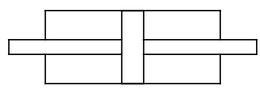


Double action cylinders

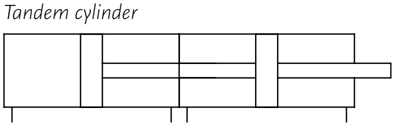
Differential double acting cylinder



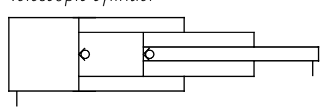
Equal area cylinder



Tandem cylinder



Telescopic cylinder



The proportion between the area of the circular bottom surface and the ring shaped top or annular surface for the 'Differential double acting' type of cylinder is approximately 2, this is called a ϕ of 2.
All symbols (Courtesy of VPH)

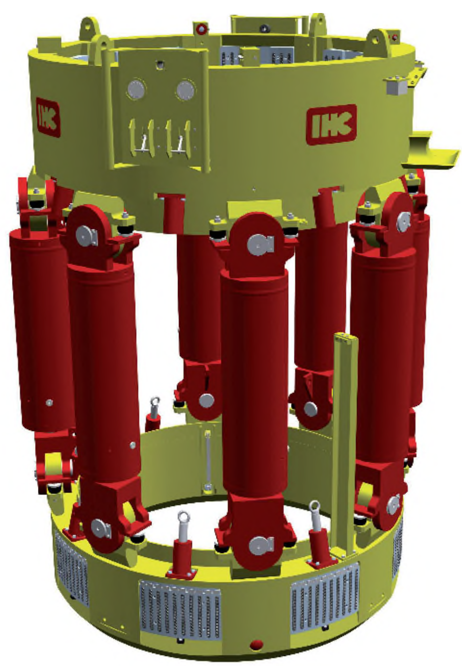


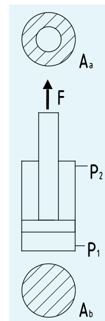
Fig 1.6.1. Application of double acting differential cylinders in a 3000t levelling tool (Courtesy of IHC)

1.6.2 Formulas for a double action cylinder

Piston area: $$A_b = \frac{\pi}{4} \cdot D^2 \qquad (1.9)$$

Annulus area: $$A_a = \frac{\pi}{4} \cdot (D^2 - d^2) \qquad (1.10)$$

A_b	=	bottom area	m²	D	=	outer diameter	m
A_a	=	annular area	m²	d	=	rod diameter m	



Surface ratio: $$\varphi = \frac{A_b}{A_a} \qquad (1.11)$$

Forces: $$F = p_1 \cdot A_b - p_2 \cdot A_a \qquad (1.12)$$

Mechanical Power: $$P_M = F \cdot v \qquad (1.13)$$

F	=	Force	N	v	=	speed	m/s

Hydraulic Power: $$P_H = p \cdot Q \qquad (1.14)$$

P	=	pressure	N/m²	P_{out}	=	output power	W
P_M	=	mechanical power	W	Q	=	flow	m³/s

For the outward movement and F > 0 the following applies:

Mechanical hydraulic efficiency: $$\eta_{m-MH} = \frac{F}{p_1 \cdot A_b - p_2 \cdot A_a} \qquad (1.15)$$

Fluid flow: $$Q = v \cdot A \qquad (1.16)$$

The fluid flow towards the cylinder is dependent on the area of the cylinder port that is driven, A_b or A_a.

Speed: $$v = \frac{s}{t} \qquad (1.17)$$

s	=	mechanical stroke	m	t	=	time	s

1.6.3 Permissible speed

The initial assumption is to adhere to a maximum piston speed of approximately 0,5 m/s. With modern seals and synthetic bearings piston speeds of up to 1 m/s are often no longer a problem and even speeds of up to approximately 3 m/s are technically possible. You will need to check this out carefully with the supplier of the cylinder seals.

1.6.4 Cylinder friction

Because of the built-in seals, the cylinder will always have a certain amount of internal friction, which results in a higher necessary pressure to drive the cylinder. The friction is highly dependent on the design of the seal and its diameter. Larger diameter cylinders have relatively less friction and therefore greater efficiency. The rod seal friction is often larger than the piston seal friction. A general guideline for practical use is that the friction causes a pressure loss of 1 – 2 % of the maximum possible working pressure. This means that for a cylinder with a design pressure of 250 bar, the internal friction of the cylinder causes a pressure loss of approximately 2.5 to 5 bar (calculated at the surface of the bottom side of the piston).

At very low operating pressures and/or speeds 'stick-slip' can occur due to the friction of the seals. This manifests itself in a stop-start type movement of the cylinder. After the cylinder starts to move, the friction of the seals will initially decrease and will then increase. Figure 1.6.4. represents the typical development of the friction of a cylinder. Some of the factors that influence that friction are: the type of seal and the smoothness of the running surfaces.

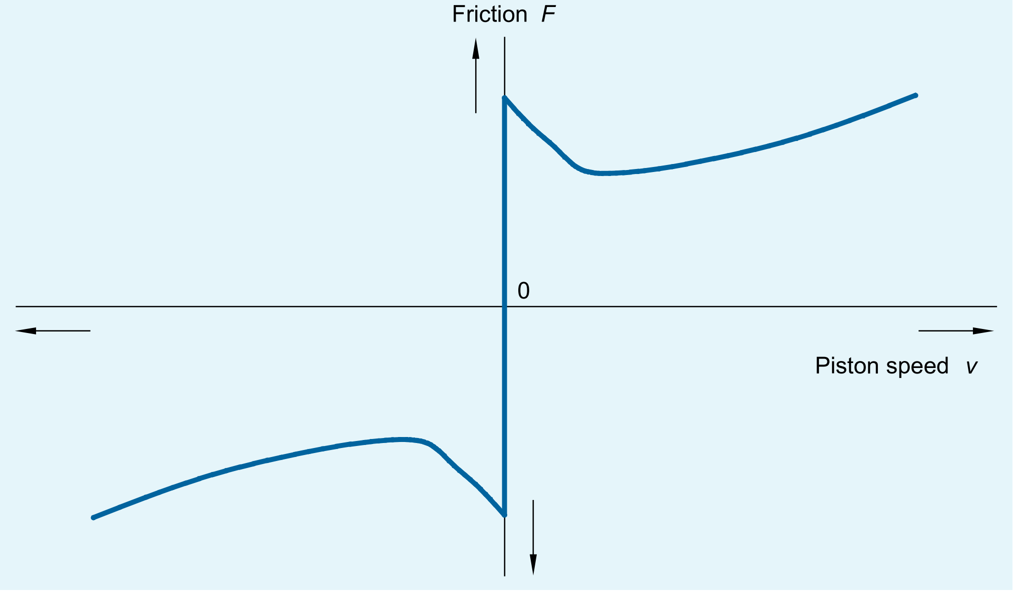
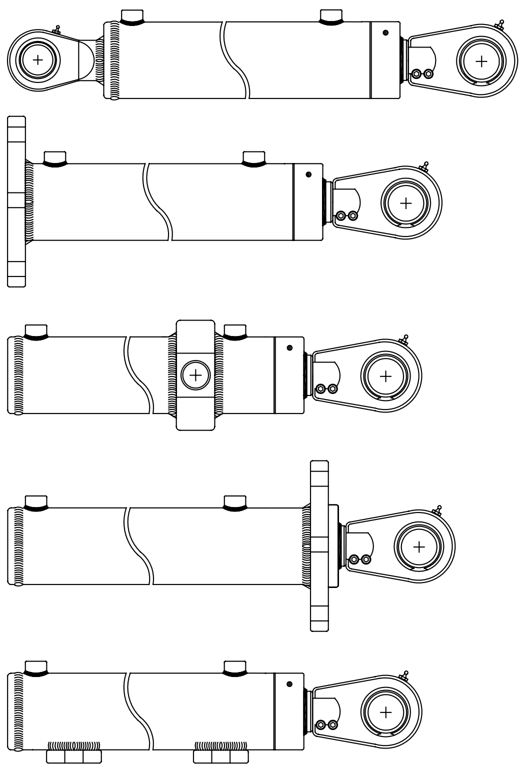


Fig 1.6.4 Cylinder friction as function of the piston speed

1.6.5 Application of the cylinder

The movements of the piston rod ensure that apart from the pure linear movement of the rod, a rotating movement of either the cylinder housing or the rod, relative to the driven machinery develops. In order to limit the torque and lateral forces that these non-linear movements cause as much as possible, a number of standard methods of incorporation is possible. Standard ISO 6099 gives definitions of several different types.



ISO type

MP5, one eye with spherical bearing at the bottom and at the rod end

MF4, swivel eye at the rod end with foot mounted flange

MT4. swivel eye at the rod end with intermediate trunnions

MF3, head mounted with circular flange, one spherical eye at the rod end

MS2, side lugs mounted, one spherical eye at the rod end

Fig 1.6.5. The most important cylinder mounting according ISO-6099 (Courtesy of VPH)

1.6.6 Buckle calculation for cylinders

The maximum thrust of a cylinder is limited to the maximum buckling load of the piston rod. To ensure that the cylinder load is nowhere near the buckling load, a safety factor is applied.

The maximum buckling load of a cylinder can be calculated with the formulas of Euler and Tetmayer. The buckling length L_k is dependent on the way the cylinder has been fitted. A number of examples is shown in the figures below. For example, for a cylinder with eye mounting at the bottom and at the rod end the rule is that L_k is equal to the built-in length of the cylinder L. The measure for L is the length of the cylinder and the rod in its most extended position.

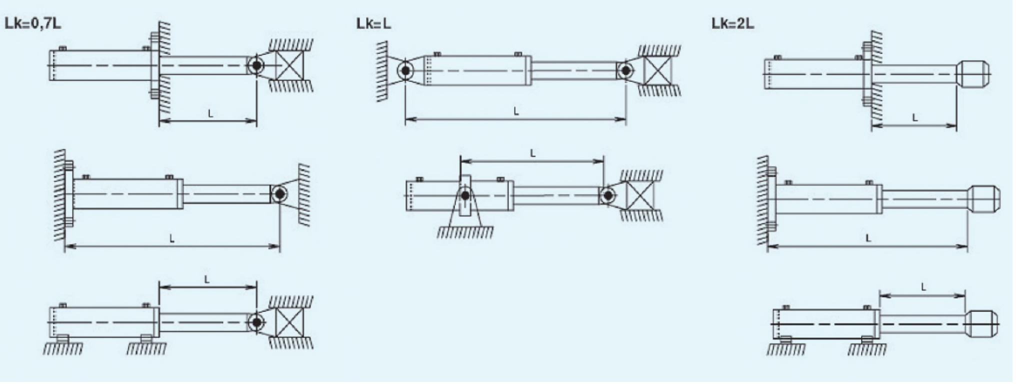


Fig 1.6.6. Buckling length L_k for different ways of fitting

Moment of Inertia of the piston rod: $$I = \pi \cdot \frac{d^4}{64} \qquad (1.18)$$

Radius of gyration: $$i = \sqrt{\frac{I}{\frac{\pi}{4} \cdot d^2}} \qquad (1.19)$$

where:

d	=	diameter of rod	mm	i	=	radius of gyration	mm
I	=	area moment of inertia	mm^4				

Slenderness ratio $$\lambda = \frac{L_k}{i} \qquad (1.20)$$

For a slenderness ratio of (λ) > 200 for horizontal build and or of (λ) > 250 for vertical build, the advice is to choose a larger diameter for the piston rod.

If λ ≥ 100: $$F_{max} = \frac{\pi^2 \cdot E_{st} \cdot I}{V_f \cdot L_k^2} \qquad (1.21)$$

With:

F_{max} = maximum pushing force	N	V_f = safety factor	2,5 3
E_{st} = modulus of elasticity	N/mm² (2,1 x 10^5)	L_k = buckling length, see figures	mm

If $\lambda < 100$:

$$F_{\max} = \frac{\frac{\pi}{4} \cdot d^2 \cdot \left(Rm - (Rm - 210) \cdot \left(\frac{\lambda}{100} \right)^2 \right)}{V_f} \qquad (1.22)$$

with Rm = Yield strength of the material (for St 52.3 : Rm = 360 N/mm²)

In the previous formulas it has been assumed that the rod is the only part that offers resistance against buckling. The housing however also has a certain rigidity, which, in practice, means that the cylinder has a larger allowable buckling load.

In Roark's *Formulas for Stress and Strain*, 7th edition, an extra correction factor for the calculation of the permissible axial force F_{max} is introduced. This can give the advantages of a higher allowable buckling load by taking the rigidity of the housing into account.

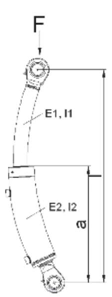
$$F_{\max} = K \cdot \frac{\pi^2 \cdot E_{st} \cdot I}{V_f \cdot L_k^2} \qquad (1.23)$$

For explanation of the different units and dimensions, see preceding pages.

We introduce:

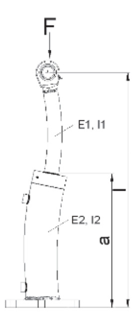
$$\frac{E_2 \cdot I_2}{E_1 \cdot I_1} = \tau \qquad (1.24)$$

The correction factor K for the different types of build, can be extracted from tables 1.6.6



Value of correction K					
a/l	1/6	2/6	3/6	4/6	5/6
$\tau = 1$	1,000	1,000	1,000	1,000	1,000
$\tau = 1{,}5$	1,010	1,065	1,180	1,357	1,479
$\tau = 2$	1,014	1,098	1,297	1,633	1,940

Table 1.6.6.A Correction K for a cylinder with eye at the bottom and rod ends



Value of correction K					
a/l	1/6	2/6	3/6	4/6	5/6
$\tau = 1$	2,046	2,046	2,046	2,046	2,046
$\tau = 1{,}5$	2,241	2,289	2,338	2,602	2,976
$\tau = 2$	2,369	2,503	2,550	2,983	3,838

Table 1.6.6.B Correction K for a bottom mounted cylinder with eye at the rod end and foot mounting

For values of τ between 1.0 and 1.5 or between 1.5 and 2, the values from this table can be interpolated.

1.6.7 Rod layers

To give a piston rod good running characteristics and to offer resistance to possible corrosion whilst the rod is extended outside the housing, a wear and corrosion resistant layer is applied. Several methods are available, each with their own characteristics, advantages and disadvantages.

Ni-Cr layers

The most common rod coating is Nickel-Chromium Ni-Cr. The Ni layer is applied to the base material first. This layer must have a very good bonding with the base material. The Cr layer is applied to this layer. Because of its soft characteristics, it can easily be polished to a very smooth surface. The thickness of the layers of Nickel and Chromium are usually in the order of 60μ and 40μ.

Chromium is porous. If the rod is moved in and out on a regular basis, a continuous oil film will stay behind on the rod's surface. But is the piston rod stays in the extended position for longer periods of time, seawater will flush the oil from the pores and will penetrate through to the Nickel layer. In the end this fills the microscopically small cracks between the nickel and the chromium with a chloride type solution.

In short, a metal-ion is formed which will dissolve in this environment and an opposite reaction takes place in which the metal-ion reacts with the water and forms a hydroxide. The chlorine in the seawater works as a catalyst in these circumstances.

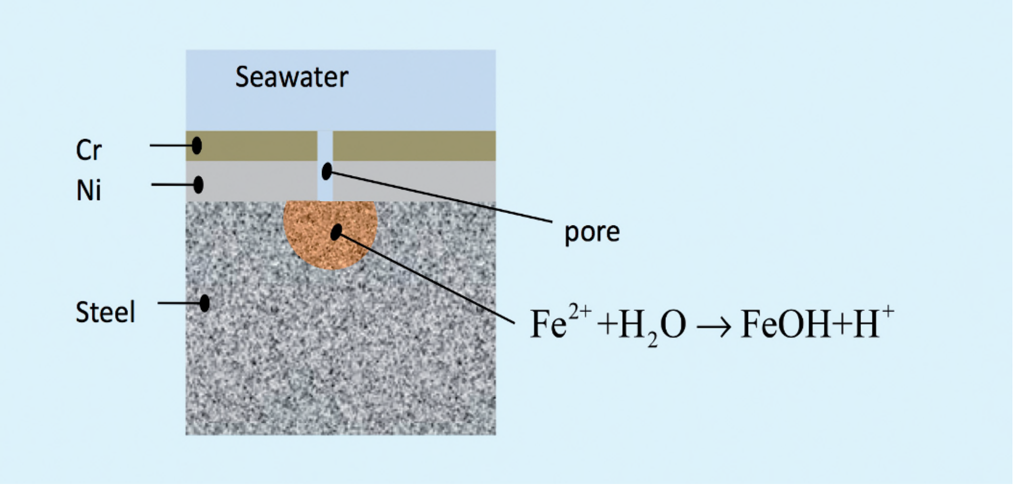


Fig 1.6.7.A The chemical reaction of steel rods and their protective layers with seawater . The same chemical reaction may occur with ceramic layers as they also may show a porous structure

Depending on the availability of fee oxygen in the water different chemical reaction will occur. The reaction as shown in figure 1.6.7.A takes place when the amount of oxygen is limited . Apart from elements of chloride, other sulphur type elements and other light impurities remain. This creates a mixture of hydrochloric acid and sulphuric acid which creates the ideal environment for the erosion of the chromium layer. As a result a light form of corrosion is formed underneath the chromium layer, a process that hardly affects the nickel layer if at all. This eventually causes small blisters to form, on the outside of the chromium layer or ceramic , which break off and leave small pits

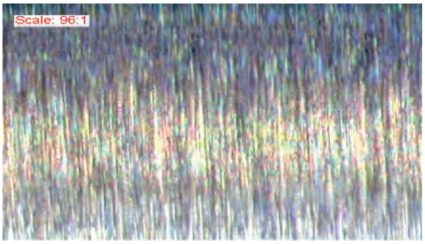


Fig 1.6.7.B Picture of a standard hard chrome plated rod (Courtesy of Parker Hannifin, Symposium Delft /Holland 2004)

Ceramic layers

An important breakthrough in the technology for piston rod coatings came when the plasma spray process was invented. This technology makes it possible to create a spray coating of Al_2O_3 and TiO_2 oxides, resulting in the development of ceramic coatings.
There are two main techniques for the production of a ceramic layer. The Plasma process and the HVOF process.

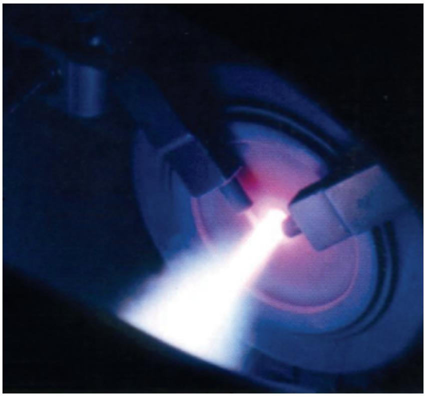


Fig 1.6.7.C The Plasma process (Courtesy of Bosch Rexroth Symposium Delft/Holland 2004)

In the Plasma process a mixture of gases, usually argon and hydrogen pass through a nozzle where a constricted arc can raise temperatures up to 30,000 ﬃC. The gas exits the nozzle to produce a high velocity of 200 m/s ionized plasma jet into which powdered material is injected. The powder is heated and accelerated to form a high velocity spray of molten material, which subsequently penetrates the prepared substrate (the piston rod) to form a dense integrated bonded coating.
The High Velocity Oxy Fuel (HVOF) coating is a process where oxygen and liquid or gas fuel are burned under pressure. The hot exhaust gases exit down a narrow bore nozzle where finely powdered material is axially introduced. The powder is accelerated and heated at 2500°C and supersonic velocities of 1200 m/sec. On impact the thermal and high kinetic energy ensures excellent bonding. The coating produced is harder and denser than those produced by the plasma process.

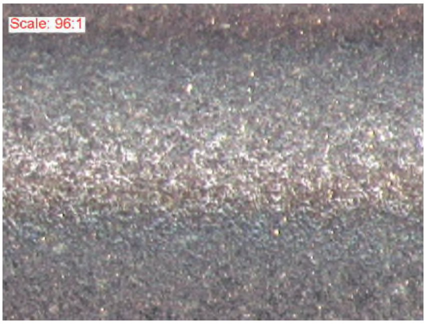


Fig 1.6.7.D Structure of a rod surface with ceramic treatment (Courtesy of Parker Hannifin, Symposium Delft /Holland 2004)

Fig 1.6.7.E Typical application of a cylinder with ceramic layer for the Abandonment and Recovery sheave on the Subsea7 Seven Oceans (Courtesy of IHC Vremac)

The experience with ceramic rod coatings is that they too can be porous and therefore have the same disadvantages as a Ni-Cr coating of corrosion between the base material and the rod coating. By applying specialist versions of the powders it is possible to create a good layer, i.e. free of porosity.
There are however a few other disadvantages with the ceramic layer. The sealing system must be made differently because ceramics are very hard and have a grainy structure, which means that normal seals wear out very fast. The costs of a ceramic layer are also higher than those of a Ni-Cr one. The costs for both methods: ceramic and Ni-Cr, will soon end at the same level because the use of Chrome is more and more restricted due to its negative environmental effects.

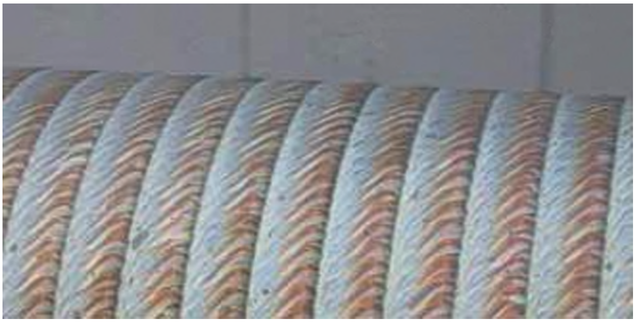


Fig 1.6.7.F Piston rod after the PTA process (Courtesy of Hunger)

Plasma transferred arc welding

The Plasma Transferred Arc welding process uses very high temperatures to weld the plasma powder. At extremely high temperatures up to 15,000 ﬄC the coating powder is melted and fused with the substrate material. It forms an absolutely non-porous layer and an excellent bonding of the coating to the substrate material. It is also possible to repair local mechanical damages on site. After this welding process the rod surface needs to be precision honed and finished.

1.6.8 Cylinder cushioning

High impact at the mechanical end position of the piston can be prevented if damping or cushioning is built in. Such cushioning is necessary for cylinders with high attached mass, combined with high terminal velocity of the piston. A cylinder cushion must only operate in emergencies, when the normal controls no longer function. The normal control of the piston needs to ensure that the piston comes to a low speed before the mechanical end position has been reached.

In principle a cushion is constructed in such a way that a certain amount of fluid is forced through a throttle valve. This throttle can operate on a fixed or adjustable basis.

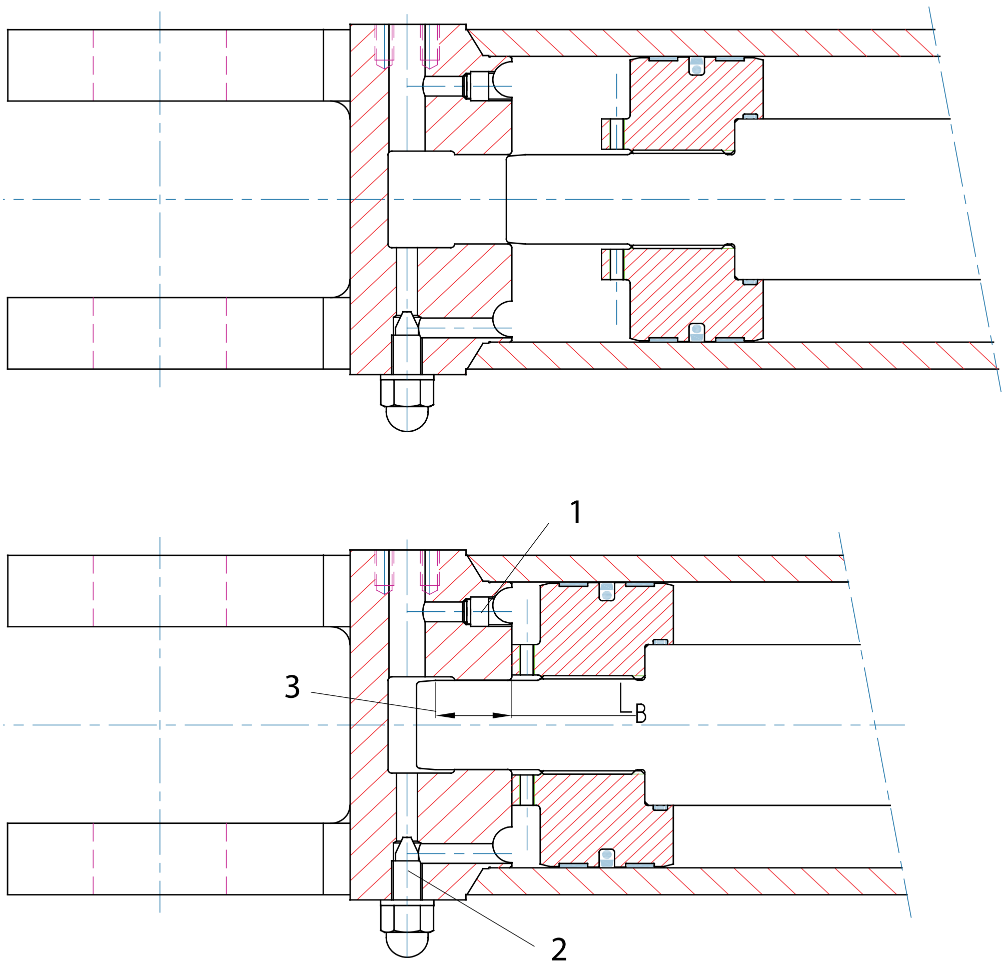


Fig 1.6.8.A A simple but efficient cushion construction at the bottom end of a cylinder. In the figure above the piston with the cushion rod is nearly entering the cushion seat in the bottom flange.

1	= check valve		h_o	= clearance between piston and chamber m	
2	= adjustable throttle		L_B	= cushion length	m
3	= cushion rod		M	= mass of rod and connected equipment	kg
A_T	= opening of variable throttle	m²	v_B	= velocity during cushioning	m/s
A_B	= buffered piston area	m²	v_K	= velocity when reaching the cushion	m/s
d_D	= diameter of cushion piston	m	x	= movement of piston from commencement of cushion	

In order that the piston doesn't need to be moved in the opposite direction to the damping direction with a reduced working area A_T, a check valve is built in. The circular groove in the bottom end cap is there to enable the pressure from the bottom end port to operate at the bottom area when the cylinder is extended again. Without such a groove the piston may be pushed against the bottom end cap by an external force where as a result the bottom area is in fact reduced to a very small area.

The following formula applies to the throttle:

$$Q_T = \alpha . A_T . \sqrt{\frac{2}{\rho} . (p_B - p_A)} = v_B . A_B \tag{1.25}$$

with:

α	=	restriction coefficient (≈ 0.61 – 0.69)		p_A	=	pressure at the bottom end port	N/m²
ρ	=	density of the fluid	kg/m³	p_B	=	pressure under the piston	N/m²
Q_T	=	flow through throttle	m³/s	Δp_T	=	pressure drop across the throttle	N/m²

From the above formula it is possible to derive the formula for the damping pressure that occurs in the case of an adjustable damping:

$$\Delta p_T = \frac{\rho}{2} \cdot \frac{1}{\alpha^2} \cdot \left(\frac{A_B}{A_T} . v_B \right)^2 \tag{1.26}$$

The pressure under the piston during the cushioning must not exceed the maximum design pressure of the cylinder. This has often happened when equipment has been newly commissioned, with the result that the diameter of the cylinder tubes has permanently expanded over a short length of tube or that the bolts which fix the bottom or piston end caps to the cylinder tube have broken.

For simple inertia loads, with no forces acting, the cushion needs to retard the mass M. If we assume an incompressible fluid and if we neglect friction we get:

$$M \cdot \frac{d^2 x}{dt^2} = M \cdot \frac{dv_B}{dt} = -\Delta p_T \cdot A_B \tag{1.27}$$

With the help of the following equation:

$$\frac{dv_B}{dt} = \frac{dv_B}{dx} \cdot \frac{dx}{dt} = v_B \cdot \frac{dv_B}{dx} \tag{1.28}$$

We combine equations 1.26, 1.27 and 1.28:

$$\Delta p_T = \frac{\rho \cdot A_B^2}{2 \cdot \alpha^2 \cdot A_T^2} \cdot v_B^2 = -\frac{M \cdot v_B}{A_B} \cdot \frac{dv_B}{dx} \tag{1.29}$$

and:

$$\int_{v_K}^{0} \frac{1}{v_B} \cdot dv_B = -\frac{\rho \cdot A_B^3}{2 \cdot \alpha^2 \cdot A_T^2 \cdot M} \int_0^x dx = -\frac{C_1}{M} \int_0^x dx \tag{1.30}$$

The solution of equation 1.30 gives:

$$v_B = v_K \cdot e^{-\frac{C_1}{M} \cdot x} \tag{1.31}$$

The result is that the velocity v_B of the cylinder during the cushioning decays exponentially, see also figure 1.6.B

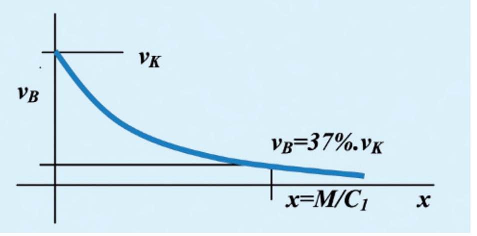


Fig 1.6.8.B Exponential decay of the cylinder velocity v_B with a throttle type cushion

The advantage of hydraulic damping with just the throttle is that it is adjustable. The disadvantage is that the induced pressure Δp_T is high at the beginning of the cushion but also decays exponentially with the velocity of the cylinder and is therefore not very efficient.

In the equations for the cushion above it has been assumed that no flow occurs between the cushion piston and the cushion chamber in the bottom end plate of the cylinder. As the piston has to enter without causing damage a minimum clearance h_o needs to be present between the piston and the chamber.

The following formula applies to the gap:

$$Q_T = \frac{2 \cdot \pi \cdot \frac{d_D}{2} \cdot h_o^3}{12 \cdot \eta \cdot x} \cdot \Delta p_T = v_B . A_B \tag{1.32}$$

or:

$$\Delta p_T = C_2 \cdot v_B \cdot x \tag{1.33}$$

with:

$$C_2 = \frac{A_B \cdot 12 \cdot \eta}{\pi \cdot d_D \cdot h_o^3} \tag{1.34}$$

with:

η	=	dynamic viscosity of fluid	Pa.s

now combining equations 1.29 and 1.33:

$$\Delta p_T = C_2 \cdot v_B \cdot x = -\frac{M \cdot v_B}{A_B} \cdot \frac{dv_B}{dx} \tag{1.35}$$

or:

$$dv_B = -\frac{C_2 \cdot A_B}{M} \cdot x \cdot dx \tag{1.36}$$

The piston velocity is found by integration:

$$v_B = -\frac{1}{2} \cdot \frac{C_2 \cdot A_B}{M} \cdot x^2 + v_K \tag{1.37}$$

The necessary stroke x or required cushion length L_D over which the mass M comes to a halt, so when $v_B = 0$, can be calculated with:

$$x(v_B = 0) = \sqrt{2 \cdot \frac{M}{C_2 \cdot A_B} v_K} \qquad (1.38)$$

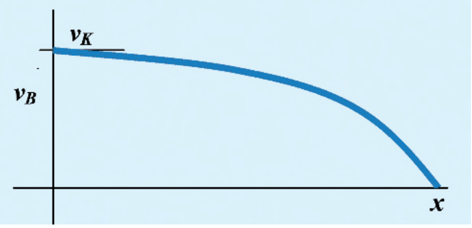


Fig 1.6.8.C Quadratic decay of the cylinder velocity v_B with viscous damping

The pressure drop during the cushioning can be found with:

$$\Delta p_T = C_2 \cdot v_B \cdot x = -\frac{1}{2} \cdot \frac{C_2^2 \cdot A_B}{M} \cdot x^3 + C_2 \cdot v_K \cdot x \qquad (1.39)$$

The value of x where the pressure drop Δp_T has a maximum is found by differentiation:

$$\frac{\Delta p_T}{dx} = -\frac{3}{2} \cdot \frac{C_2^2 \cdot A_B}{M} \cdot x^2 + C_2 \cdot v_K = 0 \qquad (1.40)$$

$$x = \sqrt{\frac{2}{3} \cdot \frac{v_K \cdot M}{C_2 \cdot A_B}} \qquad (1.41)$$

The flow across the gap as given with equation 1.32 is for the condition that the piston of the cushion is concentric with the cushion chamber. Even if the damping piston is only marginally off-centre, and this is nearly always the case due to tolerance allowed in the design diameter, the flow across the gap will increase and the achievable pressure build up during the damping process can be substantially be reduced.

For an eccentric position of the cushion piston in the cushion chamber with eccentricity e equation 1.32 becomes:

$$Q_{T,e} = \frac{2 \cdot \pi \cdot \frac{d_D}{2} \cdot h_o^3}{12 \cdot \eta \cdot x} \cdot \left(1 + \frac{3}{2} \cdot \left(\frac{e}{h_o}\right)^2\right) \Delta p_T \qquad (1.42)$$

If the cushion piston is at the most eccentric position ($e = h_o$), it can be found that the flow across the cushion piston $Q_{T,e} = 2{,}5 \,.\, Q_T$ and the pressure build up is reduced with a factor $1 / 2{,}5 = 0{,}4$

A possible alternative to hydraulic damping is the use of a pure mechanical spring in the form of synthetic rings, for example the material called Orkot. This material has good stability and durability characteristics for applications with hydraulic fluids and acts like a mechanical spring. The elasticity E_{orkot} of the material is about $2{,}5 \times 10^9$ N/m².

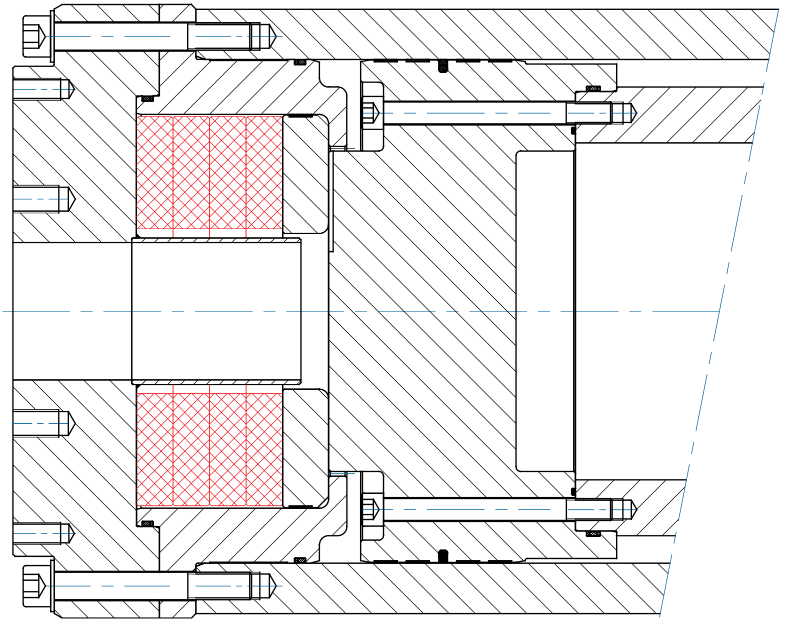


Fig 1.6.8.D Plastic "Orkot" rings as alternative for a hydraulic cushion

If the surface area of the damping material is known, it is possible to calculate the spring-stiffness C_{orkot}.

$$C_{orkot} = \frac{A_o \cdot E_{orkot}}{L_B} \qquad (1.43)$$

with:

A_o	=	surface area of orkot material	m²	E_{orkot}	=	elasticity of orkot	N/m²
C_{orkot}	=	stiffness of the plastic material	N/m²	L_B	=	length of the plastic package	m

The kinetic energy is converted by the mechanical spring in line with the following formula:

$$W = \frac{M}{2} \cdot \left(v_K^2 - v_E^2\right) = \int_0^x C_{orkot} \, . \, L_B \, . \, dx = C_{orkot} \cdot L_B \cdot x \qquad (1.44)$$

With:

x	=	linear compression of the orkot package

1.6.9 Cavitation

Cavitation can occur in hydraulic cylinders, causing damage to the seals. The cause of cavitation is the fact that there is always a certain amount of air dissolved in pressurised hydraulic oil. The dissolved air takes the form of liquid molecules, mixes fully with the hydraulic fluid molecules and is not visible. About 6% air is dissolved in the oil at atmospheric pressure. The higher the pressure the more air can dissolve in mineral fluid.

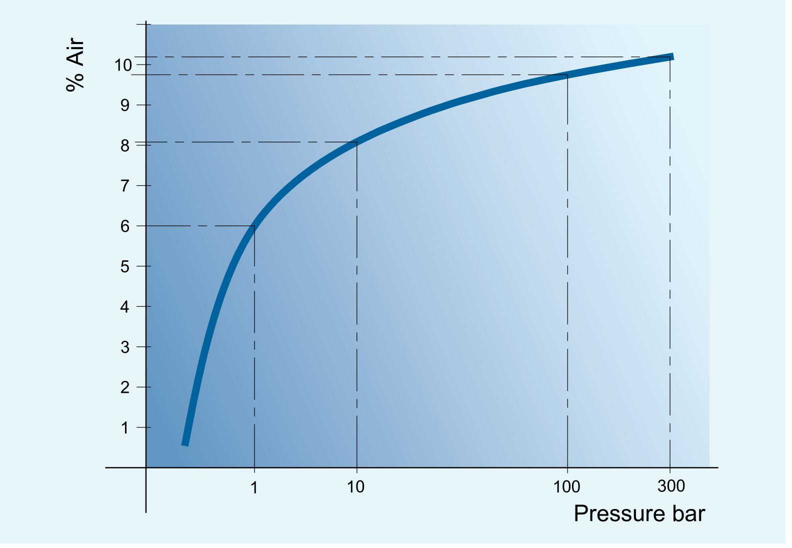


Fig. 1.6.9.A Typical curve for dissolving of air in hydraulic mineral oil

If the fluid is saturated with air at 1 bar and the pressure gets below 1 bar, an imbalance develops between the dissolved air and the fluid. The air will egress quickly in the form of 'free air'. If this free air is subsequently pressurised quickly, it is possible that cavitation occurs, the same as the cavitation occurring in hydraulic pumps, see fig 1.2.B

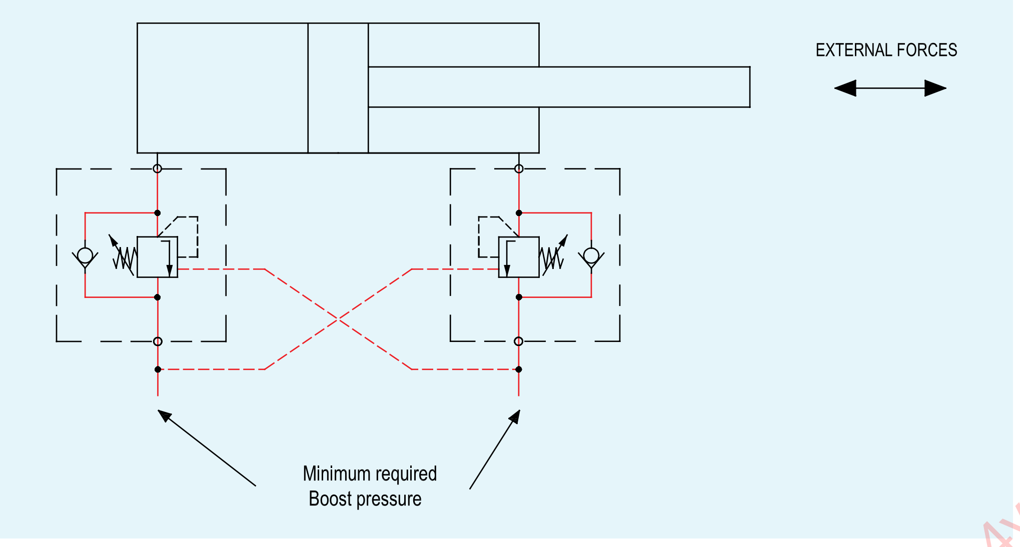


Fig 1.6.9.B Example of a cylinder that is locked in its position with high external load changes, possible causing cavitation damage to the piston seals

In figure 1.6.9.B it is shown how cavitation may occur. Here a cylinder in rest position is blocked by 2 pressure control valves. In this situation the cylinder is subject to large external load variations, i.e. pressure variations, giving a pattern of low pressure (<1 bar) followed by high pressure. The air released at low pressure implodes under the subsequent high pressure. This is also known as the 'Diesel Effect' and is coupled with high local temperatures, up to about 1200 °C.

If there is sufficient boost pressure available of between 5-10 bar in outlet ports of the pressure relief valves, then the low pressure of <1 bar cannot occur. It will then not be possible for the dissolved air to egress.

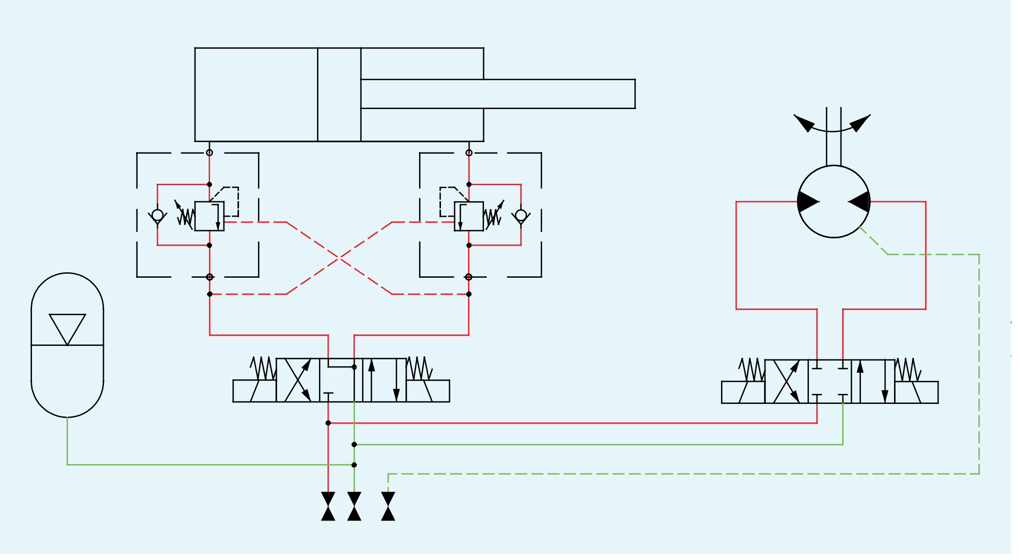


Fig 1.6.10 Pressure rise in a hydraulic system due to high temperatures

If the complete installation warms up due to solar heat, the system temperature may easy rise to a level 45 °C above the level at which it was disconnected and stored. As a result of this temperature increase, the pressure will increase by about 100 bar per 15 °C increase in temperature (See chapter 3.3). The total pressure by such a temperature increase can then rise to about 300 bar.

In the example on the drawing, the seal on the hydraulic motor shaft will fail resulting in an oil leakage or spill. If however, there is no hydraulic motor present in the whole system, then the high pressure can extend throughout the installation, even via seemingly closed middle positions of directional valves, because it is always possible that some internal leakage can occur.

A good technical solution is, as is it shown, to temporarily mount a hydraulic accumulator. This accumulator can collect the expanding oil, preventing the pressure from rising.

You need to be aware that in the somewhat bigger installations in the offshore industry the total volume of hydraulic fluid in the disconnected system can reach high levels of up to 1000 – 3000 litres. The thermal expansion by a temperature increase of 45 °C comes to 30 – 90 litres. In such cases the capacity of the hydraulic accumulator or several hydraulic accumulators will need to be chosen generously.

1.7 Actuators: Hydraulic motors

In this section we will discuss the rotating actuator otherwise known as the hydraulic motor. Where the hydraulic cylinder converts hydraulic power into a linear force and velocity, a hydraulic motors converts hydraulic power into a rotational torque and speed. In fact the pressure in the hydraulic motor is determined by the torque that the motor has to provide and the rotational speed of the motor is determined by the fluid flow that is generated in the hydraulic system.

Fig 1.7 Application of a hydraulic driven winch. Here hydraulic motors have been used that can provide the high torque of the winch without the use of an intermediate gearbox..(Courtesy of Hägglunds)

The working principle of a hydraulic motor corresponds to the working principle of a hydraulic pump. In practice, the details of a hydraulic motor are slightly different. For example, the "timing" of the pressure porting in an axial piston motor is slightly different from the timing in an axial piston pump.

All types of hydraulic motors have these common design features: a driving surface area subject to pressure differential; a way of timing the porting of pressure fluid to the pressure surface to achieve continuous rotation; and a mechanical connection between the surface area and the output shaft.

The ability of the pressure surface to withstand force, the leakage characteristics of each type motor and the efficiency of the method used to link the pressure surface with the output shaft determine the maximum performance of a motor in terms of pressure, flow, torque output, speed , volumetric and mechanical efficiencies, service life and physical configuration.

Motor displacement refers to the volume of fluid required to turn the motor output shaft through one revolution. Displacement of hydraulic motors may be fixed or variable. A fixed displacement motor provides constant torque for a constant pressure. Speed is varied by controlling the amount of input flow into the motor. A variable displacement motor provides variable torque and variable speed. With input flow and/or pressure constant the torque speed ratio can be varied to meet load requirements by varying the displacement.

1.7.1 Motor with fixed displacement or stroke volume

The external gear motor

The gear motor is the most commonly used. The workings are the same as those of the gearwheel pump. The direction of the volume stream motor's rotation is reversed when compared with the pump. In the offshore industry this type of motor is rarely used, due to the relatively low power output that can be achieved.

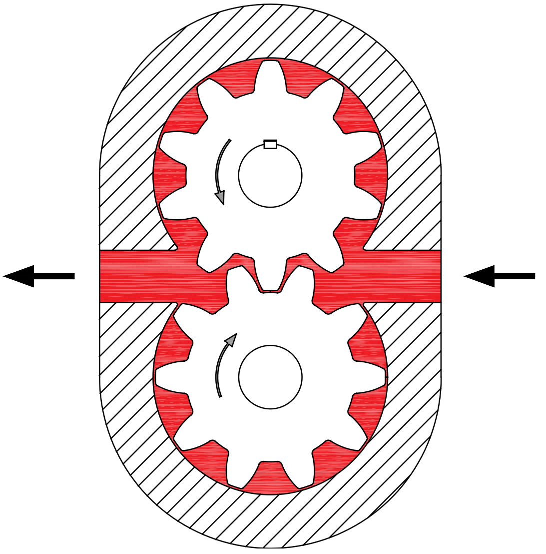


Fig 1.7.1.A Principle of a gear type hydraulic motor

For heavier applications such as mixing machinery or winch drives there is a choice between so called fast running hydraulic motors and high torque motors. The fast running hydraulic motor has a limited stroke volume (up to about 1000 cc/rev) combined with a limited torque but can be driven at high rotational speeds (up to about 4000 rpm). If the machinery that needs to be powered is a winch or a moving machine, a gearbox is fitted between the output shaft of the hydraulic motor and the input shaft of the machinery to reduce the high rotational speed and increase the input torque.

The internal gear or gerotor motor

A gerotor motor consists of an inner-outer gear set and an output shaft, figure 1.7.1.B. The inner gear has one tooth less than the outer. The shape of the teeth is such that all teeth of the inner gear are in contact with some portion of the outer gear at all times. When pressure fluid is introduced in the motor, both gears rotate. The motor has integral kidney-shaped inlet and outlet ports. The centres of rotation of the two gears are separated by a given amount known as the eccentricity. The centre of the inner gear coincides with the centre of the output shaft.

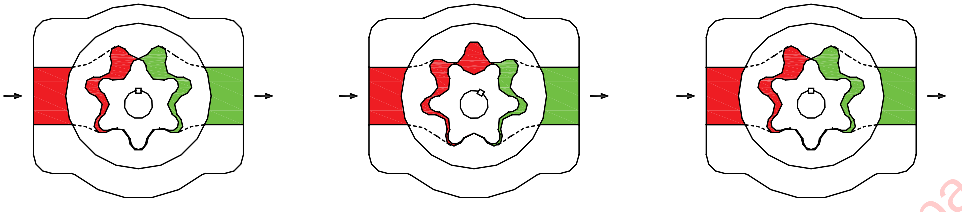


Fig 1.7.1.B Principle of an internal gear or gerotor motor. (Courtesy of Hydraulics and Pneumatics)

The axial piston motor

An example of a fast running hydraulic motor is the bent axis axial piston motor shown in figure 1.7.1.C. In the swash plate pump, the rotating movement of the pump shaft was converted into a hydraulic energy of the axial movement of the pistons. In a hydraulic motor the hydraulic energy of the axial movement of the pistons in the pump housing is converted into a rotating movement of the motor shaft.

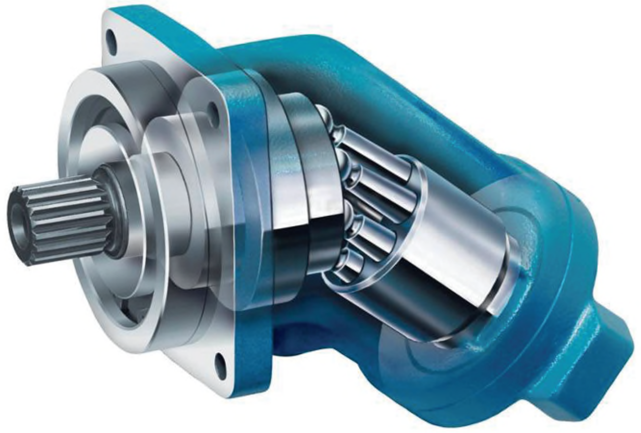
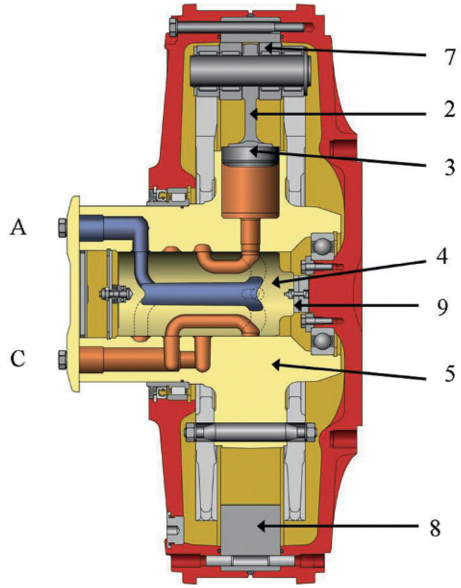


Fig 1.7.1.C Section view of a bent axis axial piston motor type A2FM (Courtesy of Bosch Rexroth)

A high torque motor is characterised by a very large stroke volume (up to about 150,000 cc/rev) but with a much more limited, lower maximum number of revolutions per minute. This type of motor has good running characteristics at low rotational speeds (0-10 rpm). This type of hydraulic motor can be applied as a drive for a winch or moving machinery, without the need to fit a gearbox between the motor and the machinery.



1 Motor case
2 Piston rod
3 Piston
4 Distributor
5 Stationary cylinder block
7 Cam roller
8 Cam ring
9 Safety coupling

Fig 1.7.1.D Section view of a radial piston motor (Courtesy of Hägglunds)

The motor in figure 1.7.1.D is of the radial piston rotating case type. The case is supported on the stationary cylinder block (5) by two main bearings. An even number of radially positioned pistons (3) work in cylinder bores in the cylinder block, which also houses the inlet and outlet ports (A and C). Each piston is coupled by a piston rod (2) to cam rollers (7). The two inner rollers press against the cam ring (8). The cam ring is anchored to the rotating case. The distributor (4) directs the input oil to the pistons during their work strokes and returns the exhausted oil back to the tank. The distributor is coupled to the rotating case via safety coupling (9). The motor can be connected to a driven machine via two mounting surfaces on the rear end of the motor.

1.7.2 Motor with variable stroke volume

In some types of hydraulic motor the stroke volume can be adjusted. Dependent on the mechanical operation of the hydraulic motor, there are a number of different ways in which the stroke volume can be adjusted. For example, for the radial piston motor shown in Fig 1.7.1.C , it is possible to halve the number of working pistons, which will reduce the stroke volume from 100% to 50%.

For a different type of radial piston motor, vane motor or axial piston motor it is possible to vary the stroke volume from 100% to 0%. The detailed documentation of the many different manufacturers will show the different possibilities in more detail.

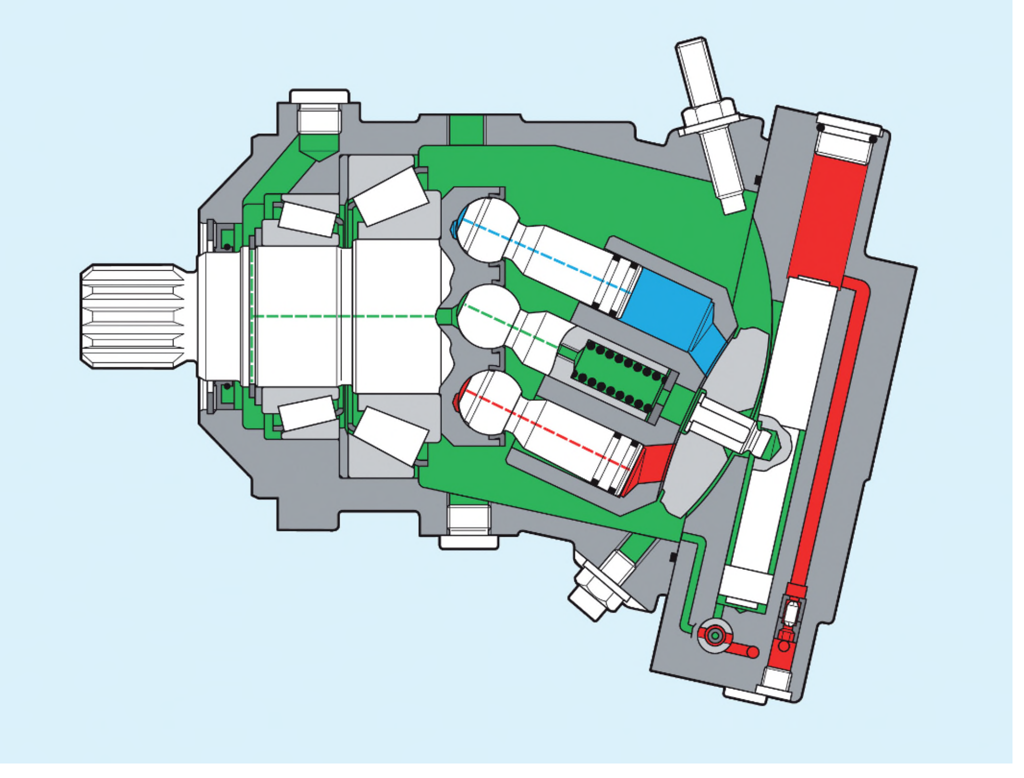


Fig 1.7.2 Section view of a variable axial piston motor type (Courtesy of Bosch Rexroth)

In the axial variable piston motor shown in figure 1.7.2 it is possible to vary stroke volume from 0% - 100%. Just like with the axial piston pump, the tilt angle of the swash plate is varied.

For example, by changing the stroke volume whilst keeping the feed rate volume the same, a different rotational speed is created. Alternatively, a different input pressure is needed to generate a certain external torque.

1.7.3 Formulas for a hydraulic motor

Theoretical Hydraulic Power:

$$P_H = \Delta p \cdot Q \tag{1.45}$$

Δp	=	pressure drop	N/m2	P_H	=	hydraulic Power	Nm/s
Q	=	flow	m³/s				

Applied Formula for Hydraulic Power:

$$P = \frac{\Delta p . Q}{600} \tag{1.46}$$

Δp	=	pressure	bar	P	=	power	kW
Q	=	flow	lpm				

Applied Formula for Mechanical power:

$$P_M = T \cdot n \cdot \frac{2\pi}{60} \tag{1.47}$$

n	=	speed	rpm	P_M	=	mechanical power	kW
T	=	torque	kNm				

Applied Formula for Volume Flow:

$$Q = \frac{V_s \cdot n}{1000 . \eta_{vol}} \tag{1.48}$$

Q	=	flow	lpm	n	=	speed	rpm
V_s	=	displacement volume cc/rev		η_{vol}	=	volumetric efficiency	0,9 - 0,98

Theoretical Torque:

$$T = \Delta p \cdot V_s \cdot \eta_{hm} \tag{1.49}$$

V_s	=	stroke volume	m³/rad	T	=	torque	Nm
(1 cc/rev = 1.592 x 10⁻⁷ m³/rad)				η_{hm}	=	mechanical Efficiency of motor	0,8 - 0,98
Δp	=	pressure drop	Pa				

Applied Formula for Torque:

$$T = 20 . \pi . \Delta p \cdot V_s \cdot \eta_{hm} \tag{1.50}$$

V_s	=	stroke volume	cc/rev	T	=	Torque	Nm
Δp	=	pressure drop	bar				

Total efficiency:

$$\eta_{tot} = \eta_{mh} \cdot \eta_{vol} = \frac{P_M}{P_H} \tag{1.51}$$

On the following page is an overview of the different motors with their characteristics. The total output efficiency mentioned in the last column is dependent on the number of revolutions per minute. For a low number of revolutions per minute or when starting from a static position the output efficiency is usually lower.

Type	Torque	Speed		Max pressure	Displacment volume	Temperature range	Viscosity range	Starting torque	Total efficiency
		Min.	Max.						
	T	nmin	nmax	p	V	Tmin-Tmax	vmin-vmax	ηmech	ηmax
	Nm	min-1	min-1	bar	cm³	°C	mm2/s		
External gearmotor	2-112	200-500	2000-4000	90-210	1,4-80	-15-+80	12-8000	0,5	0,80
Orbitmotor	0,7-25	500	5000	140	8-2000	-20-+80	60-120	0,6	0,82
Internal gearmotor	(0,7) 25-980	10-50	80-1500	140-210	(8) 60-800	-20-+80	60-120		0,85
Vane motor	140-8570	10-20	150-500	140-210	625-2500	-15-+80	30-150	0,8	0,86
Axial piston motor									
With swashplate	17-1000	(2) 200	1000-2000	180-260	20-366	-20-+80	30-220	0,6	0,88
Bent axis type	25-4500	100-150	800-5000	350-400	5-2000	-20-+80	17-80		0,95
Radial piston motor	6380-1,7·105	0,5-2	20-100	170-350	2356-50258	-35-+100	20-150	0,9-0,98	0,98

Table 1.7.3 Features of varies types of hydraulic motor

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Chapter 2

Hydraulic energy control, conductive part

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Hydraulic energy control, conductive part

To get the hydraulic energy generated by the hydraulic pump to the actuator, cylinder or hydraulic motor in a controlled way, more than just pipe work is needed. We call this the conductive part of hydraulic drive system. In this chapter we will discuss the most important basic functions of the functional valves that are applied for this purpose. Detailed information about the valves is available from the often very detailed specification and documentation of the different manufacturers.

P. Albers, *Motion Control in Offshore and Dredging*, DOI 10.1007/978-90-481-8803-1_2,

2.1 Pressure control valves

Pressure control valves are used to regulate the pressure. As far as the construction is concerned, a distinction is made between a spool and a seat construction.

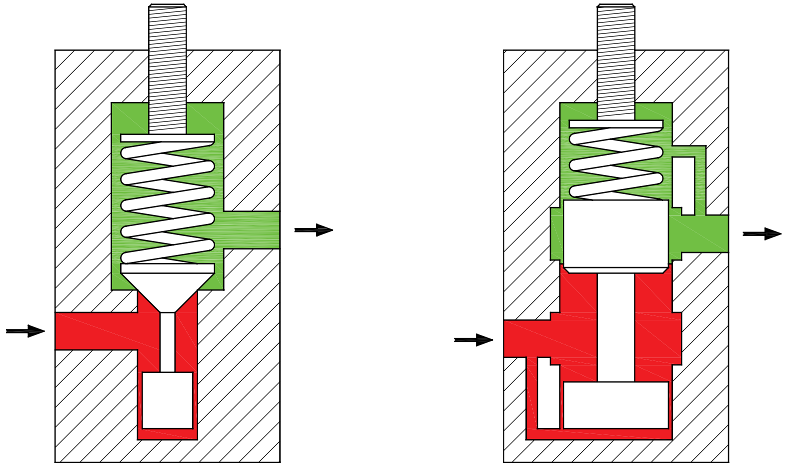


Fig 2.1 Seat type (left) and piston type (right) relief valve

2.1.1 Pressure Relief Valves

A pressure relief valve must limit the system pressure in a hydraulic installation to a maximum value. If the system pressure exceeds this value, then the pressure relief valve opens and lets oil flow from the pressurized inlet side to the return line.
Direct Acting Pressure Relief Valve, see figure 2.1.1.A

The seating valve (3) is pressed into the seat by the spring (2). The inlet pressure *P* presses underneath the pressure valve against the spring pressure. If the pressure rises above the pressure set in the spring, the valve will open to discharge the oil via the outlet port *T*.

The pre-tension in the spring can be set progressively. This type of valve is not suitable for large volumes of fluid. This valve is absolutely leak proof if its inlet pressure is lower than the set value.

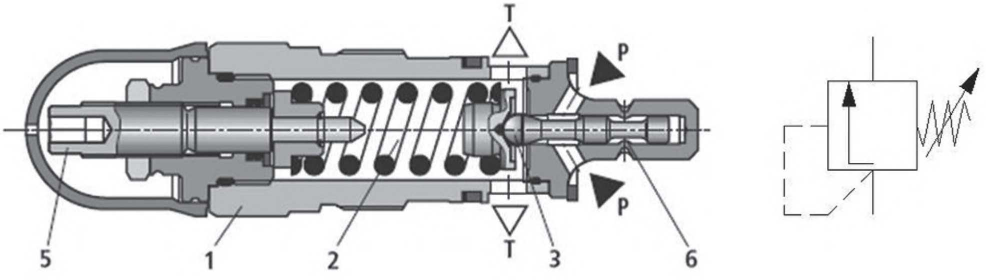


Fig 2.1.1.A Direct acting pressure relief valve with its symbol (Courtesy of Bosch Rexroth)

Pilot Operated Pressure Relief Valve, see figure 2.1.1.B
The pilot operated pressure limiter valve is suitable for large fluid flows, because this valve consists of a main valve and a pilot (indirect) valve.

The main valve consists of the cartridge (3) which is pushed into the seating by a spring. The inlet pressure Π works on the underside of the main valve. At the same time it works on the topside of the main valve, but this time via the choke (7). The main valve remains closed, because of the balanced forces working on it. However, when the inlet pressure Π rises above the level set in the pilot valve, the pilot valve (8) will open, thus lowering the pressure at the top side of the main valve. The main valve will open because the forces working on it are no longer in balance. Due to the design of the valve, a large flow capacity is possible.

This type of valve is not leak proof due to leakage via the pilot section to the spring chamber and via the outer circumference of the main piston to the return line.

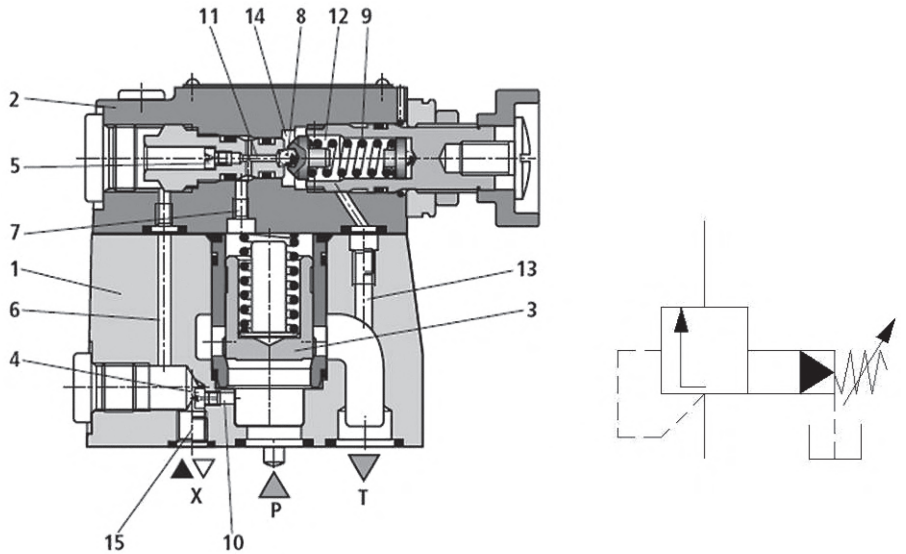


Fig 2.1.1.B *Pilot operated pressure control valve with its symbol (Courtesy of Bosch Rexroth)*

2.1.2 Pressure reducing valve

A pressure reducing valve , figure 2.1.2, is used to reduce the pressure in (part of) the system on the exhaust side of the valve to a certain, pre-determined, value. Here too we have so-called direct acting and pilot operated valves. As before, the direct acting valves have a limited flow capacity.

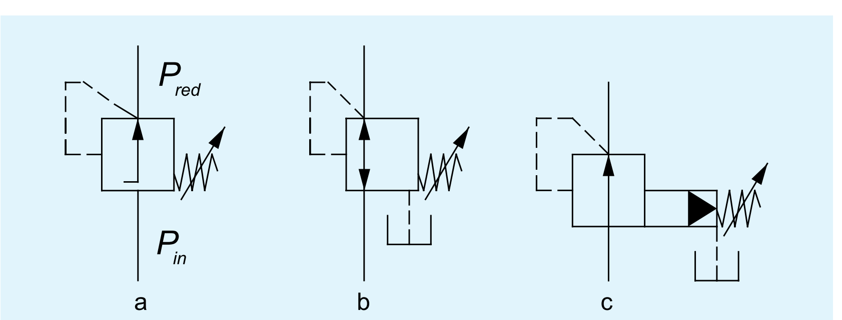


Fig 2.1.2 *Pressure reducing valves: a) direct acting pressure reducing valve, b) relieving pressure reducing valve and c) pilot operated pressure reducing valve*

A relieving pressure reducing valve is, for example, used when the pressure in a cylinder rises above the set value P_{red} of the pressure reduction valve as the piston is pushed into the cylinder. The higher pressure can be reduced through the valve via the correction outlet (often the T- or Leak oil pipe).

2.1.3 Pressure sequence valve

A pressure sequence valve opens when the inlet pressure rises above the value set for the valve. When this happens, the valve opens completely and connects the inlet side with the exhaust/outlet side, which means that the circuit on the exhaust side will be at the same pressure as the circuit on the inlet side of the valve.

In the example as shown in figure 2.1.3. the left cylinder will first move out and, when the pressure in that cylinder rises above the set value of the sequence valve, the sequence valve will open and the right cylinder will move out.

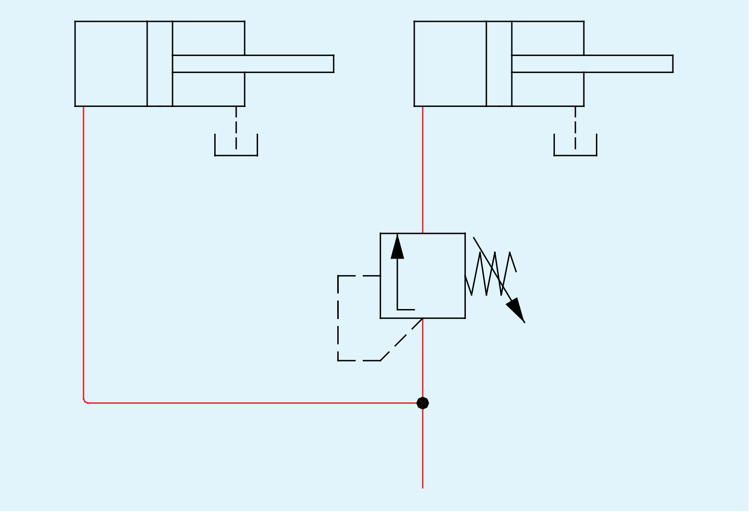


Fig 2.1.3 A pressure sequence valve to control a second cylinder

For these valves too there is a choice between direct working and piloted valves. The flow capacity of the direct working valves is again limited.

2.1.4 Brake (counter balance) valve

A variant on the sequence valve is the brake or counter balance valve. Two names for valves with nearly the same function but with different features. In paragraph 7.1 and 7.2 details are given for the function and features of the counter balance valve and the braking valve. With both valves it is possible to keep an actuator under control where an external load is applied and the actuator is driven by that load, as in the shown situation in figure 2.1.4. These valves have an external pilot control line to open the valve when the actuator has to be driven , in this case the cylinder has to be retracted.

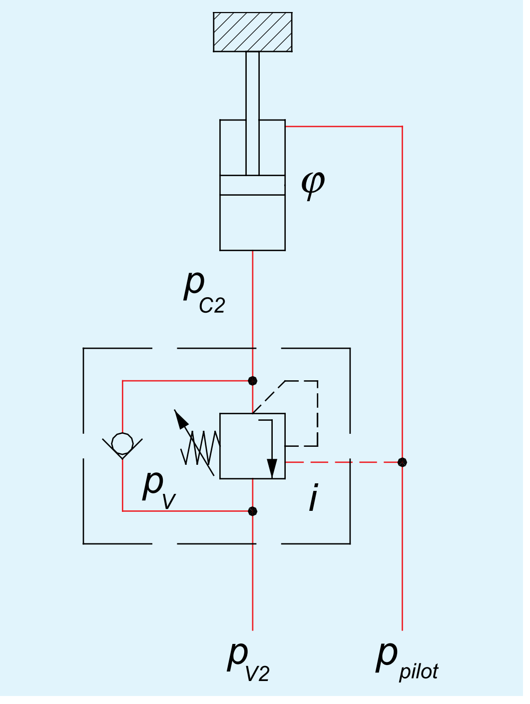


Fig 2.1.4 Function of the counter balance and brake valves with a cylinder.

With the cylinder vertical mounted the weight of the mass initiates a pressure p_L at the bottom area. The counterbalance features a relief function. The setting of this relief valve should be at a value of 130% of the maximum induced pressure p_L. If a pressure is applied at the annular end port two effects will be noticed. At first the pressure at the bottom area will increase due to the force balance on the cylinder piston. The second effect is that the counterbalance valve is gradually opened by this pressure at the annular end via the pilot pressure line.
The equation for the static pressures and dynamic behavior with the use of a counterbalance valve is in detail explained in equations 7.5. The result of the equations for the static behavior is given by:

$$p_{C2} = p_L + \left(p_V + p_{V2} - p_l\right) \cdot \frac{\varphi}{i+\varphi} \tag{2.1}$$

and:

$$p_p = \frac{p_V + p_{V2} - p_L}{i+\varphi} \tag{2.2}$$

From both equations we find that the pressure p_{C2} and the pilot pressure p_{pilot} are sensitive for back pressure of the valve p_{V2}. Optional valves are available that are non-sensitive to this back pressure. This is than achieved by venting the chamber of the main relief mechanical spring.

2.2 Directional valves

With directional valves it is possible to regulate the start, stop and direction of the fluid flow.
There are two construction groups, identical to the basic types of pressure regulator valves:

- Seating valves, which are more or less leak free. The necessary force required for switching is large because the valve has to be opened against the pressure in the spring.
- Spool valves, where a spool is controlled in different positions by springs and hydraulic pressure.

Because of the small clearance between the spool and the housing, there is always some leakage from blocked ports to other ports. In the example of the 4/3 valve as shown below leakage occurs from the pressure inlet port P to the working ports A and to B, but also from the inlet port P to the tank line port T and from the working ports A and B to the tank line port T.

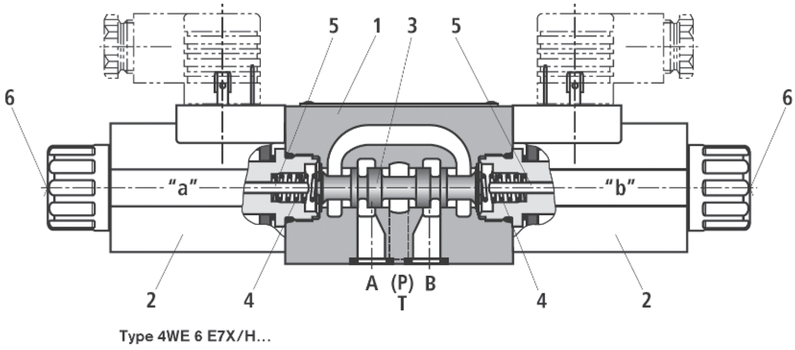


Fig 2.2.A Directly operated directional spool valve (Courtesy of Bosch Rexroth)

In this valve are two magnets a and b that can move the spool to the left or to the right. That way the central connection P is connected with either port A or port B. At the same time port A or B might be connected with port T. The size of the valves is specified according to a CETOP norm in sizes: 03, 05, 07, 08 and 10. The mounting sizes of the valves are also all standardised, which makes it possible to exchange a valve of a certain size and make with one of the same size but a different make.
The naming of the pilot valves is made up of the number of working ports (control ports not included) followed by the number of control positions.
Three examples are given in figure 2.2.B

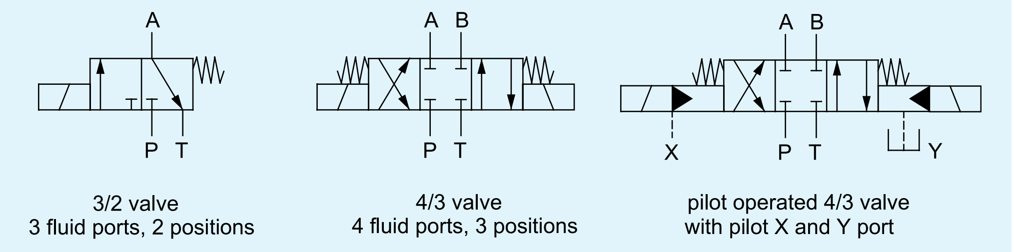


Fig 2.2.B Naming of the directional control valves, depending on number of fluid ports and control positions. P = pressure port, T = tank port, A and B = working ports and X and Y are pilot ports

The idle position is the position of the valve when no controls are applied to it. The fluid ports are indicated in this state. In the above examples a mechanical spring controls the spool to idle position. The working ports A and B and P and T port are designated by capital letters. For a pilot operated directional valves pilot ports X and Y are added. X Is the control port with a certain pilot pressure and Y is the port that is always connected to a low pressure line e.g. the tank line or a separate drain line. From the idle position of the valve it is possible to reach several different switch symbols, see figure 2.2.C. , representing the most used basic symbols. The three different symbols can be achieved by mounting different spools in the valve body.

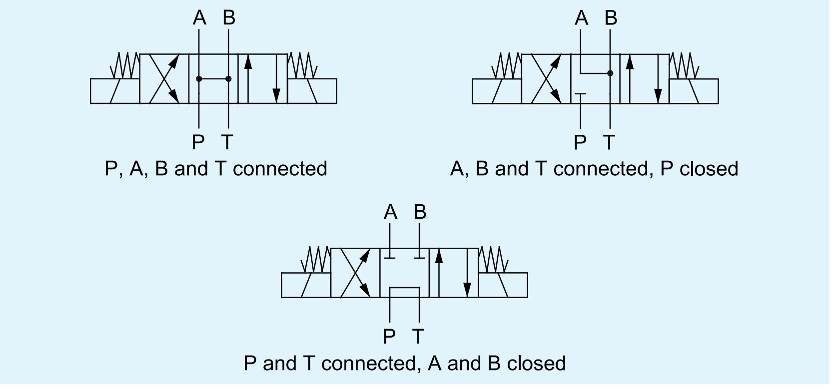


Fig 2.2.C Different flow paths in a directional valve with different spool types

The maximum flow through a valve of a certain size is limited. For a CETOP 3 valve this limit is about 30 lpm. The flow through the valve generates a force on the spool that is opposite to the operating force of the magnet. The flow forces intend to move the spool to the central position. At the same time the pressure drop across the spool increases, for example between P and A, at a rate relative to the square of the flow rate.

The maximum working pressure for currently available directional valves is about 350 bar.

The limitations on the flow rate can be largely overcome by fitting a so-called pilot valve. In that case a directly controlled CETOP 3 valve is mounted directly onto a large main valve. The connections between ports A and B and the pilot valve are used to move the much larger control spool in the main valve from left to right. This significantly increases the maximum flow allowed. For a CETOP 7 valve it goes up to about 150 lpm, for a CETOP 10 valve up to about 800 lpm.

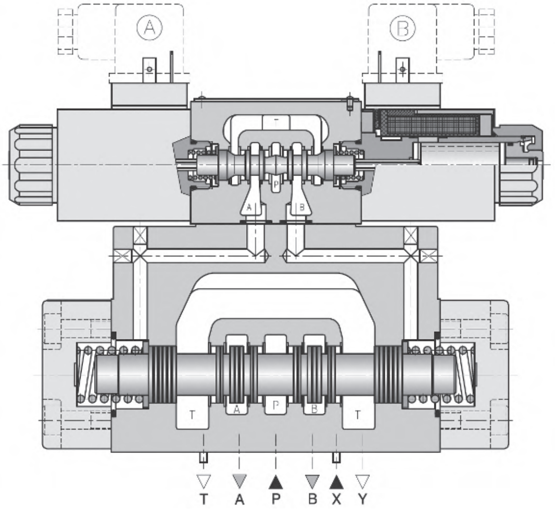
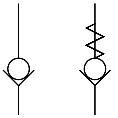


Fig 2.2.D Pilot operated directional control valve (Courtesy of Argo-Hytos)

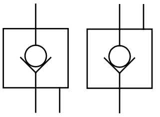
In the pilot controlled valve the pilot valve uses the pressure from port P or from port X to regulate the position of the main spool. If there isn't pressure on port P in all situations, for example if port P is connected to port T in the idle position , then it will be possible to use the external control pressure from port X to switch the valve.

2.3 Flow valves

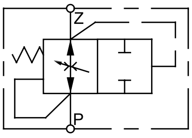
2.3.1 Non-return (check) valves



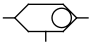
The function of check valves is to let oil flow through the valve in one direction only. There are unloaded and spring loaded valves.



A controlled check valve can be opened against the flow or closed with the flow by a separate pilot control line. In a number of cases the pilot check valves are fitted with an external oil leak connection.

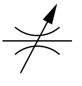


In a hose burst protection valve, the flow through the valve from Z to P can only take place during a controlled movement of the cylinder. When the flow is too large, the valve closes, stopping the flow.

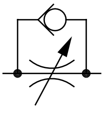


A shuttle valve has two inlet ports and one outlet port. The inlet port with the highest pressure allows flow to the exit port whilst, at the same time, closing the other inlet port. This valve is used to pass on the pressure of two inlet ports to, for example, the controls of a load sense pump.

2.3.2 Throttle valves

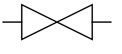


In a variable throttle valve the opening area can be varied step less. The volume flow is amongst other parameters dependent on the pressure drop across the valve, see for details paragraph 2.4.5.



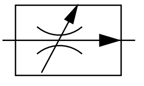
To make sure that the speed of an actuator is restricted only in one direction a check valve is fitted. This combination is also known as a speed control valve or throttle and check valve.

Shut-off (isolating) Valve



A shut-off valve is used to close a hydraulic pipe, not to choke it. The symbol shows the valve in open position. When the valve is closed the symbol is drawn solid.

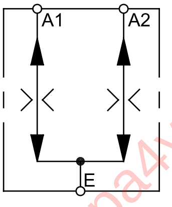
Flow control valve



The volume flow is independent from the pressure difference across he valve, which means that the flow at the outlet point stays constant , even if the pressure at the inlet or outlet points changes. For the valve to work properly a minimum pressure drop of 7-14 bar is necessary across the valve.

There are also flow control valves with temperature compensation.

Flow distribution valve



A flow distribution valve splits the volume flow into equal proportions. The volume flows stay equal, not dependent on the outlet pressures. The pressure difference across the restrictors is kept the same, which ensures that the flow volumes remain the same too. Both outlet flows are never exactly the same. Differences of 3-10% are normal.
This valve can also work in the opposite direction, in the sense that two equal flows can be accepted.

2.4 Proportional and servo valves

2.4.1 General

The varying of the volume flow to/from a hydraulic cylinder is called proportional flow control. The term control is actually wrong. No feedback of the achieved flow speed takes place. This means that this is not a case of control in line with 'control theory'.

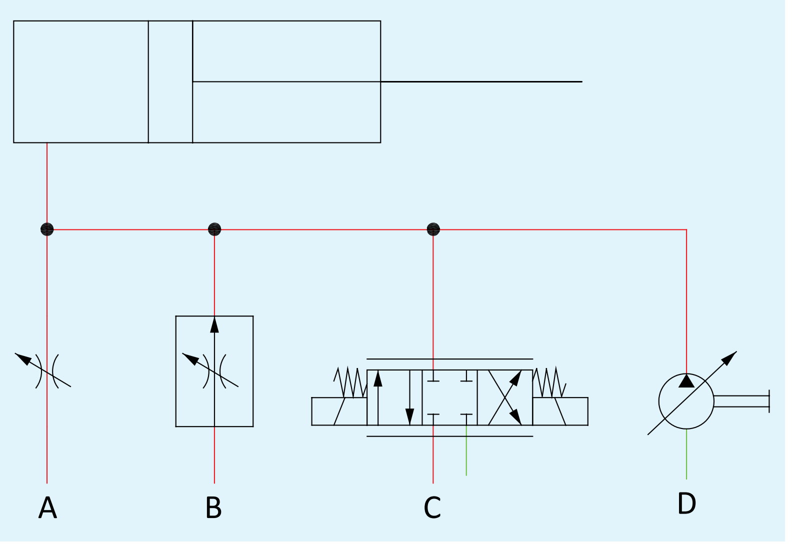


Fig 2.4.1 Proportional flow control in different ways

A proportional volume control can be achieved in several different ways. Figure 2.4.1 displays a number of options. The simplest one is the variable throttle (A), followed by the 2-way flow regulator (B), the proportional valve (C) or the variable pump (D).

2.4.2 Proportional controls

If a proportional valve is used to control a cylinder then two variable choke control valves are actually used (see figure 2.4.2.A). For example, the valve is controlled and a variable choking between inlet port P and outlet port A develops. In that case a variable choke develops from port B to port T where the passage size from P to A is equal to the passage size from B tot T.

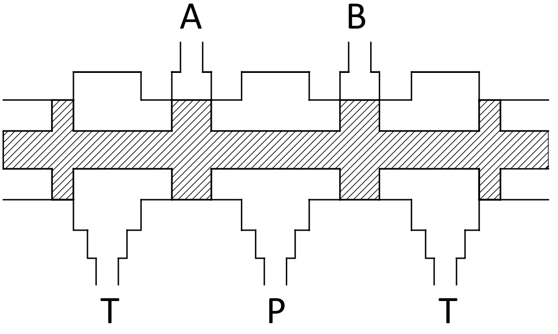


Fig 2.4.2.A 2 Chokes are always active in a proportional valve

In figure 2.4.2.B both the cylinder surface and thus the input and output volume flow are the same. Because the passage size of choke 1 is equal to choke 2 and because the volume flow Q_A is equal to volume flow Q_B the pressure drop ΔP_1 must be the same as the pressure drop ΔP_2.
* There are also designs with a volume flow ratio of 2:1 for use with a differential cylinder (φ=2), see paragraph 2.4.5

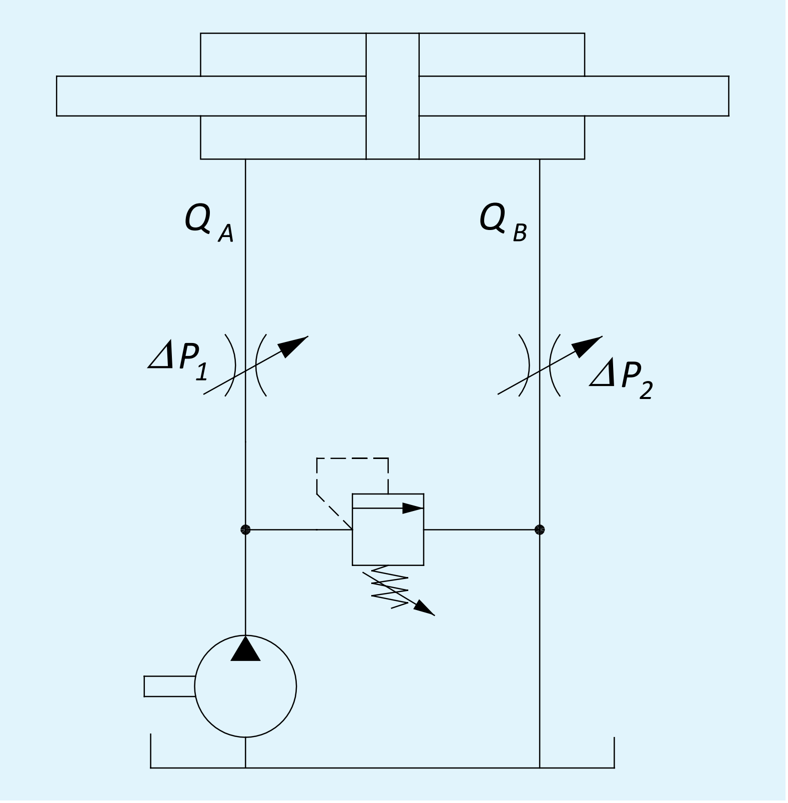


Fig 2.4.2.B A proportional valve is presented by two variable chokes

The pressure drops ΔP_1 or ΔP_2 are dependent on the bores of chokes 1 and 2. Because we are dealing with a proportional valve, the choke bore is variable. The maximum bore size is dependent on the CETOP size of the valve. The volume flow capacity of a valve is expressed in the nominal volume flow Q_n. This is the volume flow in lpm which will occur from P to A or from B to T with a 'standard' pressure drop across a full opened choke of $\Delta P = 5$ bar. In the manufacturer's documentation the pressure drop is defined by a total pressure drop across the valve of 10 bar. In this case however, they take the pressure drop from P to A + the pressure drop from B to T = 5 + 5 = 10 bar.

A proportional valve can also be used to control a drive system. In order to achieve control, a feedback takes place of a position, a speed or a force. In these cases a higher accuracy level is demanded for the valves. In the air and space industries very accurate valves have been developed for this purpose with the special name of 'servo valve'. A servo valve is in effect also proportional. Historically the standard volume flows for a servo valve were and are set at a pressure drop of 35 bar across the choke. This gives a total of 70 bar across the inlet and outlet choke. Accuracy wise and dynamic, a large difference exist between a standard proportional valve and a servo valve. Valves have now been developed which, accuracy wise, fill the gap between the proportional valve and the servo valve.

2.4.3 Higher pressure drop across the valve ports

The pressure drop across the valve can however be many times larger than the '5 bar' or '35 bar' mentioned earlier. In the example shown in figure 2.4.3.A the external load on the cylinder is *F*. This force requires a pressure drop across the cylinder of, for example, 50 bar. If the pressure in the limiter valve is set to 100 bar then 50 bar is 'left' for the proportional valve. This leaves 25 bar for each of the variable ports of the proportional valve.

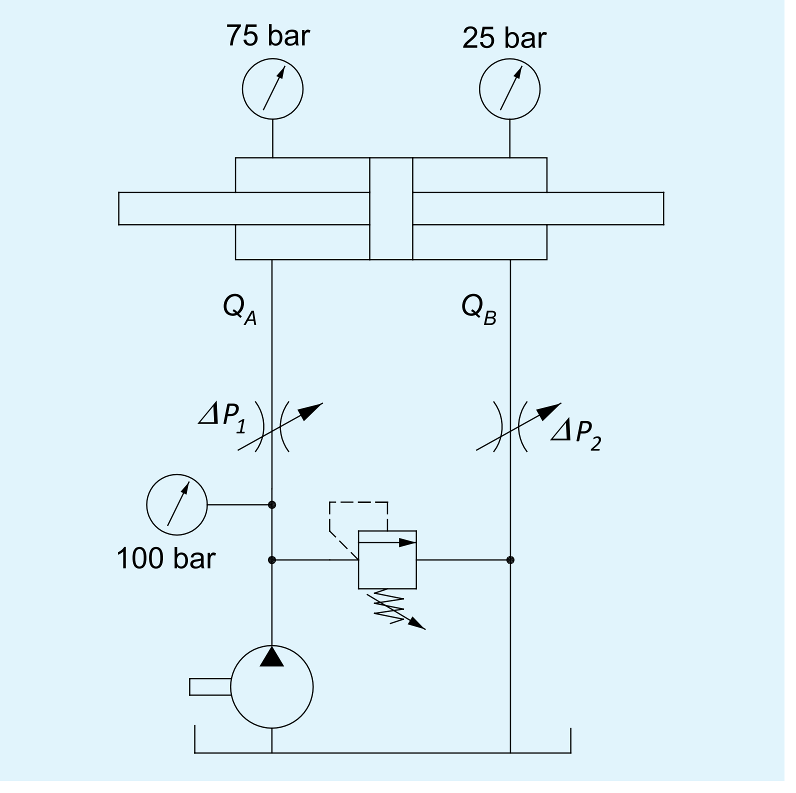


Fig 2.4.3.A Pressures in the circuit when using a proportional valve

The actual volume flow Q_{act} when the valve is fully open is then larger, as per the formula. This means that the pump needs to have sufficient capacity to supply this volume flow.

$$Q_{act} = Q_n \times \sqrt{\frac{\Delta P_{1,2}}{5bar}} \tag{2.3}$$

Where

Q_n = nominal flow	Q_{act} = actual flow.

The flow through the valve for a certain pressure drop relative to the input signal to the valve is called the "volume amplifier". Most manufacturers provide these graphs in their documentation , see figure 2.4.3.B. The diagram on the left is for a proportional valve, where the standard pressure drop of 5 bar across the choke is used. The diagram on the right is for a servo valve, where the standard pressure drop of 35 bar across the choke is used.

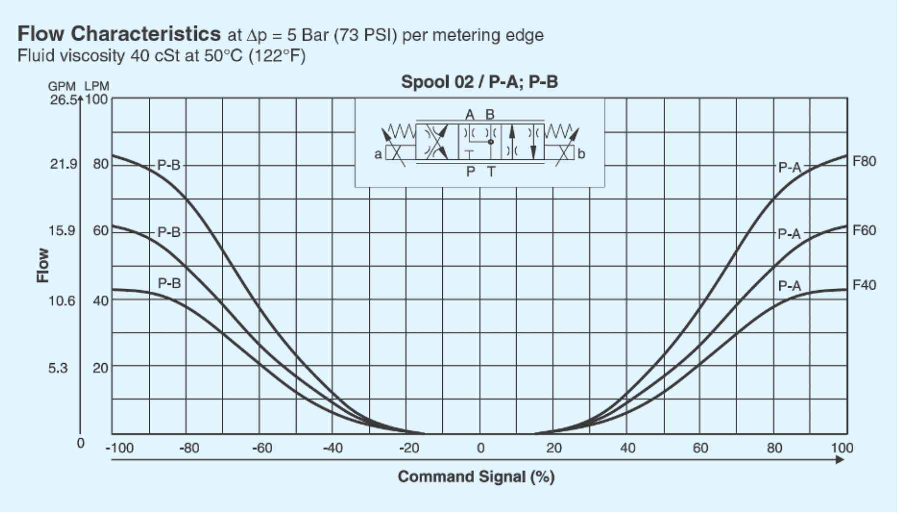


Fig 2.4.3.B *Flow of proportional valves as function of command value for different spool types and pressure drops (Courtesy of Parker)*

2.4.4 Performance curve for the proportional valve

The flow through a proportional or a servo valve is limited by the force of the magnets that control the valve or, in the case of a pilot valve, the force in the pilot. A large volume stream requires a larger piston force. The manufacturer indicates the volumetric capacity in a so-called "Performance Curve". Figure 2.4.4 shows an example of this type of graphs. The graphs for type F40, F60 and F80 are for different types of spools in the valve. The valve will be able, for a certain inlet pressure at the P port, to deliver a certain volume flow as long as the point of operation is at the left side of the graph line.

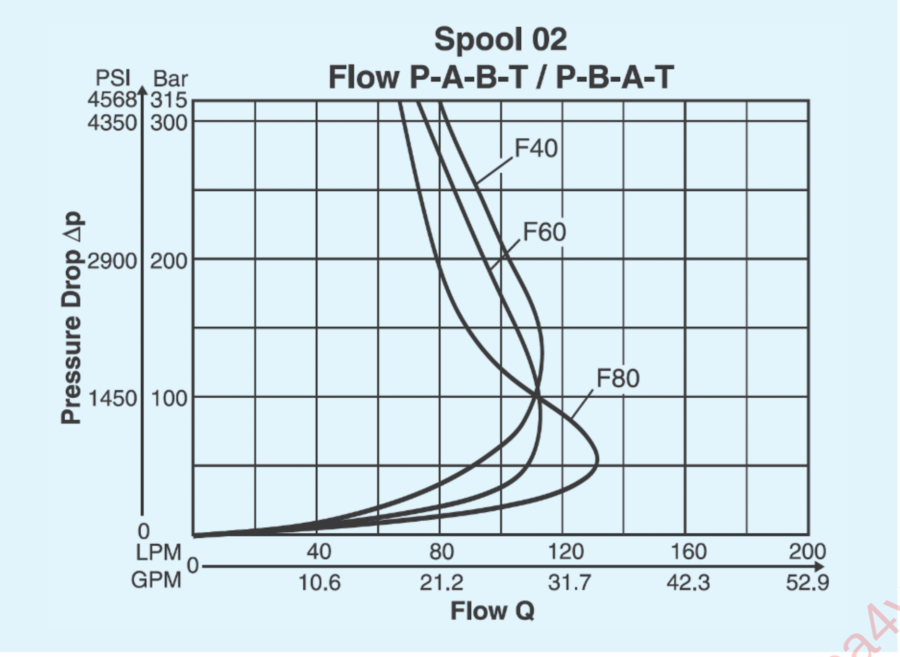


Fig 2.4.4 *Performance curve for a proportional valve (Courtesy of Parker)*

2.4.5 The asymmetrical spool

The pressure drop across variable chokes of proportional valves mentioned in earlier paragraphs are, to a large extent, determined by the volume flow through a valve. The volume flow and thus also the pressure drop for a cylinder with different piston surfaces can increase considerably.
For large volume flows this large drop in pressure also means a large loss of energy. Energy losses in a hydraulic installation are always converted directly into heat. The temperature of this type of installation can therefore rise quickly unless a sufficiently large oil cooler has been installed.
To get around this problem, a so-called asymmetrical spool is installed. For this type of spool, the bore for port P to A or port A to T is always twice the size of port P to B or port B to T, Such spools are therefore only applied for larger ports (with flow a capacity from 80 lpm up to 600 lpm).

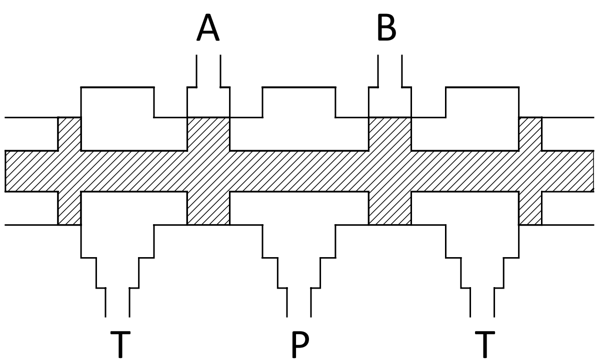


Fig 2.4.5.A Asymmetrical spool with larger port size P to A and A to T

The pressure drop across a choke is given by:

$$Q = c \cdot A \cdot \sqrt{\frac{2 \cdot \Delta P}{\rho}} \tag{2.4}$$

or

$$\Delta P = \frac{\rho}{2} \cdot \left(\frac{Q}{c \cdot A} \right)^2 \tag{2.5}$$

where

A	=	surface area of the port.

From this last formula it can clearly be seen that the pressure drop across a port increases relative to the square of the flow rate going through it. We can assume that the flow to/from the bottom side of the cylinder is twice that to/from the rod side of the cylinder. In that case, if the surface area of the choke at the bottom end always twice that of the one at the rod end is, then it is clear that the pressure drop across both ports will be the same.
Comment: The surface ratio ϕ for a cylinder is hardly ever exactly 2. For deviations from that ratio it may still be advisable to use an asymmetrical spool. The pressure drop across the ports for an asymmetrical spool with a surface ratio of 2 and a cylinder with a surface ratio of ϕ, the following pressure drops can be calculated across the ports.

$$\frac{\Delta P_2}{\Delta P_2} = \frac{4}{\varphi^2} \tag{2.6}$$

Where

ΔP_1 = pressure drop for the bottom side	ΔP_2 = pressure drop for the rod side.

2.4.6 Slowing down of a load

Proportional valves are often used to slowly accelerate and decelerate a load. When a load is being decelerated, be it with a hydraulic motor, or with a cylinder, a significantly different characteristic is important. This is easiest explained through the drive of a hydraulic motor, as shown in the diagram below.

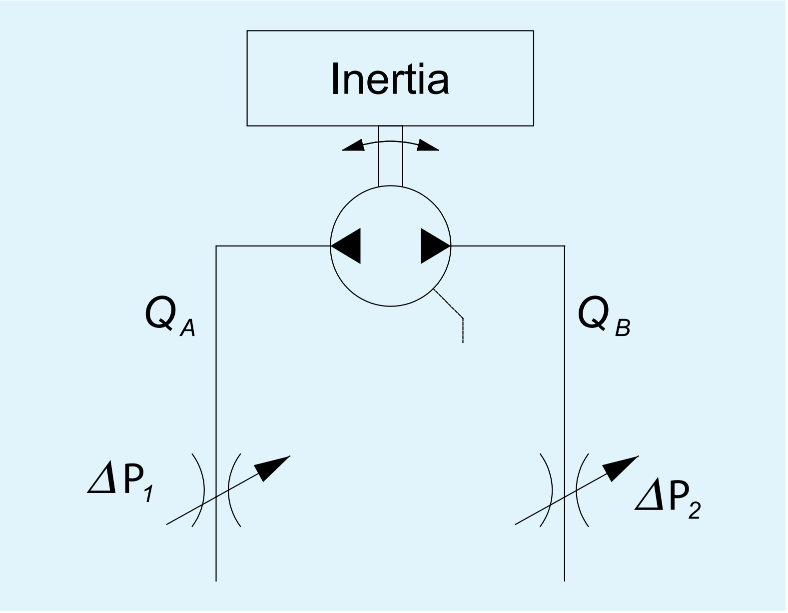


Fig 2.4.6 Deceleration of a hydraulic motor

During braking, a brake torque is required at the hydraulic motor. This can cause a high a high breaking pressure at the outlet side of the hydraulic motor, in this case a high value for ΔP_2. This is no problem for a proportional valve. The pressure drop across the outlet port can easily be generated by moving the control spool of the valve slowly towards the closed, central position. Do remember though that, apart from a small loss due to leakage, the flow rate through the inlet port of a hydraulic motor is always equals the flow rate through the outlet port. The pressure drop ΔP_1 across the inlet port must therefore always be the same as the pressure drop ΔP_2 over the outlet port. During the deceleration of this hydraulic motor there must be sufficient pressure on the inlet side of the inlet port to achieve the pressure drop ΔP_1. If the inlet pressure is not sufficient then "negative" pressure will occur at the inlet side, causing cavitation. A possible consequence of cavitation is mechanical damage to the hydraulic motor.

The same possibility of cavitation also occurs when a cylinder is used. There the pressure drop over the inlet port, as a result of the surface ratio of the cylinder, can easily be a factor higher than the pressure drop over the outlet port. In that situation is even more important that there is sufficient feed pressure at the inlet port of the proportional valve during braking.

2.4.7 2-Way and 3-Way pressure compensation, loadsensing

The volume flow through one of the ports of a proportional valve is amongst other things dependent on the pressure drop across the port:

$$Q = c \times A \times \sqrt{\frac{2 \times \Delta P}{\rho}} \tag{2.7}$$

If it is possible to keep the pressure drop across a port constant, then the flow Q is directly proportional to the size of the valve opening A. This constant pressure drop can be achieved by adding a so-called two-way compensator.

Such a compensator consists of a pressure control valve that is brought to a pressure balance by the hydraulic pressure on the inlet side of the proportional valve, by a mechanical spring and by the hydraulic pressure behind the proportional valve. In most cases the spring equals a pressure drop across the valve of 8 bar. A shuttle valve is used to sense the load pressure in the A or the B line. When the pressure balance is disturbed, then the pressure control valve will change position to the point where the balance has automatically restored itself.

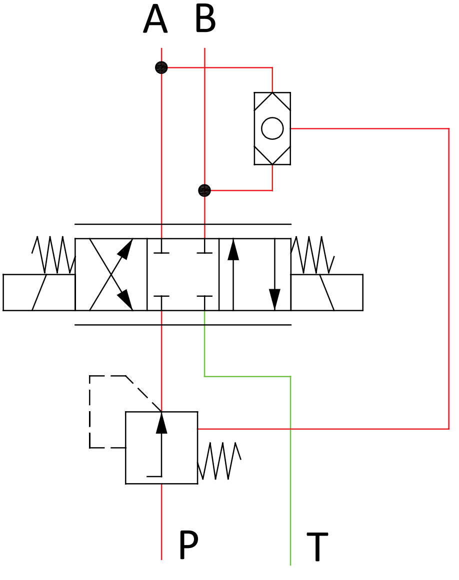


Fig 2.4.7.A A two-way pressure compensator in combination with a proportional valve

With the use of a two way pressure compensator we get:

$$P = P_A + 8\,bar \tag{2.8}$$

or

$$P = P_B + 8\,bar \tag{2.9}$$

Imagine that the pressure on the exit A or B port of the variable proportional valve rises due to for example a higher load on a hydraulic motor. This higher pressure immediately guides the pressure control valve in a direction that will open the pressure control valve further, giving a larger opening between the inlet pressure P of the two-way compensator and the inlet side of the variable proportional valve. The mechanical spring of 8 bar determines the ongoing pressure difference between the inlet and outlet side of the variable choke of the proportional valve. The pressure drop across the two-way control valve itself varies between a minimum of about 8 bar up to almost the maximum value of the feed pressure (often a constant pump pressure). This means that the two-way pressure regulator can also cause large energy losses. Several two-way pressure regulators with proportional valves can be connected in parallel for multiple actuators.

The disadvantage of a two-way pressure compensator is that the drive tends to behave in an unstable way more easily (due to oscillating movements).

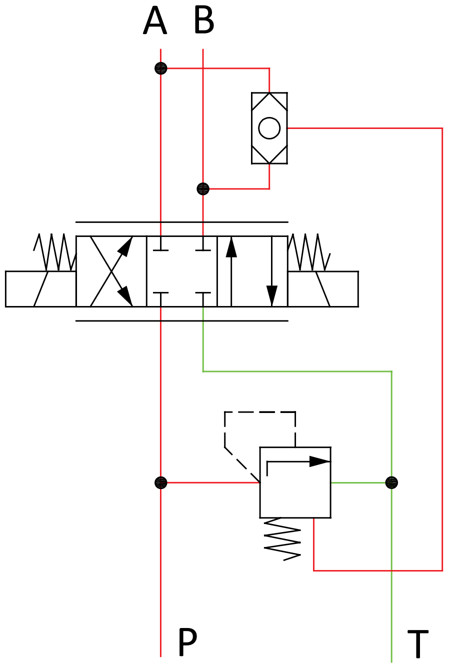


Fig 2.4.7.B A three-way pressure compensator in combination with a proportional valve

An often used variant to the two-way compensator is the so-called three-way compensator. In this case the pressure control valve also regulates the pressure at the inlet side of the valve to such a value that the pressure drop across the inlet port to the A or the B side remains constant. The surplus oil is now directed towards the return line, giving a pressure drop across the three-way pressure control valve equal to the maximum load pressure plus about 8 bar. That way, the three-way pressure regulator causes considerably less energy loss compared with the two-way pressure regulator.

Be careful to note that the most common pressure settings for two- and three-way compensators are 8-10 bar. In the meantime, the nominal flow rate through a proportional valve is specified for a pressure drop of 5 bar across the port. This means that when a two-way compensator is fitted, the volume flow will be a factor of $8/5 = 1{,}6$ times larger.

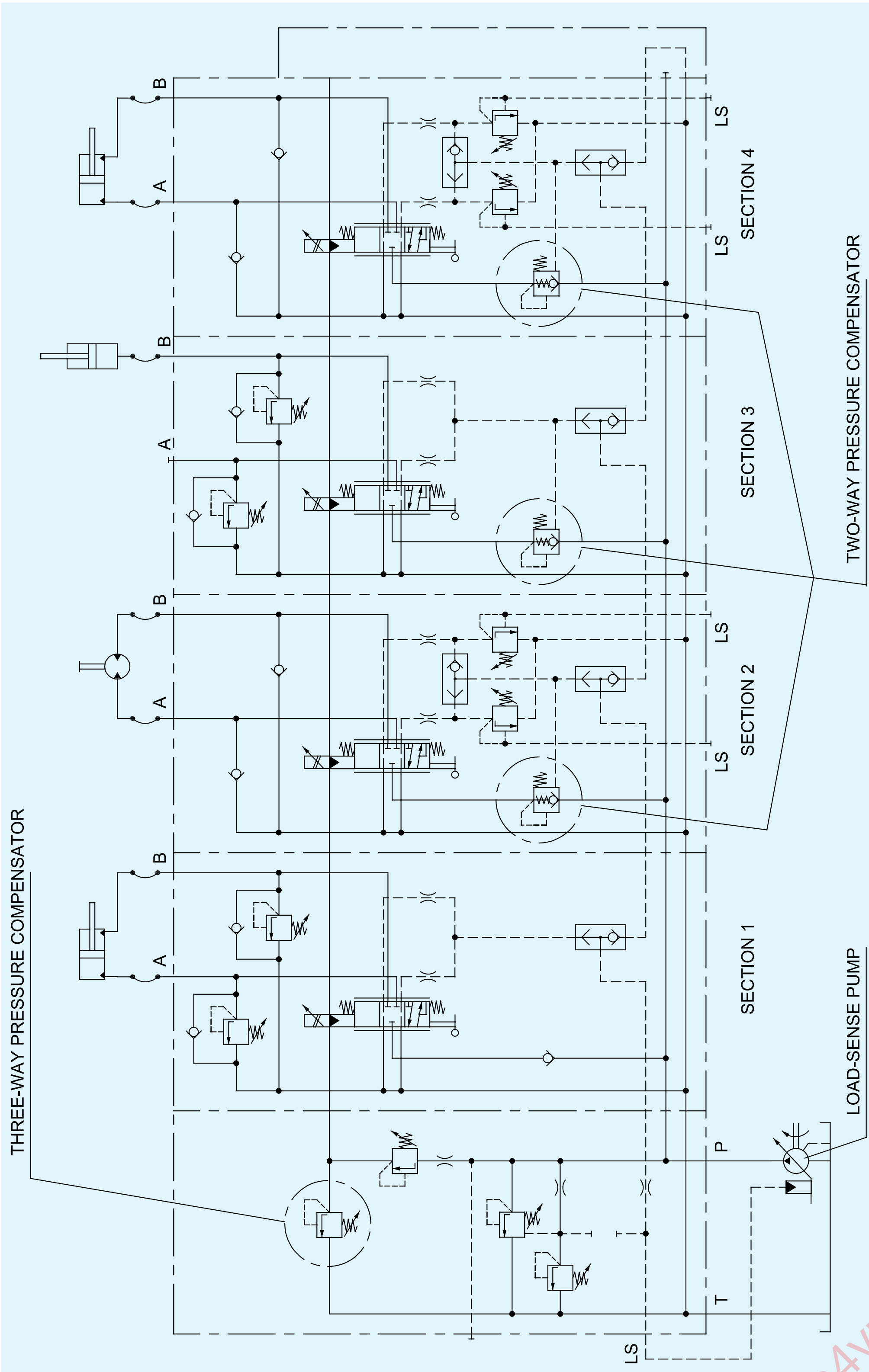


Fig 2.4.7.C A multi proportional valve as used in mobile, shipbuilding and offshore industry

Manufacturers of so-called multi body valves (valves with their sections fitted together) provide excellent examples of the design of a two-way pressure compensator. The diagram in figure 2.4.7.C shows the hydraulic schema for a multi body valve with four different proportional valves. The first section does not have a two-way compensator, the next three sections do. The second and fourth section have additionally been fitted with extra pressure control valves. This way it is possible to set the maximum secondary pressure (= the pressure after the proportional choke). When this pressure is reached, the two-way compensator closes altogether. Then it will no longer be possible for the secondary pressure to increase further. That limits the maximum pressure in the port after the proportional choke.

This example also includes, as standard, a 3-way pressure regulator. This regulator is applied when a pump with constant output is used. The surplus oil is then discharged to the tank against the highest load pressure present.

It is also possible to include a so-called load sensing control in this drawing. The highest load pressure is brought to the adjustable pump via the LS-port. The load-sensing regulator on the pump (you will need to order this specially) delivers a pressure to the P-port which is equal to the highest load pressure present, increased by about 25 bar (this is the minimum pressure drop at which the load sensing regulator can operate).

The advantage of the load sensing regulator is clear. The pump only delivers the minimum amount of energy required for the system. This also means that no discharge of surplus oil, via the pressure relief valve, to the tank takes place.

One possible disadvantage of the load sensing regulator is the risk of unstable pressure, especially with driven loads that have a low natural frequency. Another disadvantage is the relatively high minimum pressure drop of 25 bar that remains. A new development is the application of a fully electronic version of the load sensing regulator. Both the stability of the control circuit and the remaining pressure drop can be improved significantly.

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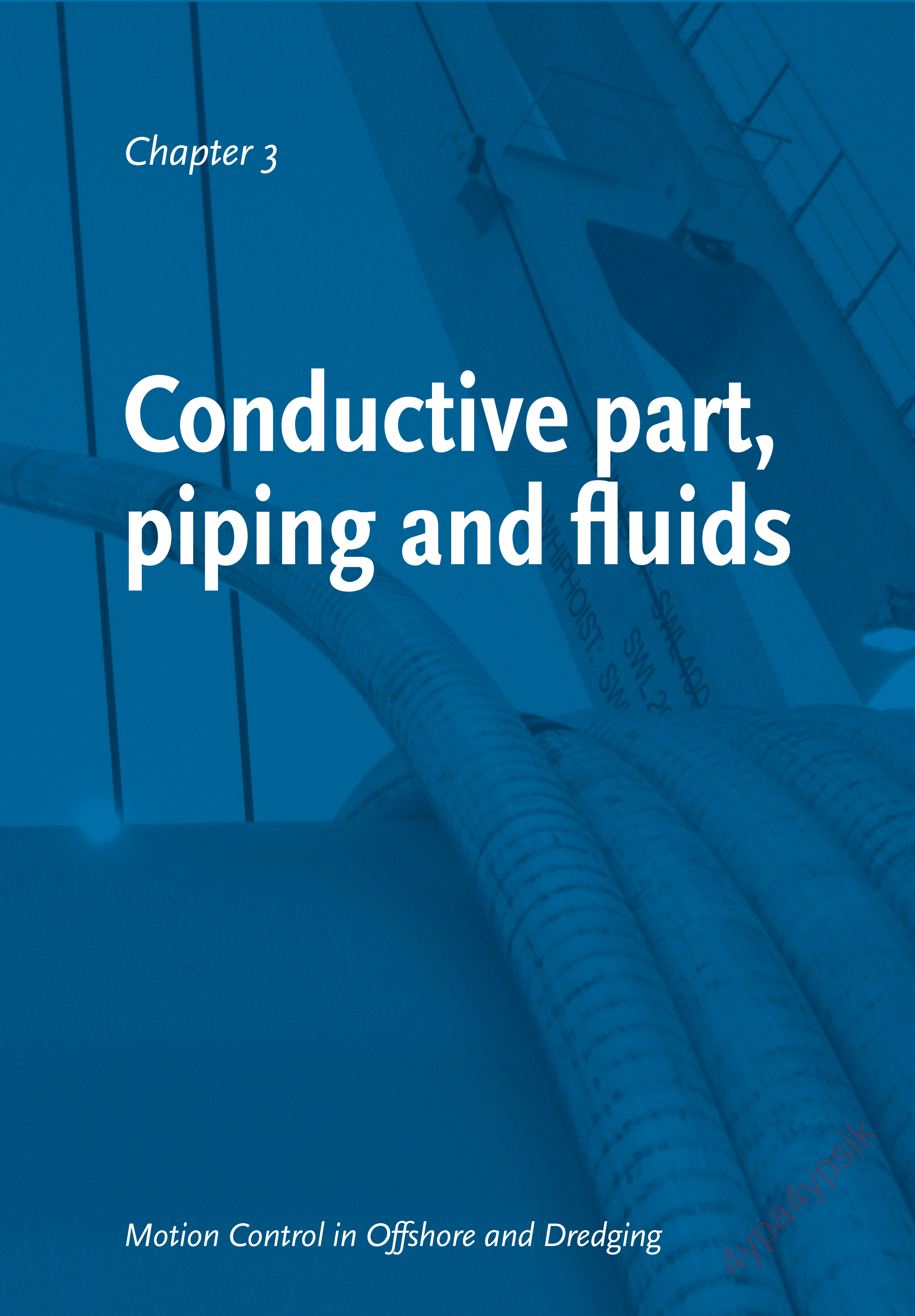
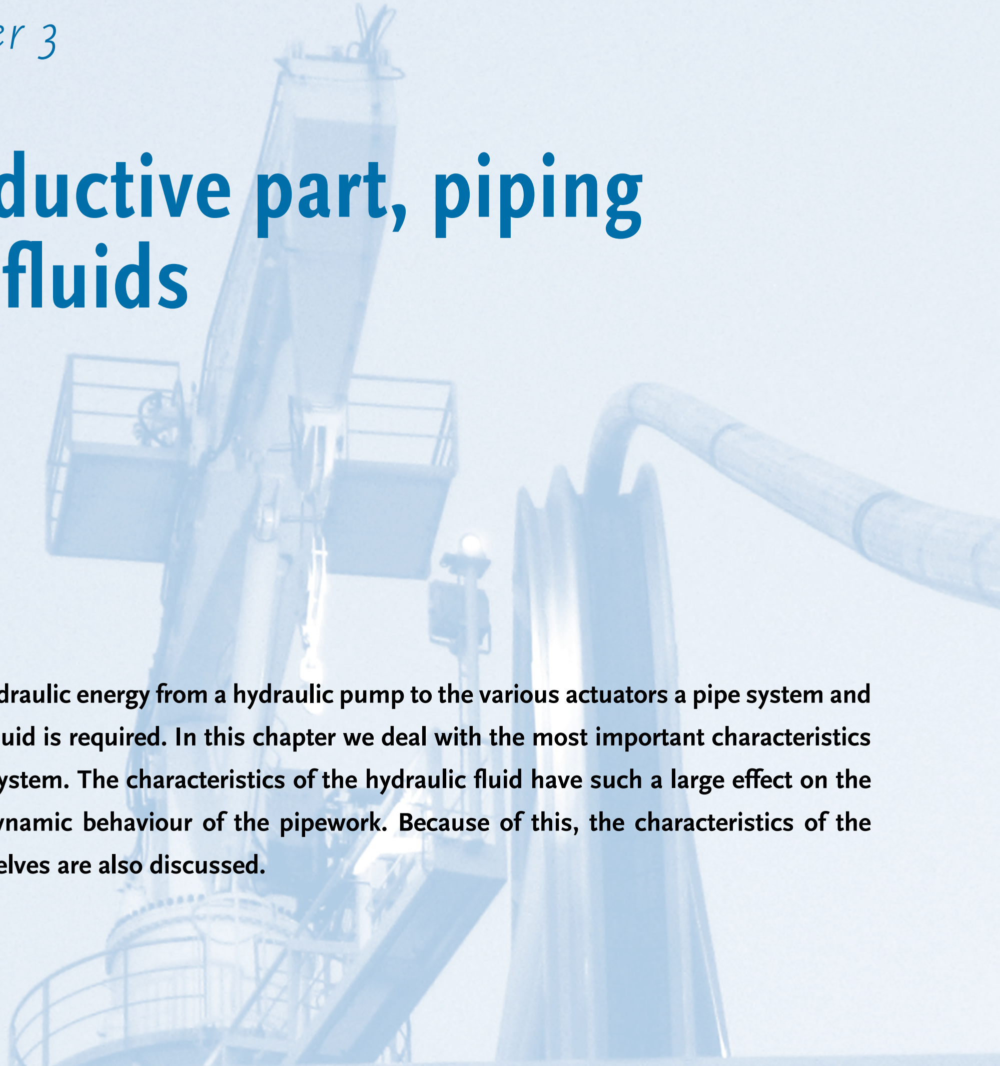


Chapter 3

Conductive part, piping and fluids

Motion Control in Offshore and Dredging



Chapter 3

Conductive part, piping and fluids

To get the hydraulic energy from a hydraulic pump to the various actuators a pipe system and of course a fluid is required. In this chapter we deal with the most important characteristics of the pipe system. The characteristics of the hydraulic fluid have such a large effect on the static and dynamic behaviour of the pipework. Because of this, the characteristics of the fluids themselves are also discussed.

P. Albers, *Motion Control in Offshore and Dredging*, DOI 10.1007/978-90-481-8803-1_3,

3.1 List of symbols

c	=	speed of sound	m/s
d_i	=	internal diameter	m
D	=	outer pipe diameter	mm
E	=	compressibility of mineral fluid	N/m^2
K	=	yield strength	N/mm^2
k	=	heat capacity	kJ/kg.°K
L	=	length	m
P	=	power	Nm/s
p	=	pressure	N/m^2 or N/mm^2
Δp	=	pressure difference	N/m^2
Δp_V	=	maximum pressure variation	N/mm^2
Q	=	volume flow	m^3/s
Re	=	reynolds number	
s_ν	=	wall thickness	mm
S	=	safety factor	
T	=	Temperature	°K
V_m	=	fluid velocity	m/s
V	=	volume	m^3
α	=	volumetric expansion coefficient	
λ	=	friction factor	
ρ	=	density of the fluid	kg/m^3
ξ	=	friction coefficient	
υ	=	kinematic viscosity	mm^2/s

3.2 Piping

Pressure drop in piping. A practical approach

ISO 4413 recommends the following maximum flow velocities for the different type of piping:

Suction Pipe : 1,2 m/s
Pressure Pipe : 5 m/s
Return Pipe : 4 m/s

If these flow velocities are applied during the design, we can assume that the pressure drop in the pipework will be so low that, as far as energy is concerned, very little power will be lost. The drop in pressure in the pipe work is very dependent on the viscosity of the fluid used. More detailed guidelines can be found in the table below.

Suction/Inlet Pipe		Pressure Pipe		Return Pipe
Kinematic viscosity ν (mm²/s)	Velocity V_m (m/s)	Pressure p (bar)	Velocity V_m (m/s)	Velocity V_m (m/s)
150	0,6	25	2,5 - 3	1.7 - 4.5
100	0,75	50	3,5 - 4	
50	1,2	100	4,5 - 5	
30	1,3	200	5 - 6	
		> 200	6	

Table 3.2 Recommended maximum fluid velocities in piping at different viscosities of the fluid.

Reliable calculation methods exist if the designer needs to know the pressure losses in a particular piece of pipework in more detail.

Pressure drop in pipe work. The theory

For the pressure drop in straight pipes and hoses that aren't smooth the following formula applies:

$$\Delta p_w = \lambda \cdot \frac{L}{d_i} \cdot \frac{1}{2} \cdot \rho \cdot V_m^{\,2} \qquad (3.1)$$

with:

Δp_w	= Flow resistance	N/m²	V_m	=	Fluid velocity	m/s
di	= Internal diameter	m	λ	=	friction factor	
ρ	= Density of the fluid	kg/m³				

The following applies to the internal diameter of the pipe or tube:

$$d_i = \sqrt{\frac{4 \cdot Q}{V_m \cdot \pi}} \qquad (3.2)$$

with:

Q	=	Flow	m³/s

To calculate the friction factor λ it is necessary to establish if the flow is laminar or turbulent first. This can be done using the Reynolds formula:

$$\mathrm{Re} = \frac{v_m \cdot d_i}{\upsilon} \tag{3.3}$$

with:

υ	=	Kinematic viscosity	mm^2/s

If Re is < 2320. then the flow is laminar and the resistance coefficient is :

$$\lambda = \frac{64}{\mathrm{Re}} \tag{3.4}$$

If Re is > 2320. then the flow is turbulent and the resistance coefficient is:

$$\lambda = \frac{0{,}3164}{\sqrt[4]{\mathrm{Re}}} \tag{3.5}$$

The Colebrook formula applies for pipes and hoses that aren't smooth:

$$\frac{1}{\sqrt{\lambda}} = -2\log_{10}\left(\frac{\varepsilon/D_h}{3{,}7} + \frac{2{,}51}{\mathrm{Re}\sqrt{\lambda}}\right) \tag{3.6}$$

with:

ε	=	roughness height	mm	Dh =	hydraulic diameter	mm

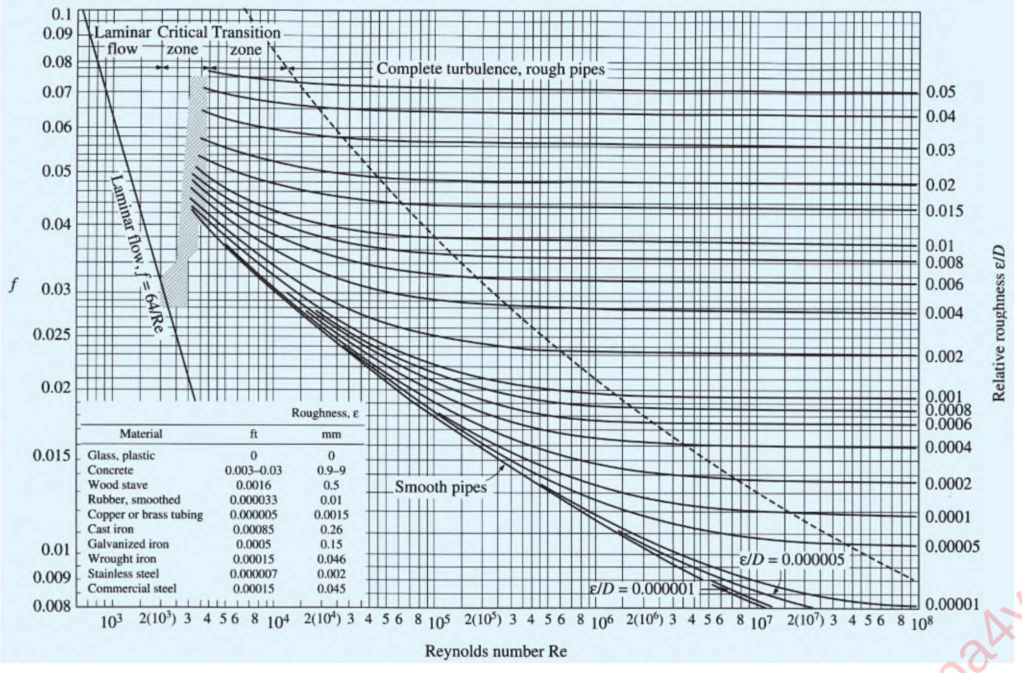


Fig 3.2.2.A The Moody graph showing the friction factor λ as function of Re and the roughness ε of the pipe

Material	Characteristic	λ in mm
Hose	'Technically Smooth'	ca. 0,0016
Seamless precision tube	With rolled skin	0,02...0,06
	Pickled Skin	0,03...0,04
	For Small Diameters	Up to 0,1

Table 3.2.2.A Practical values for the friction factor λ

The following formula applies for pressure losses in curves:

$$\Delta p = \xi \cdot \frac{1}{2} \cdot \rho \cdot V_m^{\ 2} \tag{3.7}$$

with:

ξ = friction coefficient

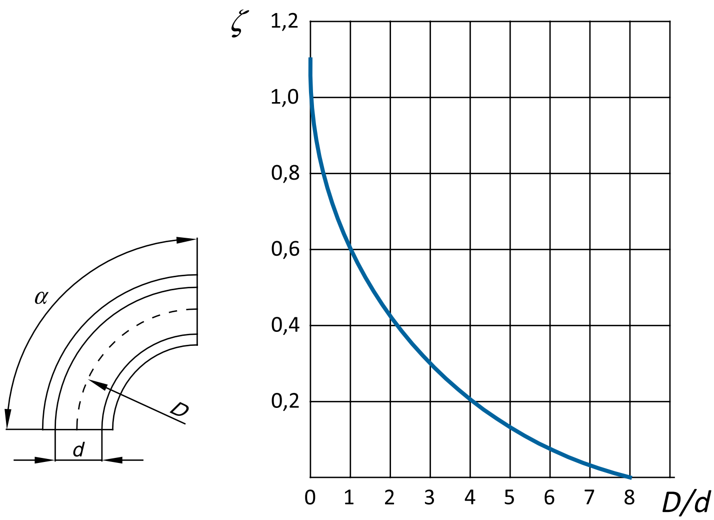


Fig 3.2.2.B Friction coefficient ξ for elbows of 90°

For bends of a different angle the following applies:

$\alpha = 45°$	$0{,}5 \cdot \zeta$
$\alpha = 135°$	$1{,}5 \cdot \zeta$
$\alpha = 180°$	$2{,}0 \cdot \zeta$

Table 3.2.2.B Friction coefficient ζ for different pipe angles

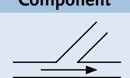
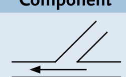
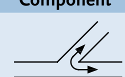
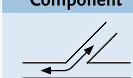
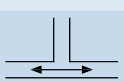
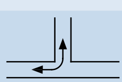
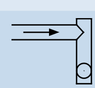
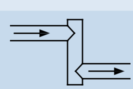
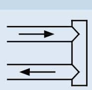
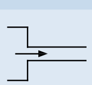
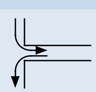
Component	ξ	Component	ξ	Component	ξ	Component	ξ
	0,15		0,05		3		0,5
	0,1		1,3		1,8		2,3
	1,5		0,3 - 0,5		0,5		

Table 3.2.2.C Friction coefficient ξ for specific component sections

The wall thickness of the tube

The wall thickness and the external diameter of a tube or pipe can be calculated on the basis of the standards issued by inspection instutions like TUV. Lloyds or Bureau Veritas. TUV uses the German standard DIN 2413 as basis for the pipe wall thickness.

Type of Load	Tube wall thickness s_v	Safety factor S for hydraulic tube		
			With material certificate DIN50049	Without material certificate
		yield	S	S
1. Static Load Temperature ≤ 120°	$s_v = \dfrac{d_i \cdot p}{2 \cdot 10^6 \cdot \dfrac{K}{S} \cdot v_N - 2p}$ (3.8)	≥ 25%		1,7
		= 20%	1,6	1,75
		= 15%	1,7	1,8
2. Dynamic Load	a) $s_v = \dfrac{d_i \cdot p}{2 \cdot 10^6 \cdot \dfrac{K}{S} \cdot v_N - 2p}$ (3.9) $s_v = \dfrac{d_i \cdot \Delta p_D}{20 \cdot \dfrac{K}{S} - 3 \cdot \Delta p_V}$ (3.10) b) Calculate both and take largest value		1,5	

Table 3.2.2.D Rules for the calculation of the pipe wall thickness sv according to the standard DIN 2413

with:

d_i	=	internal pipe diameter	mm	S	=	safety factor	
K	=	yield strength	N/mm²	s_v	=	pipe wall thickness	mm
p	=	hydraulic pressure	N/mm²	v_N	=	weld efficiency, see table 3.2.2.E	
Δp_V	=	maximum pressure variation	N/mm²				

Pipe/Hose	Material according To	Welding process *	Weld efficiency vN
For normal use Standard quality DIN 1626	DIN 17100 Quality 1	Without certificate	0,5
		With certificate	0,7

Table 3.2.2.E Welding efficiency v_N according DIN 2413*

** This is a certificate that encompasses the whole of the welding process. These establish that welding procedures will be set up. that certified welders will be used and that material certificates will be available for the materials that are used.*

If the bend is put into the pipe manually and you want to use the calculated wall thickness specified above, then the radius of the bend needs to be 3 x the diameter (3D) of the pipe. If an internal mandrel is used to bend the pipe, then the radius of the bend in the pipe can be smaller but not smaller than 2.5 D. The minimum pipe radius is to prevent the pipe becoming oval at he bend, causing high internal material stress, especially when a dynamic pressure is applied.

Types of material	Material Number	Yield strength *K* N/mm²	Max Strength N/mm²
St 35N	1.0308	235	310
St 37.4N	1.0255	255	390
St 52.3N	1.0570	355	533
RVS 316L	1.4404	234	515

Table 3.2.2.F Yield strength K for different types of pipe material

Allowable static pressure (bar)					
Tube	D mm	s_v mm	St37.4N	St52.3N	316L
3/8"	12	2	283,3	394,4	260,0
1/2"	16	2	212,5	295,8	195,0
1/2"	18	2	188,9	263,0	173,3
3/4"	20	2.5	212,5	295,8	195,0
3/4"	20	2.5	170,0	236,7	156,0
1"	25	3	204,0	284,0	187,2
1"	28	2.5	151,8	211,3	139,3
1.1/4"	30	3	170,0	236,7	156,0
1.1/4"	30	4	226,7	315,6	208,0
1.1/2"	38	4	178,9	249,1	164,2
1.1/2"	38	5	223,7	311,4	205,3
1.1/2"	42	3	121,4	169,0	111,4
1.1/2"	50	5	170,0	236,7	156,0
2"	60	3	85,0	118,3	78,0
2"	60	6	170,0	236,7	156,0
2.1/2”	75	3	68,0	94,7	62,4
2.1/2”	80	10	212,5	295,8	195,0
3"	90	3.5	66,1	92,0	60,7
3"	97	12	210,3	292,8	193,0
4"	115	4	59,1	82,3	54,3
6"	165	5	51,5	71,7	47,3

Table 3.2.2.G Allowable static pressures for different types of piping according to TUV (DIN 2413), with : D = outer diameter op the pipe and sv is the wall thickness.

Flushing and cleaning

Dirt in a hydraulic installation is mainly residue left over from the installation of the pipe work. This can be partially avoided by careful handling of ‘open’ pipes and by choosing certain types of pipe technology. The pipework will always need to be flushed and cleaned before the system can be activated. Below is an example of a procedure carrying out the flushing.

Preparation	Use seamless and passivated pipes
	Ask for the pipes to be delivered with capped ends
	Don’t use socket welded flanges but butt welded flanges
	Flushing must be incorporated during the design state

During the installation	Use a saw to cut the pipes to length, not a pipe cutter.
	Clean the pipe work after couplings and flanges have been attached using an air driven plug.
	Use an inert gas in the pipe when welding to prevent slag formation.
	Don’t use welds to connect pipes directly to each other.
	Make sure that completed sections of pipe work are fitted with blind flanges during transport to the installation site.

During commissioning	Replace servo valves and expensive proportional valves temporarily with simple directional valves and flush plates.
	Flush the pipe work with your own pump equipment or, if necessary, with a separate pump unit and where necessary in separate parts. Make use of temporary connections wherever possible to achieve a turbulent flow in the pipes.
	Increase the oil temperature. A higher oil temperature will give a higher Reynolds number and thus increases the chance of a turbulent flow. This in turn will increase particle pickup.
	Use filters with the mesh size that will help you achieve the required level of cleanliness.
	Replace the filter elements after you have flushed the installation and also during the flushing if they are contaminated
	Re-install the servo valves and proportional valves. Take care that no new dirt is being added as paint particles

3.3 Fluid properties

Hydraulic fluids are divided into a number of classifications as listed below, in accordance with ISO 6743/4. Some of them with a cross reference to DIN 51 524.

Hydraulic liquid with a mineral oil basis

ISO	DIN	Description and characteristics
HH	-	Mineral oils without additives. low cost price. only suitable for non-critical systems
HL	HL	Mineral oils with anti-corrosion and anti-oxidant additives. Longer lifespan than HH oil
HM	HLP	Mineral oil. resistant to high pressure. identical to HL but with wear-resistant additives.
HV	HVLP	Mineral oil with high viscosity index

Fire Resistant Liquids

ISO	Description
HFA	Oil in Water (maximum 20% oil)
HFB	Water in Oil (maximum 40% water)
HFC	Water/glycol (maximum 35% water)
HFD	Waterless synthetic liquids

More Environmentally Friendly liquids

ISO	Description
HEPG	Poly glycols
HETG	Triglycerides (Rapeseed Oil)
HEES	Synthetic Ester

Table 3.3.A Identification of different types of hydraulic fluids

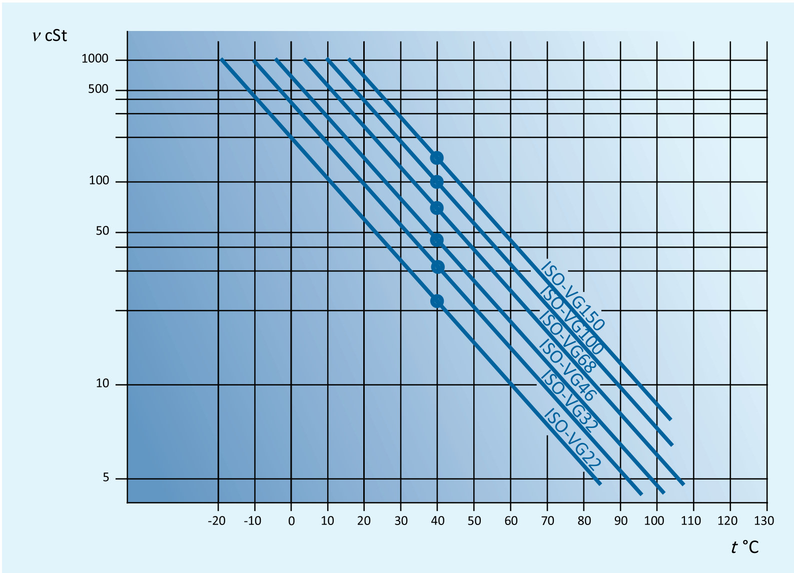


*Fig 3.3.A Viscosity **v** of hydraulic mineral oils for different viscosity grades VG (Courtesy of Shell)*

The viscosity Grade represents the normalised viscosity classes at 40° C. designated in conformance with ISO standards. They can have a variance of +/- 10% from the norm. The following classifications have been standardised: VG 15, 22, 32, 46, 68, 100 and 150.

The viscosity Index (VI) is an assessment of whether the viscosity changes faster or slower when the temperature changes. For systems with strong temperature variations, as tends to be the case with installation in the open air, oils with a even higher VI are recommended. The VI can vary from 0 to 300 but is usually between 100 and 150.

The use of motor oils

Sometimes mineral oils that comply with the Society of Automotive Engineers (SAE) standards are used. Generally speaking these oils don’t contain the additives necessary for the optimum life expectancy of the installation. The SAE numbers are designed to indicate the viscosity of engine and gearbox oils. An SAE number with the additional W designation, indicates an oil suitable for use in the winter, for temperatures down to -18oC. Without the W, the oil is mainly suitable for use in the summer. Multigrade oils can, for example, have a designation of SAE 10W30. In the summer these oils have the characteristics of an SAE 10 oil, in the winter (with lower working temperatures) those of an SAE 30 oil.

Engine Oil SAE grade	Viscosity at 40°C mm^2/s	Hydraulic Fluid ISO/VG
10	30 – 48	32 and 46
20	40 - 74	46 and 68
30	74 - 110	100

Table 3.3.B Comparison of SAE grades with ISO/VG standards

Choosing the viscosity

The choice of the viscosity of the hydraulic oil is dependent on:
- The maximum value of the viscosity at the lowest start-up temperature, as specified by the manufacturers of hydraulic pumps and motors
- The minimum allowable value of the viscosity at the highest operating temperature.

In order that components don't get damaged, a minimum and maximum viscosity for each component has been set. These can be obtained from the component manufacturers. As a rule, this needs to be controlled for pumps and hydraulic motors. The pour point for the oil can be found in the specification documentation. This is particularly important in cold regions.

Viscosity as a function of pressure

Viscosity increases with pressure. A factor that comes into play at pressures of more than 200 bar. At 350 bar it is necessary to take account of a doubling of viscosity. Figure 3.3.B shows the change in viscosity for VG 100.

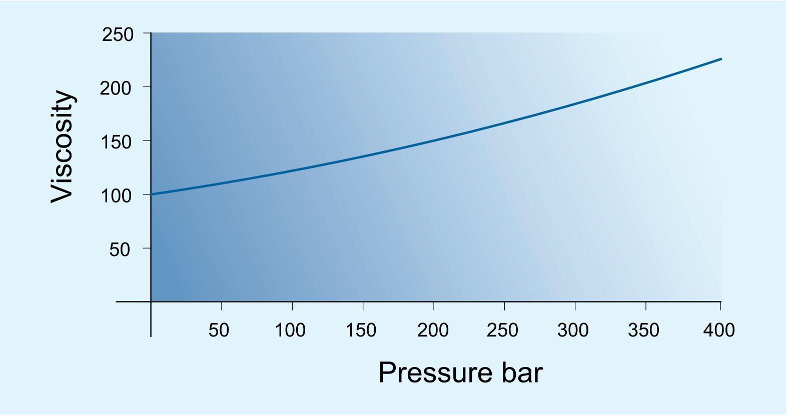


Fig 3.3.B *Viscosity ν increases with the pressure (bar), typical curve for a VG100 oil.*

Compressibility

You can use the following formula to calculate the change of volume resulting from a change in pressure.

$$\Delta V = \frac{V_0 \cdot (p_2 - p_1)}{E}$$

with:

E	=	Compressibility of mineral fluid	N/m^2 (10^9)	ΔV =	Change of volume	m^3
p_1	=	Initial pressure	N/m^2	V_0 =	Initial volume at pressure p_1	m^3
p_2		= End pressure	N/m^2			

For mineral oils you can use a compression factor of 1% per 100 bar. This takes the expansion of both pipe work and hoses and the effect of the trapped air that is present in the hydraulic fluid into account.

For hydraulic fluids that hold water (about 95% water volume) a compression factor of 0.45% is used.

Density and pressure

The density is dependent on the pressure, in line with the following formula :

$$\rho_{(p)} = \rho_{(0)} \cdot (1 - \frac{\Delta p}{E}) \qquad (3.12)$$

with:

$\rho_{(p)}$=	Density under increased pressure kg/m³	$\rho_{(o)}$=	Density under atmospheric conditions kg/m³

The density is also dependent on the temperature, in line with this formula:

$$\rho_{(T)} = \rho_{(283^o K)} \cdot (1 - \alpha \cdot (T - 283)) \qquad (3.12)$$

with:

T = temperature	°K	$\rho_{(o)}$=	Density under atmospheric conditions at 283 °K kg/m³
α = volumetric expansion coefficient (0.00067 for mineral oils)		$\rho_{(T)}$=	Density at a certain temperature T kg/m³

If we assume a value of E = 10^9 N/m2 then it is possible to combine both formulas, giving the following approximation for the temperature dependency of mineral oils in a closed system, relative to the pressure.

'The pressure increase in an enclosed space filled with mineral oil is approximately 100 bar for each 15°C increase in temperature'

AND

'Under atmospheric pressure, the volume of the oil increases by 0.76% for every 10°C increase in temperature.'

Density

Medium	Density ρ
Mineral Hydraulic Fluids	850 – 930 kg/m³
Fluids Containing Water	1000 kg/m³
HFD Fluids	1400 kg/m³

Table 3.3.C Density ρ for different type of hydraulic fluids

Capacity to dissolve air

Up to a pressure level of 300 bar, the Henry-Dalton law applies: "the amount of dissolved gas in a liquid is proportional to the partial pressure of the gas above the liquid". For air under atmospheric conditions this is 6-7% of the volume, see also figure 1.6.9.A Dissolved air is invisible and takes the form of the liquid. Dissolved air is released in the form of air bubbles when the pressure drops. When the pressure increases again the oil will absorb air again.
Free air in the tank, in the form of air bubbles, needs time to rise to the surface. The speed with which the air disappears out of the liquid is dependent on the viscosity. According to DIN 51524/2, the requirement for HLP46 (HM46) is a minimum of 10 minutes, depending on the height of the volume in the oil tank.

Resistance to ageing

The rule of thumb is that the speed at which the oil ages doubles for every 10°C increase in temperature above 70°C, in other words, the lifespan of the oil halves. The ageing process also accelerates due to pollutants: water, rust, particles released due to wear and dust. Ageing oil changes colour and becomes darker. Anti-oxidants make sure that the oil will reach the normal life expectancy.

Ability to separate out water

The water content in mineral oils is not allowed to be higher than 1000 ppm. A water content of less than 200 ppm does not normally pose any problems. Some (hygroscopic) fluids. like synthetic esters attract extra water. If that is the case dryers need to be put in place on top of the reservoir to prevent acidification.

Specific heat

The heat absorption capacity of fluids is expressed in k as [kJ/kg.°K]. For mineral oils the value of k is 1,8. The heat absorption capacity is important for the calculation of heat management in the hydraulic installation.

The example used is for the heat generated in the system when the fluid flows through a component, whilst a drop in pressure occurs across that component.

The energy loss when a pressure relief valve opens is equal to:

$$P = \Delta p \cdot Q \tag{3.14}$$

where

Q	=	oil flow across the valve	m^3/s	Δp	=	pressure drop across the valve	N/m^2

This energy transferred into heat, which results in a higher temperature of the outlet oil of the relief valve.

The temperature rise that will occur is:

$$\Delta T = \frac{\Delta p}{10^3 \cdot \rho . k} \tag{3.15}$$

with:

ΔT	=	temperature rise of the fluid	°K	k	=	heat capacity	kJ/kg.°K
ρ	=	density	kg/m^3 (≈890)				

The result of this calculation is that the temperature of mineral oil rises between 5 and 6°C for every 100 bar change in pressure. This phenomena is often used to heat up a tank in severe cold start-up conditions. In that case a dedicated pump is being used that can withstand the high viscosities.

Speed of sound

The speed of sound with which pressure waves travel through pipes is determined by the following formula:

$$c = \sqrt{\frac{E}{\rho}} \tag{3.16}$$

For mineral oils the approximate value of c is 1000 m/s.

For hoses the stretch in the hose results in a lower E value and thus in a lower speed of sound.

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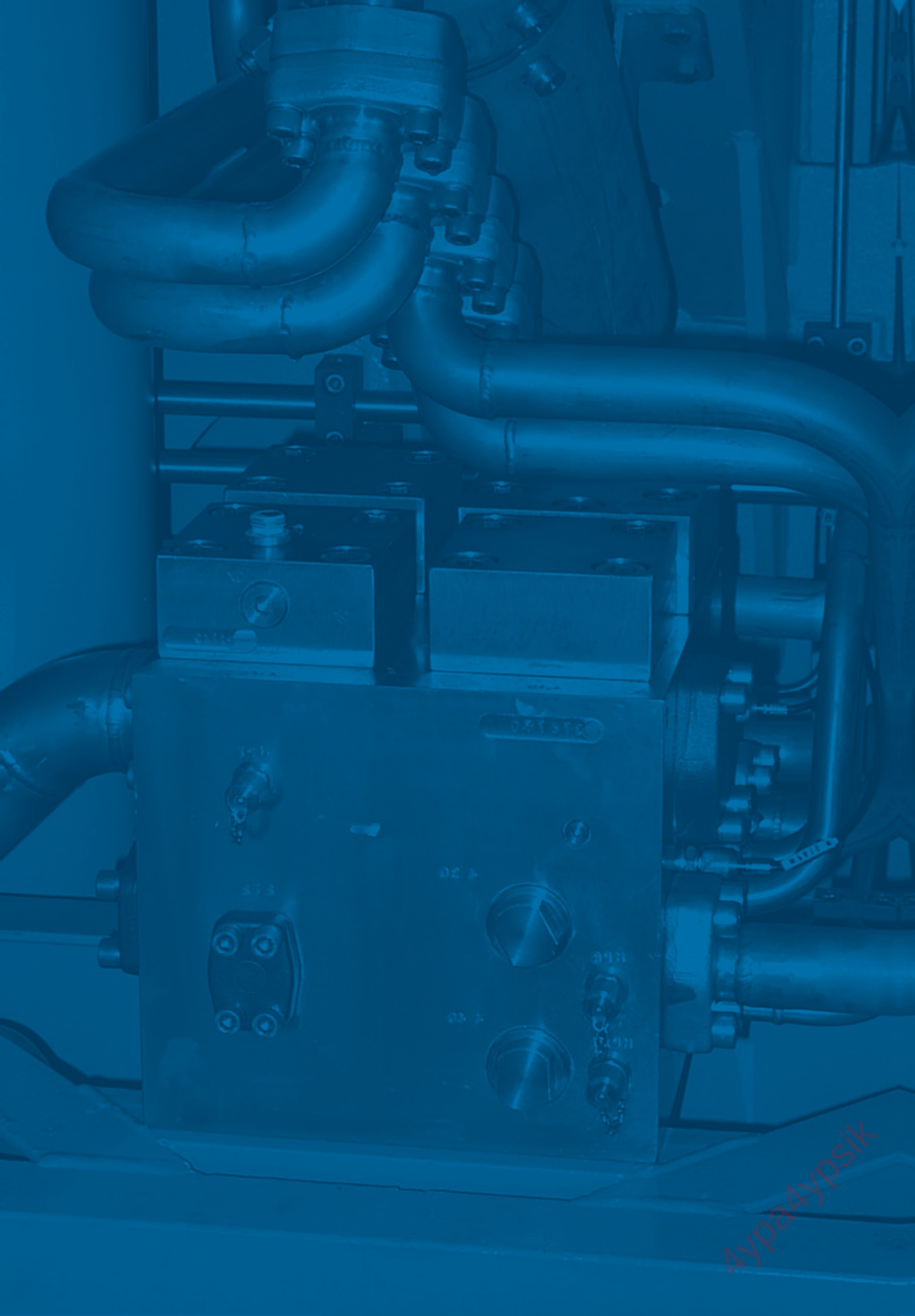

Chapter 4

Fluid conditioners and hydraulic accumulators

Chapter 4

Fluid conditioners and hydraulic accumulators

A hydraulic system does not only consist of pumps, motors, valves and hydraulic piping. This chapter deals with the parts in the hydraulic system that maintain the condition of the hydraulic fluid. A proper design for these sub systems is very important for the lifetime of the hydraulic system.

P. Albers, *Motion Control in Offshore and Dredging*, DOI 10.1007/978-90-481-8803-1_4,

4.1 List of symbols

p	=	*pressure*	N/m^2
P	=	*power*	kW
ΔT	=	*temperature difference*	$°K$
V	=	*volume*	m^3
Z	=	*heating time*	s
κ	=	*adiabatic gas constant*	
ρ	=	*density of the fluid*	kg/m^3
k	=	*heat capacity*	$kJ/kg.°C$

4.2 Reservoir

The reservoir performs several functions:

- Storage of fluids in the system. Note that the volume of the fluid present in the system can vary dependent on for instance the position of the piston rod. It is also possible that there are hydraulic accumulators in the system, which will be filled with changing quantities of oil. These variations in volume can amount to as much as 5000-15.000 dm^3 in the larger off-shore and dredging installation.

- Heat exchange. A reservoir can release a certain amount of heat because of the surface area. The amount of heat released by the tank is dependent on the wall thickness of the tank and the environmental conditions in which the tank is positioned, like for example the amount of air flowing past the tank. The maximum amount of heat radiation from the reservoir, in a naturally ventilated space, is 0,5 W/m^2. The maximum amount of heat radiation in a space with forced ventilation is 1 kW/m^2.

- If a water/oil heat exchanger is to be used a water in fluid sensor (soluted water) or a water detection switch for free water helps to detect leakage of the oil cooler.

- If a power unit is been switched off during the night the water in the humid air above the fluid level can condense onto the reservoir top plate. Free water will then assemble at the bottom of the reservoir. During the day at warmer conditions humid air may enter the reservoir etc. This process may repeat itself every 24 hours.

- The resting of the hydraulic liquid. A certain amount of air is dissolved in all fluids. Under atmospheric conditions it is possible for some of this dissolved air to be released from the liquid. The free air cannot flow to the surface of the liquid very quickly because of the viscosity of the liquid. That is why the capacity of the reservoir is such that the oil can stay in the reservoir for 3 -5 minutes before it is returned to the system by the hydraulic pump.

- To make sure that there is sufficient fluid in the tank for the suction action of the pump. This stops air from being sucked into this system via the suction pipe. If this happened, it could cause cavitation in the pump. To prevent this, a minimum fluid level needs to be maintained in the reservoir.

- In installations with a total fluid content of >100 dm^3, a partition panel must be built in to separate the oil returning from the installation from oil in the suction compartment. This panel allows the eventual available air to dissolve from the fluid and eventual available dirt to collect at the bottom. The partition panel needs to allow enough liquid to flow over the top to keep the suction compartment sufficiently full. An even better solution can be obtained if all leak oil is returned into a third separated compartment. Experience shows that the leak oil pipe contains the largest contamination.

- Temperature and level sensors are to be installed in the suction compartment.

- Drain and return lines should end below the lowest oil level. If this is not done, a turbulent flow may be introduced sucking air into the fluid. Be sure that no vent holes are installed in these pipes. These will introduce an enormous amount of free air in the fluid.

- In each suction line a shut off valve is to be installed, preferably with an electrical switch to detect an open position. The signal of this switch can be used to interlock the start of the electric motor that drives the pump.

Figure 4.2.A. shows a number of examples of the different ways in which a well functioning reservoir can be constructed.

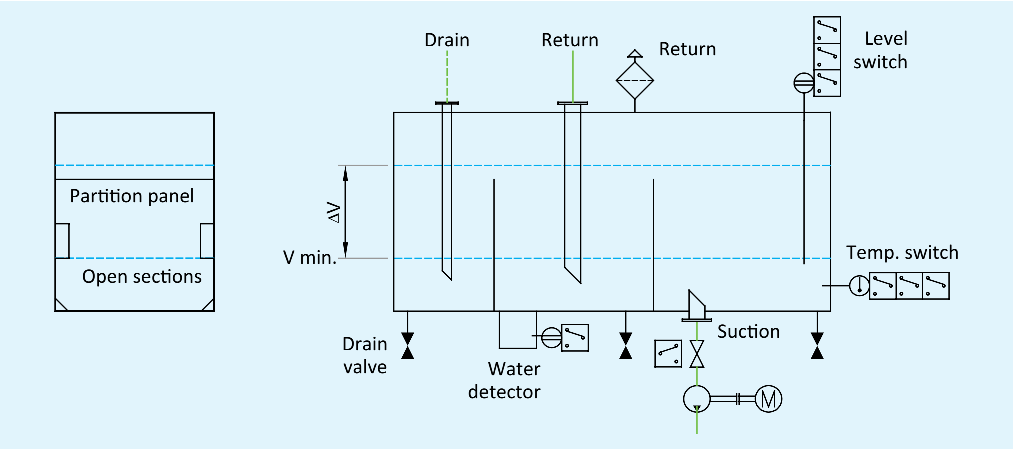

Fig 4.2.A Example of a well functioning reservoir

An Indication of the Volume Requirement for the Reservoir

The capacity of the reservoir in litres = 3-5 x pump capacity Q_p lpm + minimum volume (V_{min}) + maximum volume variation (ΔV). The mobile industry usually uses smaller reservoirs.

The viscosity of the oil needs to be lower than the maximum value specified by the manufacturer for the pump to be started successfully. The rule of thumb for axial piston pumps is a start-up viscosity lower than 300 cSt. Once the pump has started, it can not immediately be used at full hydraulic power. The viscosity of the oil needs to be much lower than the start-up viscosity before this can be done. In other words, the temperature of the oil in the reservoir needs to be higher. The optimum viscosity of the oil is between 15 and 100 cSt for many pump types.

This means that the reservoir needs to be heated before start-up if the environmental conditions are such that the pump needs to be started in low temperatures. Pre-heating the oil can be achieved in two different ways. Either by using an electric heating element or by using a separate 'conditioning' pump. The behaviour of the pre-heating cycle of the reservoir is known as a 1st order control system.
When electrical heating is applied, it is possible to calculate the electrical net power required with the help of the following formula.

$$P = \frac{\Delta T \cdot V \cdot \rho \cdot k}{Z} \tag{4.1}$$

with:

P	=	electrical net power	kW	Z	=	heating time	s
V	=	volume	m³	ρ	=	density of the fluid	kg/m³
ΔT	=	temperature rise	°C	k	=	specific heat	kJ/kg.°C

If you need to increase the oil temperature by 10°C in 30 minutes in a reservoir containing 500 dm³ of oil, then the formula shows that the power P = 4,45 kW (for ρ = 890 kg/m³ and k = 1,8 kJ/kg.°C).
The disadvantage of electrical elements is that the temperature on the element surface can easily rise above 120 °C. At this temperature the fluid's ageing process can start very quickly.

A different, frequently used, method for pre-heating is the use of an auxiliary pump that can be started at a higher viscosity (lower oil temperature). Vane pumps, gearwheel pumps and screw pumps are very suitable for this type of usage (they can work with a start-up viscosity of 1.500 – 2.000 cSt).

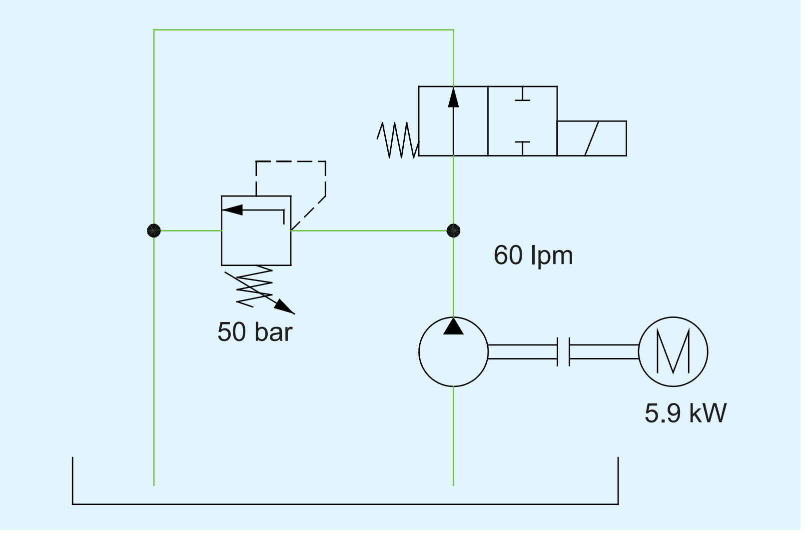

Fig 4.2.B Heating the fluid of a reservoir with a conditioning pump

In the example of figure 4.2.B a pump with a capacity of 60 lpm has been installed. Activating the 2/2 valve forces the full volume flow to pass through the pressure limiter valve. The 'heat loss' that this generates is p.Q/600 = 50.60/600 = 5 kW. If the efficiency of the pump η_{tot} = 0,85, than the electrical energy required is about 5,9 kW.

The temperature increase over one circulation through the pump and the pressure relief valve will be about 3°C. This means that the oil temperature will not reach the temperature of the electrical heating element, as described earlier, at any stage. It is also often possible to use this pump as a conditioning pump in a filtering or cooling circuit. This means that the pump has more functions than just warming the oil.

4.3 Filtration

For hydraulic fluid to be used in hydraulic installations, it needs to be of a certain cleanliness. The desired cleanliness is, on the one hand, determined by the sensitivity to dirt of the different components, but the fact that components wear out faster with contaminated hydraulic fluid is of equally importance.

Dirt in a hydraulic installation is caused in several different ways. A new barrel of oil already contains a certain quantity of contaminants, which means that the oil can only be applied in an industrial installation after pre-filtration. Dirt is also introduced into the installation by newly fitted pipe-work or newly introduced cylinders. This is why pipe-work always needs to be flushed using filters if the necessary cleanliness classification is to be achieved. If this is omitted, very large damage can occur in pumps and dirt can build up in valve blocks, cylinders and hydraulic motors.

A good example of the consequences of the addition of unclean pipe-work was a new hydraulic installation for a 'J-Lay' pipe-laying installation. In this case part of the pipe-work, 480 meters of hoses, was installed by a third party contractor, as it turned out later, without those hoses having been flushed. The result was that the dirt from the hoses spread all the way across the installation. This consisted of 150 actuators, an equal number of functional valves and 2,500 meters of pipe-work. The cost of replacing the main pumps and flushing the already assembled system came to about 600.000,- euro.

The level of impurity can be classified in line with SAE AS4059E (previously NAS 1638) or ISO 4406. Both indicators relate to a volume of 100 ml of liquid. The table below gives the relationship between the two classifications. The NAS classification measures the total number of impurities between two sizes (5 µm and 15 µm) whilst the ISO classification gives the specific number of particles larger than a particular size (> 4 µm, > 6µm and >14 µm). This means that the ISO classification consists of 3 numbers.

The rule for both classifications is that for the higher classification twice as many particles are present or allowed in the oil.

Contamination Classification		General Application	Usually recommended Filter Size
NAS	ISO		
4	13/10/7	Aircraft industry, servo technology	2-3 μm
5	14/11/8	Industrial drive systems, proportional drive technology, high pressure system (> 300 bar)	3-5 μm
6	15/12/9		
7	16/13/10	Industrial drive systems, electro-hydraulic valves, medium and low pressure system.	5-10 μm
8	17/14/11		
9	18/15/12	Small size medium pressure systems, low pressure systems with high tolerances	10-20 μm
10	19/16/13		

Table 4.3 Comparison of NAS and ISO cleanliness levels

The filtering capacity of a filter reflects its capacity to remove dirt particles of a certain size from the oil and is, amongst other things, determined by the so-called β-value. For example $\beta_{10} = 75$ or $\beta_3 = 150$. The index gives the size of particles for which the filter is suitable.

$\beta_{10} = 75$, means that of 75 particles with a size > 10μm entering the filter, only 1 particle will pass through. This means that a $\beta_{10} = 200$ filter is a little bit more efficient than a $\beta_{10} = 75$ filter.

The percentage of particles stopped by the filter can be calculated by the following formula:

$$\% = \frac{\beta - 1}{\beta} \tag{4.2}$$

For $\beta = 75$ this means:

$$\frac{75-1}{75} = 98{,}66\% \tag{4.3}$$

And for $\beta = 200$:

$$\frac{200-1}{200} = 99{,}5\% \tag{4.4}$$

The β-value given for a filter will deteriorate as the pressure drop over the filter increases (as the filter gets more and more contaminated).
Another important factor is the maximum amount of dirt that the filter can absorb. This is expressed in grams (In accordance with ISO 4572 for a pressure drop across the filter of 5 bar). You will need to look at the manufacturer's specification for more details about this.

4.3.1 Different filtration methods:

Filters can be fitted to a hydraulic system in several different ways to improve the purity of the system. Figure 4.3.1 gives the most important basic methods.

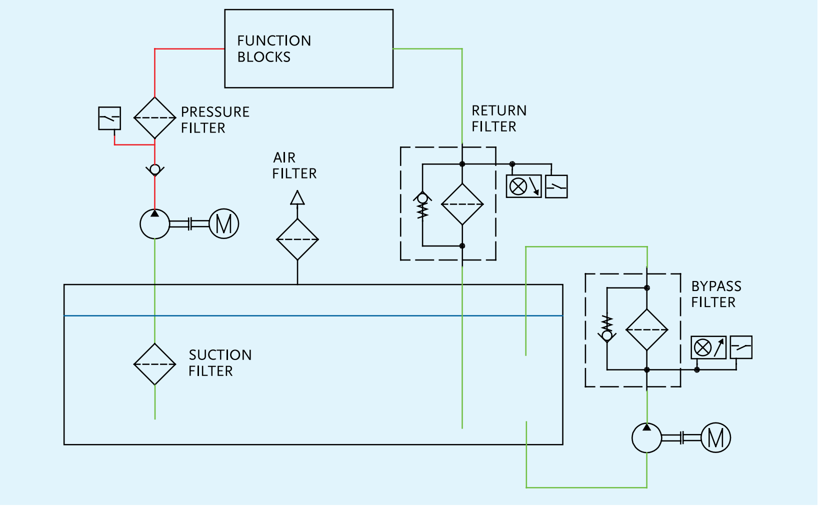

Fig 4.3.1 The most important basic methods for filtering

Suction filter

Suction filters are only used for coarse pre-filtering with a filter size of β_{150}=75. A major disadvantage of these filters is when the filter element is polluted or when the viscosity is low. In those cases the pressure drop across the filter will be high, which means that the suction pressure for the pump will be lowered thus increasing the risk of cavitation. This type of filter is not used in large industrial installations. The philosophy for those installations is that the oil in the reservoir must be clean.

Pressure filter
The pressure filter is there to protect the system beyond the filter from impurities caused by wear in the pump. This can be important for servo systems, systems with large pipe networks like in the offshore and dredging industry or in heavy industry and/or systems where idle time has a significant economic effect on the business. In these situations a β_{20}=75 filter can be applied. In such cases the filter must be fitted with an indicator for the contamination level in the filter element, not have a bypass and be able to cope with high pressure drops across the filter element.

Return filter
The return filter is usually fitted with a bypass and contamination indicator. In the drawing shown, it serves as a 'working' filter. During the design phase careful consideration needs to be given to the maximum fluid flow to the reservoir. After all, when a cylinder is used, a significantly higher fluid flow can be created due to the difference in piston surface areas. Also fast reacting valves (servo valves) can lead to pressure pulses in return lines.

Bypass filter
The bypass filter conditions the oil in the reservoir. To provide this filter with oil, an extra pump will be required with an otherwise low output. By constantly passing the oil over a bypass filter, often with a filter element with β_3>200 , it is usually possible to increase the purity of the system oil considerably. The electric motor-pump of the bypass system runs all the time, even when the main installation is switched off. The effect of the bypass filter is that the purity of the whole system can improve at least three grades.

4.3.2 Design considerations

The following design criteria are important to the choice of filters:
- β-ratio and particle size value.
- Maximum operating pressure for the filter.
- The maximum occurring liquid flow (Including effects of cylinders).
- The viscosity(ies) that will occur, taking into account start-up conditions and operating conditions.
- How to build it into the system, line mounted, tank-top mounted or manifold mounted.
- Is a bypass required or not.
- Contamination indication, visual, electronic or a combination of both.

The manufacturers will often supply details of the pressure drop of the filters under consideration. Often they also give advice on the maximum pressure drop that can be sustained. This is usually the pressure drop for a clean (= new) filter element, a certain viscosity (usually 40 cSt) and volume flow. (See figure 4.3.2.). Make sure that the pressure drop is analysed not just for the filter housing but also for the filter element itself.

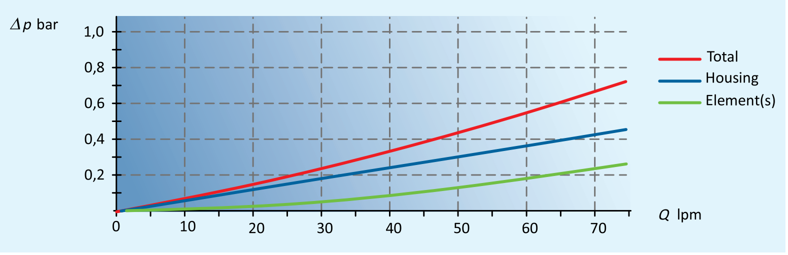

Fig 4.3.2 *Example of the pressure drop of a return filter for a maximum volume flow of 50 lpm and a viscosity of 30 cSt. (Courtesy of Hydac)*

4.4 Accumulator

4.4.1 Definition

A hydraulic accumulator (accumulator) is a pressure vessel for the storing and subsequent releasing of hydraulic energy. The compressibility of a gas (usually nitrogen) is used to store hydraulic energy and then release it later.

Because accumulators contain compressed gas and liquid in the same housing, they are covered by the directive for pressure equipment (97/23/EG). A so called CE certificate of conformity is required to build an installation that includes an accumulator in the European Community. With the certificate, the manufacturer of the accumulator declares that the required design criteria have been met. Institutions like Lloyds, RINA or DNV can set further conditions. Manufacturers can usually comply with those demands.

4.4.2 Applications

The most important application is for the storage of energy in cases where the actuators do not use hydraulic energy on a continuous basis. This means that the accumulator will then be able to supply sufficient energy when the actuators need more energy than the pump can provide and that a pump with a smaller capacity can be used. As soon as the demand for energy from the actuators drops off, the pump will be able to bring the accumulator back to the desired energy level.

A hydraulic accumulator can also deliver temporary power to the users if the primary energy source fails. On board ship for example, it is possible for a total electrical power failure or 'black ship' situation to develop. In these cases it may be important to finish a number of functions for safety reasons. It may also be necessary to have a certain amount of hydraulic energy in reserve or for a certain amount of pressure to remain present, for example to complete a clamping operation.

Pressure pulses can occur in hydraulic pipe systems, for example as the result of sudden deceleration of the liquid flow. If the accumulator is fitted in the right place and has been given the correct dimensions, most of the pressure pulses can be absorbed.

4.4.3 Types

The accumulators are subdivided on the basis of the membrane between the nitrogen and the oil side of the accumulator:

Fig 4.4.3.A Bladder type
Fitted with a closed bellow, most often found in industrial installations.

Fig 4.4.3.B Membrane type
Fitted with a bellow/membrane that is open at the top, most often found in the automotive industry.

Fig 4.4.3.C Piston type in stainless steel for offshore applications
Fitted with a free moving piston, amongst the application are the off-shore industry for heave motion compensation, in chemical industry for emergency supply.

4.4.4 Working conditions, Law of Boyle-Gay-Lussac

Because nitrogen is used as the energy carrier, the laws of thermodynamics apply. These laws describe the pressure, volume and temperature of the nitrogen gas at all times. Dependent on the speed with which the gas pressure changes, i.e.: the speed with which the accumulator is charged up or discharged, the behaviour of the pressurised gas changes too.

The behaviour of the accumulator can be described by the behaviour of the compressed gas in the accumulator. The 'Law for Ideal Gas' or Boyle-Gay-Lussac is applied:

$$\frac{p.V^{\kappa}}{R.T} = C \tag{4.5}$$

with:

p	=	gas pressure	N/m²		R	=	gas constant	
C	=	constant			T	=	absolute temperature	°K
V	=	gas volume	m³		κ	=	adiabatic gas constant	

In the following part of this paragraph we will see that the temperature T is not incorporated in the equations. This is due to the fact that temperatures in the gas cannot be measured accurately with normal temperature transmitters. For research purposes fast reacting thermocouples have to be used. What is being done in the calculations is assuming that for equation 4.5 the temperature is constant. Next we use κ values for the adiabatic constant to bring us results for the pressure p or the volume V that comply with actual measured data.

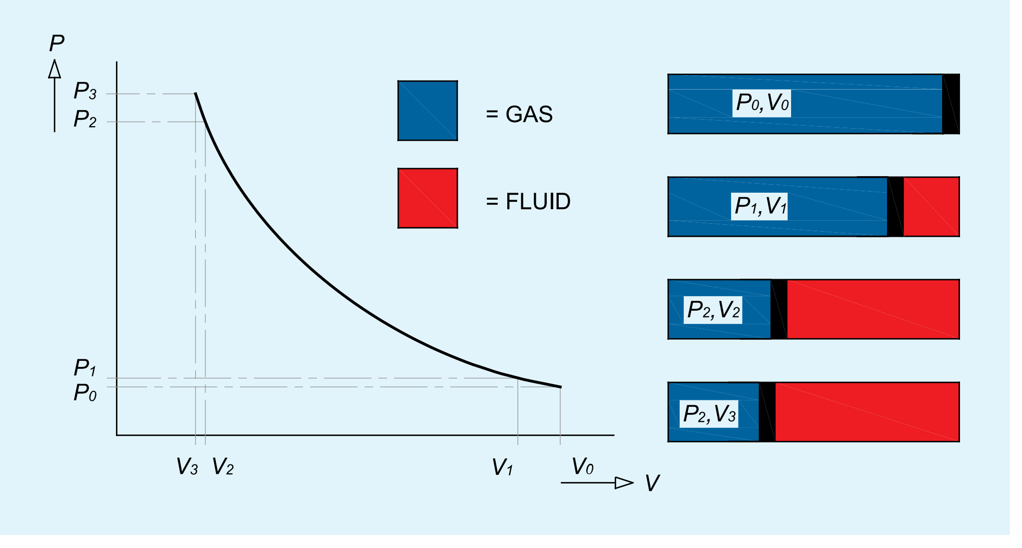

Fig 4.4.4.A Adiabatic pressure behaviour of a hydraulic accumulator

The pressure p_o is the initial pressure at which the gas in the accumulator is being charged with. p_1 is the minimum working pressure of the accumulator. p_2 is the maximum working pressure of the accumulator. p_3 is the pressure in the pressure relief valve that is often fitted to the accumulator. For power storing applications p_o is approximately 90% of the minimum pressure p_1.

The accumulator can supply a volume of $\Delta V = V_1 - V_2$ for the normal pressure range of $p_2 - p_1$. The value of

ΔV depends on the change of condition of the gas. It can easily be explained that the end pressure for a quick adiabatic expansion will be lower for an oil outflow of ΔV. This is because the temperature of the gas will reduce further than it would with a slow isothermal expansion of the gas. The change of the condition of the gas would in practice be somewhere between the two extremes of adiabatic and isothermal changes in condition.

If the condition changes slowly, with cycles of the order of 5 minutes or longer, the gas will be able to exchange heat with its environment (the accumulator wall and via the membrane to the hydraulic liquid). In these situations we talk about a so called isothermal process and ΔV can be calculated with the following formula:

$$\Delta V = V_1 \left(1 - \frac{p_1}{p_2}\right) \qquad 4.6)$$

In case of a fast expansion of the gas, with cycle times up to 30 s, no exchange of heat with the environment will take place. In those cases we talk about an adiabatic change of condition, for which the following individual calculation can be extrapolated from the ideal gas laws:

$$\Delta V = V_1 \left[1 - \left(\frac{p_1}{p_2}\right)^{\frac{1}{\kappa}}\right] \qquad 4.7)$$

The adiabatic gas constant κ is approximately 1,4 for diatomic gases like Nitrogen and Oxygen for temperatures between 20 and 40 °C and for pressure up to about 10 bar. The ideal gas law doesn't really apply to the behaviour of the gas at lower temperatures or higher pressures. In practice the formula is still applied in those conditions but higher values for adiabatic gas constant are used in these circumstances, see fig 4.4.4.B.

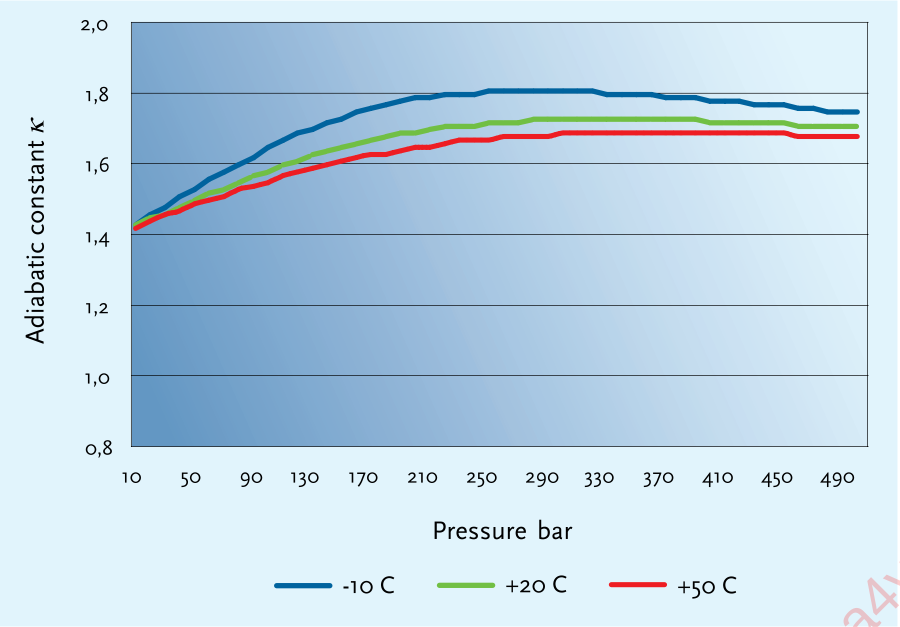

Fig 4.4.4.B Adiabatic gas constant κ at different temperatures and pressures (Source: NIST, REFPROP)

The gas pressure when filling the accumulator is usually set for a filling temperature of 20 °C. In the actual working environment the working temperature is often lower or higher. The pressure p_o at which the accumulator is filled will have a different value for these temperatures. The working $p_{o(oper)}$ can be calculated with:

$$p_{o(oper)} = p_0 \cdot \frac{T_{oper}}{273+20} \tag{4.8}$$

With:

T_{oper}	=	operating temperature	°K

From figure 4.4.4.A it is clear that the gas pressure rises steeply as the gas volume decreases. In a number of systems, like heave compensation systems this is not wanted. The gas volume can however be increased by adding a number of gas bottles and linking them to the gas system of the accumulator. The gas pressure will then rise less quickly for the same amount of oil flow ΔV. The gas pressure and thus the system pressure, will of course decrease more slowly than when the amount of oil in the accumulator decreases.

The characteristics of accumulator systems with extra linked gas volumes will be discussed in more detail in chapter 9.

In the description of equation 4.5 it has been mentioned that the temperature T of the gas is assumed to be constant during the process of compression or decompression of the gas. This is allowed only if we use the presented values for the adiabatic gas constant κ. The results for the calculated pressure p or volume V comply with actual measured data.

In real the temperature in the gas does change during the process of compression or decompression. The rate of change largely depends on the speed of the process, i.e. the cycle time of volume or pressure changes. Figure 4.4.4.C represents the measured temperature of the gas in an accumulator where the gas volume V changes from 150 litre to 200 litre in 2,5 seconds.

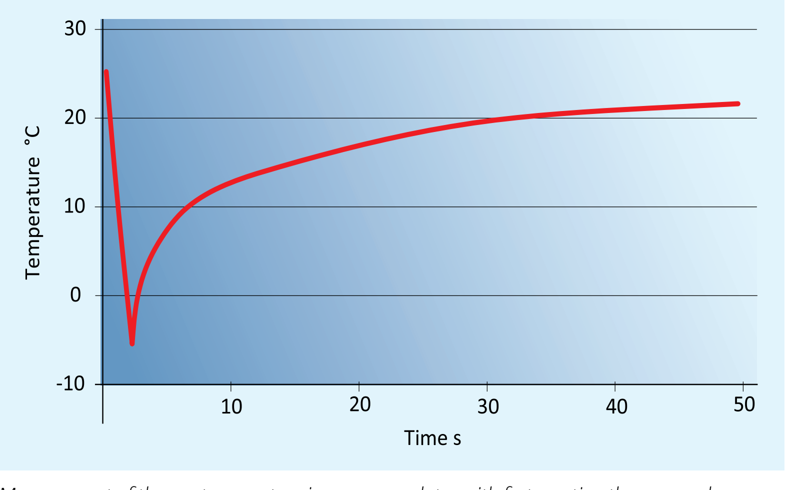

Fig 4.4.4.C Measurement of the gas temperature in an accumulator with fast reacting thermocouples.

The start temperature of the gas is 25 °C, equal to the ambient temperature. Due to the fast decompression of the gas in 2,5 seconds the temperature drops fast as well. After that the temperature slowly rises again because of the heat exchange with the ambient.

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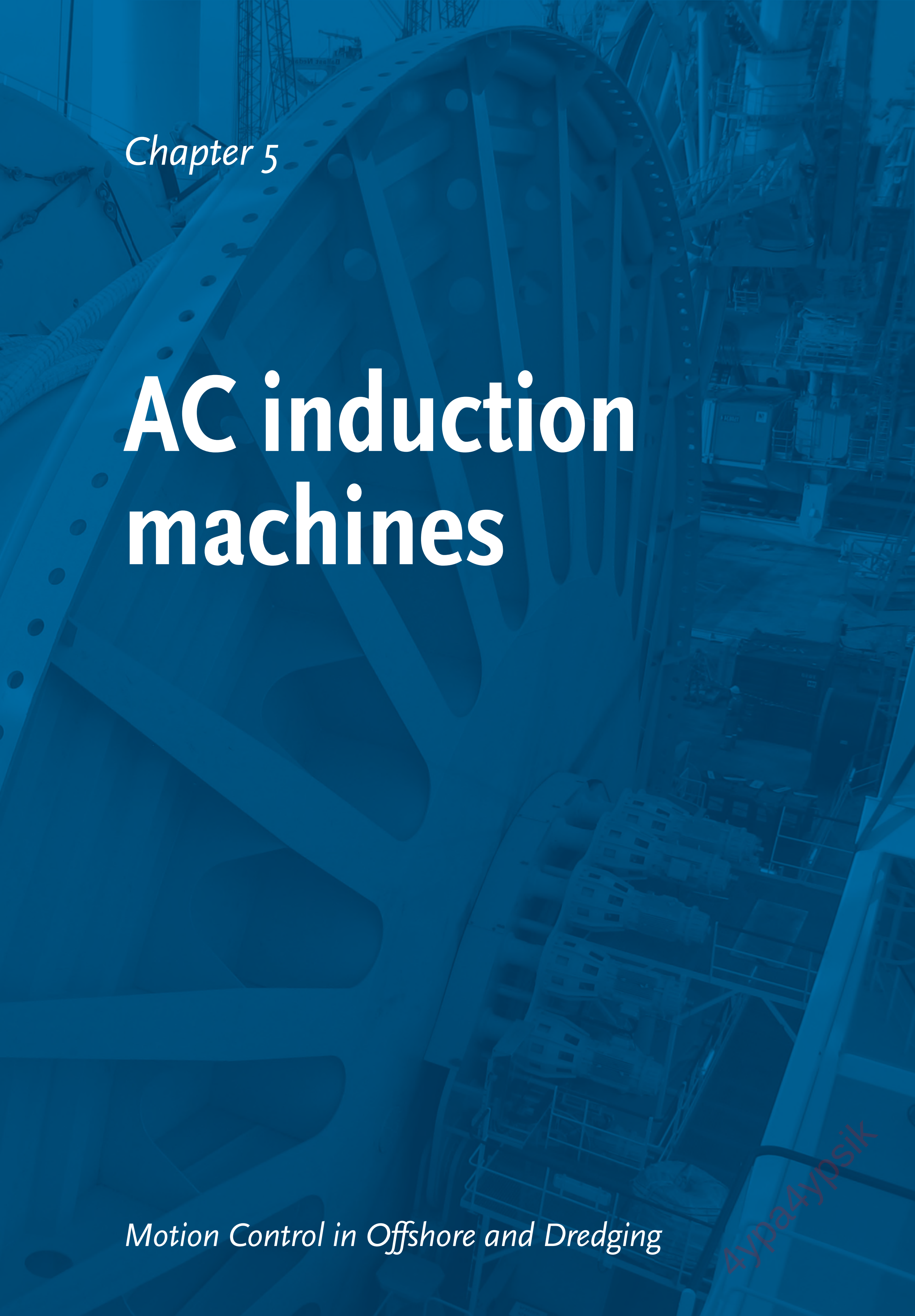


Chapter 5

AC induction machines

Motion Control in Offshore and Dredging

Chapter 5

AC induction machines

Electrical drives are used to convert electrical energy to mechanical energy. They all use a magnetic field as the medium for energy conversion. As we will see the conversion is reversible, just as it is with hydraulic drive systems.

The AC (alternating current) induction motor is the most widely used electrical drive in the shipbuilding, dredging and offshore industries. The induction machine can operate as a motor and as a generator. The induction motor is used in various sizes. In this chapter the operation and characteristic features of the three-phase induction machine are described in detail.

For those who do not yet understand the functions of the individual components of electrical systems but do have some knowledge of hydraulic systems the following analogy between hydraulic and electrical components might help.

P. Albers, *Motion Control in Offshore and Dredging*, DOI 10.1007/978-90-481-8803-1_5,

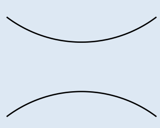
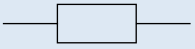
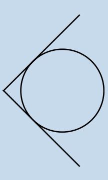
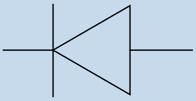
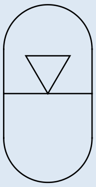
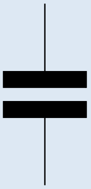
Hydraulic		Electrical	
Subject	**Symbol**	**Subject**	**Symbol**
Pipe		Wire	
Pressure		Voltage	
Flow		Current	
Orifice		Resistor	
Check valve		Diode	
Accumulator		Capacitor	

Fig 5 Two-Track tensioner for 350 mTon, each track is driven by an AC induction motor and frequency drive (Courtesy SAS)

5.1 Construction and principle of operation

The induction machine consists of a stator and a rotor. The stator is made up of laminates of high-grade sheet steel necessary for the magnetization. A three phase winding is put in slots cut on the inner surface of the stator frame as can be seen in figure 5.1.A.

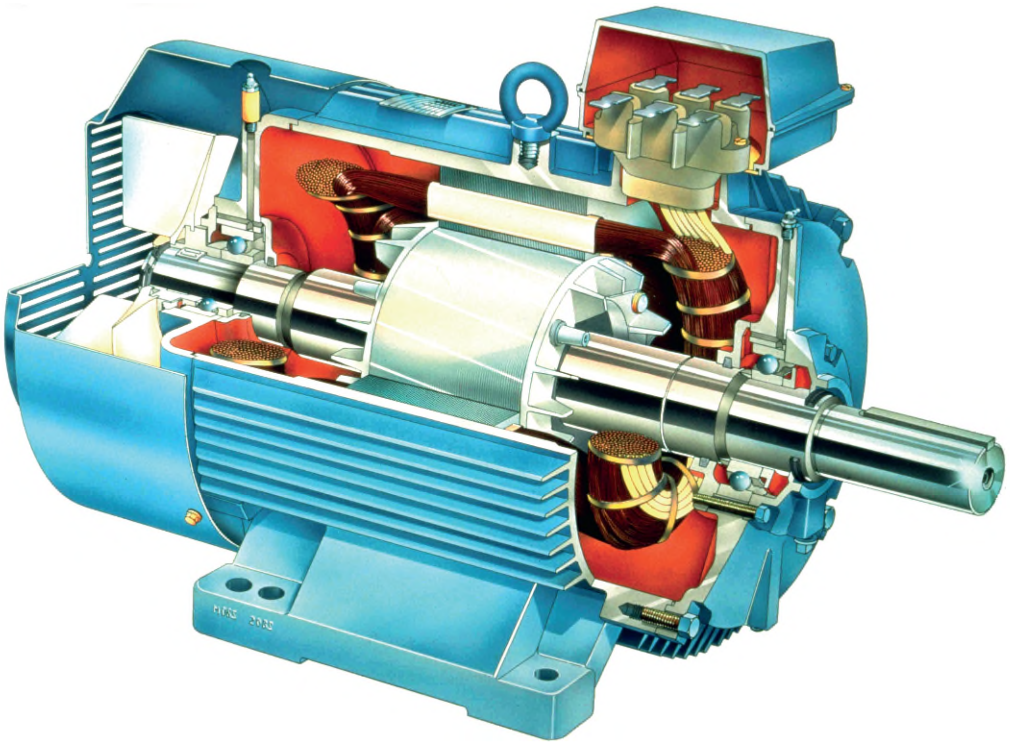


Fig 5.1.A Sectional view of a squirrel cage AC induction motor

The squirrel cage winding in the rotor consists of aluminum or copper bars embedded in the rotor slots and shorted at both ends by aluminum or copper end rings. The bars are not straight but have some skew to reduce noise and harmonics.

The field windings or 'poles' in the stator are powered by AC, creating a rotating magnetic field. The number of poles can vary but the poles are always in pairs (i.e. p = 2,4 or 6). The rotor winding current is generated by induction due to the relative speed of the rotor in the rotating magnetic field. Because the rotor is not powered externally, there is no need for slip-rings as for a DC motor.

The windings on the stator that carry the supply current to induce a magnetic field are distributed. The winding of each phase is distributed over several slots. The winding on the stator is for simplicity represented by three concentrated coils. See figure 5.1.B, coil aa' represents all the distributed coils assigned to the phase a-winding for one pair of poles. Similarly, coil bb' represents the phase b-winding, and coil cc' represents the phase-c distributed winding. The axes of these coils are 120 electrical degrees apart. The ends of these phase windings can be connected in a Y or in a Δ to form the three phase connection.

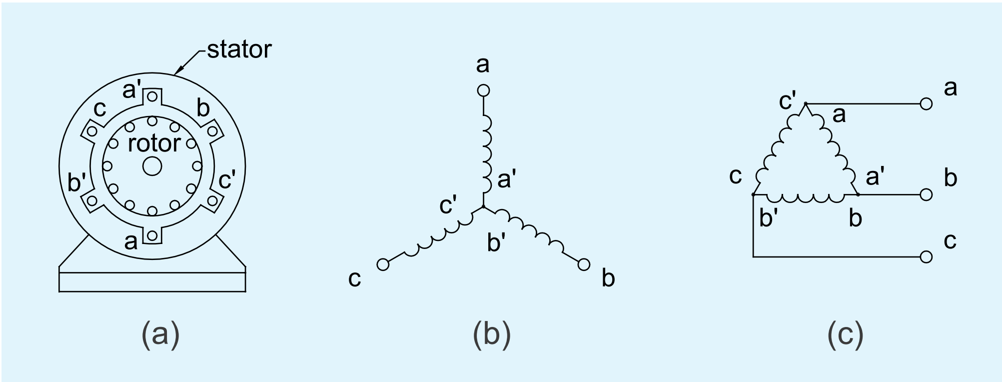


Fig 5.1.B *Three phase squirrel cage motor with its windings, (a) cross sectional view, (b) Y-connected stator winding, (c) Δ-connected stator winding*

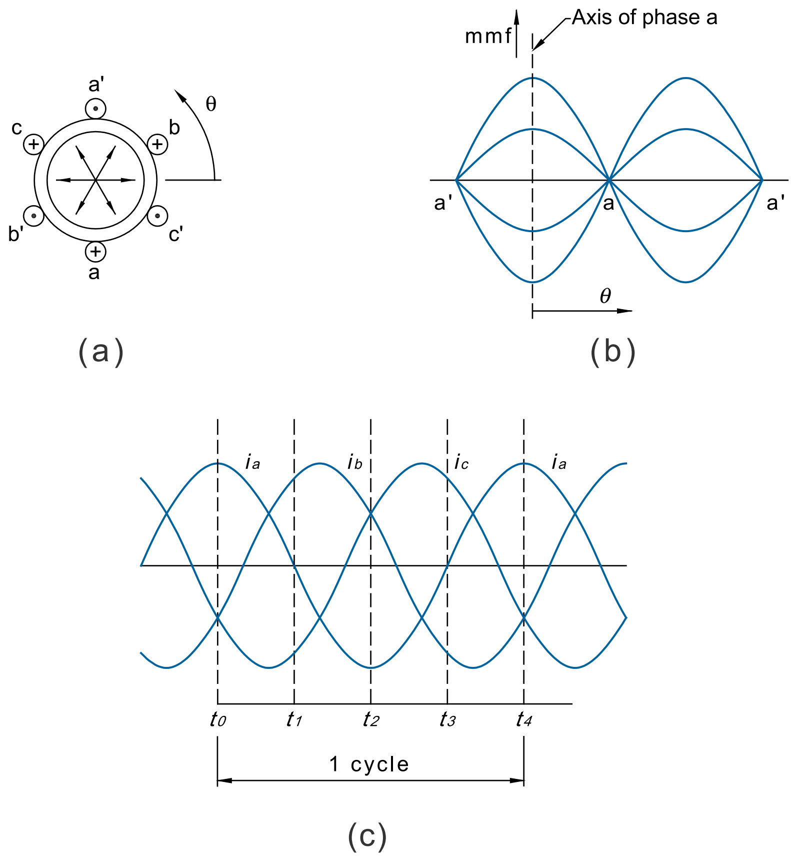


Fig 5.1.C *Three phase AC currents thru windings, (a) showing the current directions, (b) the mmf wave due to the current in coil aa', (c) a three phase current through the three phase windings*

When a current flows through a phase coil, it produces a sinusoidally distributed magneto motive force (mmf). If an alternating current flows through the coil, it produces a wave, whose amplitude and direction depend on the instantaneous value of the current flowing through the winding. Figure 5.1.C.b illustrates the mmf distribution in space at various instants due to an alternating current flow in coil aa'. Each phase will produce similar sinusoidally distributed mmf waves, but displaced by 120 electrical degrees in space from each other.

Let us now consider a balanced three-phase current flowing through the three-phase windings. The phase currents can be written as:

$$
\begin{aligned}
i_a(t) &= i.\cos(\omega.t) \\
i_b(t) &= i.\cos(\omega.t - \frac{2.\pi}{3}) \\
i_c(t) &= i.\cos(\omega.t - \frac{4.\pi}{3})
\end{aligned}
\tag{5.1}
$$

From figure 5.1.C c we can find that at instant t=to, the three currents become: ia = i, ib = -0,5i, and ic = -0,5i

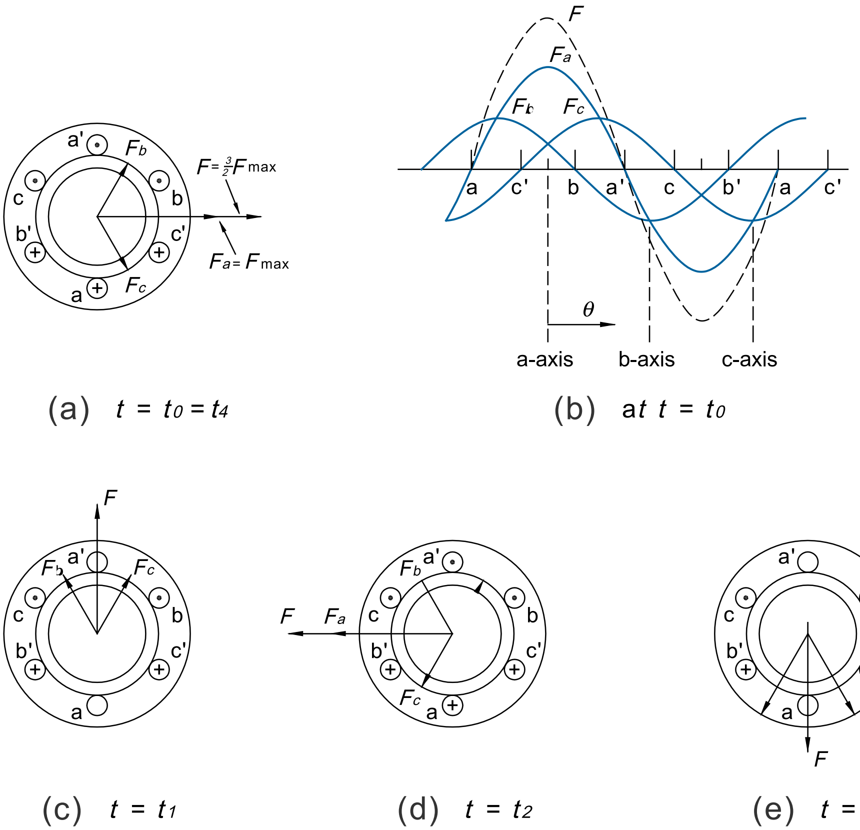


Fig 5.1.D Rotating magnetic field at various instants, (a, c, d and e) showing the vectors of the mmf at instants to – t4 , (b) the effect of the individual and total magnetic fields

The current directions in the representative coils in a 2-pole machine are shown in figure 5.1.C.a by dots and crosses. Because the current in the phase-a winding is at its maximum, the magnetic field F_a is at its maximum along the axis of phase a, also represented by a vector. The magnetic fields of windings b and c are shown by vectors F_b and F_c, each having a magnitude of 0,5F_a. The resultant of the three vectors is a vector with the magnitude of 1,5F_a, acting in the positive direction of the phase a axis. In the figure the currents and induced magnetic flux are shown for later instants of time t_1, t_2 and t_3.

It is obvious that as time passes the resultant magnetic flux retains its sinusoidal shape with constant amplitude, moving around the air gap. In one cycle of the current variation the resultant mmf vector makes one revolution. In a p-pole machine one cycle of variation of the current will make the magnetic flux wave rotate by *2/p* revolutions. From that we can conclude that the revolutions per minute *n* rpm of the travelling wave in a p-pole machine for a frequency *f* Hz are:

$$n_s = \frac{2}{p}.f_n.60 \tag{5.2}$$

with:

f_n	=	network supply frequency	Hz	n_s	=	nominal speed	rpm
p	=	pole number					

A reversal of the phase sequence of the currents in the windings makes the rotating magnetic fields rotate in the opposite direction.
The changing magnetic field pattern introduces currents in the rotor conductors. These currents interact with the rotating magnetic field created by the stator and the rotor will turn with the ability to provide output torque. However, for these currents to be induced, the speed of the physical rotor and the speed of the rotating magnetic field must be different, or else the magnetic field will not be moving relative to the rotor conductors and no currents will be induced.
The motor's mechanical output power can be calculated from speed and torque using the formula:

$$P_{out} = T \cdot \omega \tag{5.3}$$

or,

$$P_{out} = 0{,}1047 \cdot T \cdot n \tag{5.4}$$

with:

P_{out}	=	motor output power	kW	T	=	torque	Nm
n	=	motor speed	rpm	ω	=	angular speed	rad/s

The motor's input power P_{in} can be calculated from the voltage U, nominal current I_n and the power factor or $cos\phi_n$. The nominal current and power factors are taken from the manufacturer's specification data sheets.

$$P_{in} = \sqrt{3} \cdot U \cdot I_n \cdot \cos\phi_n \tag{5.5}$$

And the motor's efficiency η:

$$\eta = \frac{P_{out}}{P_{in}} \tag{5.6}$$

The slip s_n is defined at the motor's nominal point. At the nominal point the slip is:

$$s_n = \frac{n_s - n_n}{n_s}.100\% \tag{5.7}$$

Where:

n_s	=	the synchronous speed	rpm	n_n	=	speed at the nominal point	rpm

If the rotor is standing still, the slip is 100% and if it turns at the same speed as the magnetic frequency the slip is 0%. When a motor is running at its rated load (torque) the slip normally is in the order of 1-8%. The larger the motor, the smaller the slip will be.

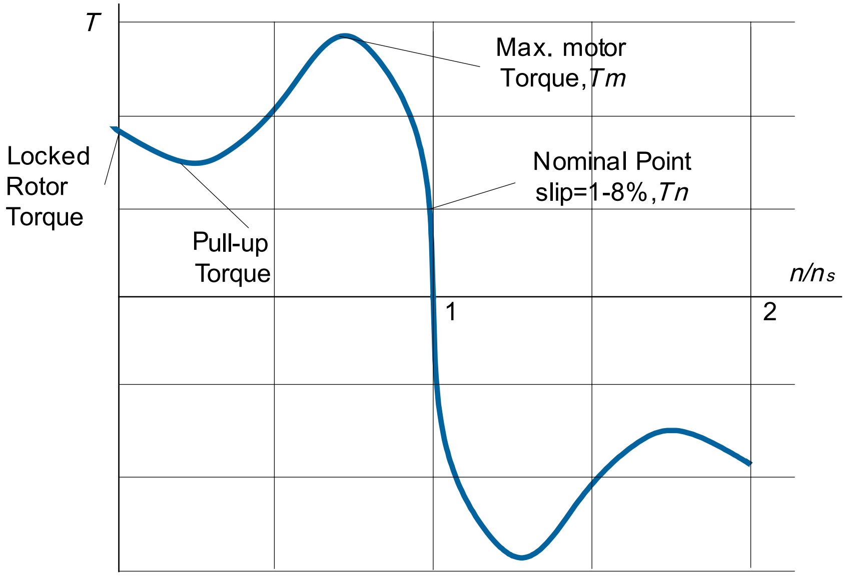


Fig. 5.1.E Torque/speed relation of an AC induction motor

The ability to provide torque depends on the speed of the rotor. At a slip value of 0% the torque is zero but with increasing slip the torque rises sharply almost proportional to the rate of slip. For the nominal slip the motor provides its nominal torque T_n . A standard induction motor's maximum torque T_m (pull-out torque) is normally 2-3 times the nominal torque. The maximum torque is available with slip s_{max}. At higher slip values the rotor inductance starts to play a role and the torque starts to fall.
The speed or frequency range below the nominal speed is called the constant flux range. Later we will see that varying frequencies can also be applied so that speeds below and above the nominal speed can be reached.

Why do we talk about the constant flux range? To determine the magnitude of the magnetic flux we will consider the equivalent of one stator winding.

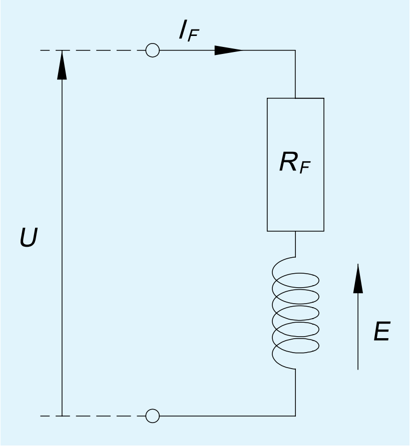


Fig 5.1.F Equivalent circuit of a stator winding

The applied voltage U causes a current I_F which generates the magnetic field. Due to the self inductance of the winding an EMF E is induced in each winding equals to:

$$E = C \cdot \Phi \cdot f \tag{5.8}$$

with:

C	=	constant		f	=	frequency	Hz
E	=	electro magnetic force	V	Φ	=	magnetic flux	V.s

The voltage U relation for the stator winding as shown in the above figure is:

$$U = I_F \cdot R_F + E \tag{5.9}$$

The winding's resistance is very low so the voltage and the induced EMF are almost equal:

$$U \approx E \tag{5.10}$$

And the magnetic flux Φ becomes:

$$\Phi = C.\frac{U}{f} = \text{constant} \tag{5.11}$$

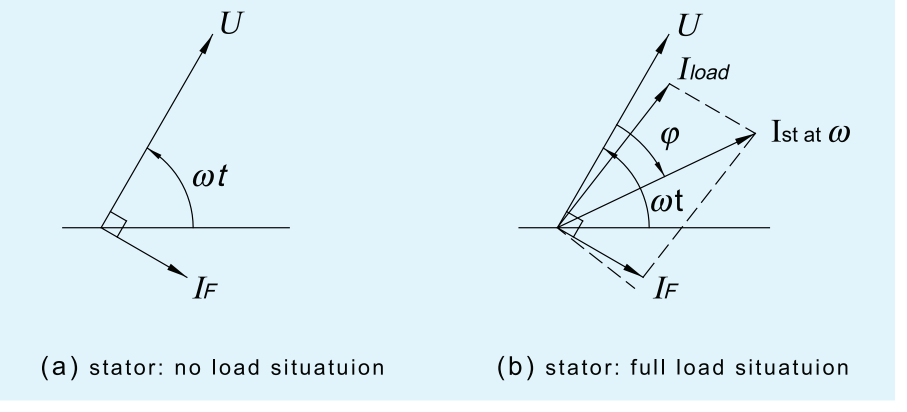


Fig 5.1.G Vector diagram of the stator

Figure 5.1.G shows the current that is supplied to the stator. In a no-load situation the only current that is present is the current to magnetize the stator (magnetizing or active current). Due to the behaviour of the field winding a phase lag of almost $\pi/2$ occurs. There is almost no real power because the power factor $\cos\phi$=0.

In the situation of a larger slip, a load current I_{load} appears (Load or reactive current). The load current has almost no phase lag with the applied voltage.

With the slip between 0% and the point where the maximum torque (pull-out torque) is provided the currents I_F and I_{load} can be approximated with:

$$I_{load} = I_n \left\{ \sin\phi_n + \cos\phi_n \cdot \left[\sqrt{\left(\frac{T_{\max}}{T_n}\right)^2 - 1} - \sqrt{\left(\frac{T_{\max}}{T_n}\right)^2 - \left(\frac{T_{load}}{T_n}\right)^2} \right] \right\} \tag{5.12}$$

and

$$I_F = I_n \cdot \frac{T_{load}}{T_n} \cdot \cos\phi_n \tag{5.13}$$

The total motor current can be calculated with:

$$I_{st} = \sqrt{I_{load}^2 + I_F^2} \tag{5.14}$$

It can be seen that with zero motor torque the active current is zero. With higher torque values the motor current I_{st} becomes directly proportional to the torque. A good approximation for the total motor current is:

$$I_{st} \approx \frac{T_{load}}{T_n} \cdot I_n, \text{ when } 0{,}8 \times T_n \leq T_{load} \leq 0.7 \times T_{max} \tag{5.15}$$

The basic parameters like power, $\cos\phi$, torques T_m and T_n and amperage I_n are usually specified at a voltage of 380 V and 50 Hz. In offshore applications higher voltages are usually used, up to 690V and frequencies of 60 Hz.

For other voltages V_N the power capacity P of the motors may be multiplied with a factor:

$$\frac{V_N}{380} \tag{5.16}$$

For a frequency of 60 Hz the nominal current I_n and torque T_n , as specified by the manufacturers at 50 Hz, are reduced by 3%. All remaining parameters like *cos*   remain the same.

5.2 Speed and torque control

We have seen that the nominal torque T_n of an AC motor is provided at the nominal speed n_s where the slip has a value of approximately 4-5%. At that point the motor works at its optimum efficiency. We have also found that the nominal speed is a function of the frequency of the power supply. The AC motor is therefore suitable for use in substantially constant-speed drive systems. Many industrial and offshore related applications however require several speeds or a continuously adjustable range of speeds. Traditionally DC motors have been used in such applications. However, DC motors are expensive, require frequent maintenance of the commutators and brushes, and are not allowed in a hazardous environment (atmosphere). The availability of solid state controllers has made it possible to use induction motors in variable speed drives.
The synchronous speed and hence the motor speed can be varied by changing the frequency of the supply. It requires a variable frequency drive.

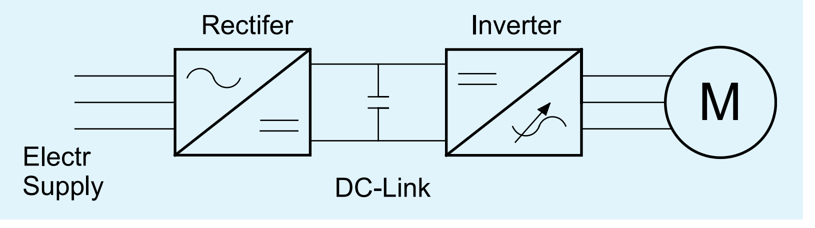


Fig 5.2.A A single frequency convertor

A typical AC drive system consists of an electric supply, a rectifier where the AC voltage is converted into a DC voltage , the DC-link and the Inverter unit where the variable frequency is being generated.
In a multi-drive system a common rectifier is often used where individual drive systems have their own inverter.

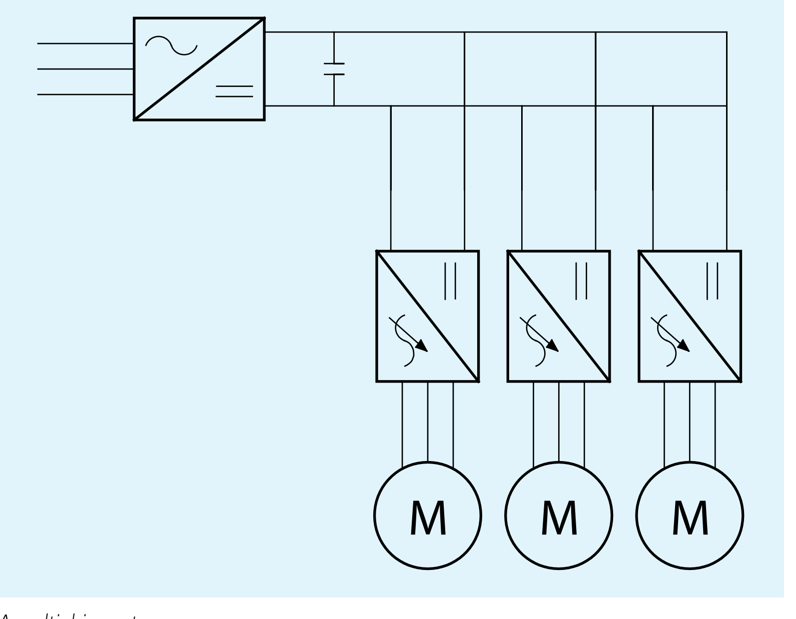


Fig 5.2.B A multi drive system

By changing the frequency of the electric supply of the AC motor the following typical torque-speed relation curves can be obtained.

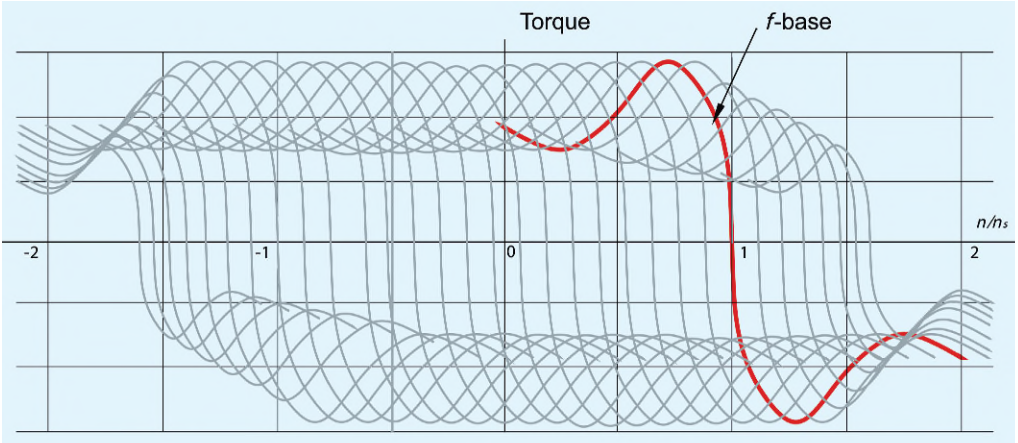


Fig 5.2.C Torque speed curves of an induction motor fed by a frequency converter

At the base frequency f_{base} the machine terminal voltage is the maximum that can be obtained from the inverter. Below this frequency (the constant flux range) the air gap flux is maintained constant by changing the *V/f* ratio so the maximum torque T_{max} in this region stays constant.

Above the frequency f_{base} (the field weakening range), since the voltage cannot be further increased with frequency , the air gap flux decreases and so does the maximum available torque. In the field weakening range constant horsepower operation is possible.

In the field weakening range the current components depend on speed. They can be calculated with:

$$I_{load} = I_n \left\{ \frac{n_n}{n} . \left(\sin\phi_n + \cos\phi_n . \sqrt{\left(\frac{T_{\max}}{T_n}\right)^2 - 1} \right) - \cos\phi_n \sqrt{\left(\frac{T_{\max}}{T_n} . \frac{n_n}{n}\right)^2 - \left(\frac{T_{load}}{T_n} . \frac{n}{n_n}\right)^2} \right] \tag{5.17}$$

and $$I_F = I_n \left(\frac{T_{load}}{T_n} . \frac{n}{n_n} \right) . \cos\phi_n \tag{5.18}$$

As before the motor current can be calculated with:

$$I_{st} = \sqrt{I_{load}^2 + I_F^2} \tag{5.19}$$

An approximation for the motor current can be used:

$$I_{st} = \frac{T_{load}}{T_n} . \frac{n}{n_n} . I_n = \frac{P_{load}}{P_n} . I_n \tag{5.20}$$

This approximation can only be used if: $$0,8 \cdot \frac{n_n}{n} \cdot T_n \leq T_{load} \leq 0,7 \cdot \left(\frac{n_n}{n}\right)^2 \cdot T_{\max} \tag{5.21}$$

and if : $$0,8 \cdot P_n \leq P_{load} \leq 0,7 \cdot \frac{n_n}{n} \cdot P_{\max} \tag{5.22}$$

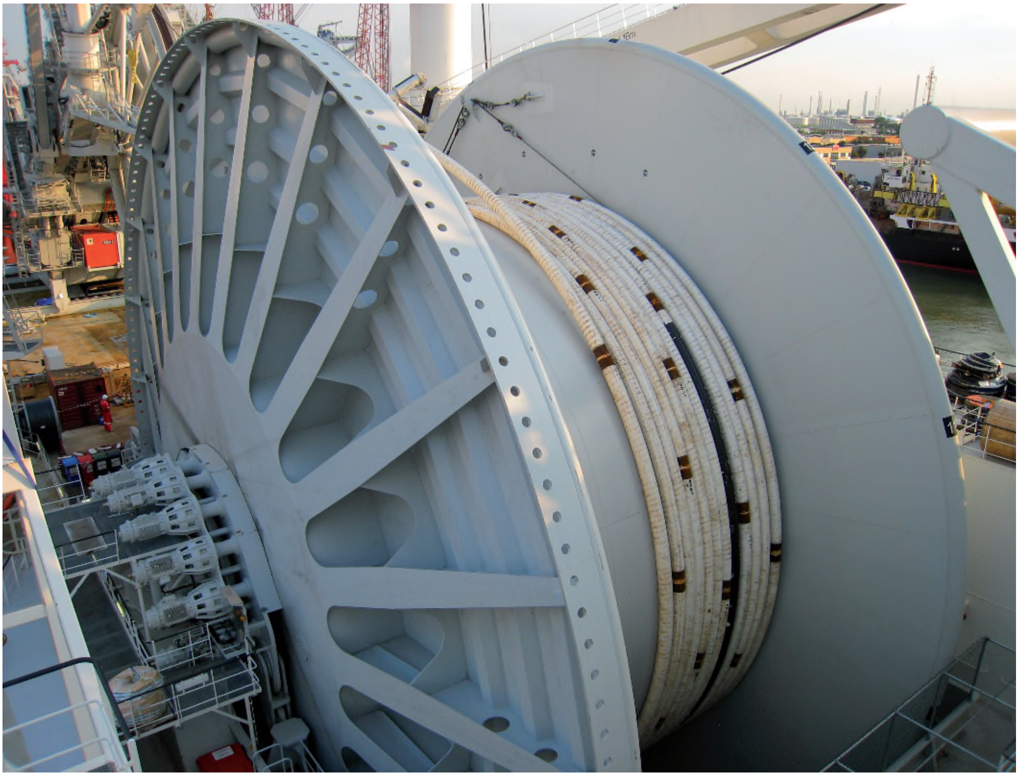


Fig 5.2.D. Electrical driven reel (AC induction motor with frequency drive) for the huge reel on the Seven Oceans of Subsea7 (Courtesy of Huisman)

5.3 Details of the AC-DC-AC

In the previous paragraph we have shown that the easiest way to create a variable speed with an AC motor is by using a rectifier, a DC link and an inverter.
In its simplest form, a rectifier consists of a number of diodes as shown in the diagram below.

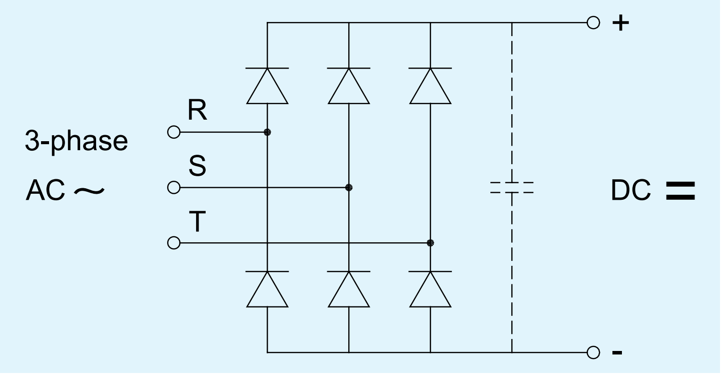


Fig 5.3.A Rectifier bridges for three-phase AC

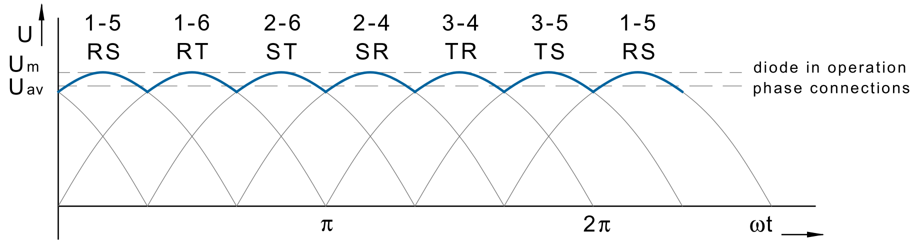


Fig 5.3.B Output voltage wave of three-phase rectifier bridge

Because the 3-phase feed gives a variable voltage, different diodes will be conductive whilst others are not. The output voltage of the diode bridge shows an almost constant voltage with a few ripples superimposed on top. A capacitor function in the DC link is required to remove the voltage ripples.

The average voltage U_{av} in the DC-link can be calculated with the following formula:

$$U_{av} = \frac{3}{\pi}.U_m = \frac{3}{\pi}.\sqrt{2}.U_e \approx 1.35.U_e \qquad (5.23)$$

with:

U_m	=	voltage amplitude of single phase supply	V	U_e	=	effective voltage of single phase supply	V

The constant flux control, or the ratio U/f forms an important basis for the conversion of the direct voltage into a variable frequency by the inverter. If the frequency f of the current to the motor changes, then the voltage U to the stator windings needs to be adapted accordingly.

Both voltage and frequency reference are fed into a modulator which simulates an AC sine wave and feeds this to the motor's stator windings. This techniques is called Pulse Width Modulation (PWM). Because there is no need to feed back the speed or the position of the motor axle, this type of control is also known as an open-loop drive. The advantage is the low cost of the frequency drive. The disadvantages are that the torque of the drive cannot be controlled accurately and that instability can occur at low speeds. This type of control is very suitable for applications such as pumps, fans, conveyors, mixers and centrifuges where the operating speed or torque does not need to be controlled very accurately. If , in addition to the PWM control, a speed or position encoder is being used to feed back the speed or position of the motor shaft the accuracy can be improved.

A different form of control is the flux vector control. In this case both the position of the motor axle and the speed can also be fed back to the inverter. Because of the feedback, this drive is called a 'closed-loop drive'. In this type of control the characteristics of the motor are mathematically modeled in the micro processors supplied with the inverter. As for a PWM drive with its flux control, the voltage, the frequency and the current are controlled independently of each other. A very good torque response, the possibility of a maximum torque when the motor is standing still and accurate speed control are the advantages of this type of control. The disadvantages are the need for feedback of both the motor axle position and the number of revolutions per minute. For only speed control a simple speed encoder suffices to control the slip of the drive.

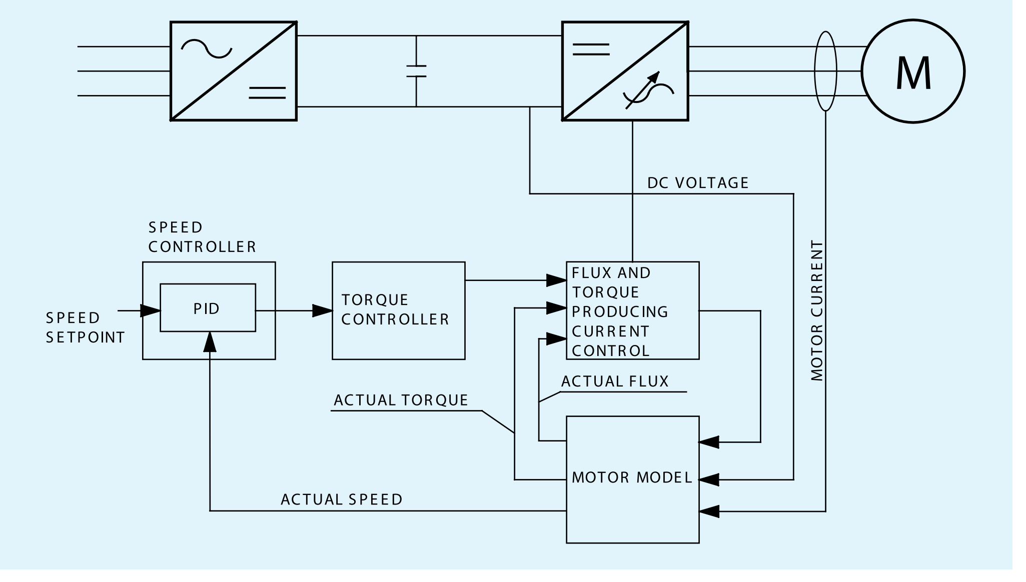


Fig 5.3.C Block diagram of DTC Direct Torque Control drive

A further improvement of the control behaviour can be achieved by the application of so-called DTC ('Direct Torque Control'). In this case an on-line model of the motor behaviour is extracted from the connected motor. This is called 'Auto-Tuning'. Data like the stator resistance, the mutual inductance, saturation co-efficients and motor inertia are applied live to the motor model. No feedback is required of the motor axle position and the number of rotations per minute of the axle. These data are simulated in the motor model and subsequently applied in the control circuit as simulated values. The position accuracy at low speeds with only this DTC is not good enough for a winch drive. Instability may occur as well. For a winch drive it should be able to control the hook of a crane at a constant and stable position. To achieve this the additional installation of a motor shaft position feedback transducer is necessary with also an additional feedback loop in the control circuit.

With these controls of the inverter, supported by computer programmes, it is possible to achieve speed accuracies to within +/- 0,5% and torque responses within 2 msec. The behaviour of the controls, including the linked AC motor, is such that we can talk about a true torque control. Such torque controls are also available in hydraulic drive technology. This means that there is, in terms of drive control technology, no longer a difference between a hydraulic or an electrical drive for, for example, a rotating drive like a winch.

There are still differences in the details of a hydraulic and an electrical drive. This applies, amongst other things, to the dynamic behaviour. This will be discussed in more detail in the following chapters.

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Chapter 6

Control technology

Motion Control in Offshore and Dredging

Chapter 6

Control technology

It is possible that using a hydraulic or an electrical actuator to drive a machine will give a 'good' result in many situations. In these cases 'good' is a qualitative statement describing situations where the number revolutions per second, the speed, power or torque (or a combination of these values) provide a satisfactory operational result for the chosen drive mechanism. Often no qualitative constraints are stipulated for the accuracy of the speed, output force, torque or dynamic behaviour of the machinery. All these drives are so-called controlled or open-loop drives. There is no feedback mechanism from the main controlled variable. Many users would, initially, rather not have a feedback mechanism because for them it means that the delivery will include extra sensors and a mysterious 'black box' with an electronic control mechanism.

When we discuss the practical application of especially hydraulic drives, we will give different examples of their 'open-loop' versions.
A controlled drive does however often show variations about the norm in its behaviour. For a hydraulic drive this can be due to the internal leakage of a control valve or a hydraulic motor, which will result in a variation in the set number of rpm as the load increases. For a frequency controlled AC-Motor a variations in the set speed can be caused by the imbalance between the torque that needs to be delivered and the flux sent out by the inverter.

Control technology offers solutions that allow us to compensate for variations that develop in the controlled machinery system. This is often achieved by feeding back the value for the controlled variable. Some readers will already be fairly familiar with the characteristics of controlled drive mechanisms. For those who don't have this knowledge yet, we will spend some time discussing conventional control technology with the help of block diagrams and the so-called Laplace transformations. The advantage of applying control technology in this way is that a block in a block diagram does indeed represent an actual machine or component of the drive mechanism.

In this chapter the basis of the control technology is explained using a hydraulic drive as a basis. Later on we will also give examples of how the same control technology can be applied in electrical drive mechanisms.

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6.1 List of symbols

A	=	*area*	m^2
C	=	*mechanical stiffness*	N/m
C_o	=	*oil stiffness*	N/m
C_H	=	*hydraulic stiffness*	N/m
E	=	*oil elasticity*	N/m2
F	=	*force*	N
H	=	*loop gain*	
I	=	*moment or inertia*	$kg.m^2$
K	=	*individual gain*	
K_f	=	*friction coefficient*	$N.s/m$
K_v	=	*velocity gain*	$1/s$
L	=	*length*	m
M	=	*mass*	kg
p	=	*pressure*	N/m^2
Q	=	*flow*	m^3/s
S	=	*stroke*	m
T	=	*torque*	$N.m$
V	=	*volume*	m^3
V_p	=	*line volume*	m^3
i	=	*imaginary unit*	
r	=	*vector length*	
s	=	*laplace operator*	
t	=	*time*	s
z	=	*complex number*	
β	=	*damping ratio*	
ε	=	*error signal*	
ϕ	=	*phase*	rad
τ	=	*time constant*	s
τ_i	=	*integration time*	s
ω	=	*frequency*	rad/s
ω_o	=	*natural frequency*	rad/s

6.2 Block diagram and laplace transformation

The speed of a cylinder can be regulated by changing the fluid flow that goes to the cylinder by means of a proportional valve or a variable volume pump. In that case we talk about a regulator, not about a control mechanism. When we regulate, we do not consider the output variable that is being realised (speed, position or force).

In a block diagram there is an indication for each block as to how the input signal is transformed into an output signal. In the following example, the output signal is achieved by amplifying the input signal with a constant factor *K*. The block diagram represents a cylinder that is regulated with the help of an amplifier and a control valve.

If we measure the output variable, in the case of figure 6.2.A the position of a piston, and compare the value with the desired output signal, we talk about control. The error signal ε is the difference between the desired signal and the measured signal.

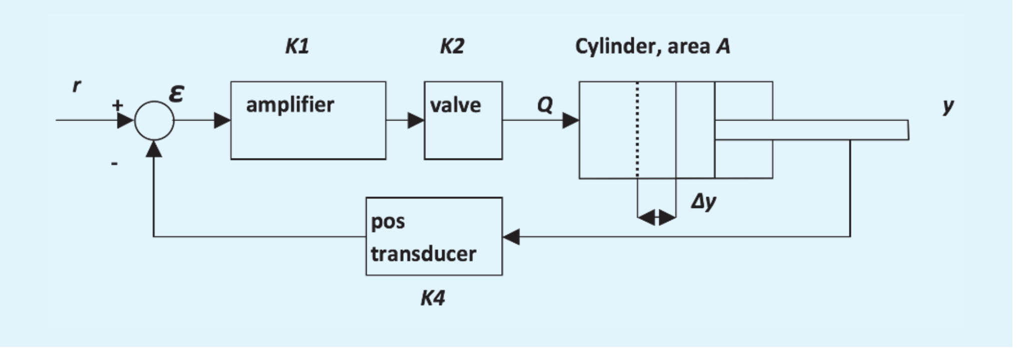


Fig 6.2.A. Block diagram of a simple position control problem

The volume flow from the proportional valve over a short period of time Δt, results in a piston movement of Δy. During the same period of time the volume of oil added to the cylinder is ΔV m³. The surface area of the cylinder is A m².

We get: $$Q.\Delta t = \Delta V = A.\Delta y \qquad (6.1)$$

or: $$Q = A.\frac{\Delta y}{\Delta t} = A.\frac{dy}{dt} \qquad (6.2)$$

The position of the piston can be found by integration of the speed against time. This means that in the block diagram, the cylinder can be represented by the blocks shown in figure 6.2.B. Each block shows how its input signal is transformed into an output signal with the use of an amplifier. The blocks can be merged into one new block with an amplification equal to the multiplier of the individual amplifications.

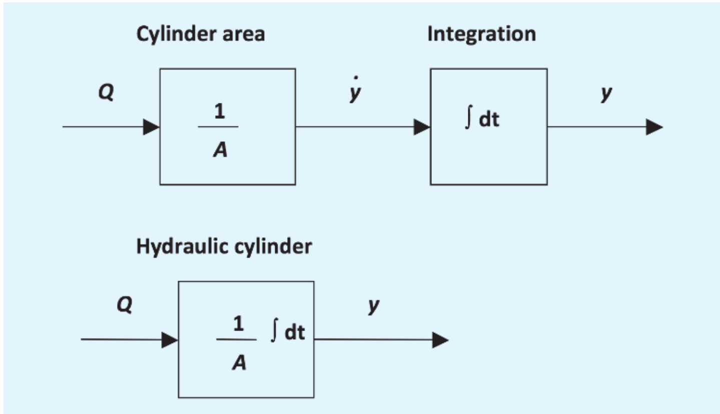


Fig 6.2.B Joining two blocks to one combined block

The differential operator *d/dt* can also be described using the 'Laplace-operator' s. As a result of this, the integration operator *∫dt* is written as *1/s*.

Note: The 'Laplace-operator' is a way of transforming a time-based differential equation into a normal algebraic one. The Laplace-operator is a complex variable, where s=i. We will discuss this in more detail in one of the following paragraphs.

The already simplified block for the hydraulic cylinder can now be simplified even further and be represented with $K_3 = 1/A$ by:

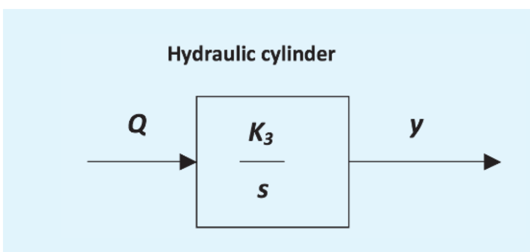


Fig 6.2.C Applying the Laplace operator for a cylinder in the block diagram

If we include the latest, simplified, block for the hydraulic cylinder into the block diagram of the feedback system we get the following result:

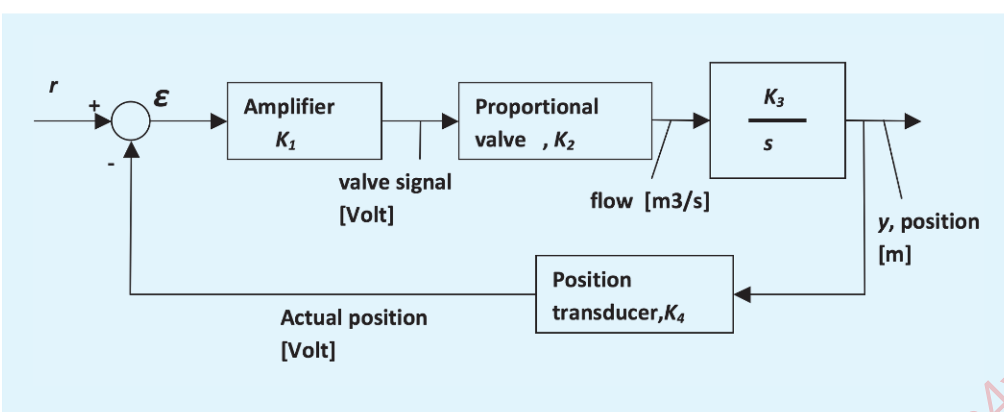


Fig 6.2.D Block diagram with its physical input and output parameters

A simple adjustment of this block diagram transforms this into a block diagram with a so-called unity feedback.

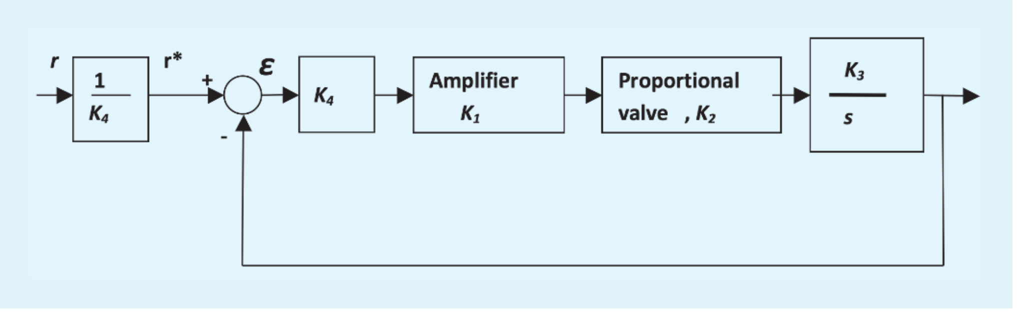


Fig 6.2.E Transformation into a unity feedback control system

The transfer function y/r* can be calculated as follows:

If $$y = K_1 \times K_2 \times \frac{K_3}{s} \times K_4 \times \varepsilon \tag{6.3}$$

and $$\varepsilon = r^* - y \tag{6.4}$$

Then: $$y = K_1 \times K_2 \times \frac{K_3}{s} \times K_4 \times r^* - K_1 \times K_2 \times \frac{K_3}{s} \times K_4 \times y \tag{6.5}$$

Or: $$\frac{y}{r^*} = \frac{K_1 \times K_2 \times \frac{K_3}{s} \times K_4}{\left(1 + K_1 \times K_2 \times \frac{K_3}{s} \times K_4\right)} = \frac{\overset{0}{H}}{1 + \overset{0}{H}} = \frac{\frac{K_V}{s}}{1 + \frac{K_V}{s}} = \frac{K_V}{s + K_V} = \frac{1}{\tau . s + 1} \tag{6.6}$$

The transfer $\overset{o}{H}$ is the multiplier of all the individual transfer functions in the circuit.
This calculation shows two things:

If the position of the cylinder is controlled with a feedback system, we have a block diagram of a 1st order system. Feedback of the cylinder position is the same as feedback from an integrator. All amplifier factors K_1 to K_4 can be combined into a new amplifier factor K_V.

The transfer value of *y/r** can be easily and quickly calculated by dividing the amplification of the circuit $\overset{0}{H}$ by the amplification $(1 + \overset{0}{H})$.

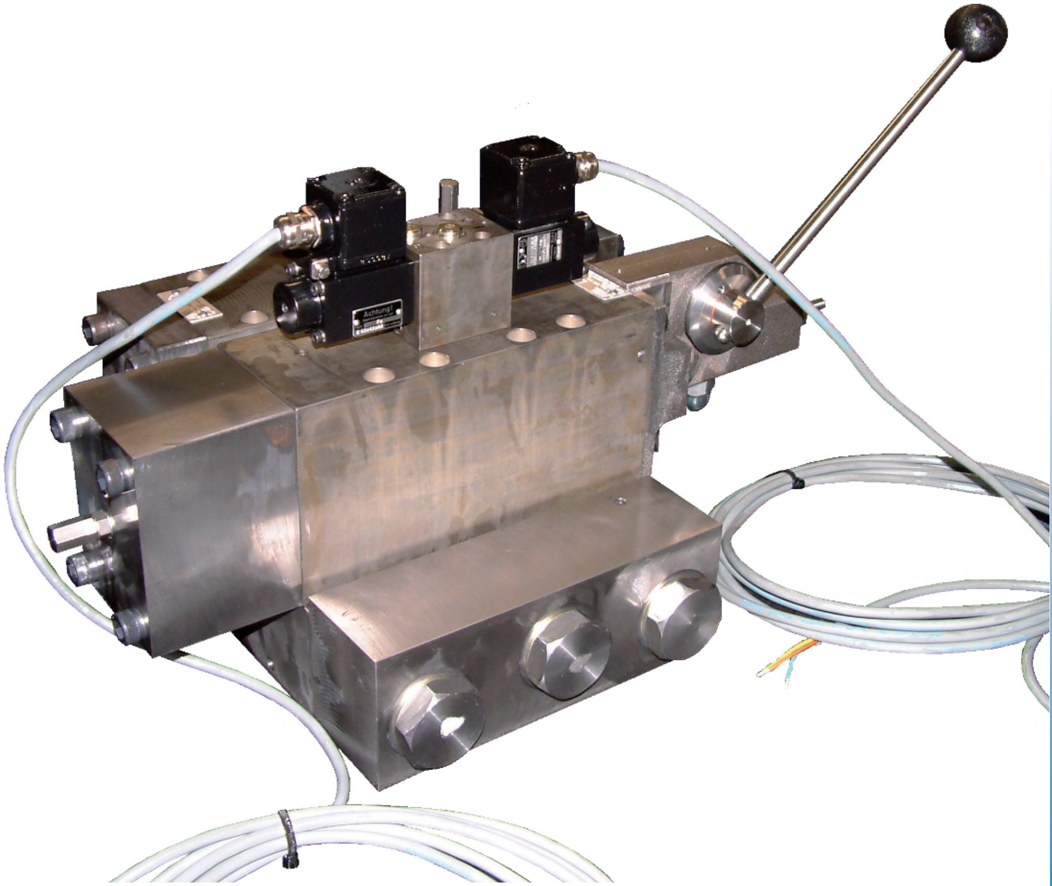


Fig 6.2.E A proportional hydraulic valve for offshore and ATEX environment that may be used to control the position of a hydraulic actuator (Courtesy of Amca)

6.3 1st Order system, step response

In the previous paragraph we proved that the block diagram of a cylinder with feedback of its position can be described as a transfer function of a 1st order system.

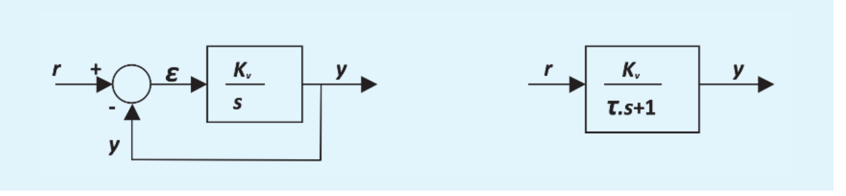


Fig 6.3.A A position feedback controlled cylinder can be represented as a first order system

The error ε is the actual error that occurs in a system with feedback. This is after all the difference between the planned and the actual position. The error ε can be approximated as follows:

From the block diagram we know that:

$$\frac{y}{\varepsilon} = \frac{K_v}{s} \tag{6.7}$$

or

$$\varepsilon = \frac{s.y}{K_v} = \frac{\frac{dy}{dt}}{K_v} = \frac{\dot{y}}{K_v} \tag{6.8}$$

The error that occurs is thus smaller the greater the amplifier factor in the circuit. In a control mechanism the amplification factor is therefore set as large as possible so that the occurring error is kept as small as possible. When we discuss higher order systems, we will show that limits have to be set for the amplification factor K_v because the controlled system will otherwise show unstable behaviour.

We get the following 1st order differential equation as the factor s can be written as *d/dt*:

$$\frac{y}{r} = \frac{\frac{K_V}{s}}{1+\frac{K_V}{s}} = \frac{K_V}{s+K_V} = \frac{1}{\tau.s+1} \tag{6.9}$$

With:

$$\tau = \frac{1}{K_V} \tag{6.10}$$

or:

$$(\tau.s+1).y = r \tag{6.11}$$

or:

$$\tau.\frac{dy}{dt} + y = r \tag{6.12}$$

The step response can also be found directly by solving the differential equation. The solution for the 1st order differential equation is:

$$\frac{y}{r} = 1 - e^{-\frac{1}{\tau}.t} \tag{6.13}$$

The easiest way to find the step response of a 1st order system is by looking at the start-up speed at the time t=0. Because the cylinder still has to start moving at that point in time, y = 0 is true too. This has the following effect on the differential equation:

$$\tau.\frac{dy}{dt} + y = r \tag{6.14}$$

if y = 0 then: $\tau.\dfrac{dy}{dt} = r$ (6.15)

and the start-up speed of the cylinder becomes: $\dfrac{dy}{dt} = \dfrac{r}{\tau}$ (6.16)

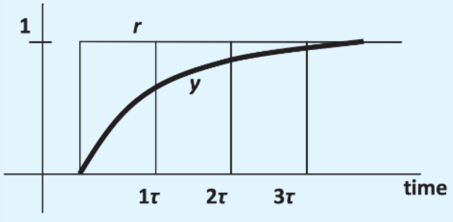
It is also quite easy to look at the static situation of the feedback where the speed *dy/dt* = 0.

$$\tau.\frac{dy}{dt} + y = r \tag{6.17}$$

with: $\dfrac{dy}{dt} = 0,$ y = r (6.18)

The response from a cylinder with a positional feedback (the response of a 1st order control system) to an input with a step size 1 looks as follows. The output signal reaches a value equal to 63% of the input signal after a time factor.

Over the next time factor τ a level of 63% + 63% of the remainder = 0,63 + 0,63 * 0,27 = 0,86 will be reached. And so on..



time	y
1τ	0,63
2τ	0,86
3τ	0,95

Fig. 6.3.B. Step response of a first order system

6.4 Elasticity of hydraulic fluids

A hydraulic motor or cylinder can be controlled with a variable pump, a proportional valve or a servo valve. The volume flow that is sent to the hydraulic motor or cylinder then determines the number of revolutions per second or the speed with which the piston moves respectively. Up to now we have assumed that there is a direct correlation between the volume flow Q and the number of revolutions per second or the speed, in other words, it is assumed that the hydraulic liquid is not compressible.
Hydraulic liquid is however most certainly compressible. This is expressed in the following formula:

$$\frac{\Delta V}{V} = -\frac{\Delta p}{E} \qquad (6.19)$$

where:

ΔV	=	decrease in volume	m^3	Δp	=	increase in pressure	N/m^2
V	=	original volume	m^3	E	=	modulus of elasticity of the liquid	N/m^2

The theoretical value for the modulus of elasticity E for mineral oil is approximately 1,6 x 10^9 N/m^2. For-comparison purposes, the modulus of elasticity for water is much higher at 2,4 x 10^9 N/m^2.

The effect can also be explained with the help of the 2 situations outlined in figure 6.4.A. The pressure on the cylinder housing increases in both situations. A rise of the oil pressure in a hydraulic cylinder will result in a compression of the oil (reduction of volume) and an expansion of the housing due to the elasticity of the material. You will need to remember though that the modulus of elasticity of steel with a value for *E* of 220 x 10^9 N/m^2 is more than 100 times higher than the value of *E* for mineral oil. This means that the increase in volume due to the expansion of the housing is negligible relative to the reduction in volume of the mineral fluid. The same rule applies to the hydraulic pipes and other components. If however hoses are used then the expansion of the hose is of significance. These extra effects on the elasticity are incorporated in the calculation by using a lower value of 1,0 x 10^9 N/m^2 or 10.000 bar. An important rule of thumb:

If the pressure increases by 100 bar and the value of *E* is 10.000 bar, then the reduction in volume is $^{100}/_{10,000} = 1\%$.

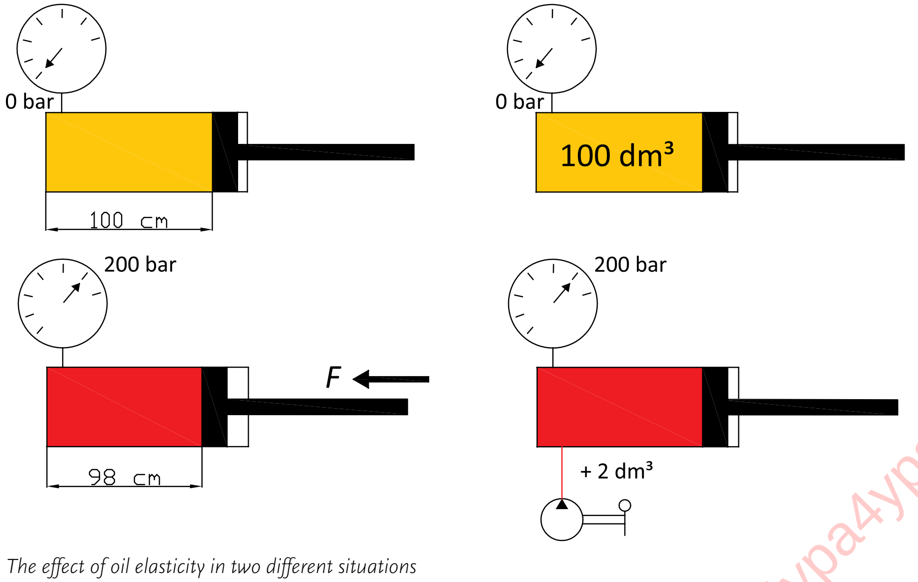


Fig 6.4.A The effect of oil elasticity in two different situations

On the left hand side a load is applied to the cylinder, which causes the pressure in the cylinder to rise to 200 bar. The displacement of the cylinder must then be 2% or 2 cm.
In the example on the right hand side, an extra 2 dm^3 is pumped into a vessel (cylinder) with an original volume of 100 dm^3. In that case the pressure increase is 200 bar.

6.5 Stiffness of a cylinder

Because of the elasticity of the oil, a hydraulic cylinder behaves like a mechanical spring with a stiffness of C_o.

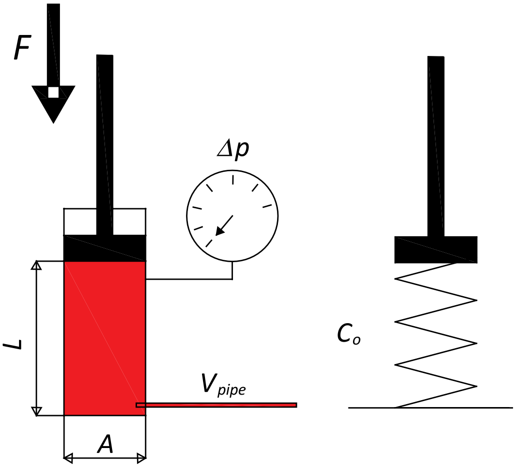


Fig 6.5.A Stiffness C_o of a hydraulic cylinder

Under the influence of an external force ΔF the displacement of the piston will be ΔL. The stiffness of a spring is defined as:

$$C_o = \frac{\Delta F}{\Delta L} \tag{6.20}$$

where:

$$\Delta L = \frac{\Delta V}{A} \tag{6.21}$$

and:

$$\Delta F = \Delta p \cdot A \tag{6.22}$$

and:

$$V = A \cdot L + V_{pipe} \tag{6.23}$$

and:

$$\Delta V = \frac{\Delta p \cdot V}{E} \tag{6.24}$$

As a result of which we get:

$$C_o = \frac{\Delta F}{\Delta L} = \frac{\Delta p \cdot A}{\frac{\Delta V}{A}} = \frac{\Delta p \cdot A^2}{\frac{\Delta p \cdot V}{E}} = \frac{A^2 \cdot E}{A \cdot L + V_{pipe}} = \frac{A \cdot E}{L + \frac{V_{pipe}}{A}} \tag{6.25}$$

with:

A =	area of cylinder	m²	L =	length	m
C_o =	oil stiffness	N/m	Δp =	pressure	N/m²
E =	fluid elasticity	N/m²	V =	volume	m³
F =	external force	N			

The stiffness of a single acting cylinder will therefore increase if the surface area A increases and decrease if the stroke L and the volume V_{pipe} of the pipework linked to it increases.

6.6 Stiffness of a hydraulic motor

As with the hydraulic cylinder, the stiffness of a hydraulic motor can also be calculated. The calculation is more or less the same. For the rotational stiffness a different set parameters of the drive unit need to be used.

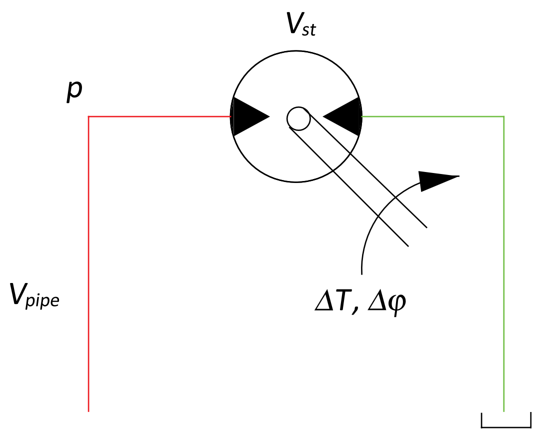


Fig 6.6.A Stiffness C_o of a hydraulic motor, note that only one port is at high pressure , the outlet port is connected to tank

Torsion stiffness $$C_o = \frac{\Delta T}{\Delta \varphi} \tag{6.26}$$

with $$\Delta V = \frac{\Delta p \cdot V}{E} \tag{6.27}$$

$$\Delta T = V_{st} \cdot \Delta p \tag{6.28}$$

$$V = \frac{\pi}{2} \cdot V_{st} + V_{pipe} \tag{6.29}$$

$$\Delta V = \Delta \varphi \cdot V_{st} \tag{6.30}$$

we get: $$C_o = \frac{V_{st} \cdot \Delta p}{\frac{\Delta V}{V_{st}}} = \frac{V_{st}^{\ 2} \cdot \Delta p}{\frac{\Delta p \cdot V}{E}} = \frac{2 \cdot V_{st}^{\ 2} \cdot E}{\pi \cdot V_{st} + 2 \cdot V_{pipe}} \tag{6.31}$$

with:

C_o =	torques stiffness	Nm/rad	V_{pipe} =	volume of piping	m³
ΔT =	External torque	Nm	$\Delta\varphi$ =	rotation angle	rad
V =	total volume	m³			
V_{st} =	stroke volume motor	m³/rad			

6.7 Results of stiffness calculations

Because of the elasticity of the hydraulic liquid, a stiffness C_o can be assigned to a hydraulic motor or cylinder. In the previous examples only one piece of pipe, connected to the hydraulic motor or the cylinder, was at pressure. In practice however there is always a return pipe too. If there is no pressure in the return pipe or if the pressure in the return pipe is low, then this pipe has no influence on the calculation. In a number of situations there can be pressure on the return pipe too. In those cases that pipe will cause a second stiffness factor. The stiffness C_o has been calculated for a number of different situations.

Only those parts of the pipe system that are directly linked to the actuator and in which the pressure varies as a result of load variations are incorporated in the volume calculation of the pipes. For example, the pipes between a pump and a proportional valve or servo valve are usually under the same, constant, pressure and are therefore not included in the volume calculation of the pipes.

For a cylinder it must be emphasized that the stiffness dependent is on the position of the piston. This was also the case for the example where a single pipe was attached. The total stiffness of the cylinder shows a minimal value $C_{o,min}$. In these examples it is that minimal value that is always mentioned.

BOTTOM END IS DRIVEN

$$C_o = \frac{A^2 \cdot E}{A \cdot L + V_{pipe}} \qquad (6.32)$$

In this case the connection to the rod side and input pipe of the valve are under constant pressure and are not included in the pipe work volume V_{pipe}

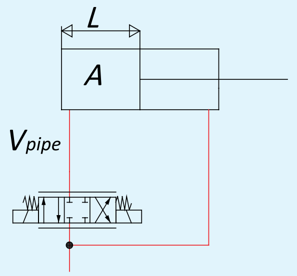


Fig.6.7.A

BOTTOM AND ROD END ARE DRIVEN

$$C_{o,\min} = \frac{E \cdot \left(\sqrt{A_1} + \sqrt{A_2}\right)^2}{S + \frac{V_{p1}}{A_1} + \frac{V_{p2}}{A_2}} \qquad (6.33)$$

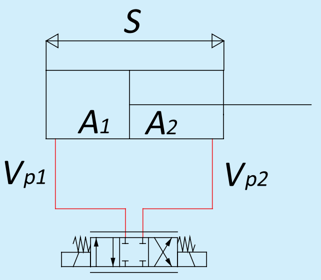


Fig 6.7.B

THROUGH PISTON ROD

$$C_{o,\min} = \frac{4 \cdot A^2 \cdot E}{A \cdot S + V_{p1} + V_{p2}} \qquad (6.34)$$

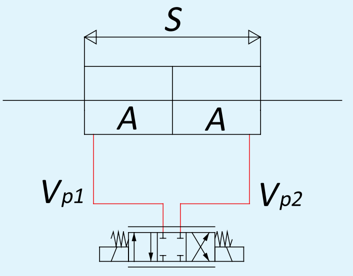


Fig 6.7.C

HYDRAULIC MOTOR WITH DOUBLE DRIVE

$$C_o = \frac{4 \cdot V_{st}^{\ 2} \cdot E}{\pi \cdot V_{st} + 2 \cdot V_p} \qquad (6.35)$$

Both pipe A and B contribute to the stiffness of the drive mechanism

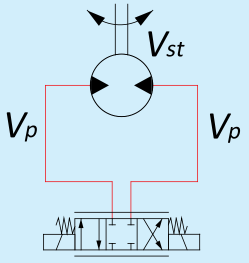


Fig 6.7.D

TRANSMISSION WITH FLUSH VALVE

$$C_o = \frac{2 \cdot V_{st}^{\ 2} \cdot E}{\pi \cdot V_{st} + 2 \cdot V_p} \qquad (6.36)$$

Be careful: with a transmission one hydraulic pipe is always set to a low pressure through a flush valve. This means that that pipe cannot contribute to the stiffness of the system.

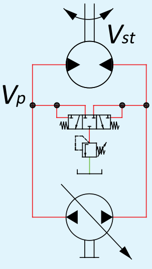


Fig.6.7.E

6.8 Mass spring system

In previous paragraphs we established that a hydraulic motor or a cylinder together with the connecting pipe-work has a spring stiffness of C_o . There are however also a mass inertia and a mass attached to the hydraulic motor and cylinder respectively. This means that the drive mechanism has the same characteristics as those of the mass spring system shown in figure 6.8.A.

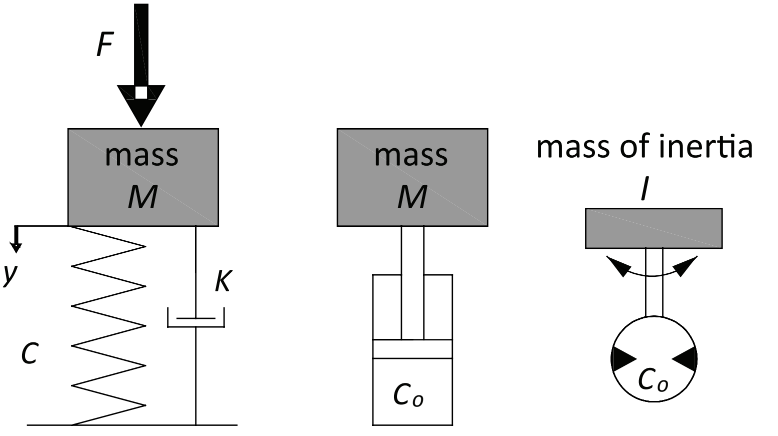


Fig 6.8.A Hydraulic drives have the same characteristics as those of a mass spring system

The hydraulic system has a stiffness C_o in the same way that a mechanical system has a stiffness *C*. You must remember though that the stiffness of a linear system (like the cylinder) is expressed in N/m, whilst the stiffness of a rotating system (like the hydraulic motor) is expressed in Nm/rad.
Any mechanical mass spring system has a certain damping *K*. This also occurs in a hydraulic drive system. Damping is caused by friction in the actuator and the pipe system. For the hydraulic motor there is also the internal leakage between the two connecting ports.

The formula for the movement of a mass spring system can be written as follows:

$$F = M.\ddot{y} + K_f.\dot{y} + C.y \qquad (6.37)$$

or with $$s = \frac{dy}{dt} = \dot{y} \qquad (6.38)$$

$$\frac{y}{F} = \frac{1}{M.s^2 + K_f.s + C} = \frac{\frac{1}{C}}{\frac{M}{C}.s^2 + \frac{K_f}{C}.s + 1} \qquad (6.39)$$

replacing $\sqrt{\frac{C}{M}}$ by ω (6.40)

and $\frac{K_f}{C}$ (6.41) by $\frac{2.\beta}{\omega}$ (6.42)

we get: $$\frac{y}{F} = \frac{1}{\frac{s^2}{\omega^2} + 2.\beta.\frac{s}{\omega} + 1} \qquad (6.43)$$

Similar to a mechanical mass spring system, hydraulic systems consisting of an actuator, a connected mass or mass of inertia also have a natural frequency of ω_o . For the cylinder and the hydraulic motor respectively the natural frequency can be calculated as follows:

$$\omega_o = \sqrt{\frac{C_o}{M}} \qquad (6.44) \qquad \text{with } M = \text{mass kg}$$

and

$$\omega_o = \sqrt{\frac{C_o}{I}} \qquad (6.45) \qquad \text{with } I = \text{mass of inertia kg.m}^2$$

In this case the natural frequency ω_o has a dimension of rad/s. A more practical/usual value of Hz can be calculated by dividing the frequency rad/s by the number 2π.

6.9 Reduced mass and moment of inertia

In many situations the mass is not connected directly to a cylinder or hydraulic motor but via a lever mechanism or a gearbox. The gearbox is used to adjust the torque and the speed of the actuator, either a hydraulic or electrical motor, to the desired speed and torque of the driven machinery.You will always need the reduced mass *M* or the reduced mass inertia *I* when you calculate the natural frequency of a drive mechanism or when you calculate the acceleration force or torque of a drive mechanism.

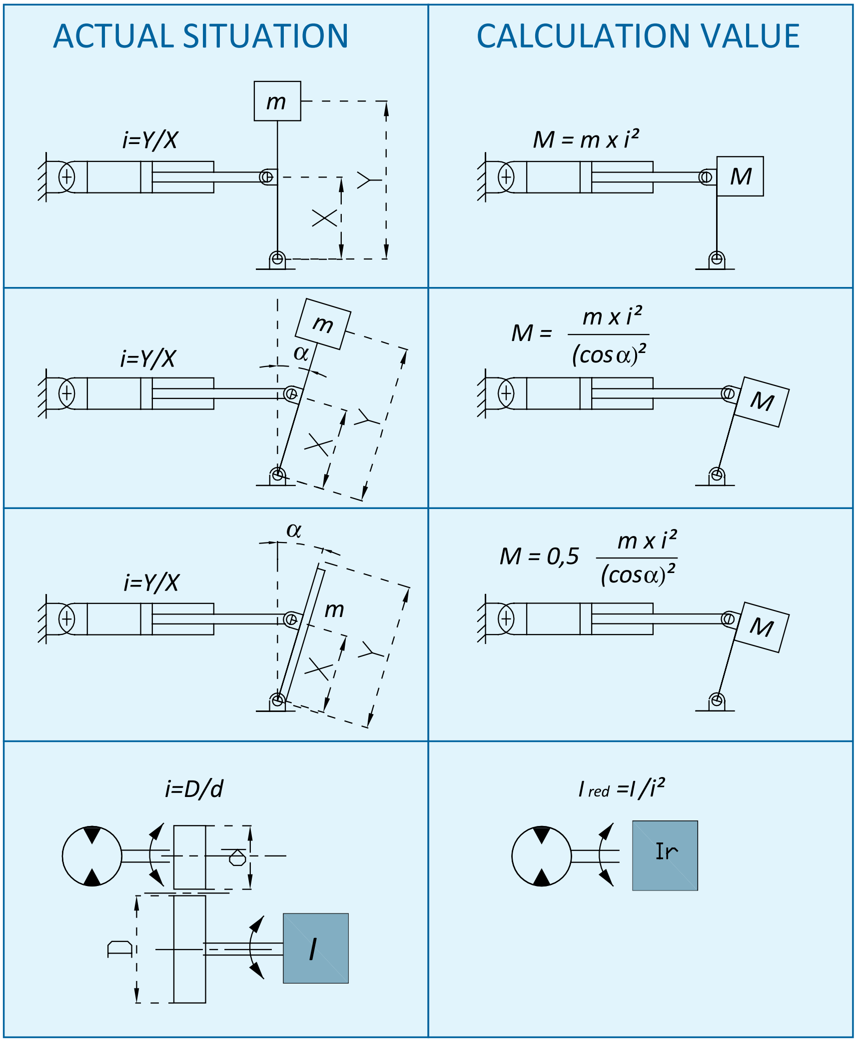


Fig 6.9.A *Calculation of reduced mass and moment of inertia for different actual situations*

The reduced mass *M* is the equivalent mass that is working on the connecting point of the cylinder and that represents the same dynamic effect as the original driven mechanism. The reduced mass moment of inertia I_{red} is the inertia in that works on the drive axle of the hydraulic motor and which represents the same dynamic effect as the original drive.

6.10 Dynamic behaviour

We showed earlier that a hydraulic actuator will show the same characteristics as a mass spring system. This also means that the simple block diagram of a 1st order system does not show the full dynamic characteristics of a hydraulic drive system.

The easiest way to derive the true dynamic behaviour of a hydraulic drive is again with the help of block diagrams. This method is always easier than the use of differential equations. In the example below we will once again use a cylinder.

The working forces on the cylinder are determined by forces of the mass (= M x s^2), with s as the Laplace operator, and friction (= K x s), as already described in the discussion about the mass spring system. F_e represents the eventual external forces on the system. The spring stiffness C_o is now included in the model. The spring stiffness determines the compression or displacement of the piston in the cylinder as a result of the sum of all the forces working on it. The hydraulic stiffness C_H represents the effect from the load forces onto the flow.

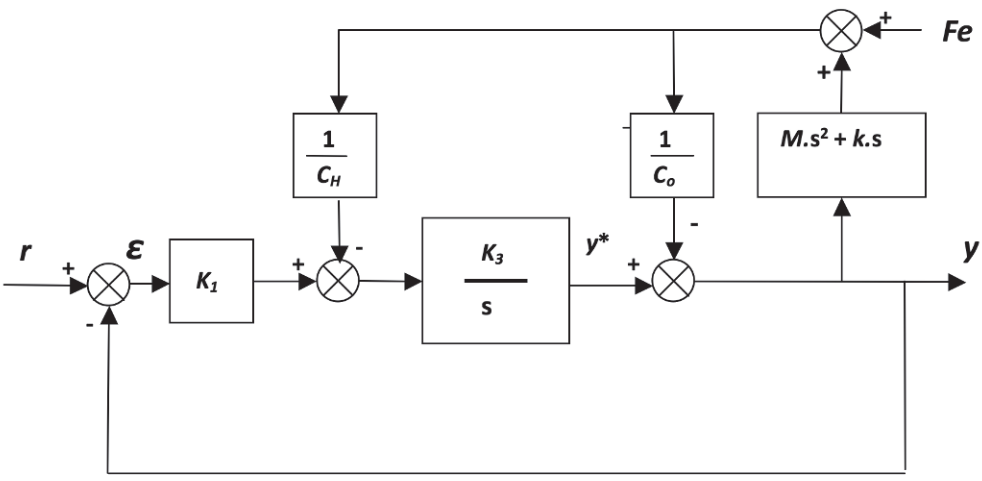


Fig 6.10.A Basic Block diagram for a mass loaded cylinder

By applying the simple calculus rules for block diagrams it is possible to convert this block diagram into a simpler one. We will give the transformation in formulaic form first.
To understand the behaviour of the hydraulic system we assume that $F_e = 0$.

$$\frac{y}{\varepsilon} = \frac{K_1 \cdot K_3}{s \cdot \left(\frac{M}{C_o} s^2 + \left(\frac{k}{C_o} + \frac{M \cdot K_3}{C_H} \right) s + 1 + \frac{k \cdot K_3}{C_H} \right)} = \frac{K_1 \cdot K_3}{s \left(\frac{s^2}{\omega_0^2} + 2\beta \frac{s}{\omega_0} + 1 \right)} \qquad (6.46)$$

Where:

$$\omega_0 = \sqrt{\frac{C_0}{M}} \qquad (6.46)$$

with:

M	=	mass	kg	K	=	friction	Ns/m
C_o	=	hydraulic stiffness	N/m	ω_o	=	natural frequency	rad/s

And for the damping ratio: $$\beta = \frac{K}{2\sqrt{MC_0}} + 0,5 \cdot \frac{K_3}{C_H} \cdot \sqrt{M \cdot C_o} \qquad (6.48)$$

The natural frequency ω_o is the same as the one defined for a mass spring system earlier. The damping coefficient β is a measure for the amount of friction that takes place. The value for β for most systems is between 0,15 and 0,35. For hydraulic motors the damping is always greater than it is for cylinders. This is because hydraulic motors have internal leakage. An appropriate value for β for hydraulic motors is between 0,2 and 0,35.

The block diagram for a cylinder with positional feedback can now be shown in a simplified form.

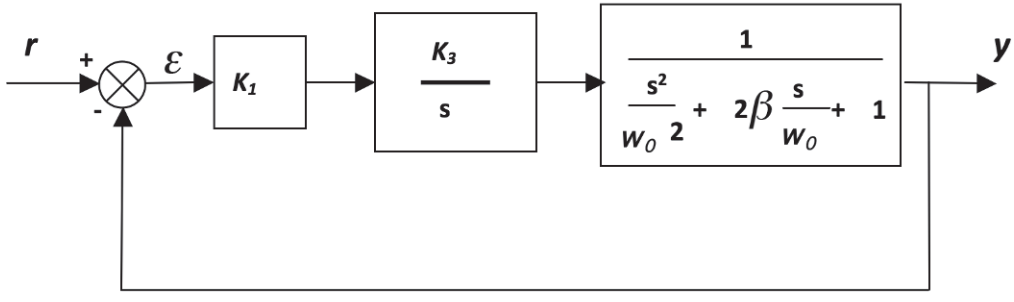


Fig 6.10.B Block diagram for a positional feedback cylinder

In a slightly different form, when factors K_1 and K_3 are combined into one factor K_v, the block diagram looks like this:

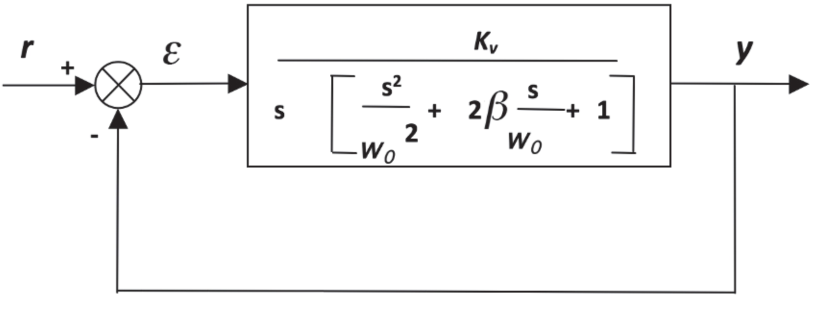


Fig 6.10.C Block diagram for a positional feedback cylinder with gain K_v

In its simplest form the dynamic behaviour of a position controlled cylinder can then be described as a series circuit of a 1st and 2nd order system.

6.11 Complex numbers, polar diagram

Complex numbers are an expansion on real numbers. A complex number z can be described in several different ways.

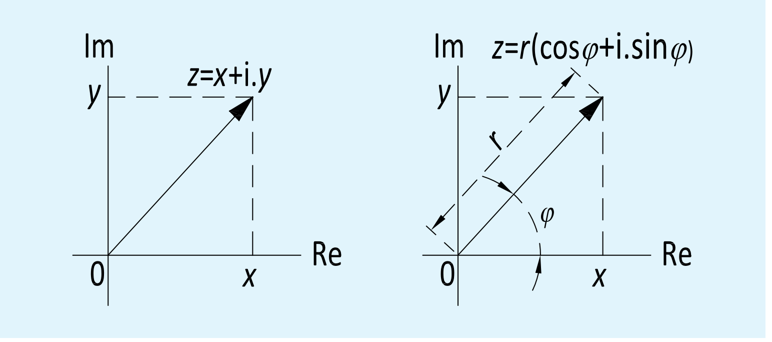


Fig 6.11.A Two different presentations of a complex number

The first form is the Cartesian form, where it is the sum of the real part *x* and the imaginary part *y*. The rule for the imaginary number *i* is that $i^2 = -1$.

$$z = x + i.y \tag{6.49}$$

The second form is the polar form, where the complex number is represented by a vector of length *r* with an angle relative to the real axis of ϕ.

$$z = r\left(\cos\phi + i.\sin\ \phi\right) \tag{6.50}$$

which can also be written as $z = r.e^{i.\phi}$ (6.51)

The conversion from the Cartesian form to the polar form can be obtained from:

$$x = r.\cos\phi \tag{6.52}$$ and $$y = r.\sin\phi \tag{6.53}$$

Also: $$r = |z| = \sqrt{x^2 + y^2} \tag{6.54}$$ and $$\phi = \arg(z) \tag{6.55}$$

The multiplication of two complex numbers z_1 and z_2 can easily be established in the polar form:

$$z_1.z_2 = r_1.e^{i.\phi_1}.r_2.e^{i.\phi_2} = r_1.r_2.e^{i.(\phi_1 + \phi_2)} \tag{6.56}$$

That way the result can also be obtained through a complex number with a vector length equal to $r_1.r_2$ whilst the angle with the real axis is determined by the sum of the angles $\phi_1 + \phi_2$.

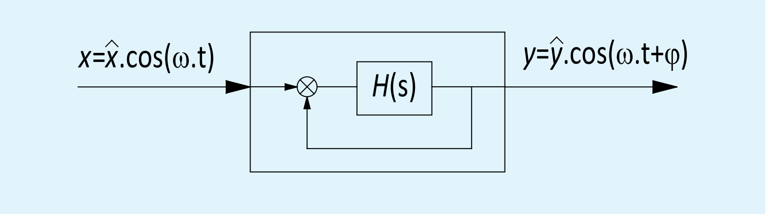


Fig 6.11.B Block diagram of a unity feedback system

We will now look at a simple block diagram with an input signal *x*, a unity feedback control system with a complex transfer function *H(s)* for the open chain and with an output signal *y*. Assume that the input signal sinusoidal is with the general description of:

$$x = \hat{x}.\cos(\omega.t) \qquad (6.57) \qquad \text{and output signal} \qquad y = \hat{y}.\cos(\omega.t + \phi) \qquad (6.58)$$

The sinusoidal form can also be seen as a complex number with just a real value, where $\phi = \omega.t$.
The output signal is then also a complex number with a different amplitude and a phase shift in the polar plain of ϕ relative to the input signal. This means that the complex transfer function *H(s)* determines what the output signal looks like if we apply a sinusoidal input signal.

From mathematics we already know that all time based signals can be described as the sum of one or more sinusoidal signals. This means that we can assess the dynamic behaviour of a system by assessing the transfer function H(s).

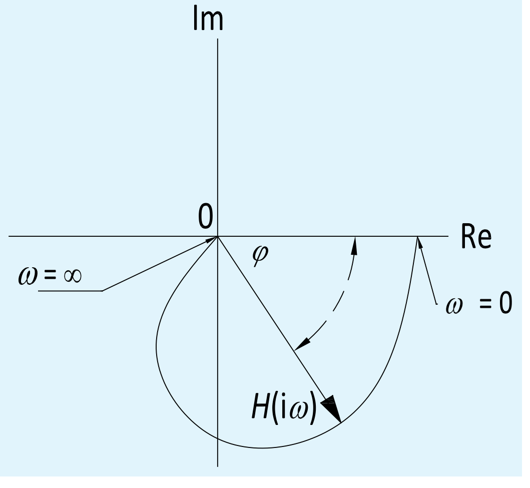


Fig 6.11.C The polar diagram of H(s)

In that case the transfer function H(s) of an open chain is a complex function with a real value and an imaginary value. Using simple algebraic methods, the values of the real part and the imaginary part can be determined by replacing s with $i.\omega$. The frequency ω is then the variable in this new equation.

The result for a particular application can be represented in a so-called polar diagram, figure 6.11.C where the real axis and the imaginary axis are shown again. If $\omega = 0$ then the transfer function has a purely real value. As the frequency ω increases a phase shift (phase lag) ϕ will develop.
When $\omega = \infty$ the transfer function will once again have a real value of zero but with a phase lag of approximately 135°.

What does it mean for the controlled system if H(s) has a real value of -1 for $\omega = \infty$
The formula for the controlled system was:

$$\frac{y}{x} = \frac{H}{1+H} = \frac{-1}{1-1} = \frac{1}{0} = \infty \qquad (6.59)$$

The conclusion is that a controlled system will be unstable if the polar diagram of the open chain for a certain frequency ω goes through the real point -1. With the help of control theory it is possible to show that the polar diagram must be sufficiently far removed from the -1 point, not just to get a stable system, but also to obtain a sufficiently dampened system.

6.12 Stability of 2nd and higher order systems

Now that the concepts of the transfer function H(s), complex numbers and polar diagrams are understood, we can look at the control behaviour of the system that monitors the position that was set up earlier for a cylinder loaded with a hydraulic mass.

We have already got the transfer function for the open chain:

$$H(s) = \frac{K_v}{s.\left(\frac{s^2}{\omega_o^2} + 2.\beta.\frac{s}{\omega_o} + 1\right)} \qquad (6.60)$$

If you replace the Laplace operator with $i.\omega$, you can obtain the algebraic equations for the polar diagram.
The length of the vector can be derived to:

$$\left|H(i.\omega)\right| = \frac{K_v}{\omega.\sqrt{\left(1-\frac{\omega^2}{\omega_o^2}\right)^2 - \left(2.\beta.\frac{\omega}{\omega_o}\right)^2}} \qquad (6.61)$$

And the phase ϕ can be found with:

$$\phi = ArgH(i.\omega) = -\frac{\pi}{2} - \arctan\frac{2.\beta.\frac{\omega}{\omega_o}}{1-\frac{\omega^2}{\omega_o^2}} \qquad (6.62)$$

The polar diagram of the transfer function is represented in figure 6.12.A
A phase $\phi = 90°$ occurs at $\omega = \omega_o$ where $H(i.\omega) = K_v/2.\beta.\omega_o$. To be stable the polar plot should not encircle the point -1,0. Thus for a position feedback controlled hydraulic cylinder the stability condition is:

$$K_v \leq 2.\beta.\omega_o \qquad (6.63)$$

Another criterion is that sufficient damping should be present. This condition can be met when the polair circle does not cross the M=1.3 circle, see figure 6.12.A. As showed in the figure the damping condition is:

$$K_v \leq \beta.\omega_o \qquad (6.64)$$

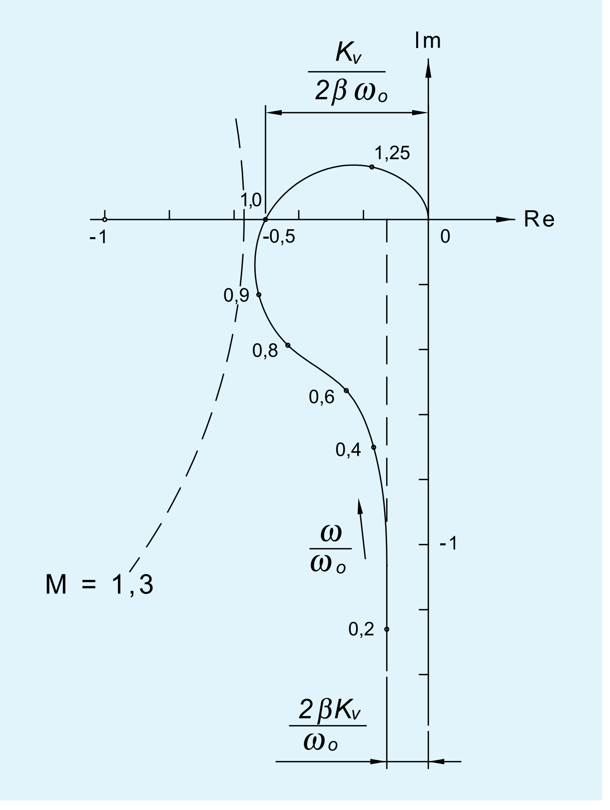


Fig 6.12.A *Polair diagram for a mass loaded hydraulic cylinder with position feedback*

Adjusting the controls for a system that monitors the position of the cylinder or any other random control mechanism can in practice be done by setting the maximum amplification factor K_v calculated above in the software of the PLC. In practice however, there are often several more factors that influence the shape of the polar diagram. It is for example possible that a proportional or a servo valve itself can introduce a phase shift. The position sensor itself can introduce phase shift too. In those cases the theoretical maximum gain is only a rough approximation of the gain that needs to be set.

Increasing the gain until one of the measurements shows that instability occurs in the controlled system, is a very practical way to set the gain. After that the gain needs to be set at 50-60% of this value. This method may appear to be rather unscientific for the setting of a control mechanism, but it does work very well in practice.

6.13 Control errors

The velocity error

Earlier on in this chapter we deduced the quasi static behaviour of a hydraulic position control mechanism.

The block diagram of this behaviour can be represented as follows:

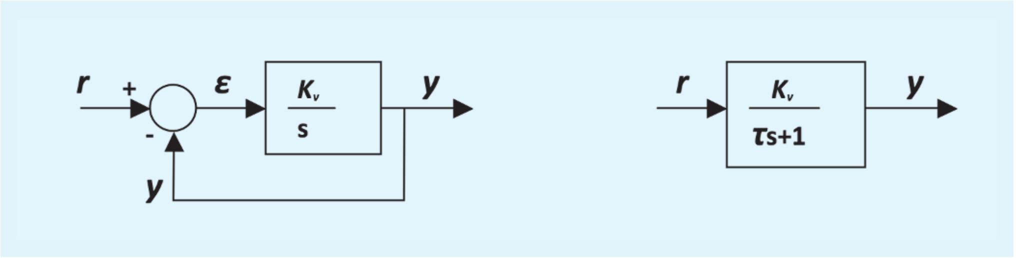


Fig 6.13.A The block diagram for the quasi static behaviour of a hydraulic position control system.

When we discussed the stability of higher order system, we also showed that the gain K_v is limited to a maximum value.
In a very simplified form, the block diagram can also be drawn like this:

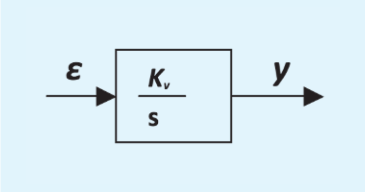


Fig 6.13.B Part of the block diagram 6.13.A

The formula for this block diagram can also be written as follows:

$$y = \frac{K_v}{s}.\varepsilon \qquad (6.65) \qquad \text{or} \qquad \varepsilon = \frac{dy/dt}{K_v} \qquad (6.66)$$

The error ε is then directly proportional to the speed dy/dt that actually occurs in the system. The larger the gain K_v, the smaller the error ε. Because there is a limit for the value of K_v it follows that the positional error always exists when the system is moving. You could even say that an error ε is needed to be able to obtain the speed dy/dt.

This situation develops often in tracking systems, where the cylinder or the hydraulic motor needs to follow a preset movement pattern as precisely as possible. Control theory does however offer an excellent solution for this problem. This solution is called the feed forward control mechanism. In that case the block diagram looks as follows:

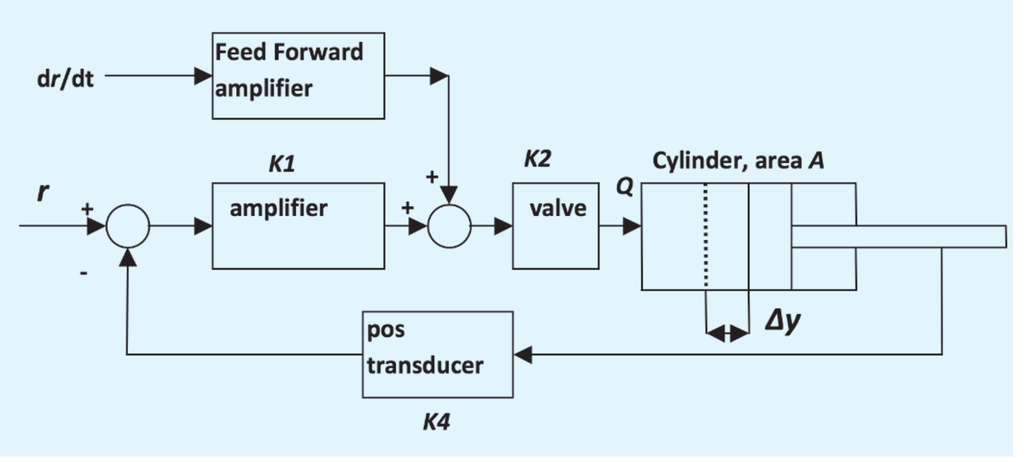


Fig 6.13.C Feed Forward control to correct for the velocity error

In the block diagram of figure 6.13.C the speed d*r*/dt of the input signal is fed directly to the valve, via a separate amplifier. The valve is in fact governed to a value that ensures that the cylinder already has the speed d*y*/dt that corresponds with the speed d*r*/dt of the input signal.

The closed position control circuit remains the same and does in fact take over the control as soon as the input signal constant of the controlled system needs to be guided to a specific position.

The interesting part of the Feed Forward control is that such a control mechanism has no negative effect on damping and can cause no instability.

The friction error

Another error that can occur in a controlled system can be caused by the static friction in for example a cylinder or a hydraulic motor or the static friction between the driven mechanism and its environment. This friction stops the cylinder or the hydraulic motor from moving until the pressure difference across the actuator is big enough to overcome that friction.

This means that it is possible that there is an equilibrium situation where the cylinder is static whilst there is a pressure difference Δp across it. The valve will however be guided as a result of the small error signal ε. In other words, this tracking system will not start to move when the positional error is small. This means that the positional error will be larger when the static friction is larger.

In earlier examples we have only discussed the flow characteristics of the proportional or the servo valve. These characteristics are not suitable to show the pressure difference created by the valve as a result of a small control signal due to the small signal error ε.

To do that it is necessary to introduce a so-called pressure amplification characteristic. This is shown in figure 6.13.D. This graph gives the pressure difference that develops across the valve as a function of the percentage of the opening of the valve. The graph is recorded whilst no volume flow passes through the valve, in other words, with just a pressure gauge at the outlet ports of the valve.

A pressure difference of 100% across the outlet ports of the valve will then correspond with the maximum pressure difference that will be made available across the inlet ports. Assuming that the inlet pressure of a valve is 250 bar and that the T-connection is connected to the reservoir, the valve can then generate a maximum pressure difference of 250 bar across the cylinder.

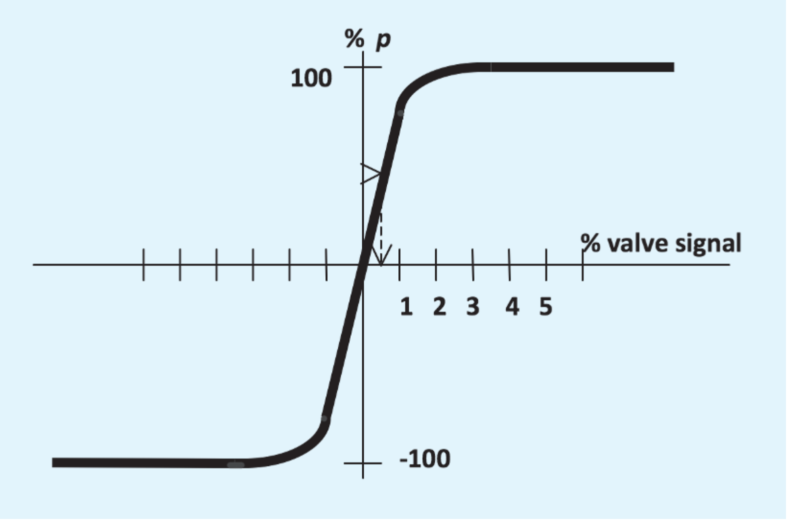


Fig 6.13.D Pressure gain of control valves, correcting the friction error

If, for example, we know that the actuator together with the driven machinery has a static friction of 35 bar, then we can find from the graph at which setting of the valve this pressure difference can be generated. Using the amplification factors from the bode diagram it is possible to calculate what the positional error is caused by the static friction.

6.14 The integrating controller (I-controller)

The I-controller or integrating controller

Earlier in this chapter we showed that positional errors can develop as a result of, for example, the speed of the moving actuator or the static friction of a cylinder or hydraulic motor. A normal proportional controller is not able to correct these errors. In this section we introduce a controller that will be able to do this, the so-called I-controller or Integrating Controller. The working in the frequency domain can easily be explained with help from the classical theories of control engineering. In this case however, we have chosen to explain this in the context of a time domain.
The I-controller is described with:

$$V_I = \frac{1}{\tau_i} \int_0^t \varepsilon \cdot dt \qquad (6.67)$$

where:

τ_i	=	integration constant	s

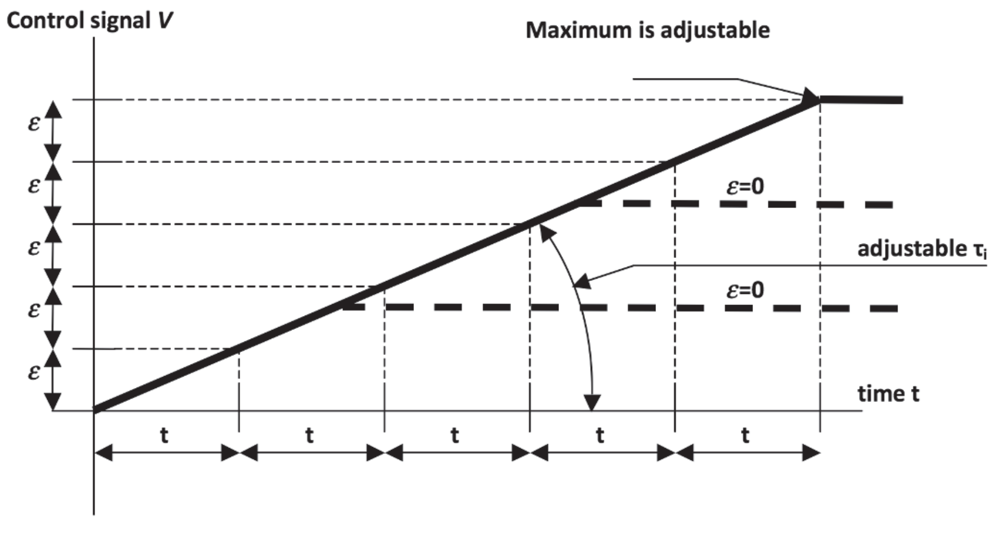


Fig 6.14.A The function of the integrator in the time domain

With an I-controller, the input signal is integrated against time. The output signal will increase continuously over time if the input has a constant value of ε. If, after a certain amount of time, the input signal becomes 0 then the output signal will be maintained at the value that it reached just before that point in time.
On an I-controller, both the integration constant τ_i (the angle of the I-graph) and the maximum output V_I signal can be set. The output signal will vary quickly if the integration time is short. If, on the other hand, the integration time is slow, then the output signal will change slowly. It is important to remember though that, as a rule, the output signal of an I-controller is limited to a range between -10 and +10 Volt. In many cases though, one may want to limit the range of an I-controller to a range between -1 and +1 Volt.

This means that, as long as a positional error ε is present, the I-controller will create an output signal that will make the needed positional error smaller. The system that is being controlled (the servo valve and the cylinder) do however need to realise that position. The I-controller can be built into the block diagram for a position controlled cylinder in the following way:

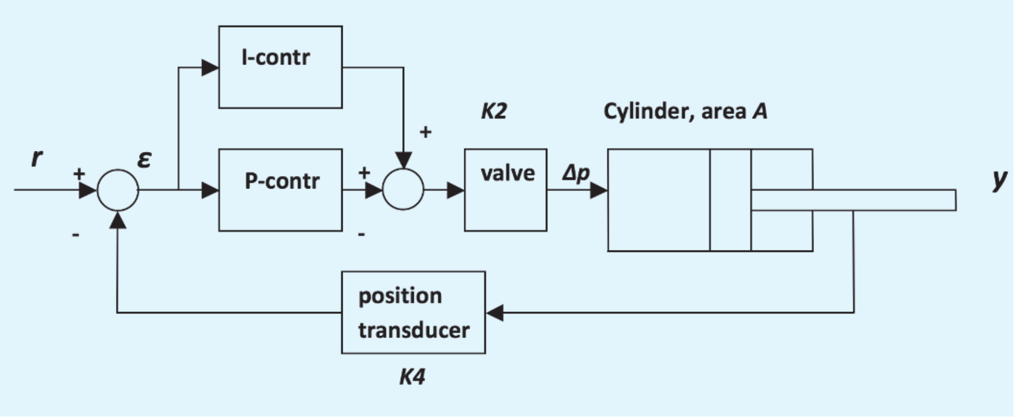


Fig 6.14.B Integrating amplifier in parallel with proportional amplifier

An I-controller is always installed together with a proportional or P-controller. The controllers are always fitted in parallel in circuit. Large positional errors are controlled by the P-controller, whilst the small errors are controlled by the I-controller.

There is an important condition for the application of an I-controller. Imagine that there is no hydraulic power, for example because the pump has been switched off whilst the system is out of use. In those cases the measured position is often different from the set point that is sent to the control electronics. This means that the system reads a positional error of ε. The I-controller will react by sending, after the integration time, a maximum output signal to the valve.
So far, so good. If however, the pump is started in this position, then the effect will be that the valve will be fully opened by the controller and the actuator will move in a particular direction with high speed. This is a very dangerous situation which can easily lead to physical damage to the installation or a person. To prevent this, industrial I-controllers are always fitted with an extra 'enabling' signal that will only release the I-controller from the moment that the installation is operational.

Stability condition if an I-controller is applied.

An integrator will give an extra phase shift of -90° in the polar diagram. It is relatively easy to understand when you realize that the integration of a cosine always is a sine function. And in a time frame, the sine always lags a phase of 90° behind the cosine.

The extra phase shift of 90° in the polar diagram ensures that there is an influence on the allowable gain of the system. The further the integration time for the system is reduced, the lower the setting for the gain K_v needs to be.
A short integration time τ_i provides a quick correction of the postional error ε, but because of the lower value of K_v, the speed of the feedback error will increase.

The general stability condition when applying an integrating control system in a hydraulic system with positional feedback is:

$$\omega_o . \tau_i \geq 25 \qquad (6.68)$$

with:

ω_o	=	natural frequency	rad/s	τ_i	=	integration time	s

6.15 The dynamics of the AC-induction motor

The AC induction motor can build up torque in a very short period of time (within a few msec). In an open-loop system the flux (U/f) and the frequency will be changed to achieve the desired torque and revolutions per second respectively.

An open loop system with an inverter a an AC induction motor behaves, in principle, like a 1st order system.

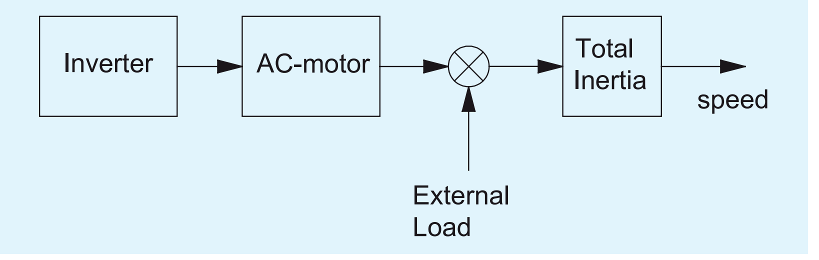


Fig 6.15.A Block diagram for an inverter with an AC induction motor

The difference between the torque T that is delivered by the AC motor and the torque T_l by the external load will be available to either accelerate or slow down the mass inertia *I*.

$$n_m = \frac{1}{I}\int_0^t (T - T_l)dt \qquad (6.69) \qquad \text{or:} \qquad I.\frac{dn}{dt} = T - T_l \qquad (6.70)$$

The torque created by the motor is proportional to the slippage s of the motor when the number of revolutions per second is approximately equal to that which prevails when there is no load on the motor. The formula for that situation is as follows:

$$T = K.s = K.\frac{n_s - n}{n_s} \qquad (6.71)$$

From both equations we get:

$$I.\frac{dn}{dt} = K - \frac{K}{n_s}.n - T_l \qquad (6.72) \qquad \text{or:} \qquad I.\frac{dn}{dt} + \frac{K}{n_s}.n = K - T_l \qquad (6.73)$$

Or with the Laplace operator s:

$$\frac{n}{(K - T_l)} = \frac{1}{\tau.s + 1} \qquad (6.74) \qquad \text{with the time constant} \qquad \tau = \frac{I.n_s}{K} \qquad (6.75)$$

If the external torque varies, then the number of revolutions per second of a VFD (Variable Frequency Drive) will show fluctuations as a result of the 1st order system.

A more or less constant number of revolutions per second can be achieved by measuring the speed of the motor axle and feeding it back via a control mechanism. Variations in the revolutions per second will then be compensated for by the controller. The flux *V/f* that is generated by the inverter will result in a constant flux across the air gap if the resistance of the stator is negligible. This is also the case when the number of revolutions is equal to that of the unloaded position. But the voltage losses across the stator for a low frequency (and thus a low number of revolutions per second) are considerable since the voltage losses across the stator are more or less constant. The result is that an extra phase shift of the total motor current occurs, relative to the input voltage. This is the cause of unstable behaviour in situations of low revolutions per second combined with high output torque when a speed control with feedback is used.

A current limiter is often added to a standard controller. If a large increase in the required number of revolutions per second occurs, then the maximum motor current and the maximum motor torque will be reached as a result. In those situations the limiter function will limit the increase in flux. This will of course have a negative impact on the dynamic speed behaviour of the drive system.

Because the suppliers of frequency drives have recognised this problem, they now supply feedback of both the actual speed and the actually generated torque in the micro processors of their frequency drives.
Existing controllers are known as DTC (Direct Torque Control, ABB) or RFC (Rotor Flux Control, Control Techniques). In these controllers the number of revolutions per second and the torque of the motor are simulated which means that feedback can take place without the need for an actual recorder of the position or the number of revolutions per second.

A DTC or a RFC frequency controller can then in fact be considered as the ideal torque generator. In that case the reaction time in drives where the number of revolutions per second is controlled, will only be limited by the torque that is present in the motor and total reduced mass inertia of the drive to the motor axis. They claim that the reaction times of these drives will be between 0,0005 and 0,002 s for a torque increase from 0 to 100%.

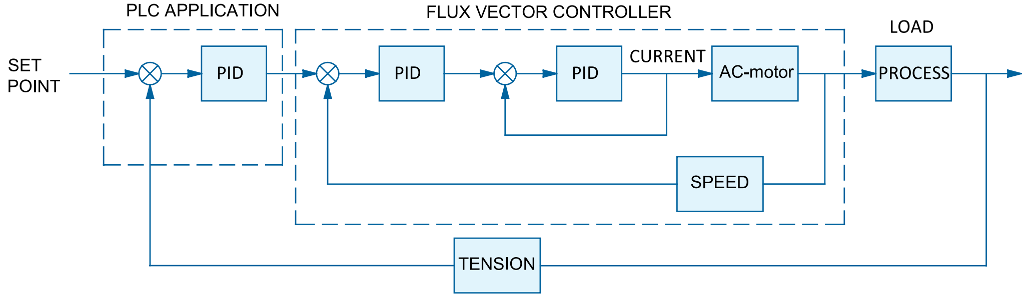


Fig 6.15.B Block diagram for a DTC or RFC frequency controller

In practice these DTC and RFC frequency controllers are sometimes included in much larger control circuits. In those cases it will be possible to select a random parameter of the drive process as feedback parameter or control measure. The example above shows the control for a pipe tensioner. The pipe tracks are electrically driven by electric AC motors with a Flux Control frequency drive. The force that is applied to the pipe by the tensioner is measured by a load sensor.

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DJN

Chapter 7

Linear drives, open-loop

Motion Control in Offshore and Dredging

Chapter 7

Linear drives, open-loop

In this chapter we will discuss examples of linear drives from the offshore and dredging industries. The examples are limited to the controlled drives, so without feedback of one of the system variables such as position, speed, force etc. Those drives will be discussed extensively in the next chapter.

We will give many practical hints with the examples. Where necessary we will also show the theoretical basis for the correct calculation of the drive parameters. There are several different ways to engineer and design a linear drive.

- The screw spindle drive in which the screw is often driven by an electric motor. The drive is characterised by a high degree of control accuracy whilst the actuator, the spindle, has a high degree of rigidity. Because of the self-braking characteristics of the screw mechanism and the spindle, the spindle will remain in its actual position if the drive power to the spindle fails. In this chapter we will give an example of the application of a spindle mechanism combined with a linear hydraulic drive.
- The pneumatic cylinder is often used in the process automation industry for a number of reasons. The readily available and easy distribution of the necessary air pressure, the relatively low cost of the components and the high piston speeds (of up to 10 m/s) that can be reached. This type of application is not suitable for the offshore and dredging industries because of the limits in the maximum pressure that can be generated. A force of 15 kN is about the maximum that can be achieved with a pneumatic drive. In the next chapter we will show the exception that can be made in heave compensation system, where much higher air or gas pressures of up to 280 bar are applied. These cylinders can however not be considered as a standard pneumatic application.
- The hydraulic cylinder is particularly suited to generate the high forces required in the offshore and dredging industries. Hydraulic actuators can also achieve a high degree of positional accuracy. They combine this with sufficiently responsive dynamic characteristics.

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7.1 List of symbols

A	=	full bore area of cylinder	m^2
C_H	=	hydraulic capacity	m^5/N
F_{pipe}	=	total pipe force	kN
F_{sq}	=	squeeze force	kN
G_{DCV}	=	flow gain proportional valve	$m^5/N.s$
G_{rel}	=	flow gain of pressure relief valve	$m^5/N.s$
G_{pilot}	=	flow gain valve as function of pilot pressure	$m^5/N.s$
M	=	mass	kg
p_l	=	load pressure	N/m^2
p_p	=	pilot pressure	N/m^2
p_v	=	bias setting pressure relief valve	N/m^2
Q_p	=	pilot flow	m^3/s
T	=	temperature	°K
V	=	volume	m^3
d	=	orifice diameter	m
f_{fr}	=	friction coefficient	
i	=	pilot ratio counterbalance valve	
n	=	number of squeeze cylinders	
ϕ	=	area ratio cylinder	

7.2 Mass loaded cylinder with counterbalance valve

An example of a simple cylinder drive is shown in the hydraulic diagram , see figure 7.2.A. In practice this could also be a vertical movement for a pipe handling mechanism or the operation of an A-Frame for a suction pipe winch on a hopper dredger.

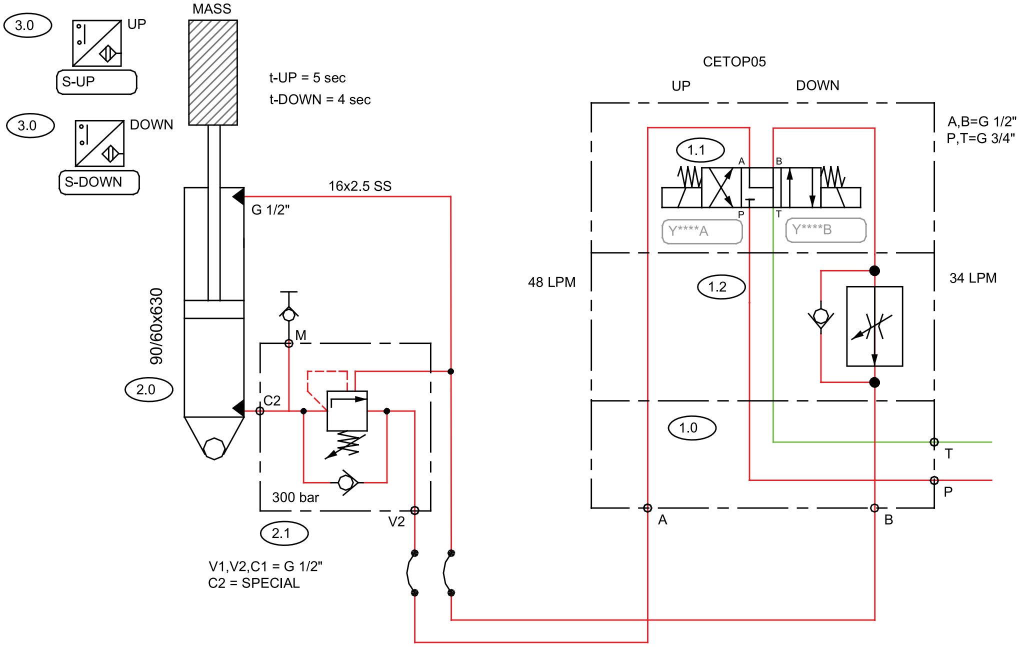

Fig 7.2.A Mass loaded cylinder with counterbalance valve

The only load in the vertical direction is the weight of the pipe. The cylinder is controlled with the help of a 4/3 directional valve in position 1.1. Make sure that the A-connection of function block 2.0 always connects with the bottom side of the cylinder. Also make sure that the B-connection always connects with the rod side of the cylinder. There is no international standard for this. However, if this rule is applied consistently, it will be easier for the service engineer when he is looking for a malfunction. The volume flow with the to-and-fro movement is provided by a two-way flow regulating valve in position 1.2. A counterbalance valve is mounted directly onto the cylinder. This valve makes sure that the suspended mass, during the downwards movement, doesn't leads on the controls from the 4/3 way valve.

Proximity switches give a signal when the top or bottom position of the piston is reached.

Please note: in these situations, the two-way flow regulator valve must always be applied as a meter-in valve. In these situations the valve is used to direct a constant flow towards the cylinder, independent of the load. Later on we will also find out that the application of a two-way flow regulating valve can lead to unstable behaviour in a cylinder drive.

The volume flow required for the up and down movement can be calculated with the help of simple formulas that are available. Special attention needs to be paid to the calculation of the load pressure when a counterbalance valve pos 2.1 is applied. Opening this valve is achieved by creating a balance of the forces working across the control piston in the valve.

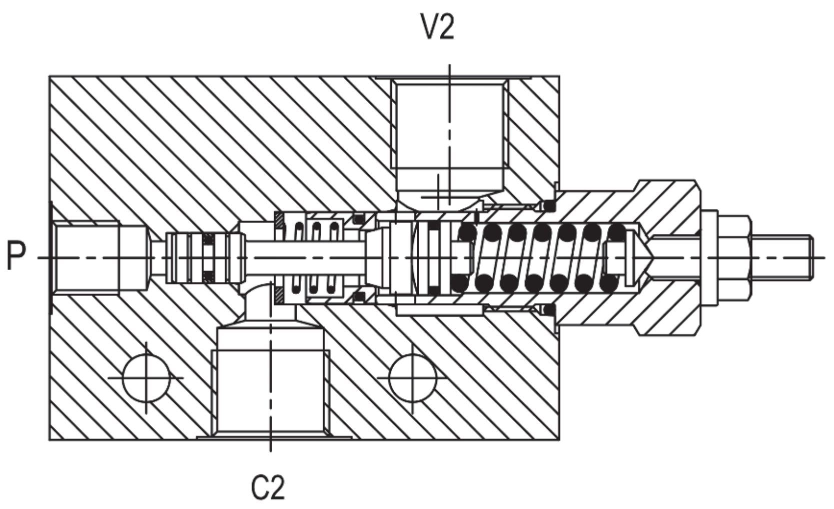

Fig.7.2.B. Section view of a counterbalance valve (Bosch Rexroth Oil Control)

When the pressure at V2 rises above the small spring pressure , the check valve seat is pushed away from the piston and flow is allowed from V2 to C2. When load pressure at C2 rises above the pressure setting, the directly operated, differential area relief function is activated and the flow is from C2 to V2. With pilot pressure at P, the pressure setting is reduced in proportion to the stated ratio of the valve, until the valve opens and allows the fluid to flow from C2 to V2. The spring chamber is drained to V2, and any back-pressure at V2 is in addition to the pressure setting in all functions.

An important parameter for these valves is the pilot ratio *i*. A range of ratios between 3,5 and 10 is available. The higher the ratio, the lower the pilot pressure at which the valve will be opened. There is thus a tendency to choose the highest possible pilot ratio. However, for the mass loaded cylinder shown in the example, it is necessary to choose the lowest possible pilot ratio. If too high a ratio is chosen, unstable behaviour can develop due to the combination of the dynamic behaviour of the load and elasticity of the cylinder. The other reason is that as a general rule the pressure setting p_v needs to be set to at least 130% of the maximum load pressure p_1.

The static pressures that can develop in the cylinder when a counterbalance valve is applied can be calculated as follows:

The load pressure π_{Λ} in the cylinder is:

$$p_L = \frac{M \cdot g}{A} \tag{7.1}$$

where:

A	=	bottom area of cylinder	m^2	M	=	mass	kg
g	=	gravity	m/s^2	P_L	=	load pressure	N/m^2

Note:
Factors other than the pure mass load, like for example external forces, may influence the load pressure p_L.

Due to the force balance in the counterbalance valve we get:

$$p_p.i + p_{C2} = p_V + p_{V2} \tag{7.2}$$

with:

P_{C_2}	=	pressure at port C2 of the counterbalance valve	ϕ	=	area ratio of the cylinder (bottom/annular area)
p_V	=	pressure setting of the valve (=130% of p_l).	i	=	pilot ratio of the counterbalance valve (from manufacturer)
p_{V_2}	=	back pressure of the directional control valve			

For the cylinder force balance we get:

$$p_{C2} = p_l + p_p.\varphi \tag{7.3}$$

and:

$$p_p = \frac{p_V + p_{V2} - p_L}{i+\varphi} = \frac{0,3.p_L + p_{V2}}{i+\ \varphi} \tag{7.4}$$

and:

$$p_{C2} = p_L + \left(p_V + p_{V2} - p_l\right).\frac{\varphi}{i+\varphi} = p_L + \left(0,3.p_L + p_{V2}\right).\frac{\varphi}{i+\varphi} \tag{7.5}$$

These formulas above provide the result for the pressures in the counterbalance valve at static load conditions. Under variable external loads the dynamic behaviour of the valve becomes even more important.

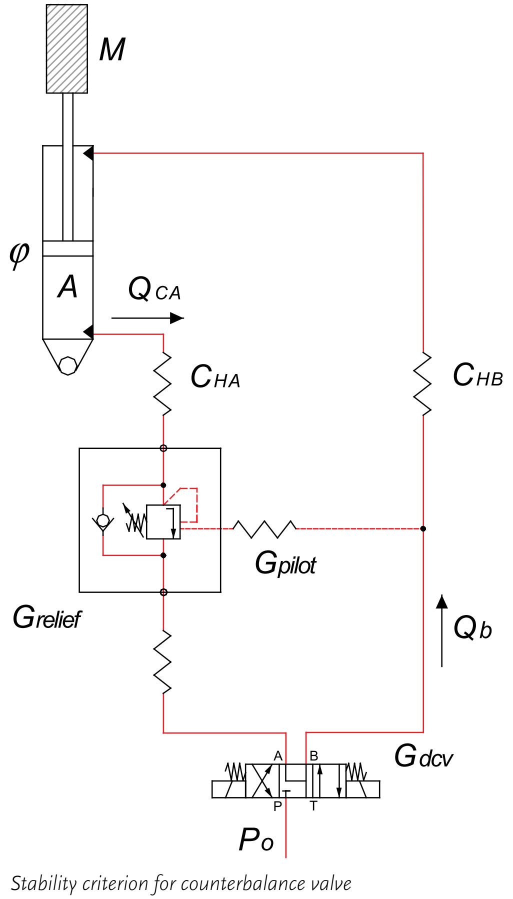

Fig 7.2.C Stability criterion for counterbalance valve

If we use the pipe capacity given in the figure above and create the formula for the calculation of the movement in the cylinder, we can approximate the stability of a counterbalanced valve in a cylinder diagram.

Equilibrium in the forces on the cylinder gives:

$$M.\ddot{x} = p_b.\frac{A}{\varphi} - p_a.A \tag{7.6}$$

and:

$$\dot{P}_b = \frac{1}{C_{HB}}(meter.in.flow - meter.out.flow) \tag{7.7}$$

$$\dot{p}_b = \frac{1}{C_{HB}}.\left(G_{DCV}.(p_0 - p_b) - \frac{A}{\varphi}.\dot{x} \right) \tag{7.8}$$

Where:

A	=	bottom area	m^2	p_a	=	pressure in A-line	N/m^2
M	=	mass of driven load	kg	p_b	=	pressure in B-line	N/m^2
C_{HB}	=	capacity of line B	m^5/N	p_o	=	inlet pressure of valve	N/m^2
G_{DCV}	=	flow gain of directional control valve	$m^5/N.s$	ϕ	=	area ratio of cylinder	

The hydraulic capacity C_H of a particular volume like lines A and B is defined as the necessary additional volume dV to reach a pressure increase dp, in formula: $C_H = dV/dp$

This leads to a 3rd order differential equation:

$$a_3.\dddot{p}_b + a_2.\ddot{p}_b + a_1.\dot{p}_b + a_o.p_b = 0 \tag{7.9}$$

The solution for this equation provides a stable behaviour if the following condition is met:

$$\frac{G_{DCV}.G^2_{relief}}{CHB} + \frac{G_{DCV}.A^2.C_{HA}}{\varphi^2.M.C_{HB}} + \frac{G^2_{relief}.G_{DCV}}{C_{HA}} + \frac{G_{relief}.A^2.C_{HB}}{M.C_{HA}} > \frac{A^2.G_{pilot}}{\varphi.M} \tag{7.10}$$

The stability condition is not absolute. It is however possible to determine a number of trends as a result of these boundary conditions.

- A small factor G_{pilot} is good for stability. (Smaller change in flow when the pilot pressure changes, lower pilot ratios i usually mean smaller G_{pilot})
- Small capacities (C_{HA} and C_{HB}) on both sides of the cylinder are good for stability
- A pressure compensated flow control valve instead of a directional valve with throttle valve is bad for stability (G_{DCV} low).
- A flat relief curve of the counterbalance valve is good (G_{relief} high). This also means that the flow capacity of the valve should be chosen as small as possible. When selecting a valve the user often takes a somewhat larger valve (to be sure that the valve capacity is always sufficient).
- Mass is good for stability. In vertical applications changing masses affect the operating point and therefore parameters of the valves. In a horizontal application one can expect better stability with more mass

7.3 Mass loaded cylinder with load control valve

Stability can be a problem for a cylinder with a counter balance valve like the one described in paragraph 7.1. For larger installations, like for example the topping cylinder drive on a tower crane, unstable behaviour can be dangerous. If unstable behaviour were to happen, the driver may lose control of the load whilst the load on the jib can become so large that the jib or its mounting fails altogether.

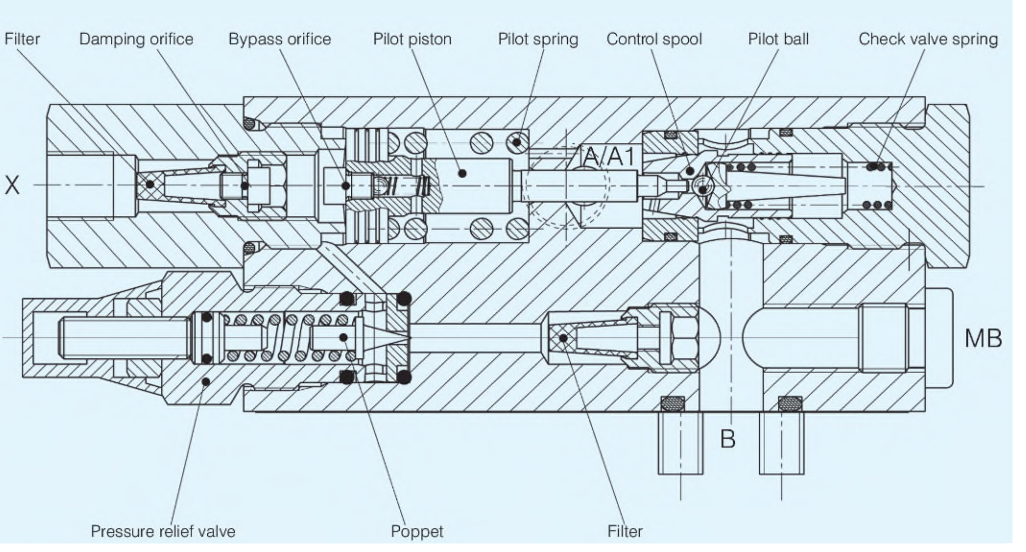

Fig 7.3.A Load control valve for larger flow capacities (Courtesy of Bucher Hydraulics)

In that type of installation so-called load control valves are used instead of the counter balance valve. A detailed function is shown in figure 7.3.A.

The valve shown is of a design that is often mounted directly underneath a cylinder or hydraulic motor. The B-port is loaded with high pressure, the A-port is connected (via a directional control valve) to tank. The control spool is pulled into the seating by the force of the check valve spring, thus blocking the connection between B and A. With an external pilot pressure at port X it is possible to slowly and proportionally open the control spool against the pressure of the spring. The load control valve is thus in effect a proportional flow valve. The pressure relief valve can, for example, be used to limit the pressure in inlet B when this, for example due to thermal expansion, gets too high.

The same conditions as for the counter balance valve apply to the stability of the load control valve. The load control valve does however have a much larger pilot ratio. Whilst the maximum value was i=10 for the counter balance valve, pilot ratios of up to i=66 are available for load control valves.

One important advantage of the load control valve over the counter balance valve is that the amount of oil needed to open the valve is much greater. This is because the control piston itself is bigger and thus has a larger control area and stroke length. Chokes (orifice valves) have also been fitted to the load control valve, which allow the open-time of the main control piston to be slowed down. In this example, the closing time will not have been slowed down. This is logical, since the valve still needs to be able to close quickly (to stop crane movement).

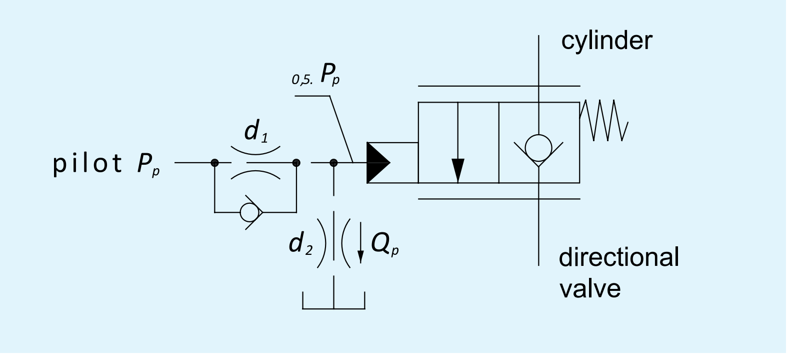

Fig 7.3.B Pilot flow restriction for load control valves

The symbol for the rest position of the non-return valve in the drawing indicates that the valve is leak free in that position. This is a strict requirement for a load bearing cylinder. The two parallel lines above and below the symbol indicate that the valve is proportionally controlled (this means that there are an infinite number of positions between 0% and 100%).

In this example an extra orifice d_2 has been drawn that runs from the control piston to the drain line for the oil leakage. Assume that d_1 and d_2 are the same size. If the pilot pressure is P_p there will be an oil flow from the control mechanism of Q_p across the two orifices, with 50% of the loss of the total control pressure taking place across each of the two orifices. In other words, the control pressure will be halved. The original control pressure ratio i of the valve will, as a result of the opening d_2 be increased to $2i$. By varying the relative size of the openings d_1 and d_2, it is possible to increase the control pressure ratio by a different factor. All this is done to enable the engineer to correct afterwards if the specific drive was not sufficiently stable.

If we use the same stability condition formula as in paragraph 7.1 then one of the very important conditions is that G_{relief} (flow gain of the valve as function of pressure p_a with p_p=0) is equal to zero too. In that case the formula for the stability condition is reduced to the following simplified form:

$$\frac{G_{DCV}.C_{HA}}{\phi.C_{HB}} > G_{pilot} \tag{7.11}$$

This means that stability can be obtained if the value for G_{pilot} can be kept low. In practice this means that a sufficiently high value for the pressure drop across the valve needs to be chosen when the control pressure reaches 100%. The pressure drop across the valve is specified in great detail by the manufacturers, for example as shown in the graph of figure 7.3.C

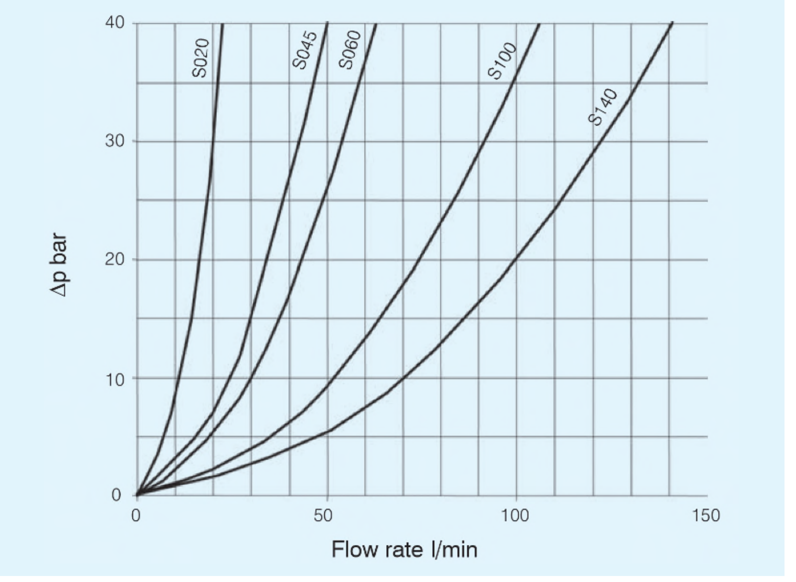

Fig 7.3.C *Pressure drop curves of standard braking valve with different main spools (Courtesy of Bucher Hydraulics)*

In the graph, the pressure drop is specified on the basis of the varying sizes of the main piston that can be chosen for the same cylinder. The rule of thumb for the choice of the correct piston size is that the pressure drop across the piston for the maximum volume flow across the valve (at 100% control pressure) must be at least 35 bar. For a flow capacity of 140 lpm for example, the S140 piston will need to be chosen. The pressure drop across the valve can certainly be much higher than the 35 bar mentioned earlier. If the pilot pressure p_p is smaller than the pressure drop across the valve, then it is possible that the pressure increases to the maximum design pressure of the valve, for example 350 bar.

As with the counterbalance valves also here a very important condition is that the load control valve should be installed as close as possible to the actuator.

The flow capacity of load control valves (up to 650 lpm and a pressure drop ΔP of 35 bar) is higher than that of counter balance valves (up to approximately 450 lpm). If even higher volume flows are required, then it is possible to fit these valves in parallel.

7.4 Mass loaded cylinder with flow control valve

The chance of unstable behaviour in a cylinder fitted with either a counter balance valve or a brake valve will be less if the design complies with the stability conditions. In this context we deliberately talk about "chance". From practical experience we know that the chance of unstable behaviour is larger if the natural frequency ω_o of the driven system is <2 Hz.

Fig 7.4.A J-Lay tower with adjuster system for the Saipem S7000 (Courtesy of Huisman)

The example shown is of the drive for the adjuster system for a 110 meter high J-Lay tower. This tower is used to install steel pipes at the seabed. The vertical angle of the J-Lay can be adjusted to make sure that the ideal angle relative to the seabed is obtained during the pipe laying process (for shallower waters a larger angle from the vertical). This is done with the help of two adjuster systems attached to the tower.

Such an adjuster system consists of a fixed section that is connected to the deck as well as a moving section that is connected to the lugs. The moving section can be adjusted in steps of about 900 mm relative to the fixed section. Fixing into the different positions is achieved by a number of lockable pins. One adjuster system consists of four sets of mechanically linked cylinders that take care of the longitudinal adjustment.

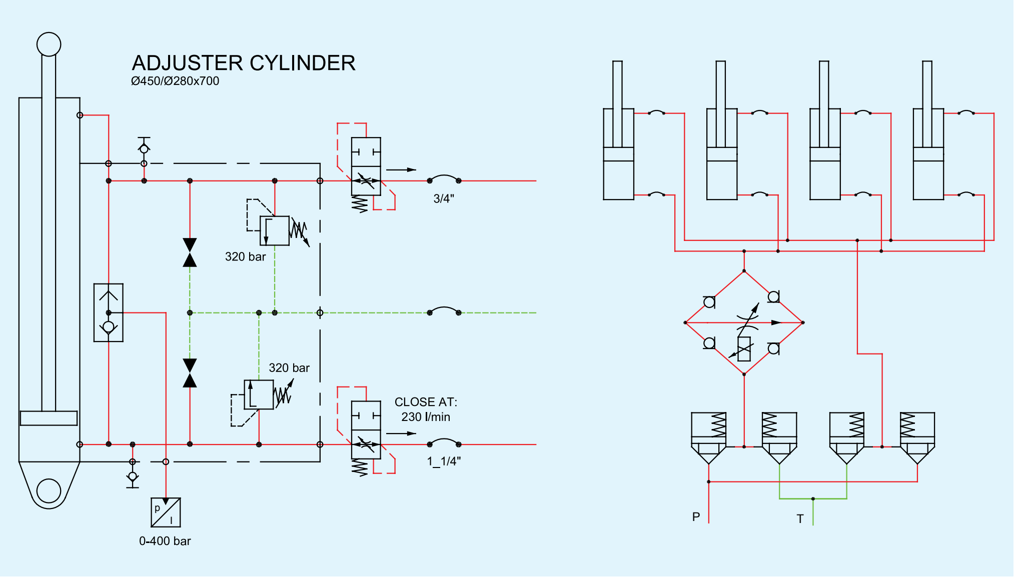

Fig 7.4.B Details of the adjuster system drive, the use of flow control valve instead of braking valve

The four cylinders with a bore of 450 mm are linked in parallel. The maximum total load of the mass for each set of 4 cylinders is here 12.000 kN. These high forces are not so much determined by the static load of the mass of the tower but much more by the dynamic load caused by the pitch movement of the crane vessel. Hose burst valves have been fitted to each cylinder because of the presence of hoses in the connecting pipe work and the possible danger of a break in one of the hoses. Experience teaches that the setting at which the hose burst valve closes must be approximately 50% higher than the maximum volume flow through the valve under operating conditions. If this extra margin is not applied then it is possible that the valve will close under normal operating conditions when the temperature is low and the viscosity is high, even if no hose burst has occurred. A hose burst is detected by the pressure sensor that has been fitted to the system.

Unstable behaviour of the adjuster setting as a result of the use of load control valves on the cylinders can cause very significant damage to the J-Lay tower. That is why this type of valve is not used for this type of application. An electrically adjustable two-way flow control valve is applied instead. This valve is incorporated in a so-called 'Wheatstone Bridge' (consisting of four check valves), which means that the valve gets both the meter-in and meter-out. The direction of the movement is achieved through the use of four cartridge valves instead of one 4/3 directional control valve. Cartridge valves have a much higher flow capacity than directional valves.

A two-way flow control valve combined with one or more cylinders cannot cause any unstable behaviour. This holds as long as the speed of the cylinders is controlled and the cylinder is not included in a positional feedback control circuit. Tests have been carried out on the solution shown, where the position of the moving part was fed back into a positive control circuit. The result was an oscillating movement with a frequency of approximately 1,3 Hz, even by a very low gain in the control circuit. To see a tower move with such a low natural frequency makes a deep impression on every interested engineer. As the proportional flow controller without feedback loop already performed very well, it was decided very quickly that a control circuit with position feedback must not be applied for this application.

Fig 7.4.C *Picture taken during the float over project from Heerema at the west coast of Nigeria. Eight of the shown jacks were used during the float over to lower the 12.300 mTon weighing topside onto its jacking structure. The cylinders were also lowered making use of large flow control valves. (Courtesy of Heerema)*

The solution with flow control valves is used just for J-Lay towers only. In many other applications where the natural frequency ω_o of the drive is low, for example large mobile cranes, jacking systems and backhoe dredgers, flow control valves provide a stable control system.

7.5 Pressure/force control

In pipe laying installations pipe or cable tensioners are often used. A tensioner consists of two or more tracks into which the flexible hose or cable ('flexible') or rigid pipe is clamped. Because of the large weight of the pipe or the flexible that hangs overboard the tensioner must be able to generate a large axial force whilst it must at the same time be possible to let the pipe or flexible sink to the seabed in a controlled way.

The rotational drive for the track can be with hydraulic or electric motors. Either a large torque hydraulic motor or a fast running hydraulic motor with a gearbox, connected directly to the track drive axle, can be used if a hydraulic drive mechanism is required. In this example an electrical drive with gearbox is used.

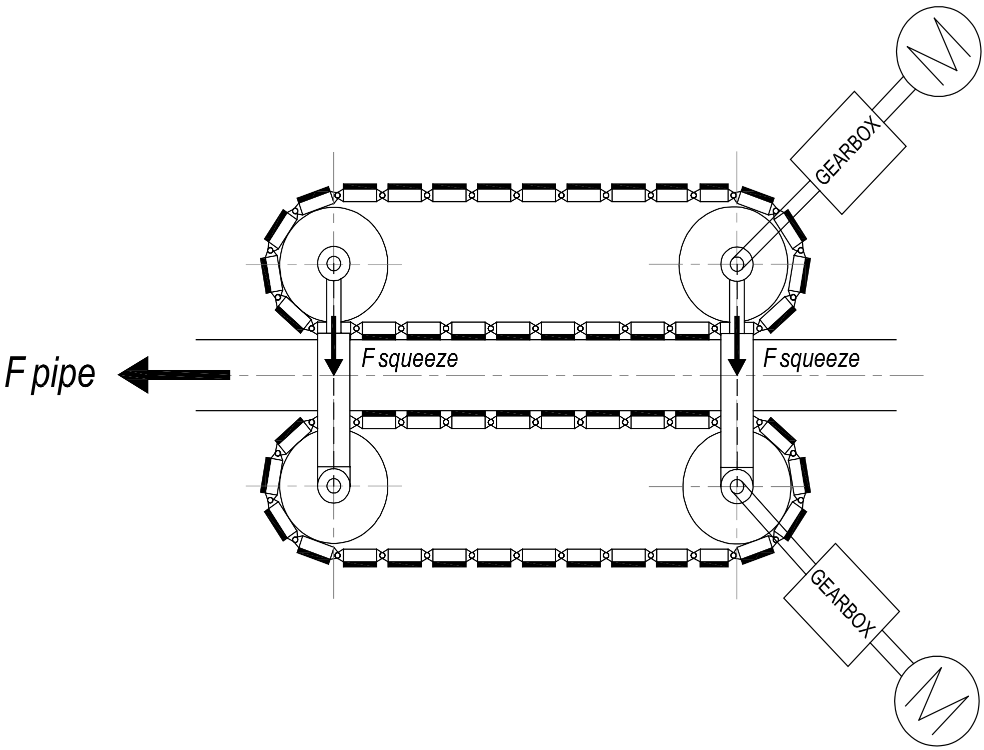

Fig 7.5.A Drive principle for a pipe or flexible lay tensioner

In order to generate a sufficient friction force between the track and the pipe or the flexible, a minimum normal force (also known as the squeeze force) needs to be available. In this case, the squeeze force is generated by the use of two hydraulic cylinders that have a tension load applied to them. There is a limit to the maximum squeeze force to make sure that the pipe or flexible doesn't get damaged.

You will also need to remember that one type of tensioner needs to be able to lay a large number of different diameters and types of pipes and flexible. With the cylinders the gap between the two tracks can be adjusted for the pipe diameter.

The squeeze force that is to be applied is calculated with:

$$F_{sq} = \frac{F_{pipe}}{f_{fr}.n} \tag{7.12}$$

Where:

f_{fr} =	friction coefficient between pipe/ flexible and tensioner pads	F_{sq} =	necessary squeeze force for each cylinder
F_{pipe} =	total longitudinal force in the pipe	n =	number of squeeze cylinders

The friction coefficient f_{fr} can, dependent on the material of the pipe or flexible and combined with the material of the pads on the tensioner, vary from 0,2 to 0,5.

This means that the squeeze force for each cylinder needs to be adjustable, often in a ratio of 1:10 in a very usable range of 50-500 kN.

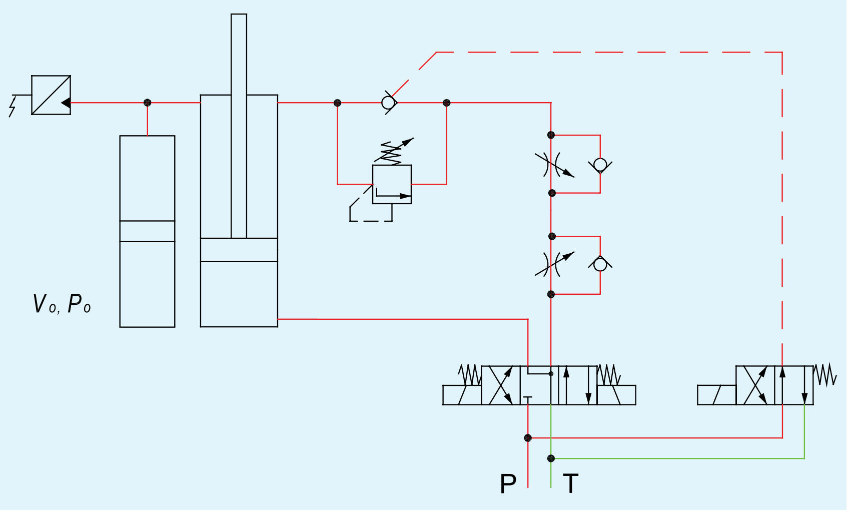

Fig 7.5.B *Principle diagram for the squeeze system of a pipe tensioner*

The basic method for the control of the squeeze cylinder of a tensioner is shown in the diagram in figure 7.4.B. The pressure on the rod side can be increased or reduced with a 4/3 valve. A controlled non-return valve keeps the rod side under pressure. This non-return valve can be opened with a 4/2 valve. Because of the maximum design pressure of the cylinder, a direct acting pressure relief valve has been fitted, to limit the pressure on the rod side. The volume flow to and from the rod side of the cylinder can be set with the help of two individual throttle and check valves. The pressure sensor is there to measure the pressure that has actually been set.

The pressure in the cylinder is influenced by two important factors. The piston makes small movements all the time because of the small diameter variations in the pipe or the flexible. During these movements the pressure in the cylinder needs to be kept as constant as possible. A second and possibly much bigger effect is present in a so called dead-ship situation. In such a situation there is no hydraulic energy available at all, which means that the squeeze pressure cannot be changed at all. In this situation it is possible that the external temperature decreases, for example during the night. This would mean that the temperature of the cylinder and the fluid in it will decrease too, which would reduce the squeeze pressure. The squeeze force can reduce a few percentage points, but too large a drop in the squeeze force can lead to complete loss of control over the pipe or flexible. The decrease in the pressure, approximately 100 bar for every 15°C, is so big that extra measures are required to stop this from happening.

Fig 7.5.C A hydraulic driven pipe tensioner, with the two squeeze cylinders on top, note the small accumulator alongside the cylinders (Courtesy of SAS)

For this application both bladder and piston accumulators are in use. The bladder accumulator is used because it is a cheaper solution. The disadvantage of the bladder type accumulator is that the bladder gets damaged if the pre-charge pressure p_o of the accumulator has not been set correctly, relative to the operating pressures. For a good operation the pre-charge pressure p_o has to be set at approximately 75% of the squeeze pressure p_{sq} . If a piston accumulator is used the same pre-charge pressure has to be used, but a wrong pressure setting will not damage the accumulator.

The temperature of the fluid in the cylinder and the gas in the accumulator are the most important factors in the calculation method. Both fluid volumes, gas and oil, will shrink due to the temperature drop that may occur during the night in a 'dead-ship' situation.

The enclosed oil volume ΔV_{fluid} (cylinder annular end volume + accumulator oil volume at pre-charge conditions) decreases if the temperature of the fluid drops. This oil volume decrease is a gas volume increase ΔV_{gas} . Note that the annular volume of the cylinder differs depending on the diameters and stroke of the applied cylinder. For the calculation the worst case shall be used, i.e. the cylinder is fully retracted, see figure 7.5.B

$$\Delta V_{fl} = -\Delta V_g = V_{fl} \cdot \frac{1}{100} \cdot \frac{(\Delta T)}{15} \qquad (7.13)$$

with:

ΔT =	assumed temperature drop (negative value)	°K	V_s =	gas volume of accumulator	m³
V_{fl} =	volume of oil in cylinder + accumulator	m³			

For the accumulator the standard isothermal calculation method is used according to the ideal gas law.

$$\frac{p_o \cdot V_o}{T_o} = \frac{p_{sq} \cdot V_g}{T_1} \tag{7.14}$$

with:

p_o	=	pre charge pressure of accumulator	N/m²	T_o	=	temperature during precharge	K
p_{sq}	=	squeeze pressure	N/m²	T_1	=	operating temperature	K

At each squeeze pressure and operating temperature the gas volume V_g can be calculated.
The pressure drop Δp in the clamp system at a temperature drop of ΔT can now be calculated with:

$$\frac{p_{sq} \cdot V_g}{T_1} = \frac{(p_{sq} - \Delta p) \cdot (V_g + \Delta V_g)}{(T_1 + \Delta T)} \tag{7.15}$$

$$\Delta p = p_{sq} \cdot \left(1 - \frac{(T_1 + \Delta T)}{T_1} \cdot \frac{V_g}{\left(V_g - \left(V_{fl} \cdot \frac{(\Delta T)}{1500} \right) \right)} \right) \tag{7.16}$$

With the condition that the pre-charge pressure p_o of the accumulator needs to be set at 75% of the operating squeeze pressure p_{sq} in the cylinder, formula 7.16 can be re-written in a more useful form where the necessary gas volume V_o of the accumulator can be calculated from the fluid volume $V_{fl,cyl}$ of the cylinder in its retracted condition.

$$V_o = \frac{V_{fl,cyl}}{\left[0,75 \cdot \frac{T_1}{T_o} \cdot \left(1 + \frac{1}{C} \right) \right] - 1} \tag{7.17}$$

with: $$C = \left(\frac{-T_1 \cdot \frac{\Delta T}{1500} \cdot (p_{sq} - \Delta p)}{p_{sq} \cdot \Delta T + \Delta p \cdot T_1} \right) \tag{7.18}$$

with:

C	=	calculation value	$V_{fl,cyl}$	= volume of cylinder, retracted	m³

For example:

p_{sq}	=	Required squeeze force	400 kN	ΔT	=	Maximum expected temperature drop	-35 °K
S	=	Required mechanical maximum stroke	1,2 m	T_o	=	Pre-charge temperature	293 °K
p_{max}	=	Maximum design pressure	250x10⁵N/m² (250 bar)	T_1	=	Operating temperature	308 °K

The maximum required squeeze force of 400 kN and the maximum design pressure of 250 bar provide the dimensions for the cylinder.

with:

D	=	0,210	m	d	=	0,125	m

We get for the annular cylinder area:

$$A_{ann} = \frac{\pi}{4}\left((D)^2 - (d)^2\right) = 0,02237 \qquad \text{m}^2 \qquad (7.19)$$

And for the squeeze pressure:

$$p_{sq} = \frac{400.10^3}{0,02237} = 179x10^5 \qquad \text{N/m}^2 \text{ (179 bar)} \qquad (7.20)$$

And the pre-charge pressure:

$$p_o = 75\%.p_{sq} = 134x10^5 \qquad \text{N/m}^2 \qquad (7.21)$$

The hydraulic volume that is present in the hydraulic cylinder when its fully retracted can be calculated

with:

$$V_{fl,cyl} = A_{ann} \cdot S = 0,02237 \cdot 1,2 = 0,0268 \qquad \text{m}^3 \qquad (7.22)$$

Our calculation value C becomes:

$$C = \left(\frac{-T_1 \cdot \frac{\Delta T}{1500} \cdot \left(p_{sq} - \Delta p\right)}{p_{sq} \cdot \Delta T + \Delta p \cdot T_1}\right) = \left(\frac{-308 \cdot \frac{-35}{1500} \cdot \left(179 \cdot 10^5 - 30 \cdot 10^5\right)}{179 \cdot 10^5 \cdot (-35) + 30 \cdot 10^5 \cdot 308}\right) = 0,3599 \qquad (7.23)$$

and: $$V_o = \frac{V_{fl,cyl}}{\left[0,75 \cdot \frac{T_1}{T_o} \cdot \left(1 + \frac{1}{C}\right)\right] - 1} = \frac{0,0268}{\left[0,75 \cdot \frac{308}{293} \cdot \left(1 + \frac{1}{0,3599}\right)\right] - 1} = 0,01354 \qquad (7.24)$$

The volume V_o of the accumulator that needs to be selected should thus have a minimum value of 0,01354 m³ = 13,54 litre to guarantee a maximum pressure drop in the cylinder of 30 bar at a temperature drop of 35 °K.

7.6 Suspended force

In the introduction to this chapter we have already mentioned that an electrically driven spindle motor has a very accurate adjustment. The mechanical transmission and spindle axle result in a rigid linear drive mechanism. It was this last characteristic that was particularly important for the development of a 'Line Up Tool' (LUT) for the J-Lay tower. A LUT is used to align a new piece of pipe (with a dual, quadruple or hexagonal joint) with the main pipe. The new joint is welded to the main pipe when the alignment has been achieved. It is very important that the alignment position of the new pipe doesn't drift during the welding process.

If a hydraulic cylinder is used for the adjustment mechanism of the LUT then the stiffness of the oil column becomes a problem. A hydraulic piston that is locked in position, moves 1% of its stroke length for every 100 bar increase in load under the influence of an external load.

Fig 7.6.A *An example of a electrical driven mechanical spindle drive. The drive can be driven by a hydraulic or electric motor. Shown is a spindle with maximum actuation force of 1000 kN (Courtesy of Groneman)*

A mechanical spindle does not have the same disadvantage of elastic compression as the hydraulic cylinder. An additional advantage is that the spindle drive can be designed self braking, which means that its position will be maintained, even if the control drive fails.

A possible disadvantage of the spindle drive is that, as with a hydraulic cylinder, the buckling load for the spindle axle can be a problem for the maximum length that can be applied.

It is possible to limit the lateral force on the spindle by combining a mechanical spindle drive with a parallel hydraulic cylinder. This will give a drive mechanism that is very rigid, has a very accurate position control and that can maintain its position in the case of a power failure.

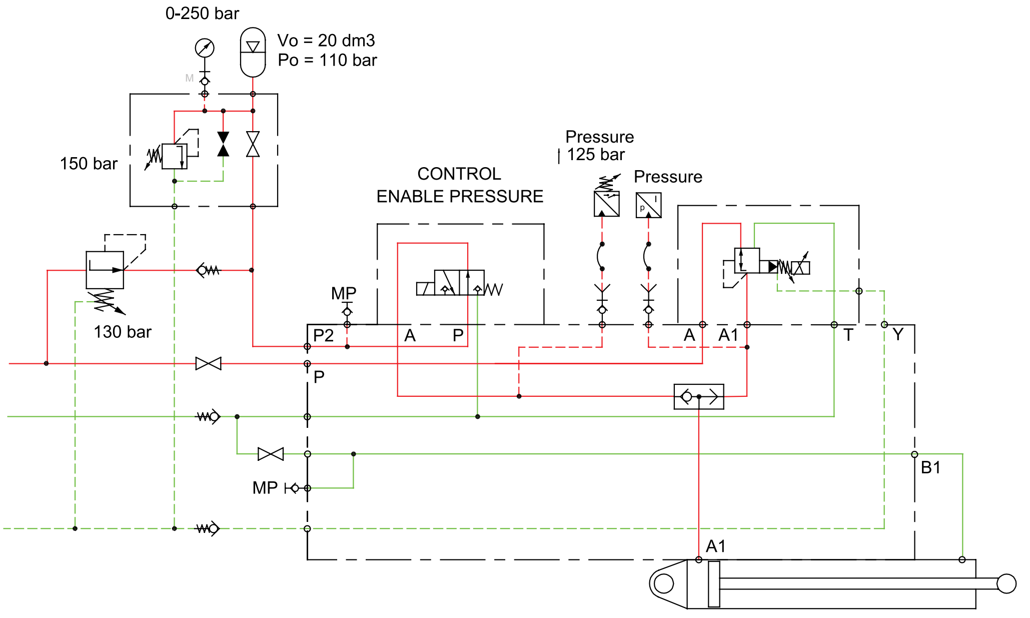

Fig 7.6.B Hydraulic suspension system for an electric driven spindle drive

A manifold has been fitted to the hydraulic cylinder. A proportionally adjustable pressure reducing valve has been fitted on the right hand side. With this valve it is possible to make step less adjustments to the pressure at the bottom end of the cylinder, and thus the supporting force of the cylinder. A pressure sensor and a pressure switch measure the bottom pressure that has been set. The possible danger of an electrically adjustable valve is that if there is a power cut or if the PLC fails, then the signal to the valve will disappear and as a result the hydraulic pressure at the bottom side will fall away too. The spindle drive would then fail immediately because of the too high buckling load. The fail safe position of the 3/2 valve in case of a power failure is the shown idle position.

At low external forces, and thus at low hydraulic pressure, the 3/2 valve is energized. When the pressure sensor signals a pressure rise in the cylinder above 125 bar, the hydraulic accumulator that has been preset to 130 bar will be switched to the bottom side of the piston by the electrically operated 3/2 valve (3/2 valve is de-energised). This will create a sort of safety net of 130 bar. This in turn means that the maximum allowable load on the spindle will not be exceeded.

Make sure that the accumulator has been fitted with a safety relief valve. This valve is a legal requirement in the EU if the installation is to be traded or put into operation in the EU. This legal requirement does not for example apply to mobile offshore units. Inspection institutions like Lloyds, DNV and RINA do however incorporate this requirement where it concerns approval for a hydraulic installation.

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Chapter 8

Linear drives, closed loop systems

Motion Control in Offshore and Dredging

Chapter 8

Linear drives, closed loop systems

In open loop systems, the value for one of the system variables, like position, speed or force, will be reached by controlling the volume flow or pressure with a proportional valve or a variable pump. The process variable is often measured to give the system operator information over the actual position or value of the variable. This signal is however not used for a real feedback control loop. You might say that the system is being regulated with eye feedback.

No stringent requirements are set for a high degree of accuracy of the actual process function. The accuracy is dependent on, amongst other things, the accuracy (hysteresis, linearity or dead time) of the valves or pumps that are being used and the internal leakage that occurs in the valves or actuators as well as the friction of the actuators and the internal system friction itself.

In this chapter we will discuss linear drives where the accuracy and/or control of the movement do play an important role.

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8.1 List of symbols

A	=	full bore area of cylinder	m^2
A_b	=	bottom area of cylinder	m^2
A_{ann}	=	annular area of cylinder	m^2
A_r	=	rod area of cylinder	m^2
$C_{o,min}$	=	minimum stiffness	N/m
D	=	cylinder bore size	m
F_c	=	cylinder static load	kN
F_{dyn}	=	cylinder dynamic load	kN
K_x	=	individual gain	
K_v	=	velocity gain	s^{-1}
L	=	torque arm for cylinders	m
M_r	=	reduced mass	kg
p_L	=	load pressure	bar
p_s	=	maximum system pressure	bar
Q_m	=	maximum valve flow	m^3/s, lpm
Q_n	=	nominal valve flow	m^3/s, lpm
P	=	pressure	N/m^2
S	=	mechanical stroke	m
S_{act}	=	actual stroke	m
T	=	reriod of motion	s
a	=	dimension	mm
b	=	dimension	mm
d	=	cylinder rod size	m
i	=	dimension ratio	
v	=	linear speed	m/s
β	=	damping	
ε	=	tracking error	m
ϕ	=	cylinder area ratio	
ϕ	=	angular rotation	rad
ω_o	=	natural frequency	rad/s

Fig 8.2.B *Photo of the loader system (in black) with the loader arms (in yellow), where a new 48 meter long new pipe has just been lifted from the deck (Courtesy of Huisman)*

8.2 Proportional control

The first example is the drive for a number of loader arms for the J-Lay system of the Balder from Heerema. Figure 8.1.A shows a cross section of the truss construction of the loader system. The function of the loader system is to pick up a pipe with 5 loader arms at the same time from the level of the deck. After the pipe has been picked up, the loader arms all move, in a synchronised way, from the lower to the upper position. After this the whole loader system, together with the attached pipes, moves into a vertical position next to the J-Lay tower, where the pipe is presented to the pipe clamps in the tower by the same loader arms.

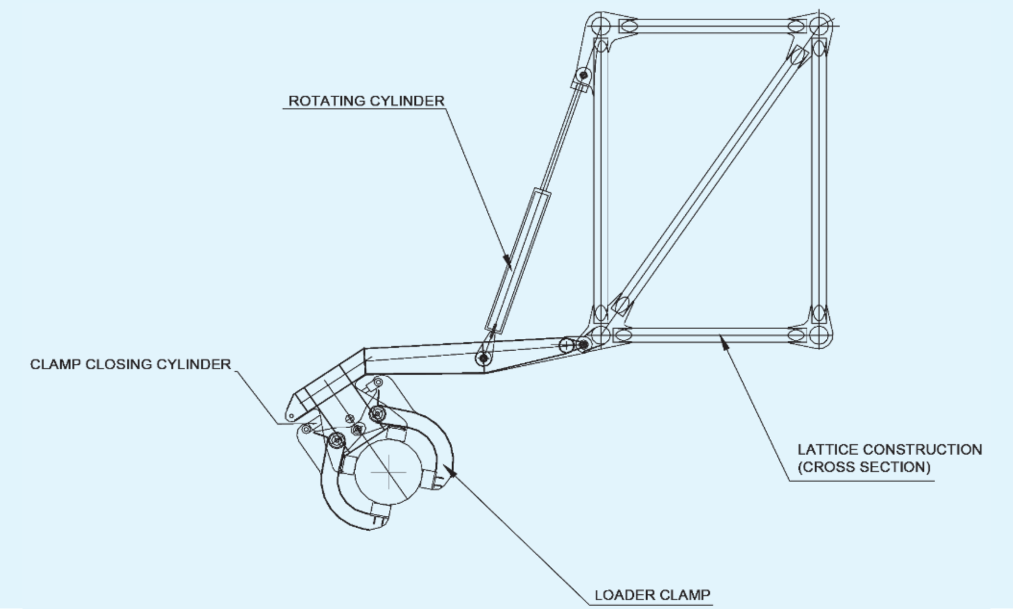

Fig 8.2.A Section view of loader system with one hydraulic driven loader arm (Courtesy of Huisman)

To be able to make a good design of the cylinder drive, it is important to look at the load cycle first.

- The cylinder will be in pulling mode when the pipe is lifted from the deck.
- The necessary accuracy range of the end positions of the cylinder is +/– 5 mm. The synchronicity of the 5 cylinders of the individual loader arms is also in the range of +/- 5 mm. You also need to remember that the loader arms are mechanically coupled to each other through the pipe.
- The position of the loader arms is measured by measuring the angle between the loader arms and the trusses. The reason is that the instruments used to measure the angle wear out less and are less vulnerable in the damp and saline conditions than a linear sensor. (In later examples position sensors are used that are fitted either externally or internally to the cylinder).
- The static force on the cylinder caused by the mass is dependent on the position of the loader arms because the angle of the cylinder relative to the lattice construction changes and because the resultant force of the weight changes as a result of the lever operation too.
- When the cylinder extends, there is normally no load. At that point, the pipe has been delivered to the J-Lay tower. The process of picking up the pipe must however be reversible, which means that it must be possible to put the pipe down under full load conditions.
- Because the 5 loader arms need to move in parallel it is very important to keep the volume flow to the drives to a minimum because the power unit that has been installed has a limited flow capacity. The maximum combined volume flow is limited to 1200 lpm. The limit on the volume flow must not have any consequences for the stipulated cycle times. Cycle times are of great importance on a J-Lay system.

8.2.1 Static and dynamic parameters

The static and most important dynamic parameters can be calculated on the basis of the measurements and angles that can be determined graphically or that can be calculated in, for example, a conversion table or even with the help of a simulator model. The conversion table and the simulator model have the advantage that the parameters for the drive can be determined for the whole stroke of the cylinder.

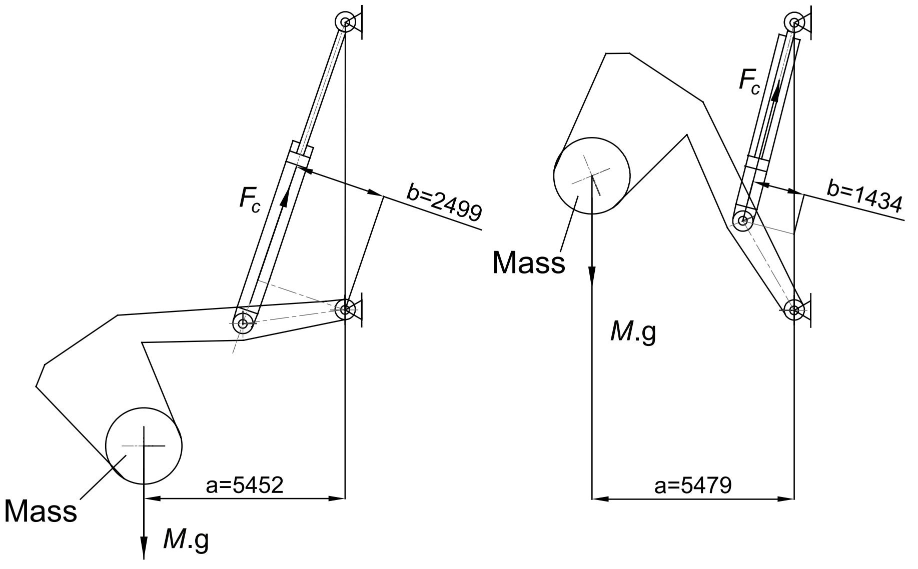

Fig 8.2.1.A Parameters for the static and dynamic calculations

In this example we have used the graphical method with the help of a drawing of the extreme lower and upper positions. This gives the reader a detailed picture and sufficient insight into the effects of the position of the cylinder and for the static and dynamic characteristics.

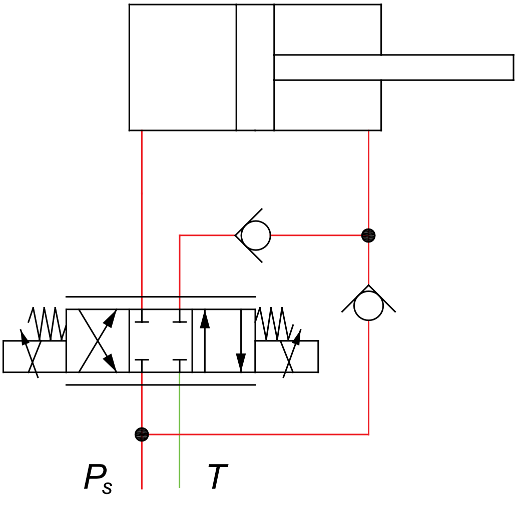

Fig 8.2.1.B Principle diagram for the differential mode

Static parameters in lower/upper position				
Variable	**Equation**	**Value**		**Unit of Measure**
Maximum system pressure *Ps*	For equations see chapter 1	280		bar
Bore diameter, *D*		0,250		m
Rod diameter, *d*		0,125		m
Mechanical stroke, *S*		2,280		m
Mass, *M*, including pipe weight		8000		kg
Bottom area, A_b		0,0491		m2
Rod area, A_r		0,0123		m2
Annular area, A_a		0,0368		m2
Cylinder ratio,		1,33		
Time allowed for extension		20		sec
Time allowed for retraction		25		sec
Fluid flow to extend		336		lpm
Fluid flow for differential extending		84		lpm
Fluid flow to retract		201		lpm
		Position		
		Lower	**Upper**	
Cylinder static load, F_c	$F_c = \frac{M \cdot 9{,}8}{1000} \cdot \frac{a}{b}$ (8.1)	171	299	kN
Load pressure cylinder, P_L when cylinder differential out	$p_L = \frac{F_c}{A_b} + \frac{p_s}{\varphi}$ (8.2)	245	271	bar
Load pressure cylinder, P_L normal retracting	$p_L = \frac{F_c}{A_a}$ (8.3)	46	81	bar

Table 8.2.1.A Static parameters in Upper and Lower position

Explanation of the table:

- The available pump pressure is 280 bar. It is a constant pressure system and is also used for many other drives in the J-Lay tower. This means that the 280 bar inlet pressure of this drive is given.
- For the calculation of the load pressure p_L during the movement from the lower to the upper position, the ring side is controlled whilst the bottom side is connected to the T-line via the proportional valve.
- The cylinder is always under a tension load. This means that buckling load is not an issue.
- The volume flow necessary to fully extend the cylinder in a normal cycle time of 20 sec is approximately 336 lpm. For 5 cylinders working in parallel this would come to a total of 1680 lpm, much more than the available volume flow from the HPU. That's why the volume flow has also been calculated for a differential out-movement. The volume flow per cylinder is then only 84 lpm, a substantially lower demand for the HPU.
- The static cylinder force F_c resulting from the weight is determined by the measurements a and b.
- In the differential circuit, see figure 8.2.1.B, the oil on the annular side is charged to P_s = 280 bar during the out-movement of the cylinder. This means that the maximum system pressure of 280 bar is available on the annular side during the whole of the out-movement. The bottom pressure for the out-movement is thus higher than in the normal out-movement. This is why the load pressure p_L is higher for the differential unit.
- The maximum load pressure p_L = 271 bar in a differential circuit gives an available pressure drop 280 – 271 = 9 bar for the proportional valve. This is just sufficient for the use of a two-way pressure compensator. If such a valve is being used the result is a constant pressure drop across the proportional valve notches of ca 8-10 bar. It should also be mentioned that the load during lowering is normally much lower as during that part of the cycle the pipe is not present.

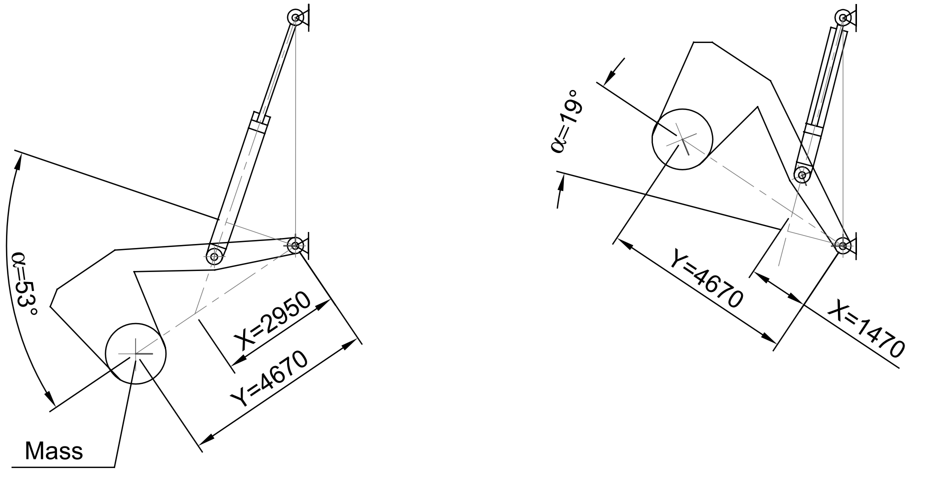

Fig 8.2.1.C Graph to indicate the parameters Y and X for the calculation of the reduced mass M_r

Apart from the calculation of the static characteristics of the drive it is also necessary to determine the dynamic characteristics. The cylinder and the mass of the pipe together form a mass spring system. This means that there will also be a natural frequency of the drive mechanism. The lowest natural frequency is dependent on the reduced mass and the actual stroke length of the cylinder. The reduced mass is not constant because the working angle between the cylinder and the loader arm changes and also because the dimensions of the lever change. This means that it is necessary to find out where the lowest natural frequency occurs. The lowest natural frequency determines the setting for the gains in the positional control circuit. This will also determine what the maximum acceleration is with which the loader arm cylinders can be driven.
The dynamic parameters are also calculated for the lower and upper positions. For the used formulas see paragraph 6.7. and 6.9

Dynamic parameters in lower / upper position				
		Lower	Upper	
a		4670	4670	mm
b		2950	1470	mm
i	$i = \frac{a}{b}$ (8.4)	1,58	3,18	
α		53	19	
cos(α)		0,60	0,95	
Reduced mass M_r	$M_r = M \cdot \frac{i^2}{(\cos\alpha)^2}$ (8.5)	55355	90313	kg
Actual Stroke S_{act}		2,28	0,1	m
$C_{o,min}$	$C_{o,\min} = \frac{A_{ann} \cdot E}{S_{act}}$ (8.6)	$1{,}61 \times 10^7$	$36{,}8 \times 10^7$	N/m
$\omega_{o,min}$	$\omega_{o,\min} = \sqrt{\frac{C_{o.\min}}{M_r}}$ (8.7)	17,1	63,8	rad/s
$W_{o,min}$ Hz		2,7	10,2	Hz

Table 8.2.1.B Dynamic parameters in Upper and Lower position

Explanation of the results.

- The reduced mass M_r is dependent on the measurements *a* and *b* as well as cos α.
- The oil stiffness $C_{o,min}$ is dependent on the effective length of the oil column ie: the actual stroke *S* and the annular area A_r.
- The volumes of the piping lines are neglected.
- The minimum natural frequency $\omega_o = 2,7$ Hz has its lowest value in the lower position. This means that this condition determines the gains that can be set for the position control system. No detailed attention is paid to the gains and the reaction speeds of the drive for this application. A good rule of thumb often used in practice is that the acceleration and deceleration time of the drive must be a factor 5 higher than the lowest vibration period t_L of the drive mechanism. The vibration period is $t_l = 1/\omega_{o,MIN} = 1/2,7 = 0,37$ s. This means that the drive can be accelerated and decelerated from 0 – 100 % in 5 x 0,37 = 1,85 s.

8.2.2 Detailed hydraulic schema

In the detailed hydraulic schema, the differential mode has been built as a sandwich plate. In order to keep the volume flow across the proportional valve as constant as possible, irrespective of the load pressure on the cylinder, a two-way pressure compensator has been installed. The highest load pressure from either the A or B line is fed back to the compensator via a shuttle valve.

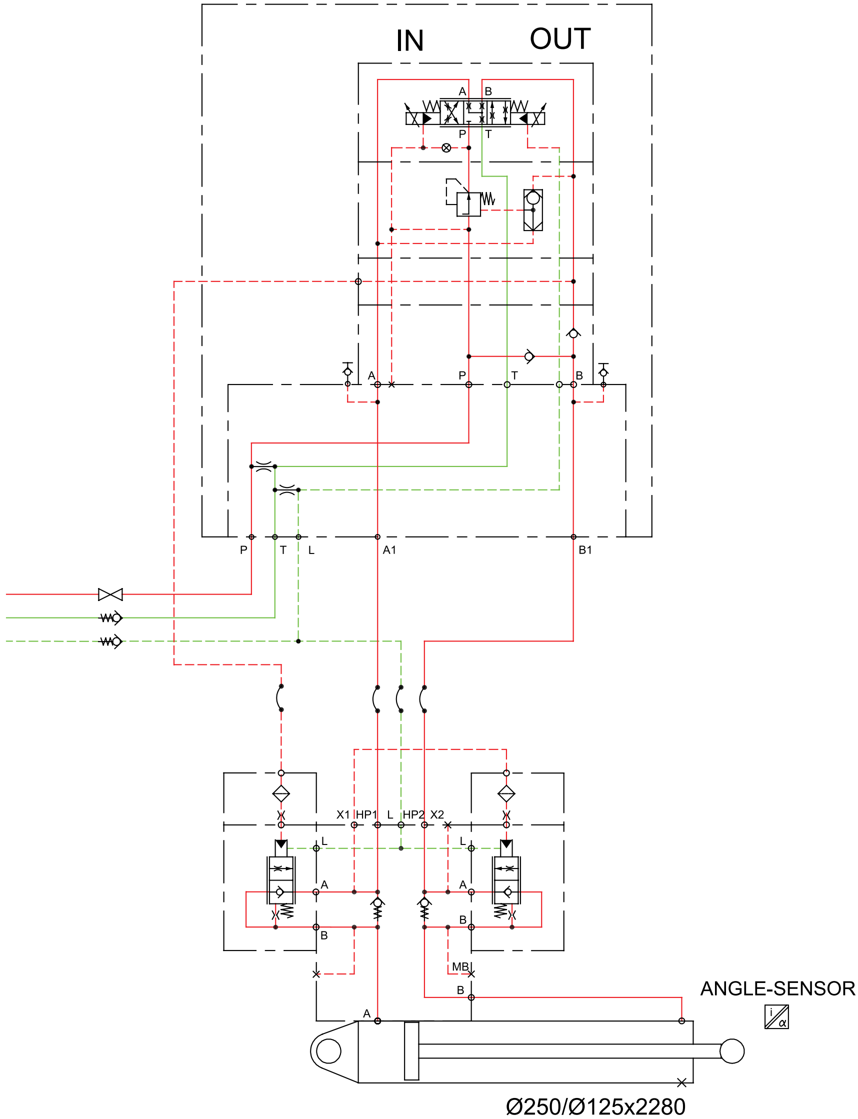

Fig 8.2.2.A Hydraulic diagram for the loader arm drive

The cylinder is connected to the manifold with hoses. This allows easy access to the manifold for maintenance purposes. The disadvantage is that extra valves need to be fitted to block the cylinder in case of a hose burst. A proportional valve does in practice always show leakage from A to B and then to T, which means that brake valves are always required in this type of drive system, where a load has to be kept at a certain level. In this case a braking valve was chosen for both the bottom and ring side of the cylinder. The pilot line for the valve at the bottom side of the cylinder has to be taken from the opposite working line with the help of an extra sandwich plate between the proportional and the differential valve.

Small throttle valves have been fitted between P and T and between T and L. This has been done to create a continuous small 'leak' fluid flow in the system that will keep the manifold at a constant operating temperature. You need to remember that this installation also needs to operate in lower environment temperatures of approximately 0° C.

The following parameters are important for the design of the proportional valve:

- A maximum volume flow through the valve of 201 lpm on the annular side and $Q_m = 201 \times \phi = 201 \times 1{,}33 = 267$ lpm on the bottom side of the cylinder. The nominal flow through a valve is determined by the pressure drop of 5 bar for each pilot valve. Because there is a 2-way compensator with a standard pressure drop of 9 bar, the nominal volume flow through the valve for the design will be:

$$Q_n = Q_m \cdot \sqrt{\frac{5}{9}} = 267x0,74 = 197,6 \quad \text{lpm} \tag{8.8}$$

This volume flow is available with a NG16 or a CETOP7 pilot operated valve.

- Due to the same ϕ ratio the pressure drop across the notch that controls the flow from the bottom end to the T-line is $\phi^2.9$ bar = $1.33^2.9 = 15.9$ bar
- To avoid a detrimental effect on the control mechanism the bandwidth of the valve needs to be at least twice the natural frequency of the system. In this case at least 2 x 2,3= 4,6 Hz. The valve that has been chosen here has a bandwidth of 7 Hz, more than sufficient for this drive mechanism.
- In this case a proportional valve has been chosen where the position of the main spool is measured and fed back electronically. The hysteresis of the valve, which would have been about 6% without the feedback, can thus be reduced to < 0,2%. This means that, together with the 2-way compensator, it is possible control this drive accurately to within 0,5 % of the maximum speed.

8.3 Servo control, application 1

8.3.1 Functional requirements

This is an example of a J-Lay tower that has been placed on a barge. The J-Lay tower moves as a result of the pitch and roll movements of the barge, this means that there will be bending force in two directions on the pipe that goes to the seabed under the tower. If, for example, the pipe laying process is stopped for a certain period of time, say an hour, due to a malfunction, the pipe will be frequently stressed locally as a result of the bending moment. This has a detrimental effect on the fatigue strength of the pipe.

By placing the J-Lay tower on a two directional gimbal ring whilst at the same time actively controlling its movement with the help of hydraulic cylinders, it is possible to neutralise the movement of the tower relative to the earth.

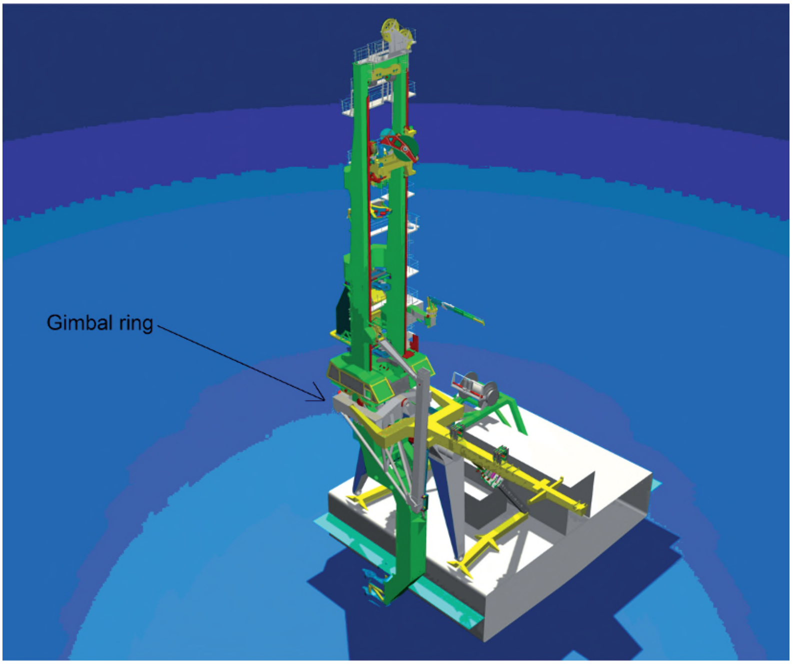

Fig 8.3.1.A J-Lay tower with gimbal ring (Courtesy of Huisman)

The pitch and roll movement (both the position and the speed) of the barge is measured by a Motion Reference Unit (MRU). In essence an MRU measures the acceleration in the direction of the movement that is under consideration. If a large number of electronic filters and integration of the acceleration is applied, the rotational speed signal can be obtained. Further integration will give the angle of the rotation. By making sure that the cylinders make an equal but opposite movement in a positional control circuit, it is possible to compensate for the pitch and roll movement of the tower.

A total of four cylinders for each direction of movement has been chosen. The bottom side of each cylinder is always connected with the ring side of the opposite cylinder. In terms of hydraulics this creates a drive of one actuator with a piston rod that goes all through the cylinder. Per two cylinders the rod ends are connected to each other. Each of these mechanical joints drive the gimbal ring via a lever with an arm of 3,14 metres.

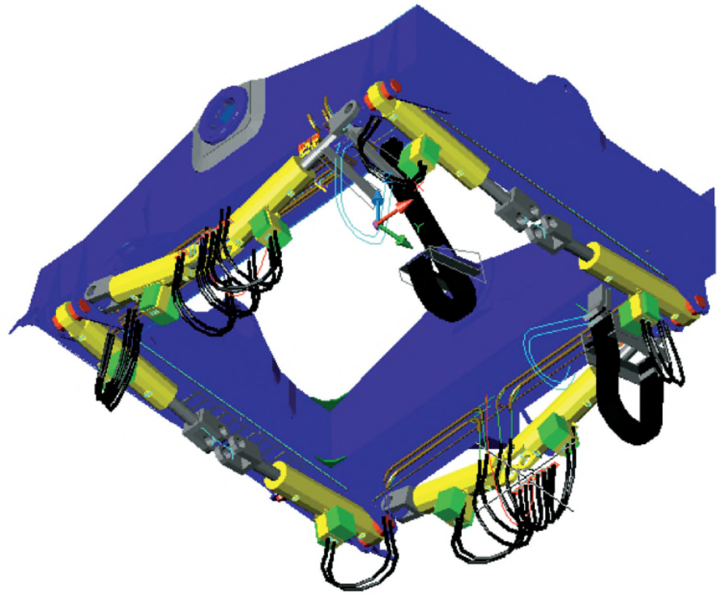

Fig 8.3.1.B Bottom view of the gimbal ring with the total eight cylinders (Courtesy of Huisman)

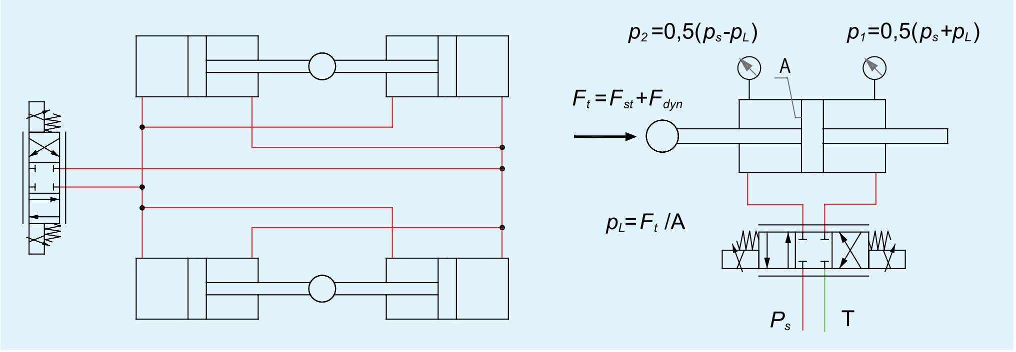

Fig 8.3.1.C Principle of the hydraulic connections of the drive cylinders

Some general functional details:

Total mass of the tower	1200	mton
Period of movement	9.2	s
Pitch amplitude	1.5	degr
Roll amplitude	2	degr
Max static load on cylinders	4500	kN
Mass system pressure HPU	300	bar

As can be seen from the diagram, the load on the cylinders is divided over the two piston areas. If there is no external load, both cylinder compartments will be under half the pump pressure 0.5 x p_s. The load pressure will be divided across the two cylinder compartments in the same way:

$$p_1 = 0,5 \cdot (p_s + p_L) \tag{8.9}$$

and

$$p_2 = 0,5 \cdot (p_s - p_L) \tag{8.10}$$

8.3.2 Calculation of the drive system

As in the previous example, there is a static load F_{st} and a dynamic load F_{dyn} in the two individual directions of movement. The static load is determined by a combination of the static torque on the gimbal resulting from the weight components and the static load from the pipe. The dynamic load is made up by the, in this case very large, mass inertia of the tower and the movement that needs to be achieved. If the movement of the tower relative to the movement of the barge is well compensated, then the dynamic load is, in principle, zero because the tower no longer moves relative to the earth. This means that the rotational acceleration is also equal to zero. Because it needs to be possible to start the system from any possible operational position, the drive needs to be calculated on the basis of the maximum combined static and dynamic loads.

In the calculation used in the example only the roll movement has been assessed. The roll movement is controlled with four cylinders. The calculation for the pitch movement is identical but with a different set of parameters.

Static and dynamic parameters			
Parameter	**Symbol and formula**	**Value**	**Unit of Measure**
Bore diameter, *D*		0,450	m
Rod diameter, *d*		0,250	m
Mechanical stroke, *S*		1,500	m
Bottom area 1 cylinder	$A_b = \frac{\pi}{4} \cdot D^2$ (8.11)	0,159	m²
Annul area 1 cylinder	$A_{ann} = \frac{\pi}{4} \cdot (D^2 - d^2)$ (8.12)	0,11	m²
Total cylinder area *A*	$A = 2 \cdot (A_b + A_{ann})$ (8.13)	0,538	m²
Period of movement	T	9,2	s
Frequency of movement	$\omega = \frac{2\pi}{T}$ (8.14)	0,68	rad/s
Angular amplitude of movement	$\hat{\phi}$ (8.15)	1,5	degr
Angular amplitude of movement	$\hat{\phi}$ (8.16)	0,026	rad
Angular speed	$\hat{\dot{\phi}} = \omega\hat{\phi}$ (8.17)	0,018	rad/s
Torque arm for cylinders, length	L	3,14	m
Linear speed of the cylinder, *v*	$\hat{v} = \hat{\dot{\phi}}.L$ (8.18)	0,056	m/s
Angular acceleration	$\hat{\ddot{\phi}} = \omega\hat{\dot{\phi}}$ (8.19)	0,012	rad/s²
Linear max acceleration of the cylinder, *a*	$a = \hat{\ddot{\phi}}.L$ (8.20)	0,038	m/s²
Reduced mass Mr (given parameter)	M_r	2,1x10⁸	kg
Maximum dynamic load *Fdyn*	$F_{dyn} = M_r \cdot a$ (8.21)	8052	kN
Maximum static load *Fst*	F_{st}	4500	kN
Supply pressure ps	p_s	300	bar
Maximum load pressure *pL*	$p_L = \frac{F_{dyn} + F_{st}}{A}$ (8.22)	233	bar
Available for servo valve(s)	$= p_s - p_L$ (8.23)	67	bar
Total max flow (4 cylinders)	$Q_m = \hat{v}.A$ (8.24)	30,2X10⁻³ 1812	m³/s lpm
Stiffness of cylinders	$C_o = \frac{4.A.10^9}{S}$ (8.25)	1,43x10⁹	N/m
Natural frequency of drive	$\omega_o = \sqrt{\frac{C_o}{M_r}}$ (8.26)	2,61	rad/s

Table 8.3.2.A Specification and Calculation of the static and dynamic parameters for the gimbal drive system

Explanation of the table:

- The available pressure drop for the servo valves of 67 bar is the sum of the pressure drop across the individual control ports (notches). A pressure drop of 25 – 30 bar per port (notch) is generally sufficient for servo valves.
- The stiffness C_o is calculated with an elasticity modulus of $E = 10^9$ N/m². You can find more details about the formula that has been used in paragraph 6.6. Instead of the single servo valve a total of 4 servo valves has been applied in the application. The volume of the pipes between the cylinders and the valves has been ignored because the servo valves have been mounted directly onto the manifold.
- A mechanical stroke length of 0,16 mtr is sufficient to compensate for the dynamic movement of the barge. What we haven't mentioned earlier is that it is possible to put the whole tower at a static angle of 15°.
- The design of the hydraulic drive and control mechanism needs to be such that no dangerous situation develops if one of the cylinders fails. In that situation the three remaining cylinders for each direction of movement must be able to continue to control the movement of the tower. The actual design therefore featured logic valves in the A and B line of each individual servovalve, These logics were able to block any of the servovalves in case of a failure.
- The operating maximum flow for this application ≈ 1800 lpm. As we do not want to have the servovalves being operated at 100% or in some cases maybe just above 100% of their capacity, we choose the nominal flow capacity approximately 30% higher, in this case ≈ 2400 lpm.

Fig 8.3.2 *The J-lay tower with active roll and pitch control in operation on the Polaris of Stolt Offshore (Courtesy of Huisman)*

Control parameters			
Parameter	**Symbol**	**Size**	**Unit of Measure**
Natural damping coefficient (assumption)	β	**0,15**	
Velocity gain K_v	$K_v = \beta.\omega_0$ **(8.27)**	**0,392**	**1/s**
Tracking error	$\varepsilon = \dfrac{\hat{v}}{K_v}$ **(8.28)**	**0,143**	**m**

Table 8.3.2.B Calculation of the control parameters for the gimbal drive system

Explanation of the table:

- The total gain K_v in a position control circuit is limited because of the danger of instability. For the formulas for this calculation, see paragraphs 6.12 and 6.13.
- The tracking error of 0,14 m relative to the total effective stroke for the dynamic compensation of about 0,18 m is very large. This tracking error is purely the result of the limited amplification factors in the position control circuit. In order to improve on the tracking error, a feed forward control mechanism has been applied in this system. For further details about this method, see paragraph 6.13.

8.3.3 Details of the drive system

Several different principles can be applied for the measurement of the actual position. Only sensors that work on the basis of the magnetostrictive principle and that are built into the cylinder are used in the industrial, shipping and offshore industries. It is a very sturdy design that is marketed by several different manufacturers. They are available for working pressures of up to 600 bar.

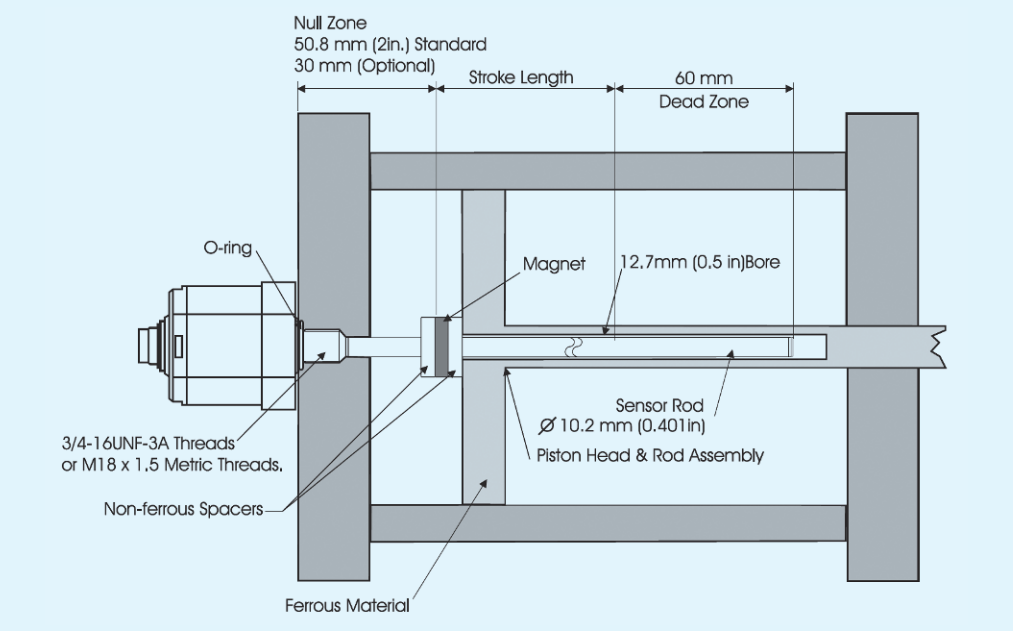

Fig 8.3.3 Example of built-in position transducer (Balluff)

The sensor is built into the cover of the bottom side of the cylinder. The pen of the measuring transducer can be inserted into a hole that is drilled all along the length of the piston rod. A magnet that moves with the piston along the measuring transducer is fitted onto the piston. There is a copper wire inside the measuring transducer. A magnetic field is generated by an electric pulse that is sent through this copper wire by the electronic control system. The pulse moves towards the magnet at a very high constant speed. An opposite pulse is generated as soon as the magnetic field of the pulse reaches the magnet. The electronic system measures the time it takes before the opposite pulse comes in. This time is in the order of nano seconds and is a measure for the distance of the magnet into the cylinder head.

This type of sensor can deliver a speed signal as well as the position signal. The accuracy of the measured position is, dependent on the type of output signal and the length of the sensor (this can be up to 7,5 meters), in the order of 0,1 mm.

One of the disadvantages of building the sensor into the cylinder is that the whole cylinder needs to be taken apart in case of a malfunction. The solution applied in this case is to put the sensor in its own housing and then fit the sensor mechanically alongside the main cylinder. With new designs it is possible to interchange the electronics without having to remove the control pen.

It is not possible to show the detailed information of the hydraulic schematics because of their commercially confidential nature. When this technology is applied to such a large object, where several people will be working on the tower whilst it is actively controlled, a number of extra and special conditions apply for the design of the control mechanism, the hydraulics, the hardware and the software. If one of the components fails, the movement of the cylinders cannot suddenly be stopped. If this happens, the top of the tower, approximately 85 meters high, would suddenly be accelerated horizontally within a very short period of time, which would mean that staff would not be able to remain upright/standing. To prove that this will not happen, a Failure Mode and Effect Analyses (FMEA) needs to be carried out in which the effect of failure is checked for each component and what detection methods are available to detect the failure as early as possible. The result of this type of analysis often leads to change of the concept design. For details on FMEA see chapter 12.

8.3.4 Commissioning

The commissioning of an active control mechanism for such a large tower demands special precautionary measures. A single fault in the software or a faulty setting for the amplification factors in the positional control circuit can lead to wrong movements of the cylinders with the potential for a total force capacity of 15.000 kN for each direction of movement.

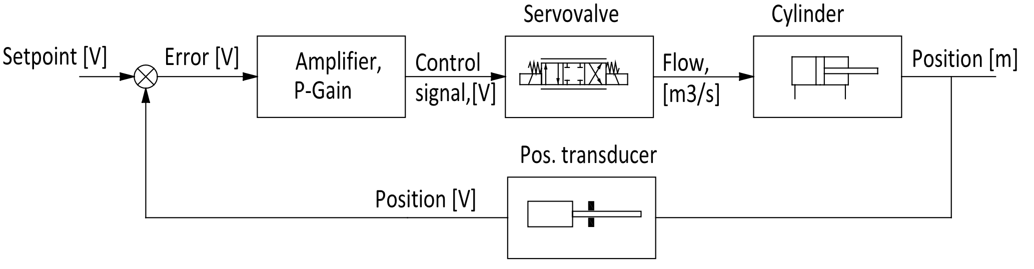

Fig 8.3.4.A Block diagram of the position control loop with their individual gains

The maximum total gain $K_v=K_1.K_2.K_3.K_4$ for the whole control circuit was already calculated, see table 8.2.2. It is the product of the individual gains of all the control components that form part of the design. The individual gains of each component were already established during the detailed design. This means that the maximum achievable amplification factor K_1 for the P-gain can be calculated with:

$$K_2 = \frac{2400[lpm]}{10[V]} = 0,004 \quad m^3/V.s \qquad (8.29)$$

$$K_3 = \frac{1}{A} = \frac{1}{0,538} = 1,86 \quad m^{-2} \qquad (8.30)$$

$$K_4 = \frac{10[V]}{1,5[m]} = 6,66 \quad V/m \qquad (8.31)$$

Where:

$$K_v = K_1.K_2.K_3.K_4 = K_1 x 0,004 x 1,86 x 6,66 = 0,392 \qquad (8.32)$$

Which means that:

$K_1 = 7,91$ [V/V]

The setting for the gain K_1 has to be achieved in steps. The first test for the drive system is to generate a stepped input signal with a very low amplification factor $K1$ (maximum 10% of the calculated value). By measuring either the positional input signal or the hydraulic pressures in the cylinders the natural frequency becomes immediately apparent.

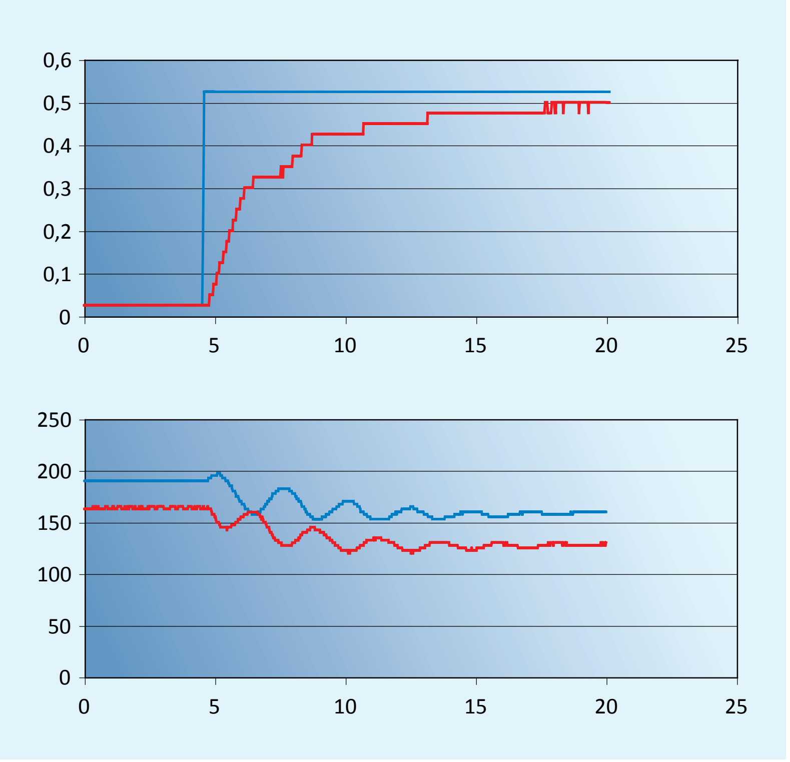

Fig 8.3.4.B *Stepresponse of the hydraulic drive. In the upper graph a step setpoint has been introduced at t=5 s. The position response shows the behavior of a first order system. In the lower graph the pressure measurements of the cylinder are shown.*

The drive system oscillated towards the desired position with a period of vibration of approximately $T = 2{,}5$ sec. The corresponding natural frequency is: $\omega_o = 2.\pi/T = 2{,}5$ rad/s. This shows that the measured natural frequency of the whole drive system is 4% lower than the calculated value for which assumptions had to be made about the rotational inertia of the whole tower. With the new, measured, natural frequency ω_o it is possible to calculate the new maximum amplification factor K_1.

In practice however, the amplification factor is gradually increased until a stepped response from the closed control circuit becomes visible, after which the eventual value of the amplification factor is set at 50% of that value.

Because the tracking error in this type of installation is very large due to the low natural frequency and achievable amplification factors a feed forward control system has been installed. After all, the MRU has access to the rotational speed of the barge as an output signal. After a bit of re-calculation it is possible to determine the desired speed of the piston, after which the speed signal can be fed directly into the servo valves.
With the feed forward implemented, this control system was able to correct for approximately 90% of the original motions, i.e. the remaining motions of the tower were reduced by 90%.

8.4 Servo control, application 2

This application is the drive system for a platform where the movement is controlled with the help of six actuators, also known as a hexapod. The base frame of this platform is put onto a ship that moves in all six degrees of freedom due to the waves. If the correct control mechanism is used for the six cylinders, it is possible to keep the platform in a static position relative to the earth. This system is used to transfer service personnel from a moving ship to a fixed object; the mast of a windmill or a fixed platform. The transfer of personnel by any other means is often limited by the actual wave condition of the waves and the large ship movements resulting from it.

8.4.1 Functional requirements

The reason why we discuss this application is to show that the design of a seemingly complicated drive system can be simplified to the level of the simple calculations that have been discussed in earlier chapters of this book.

Fig 8.4.1 The active controlled platform Ampelmann in offshore operation (Courtesy of Ampelmann)

The six different movements of the ship consist of:

- Moving up and down (heave)
- Moving left and right (sway)
- Moving forward and backward (surge)
- Tilting forward and backward (pitch)
- Turning left and right (yaw)
- Tilting side to side (roll)

These individual movements are converted by a computer system into the 6 individual movements of the hydraulic actuators.

The important functional parameters of this system are:

• Mechanical stroke for each cylinder	2	m
• Maximum linear piston velocity	1,5	m/s
• Shortest period of movement	1,5	s
• Maximum linear dynamic load		
• Pushing	78	kN
• Pulling	36	kN
• Mass for dynamic acceleration	2000	kg per cylinder
• Supply pressure	245	bar

The starting point for the movement is that we are dealing with sinusoidal type movements. Although the wave movement of a ship doesn't appear to be sinusoidal, this movement can be described as the sum of a number of sinusoidal waves in which higher order frequencies exist on top of the base harmonic frequency. The shortest period listed, describes the maximum frequency (= 1/1,5 = 0,66 [Hz]) that the drive mechanism needs to achieve from a sinusoidal type input signal.

The hydraulic power for this installation is generated by 2 sets of diesel-hydraulic pumps. The advantage of the diesel powered pumps is that the whole installation can be put onto any random ship without being dependent on the presence of a hydraulic pump or electric power. Because the platform can only be held still as long as there is hydraulic power, it is necessary to make sure that there is a provision to ensure that system will continue to operate for 20 seconds after failure of one of the diesel-hydraulic sets. A hydraulic accumulator appears to be a suitable solution for this problem.
These 20 seconds allows any passenger that is passing the gangway to either stay on the supply vessel or to go to the platform or windmill mast.

8.4.2 Calculation of the control system

The hydraulic design of the actuators for this type of control mechanism can be compared with the design for a mass loaded cylinder, controlled with a servo valve. The task is to make use of standard available industrial components as much as possible. In a flight simulator for example, actuators with hydrostatic bearings are applied to keep the friction to a minimum. This restriction does not apply in this case, which means that is possible to use standard industrial cylinders with a good offshore coating for the piston rod.

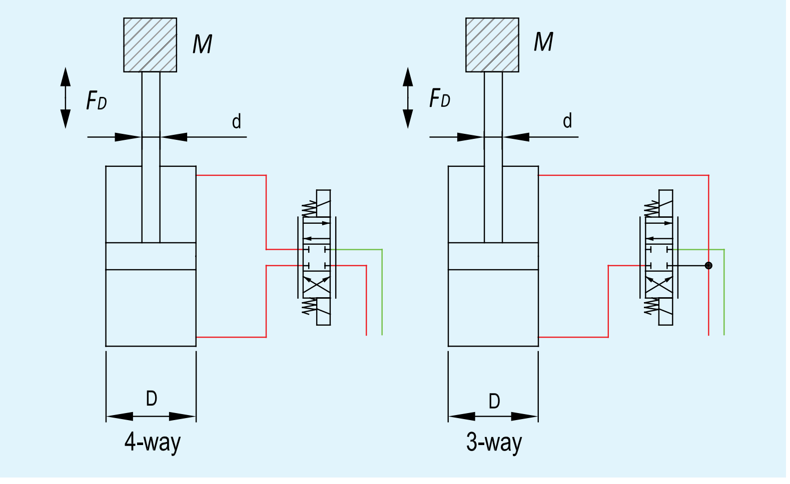

Fig 8.4.2.A Comparison of 4-way and 3-way control of the actuator

The next choice is between a so-called 4-way or 3-way control mechanism. Both systems use a constant pressure system for the inlet pressure. A number of important hydraulic parameters can be determined on the basis of a comparison between these two types of system.

Static and dynamic parameters				
Parameter	**Symbol/Formula**	**4-way**	**3-way**	**Unit of Measure**
Bore diameter	D	120	120	mm
Rod diameter	d	90	90	mm
Mechanical stroke	S	2000	2000	mm
Bottom area	A_b	0,113	0,113	m²
Annular area	A_a	0,495	0,495	m²
Maximum longitudinal velocity	v	1,5	1,5	m/s
Bottom flow	Q_b	$16{,}9\text{x}10^{-3}$ 1018	$16{,}9\text{x}10^{-3}$ 1018	m³/s lpm
Annular flow	Q_a	$7{,}41\text{x}10^{-3}$ 445	$7{,}41\text{x}10^{-3}$ 445	m³/s lpm
Nett pump flow	Q_p	$16{,}9\text{x}10^{-3}$ 1018	$9{,}55\text{x}10^{-3}$ 573	m³/s lpm
HPU supply pressure	p_s	$245\text{x}10^{5}$ 245	$245\text{x}10^{5}$ 245	N/m² bar
Hydraulic power consumption	(8.33)	416	234	kW
Reduced mass	M_r	2000	2000	kg
Stiffness of the actuator, 4-way control mode	$C_{o,\min} = \dfrac{E \times \left(\sqrt{A_1} + \sqrt{A_2}\right)^2}{S + \dfrac{V_{l1}}{A_1} + \dfrac{V_{l2}}{A_2}}$ (8.34)	$1{,}56\text{x}10^{7}$		N/m
Stiffness of the actuator, 3-way control mode	$C_{o,\min} = \dfrac{A^2 \times E}{A \times S + V_l}$ (8.35)		$0{,}56\text{x}10^{7}$	N/m
Natural frequency	$\omega_o = \sqrt{\dfrac{C_{o,\min}}{M_r}}$ (8.36)	88,4	53,2	rad/s
Natural frequency	$\omega_o^* = \dfrac{\omega_o}{2\pi}$ (8.37)	14,1	8,5	Hz

Table 8.4.2.A Specification and calculation of the static and dynamic parameters of the Ampelmann drive system in two different control mode

Explanation of the table:

- A 4-way control mechanism delivers a higher total stiffness C_o due to the spring stiffness on both sides of the piston. This in turn will give a higher natural frequency ω_o. The maximum flow from the HPU which is determined by the bottom flow, is however much higher than with a 3-way control mechanism.
- By the outward stroke of a 3-way control mechanism the volume flow is added to the inlet side of the valve port. This means that the maximum net volume flow and the net hydraulic power can be reduced considerably.
- The lowest natural frequency of the 3-way control mechanism of 8,5 Hz is more than adequate for the highest possible movement frequency of 0,66 Hz. The rule of thumb is that the natural frequency of an actuator needs to be at least twice that of the maximum movement frequency that needs to be generated.
- On the basis of these simple calculations it is advisable to design the 3-way control mechanism in more detail.

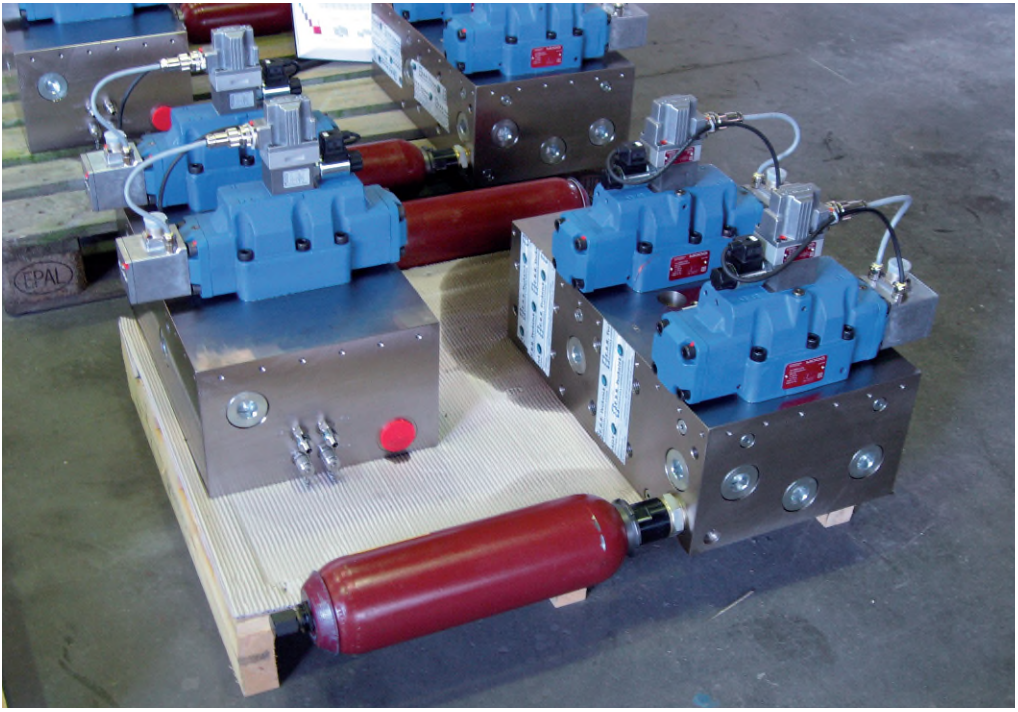

Fig 8.4.2.B Two of the six hydraulic control manifolds during assembly, clearly shown is the dual servovalve arrangement (Courtesy of Ampelmann)

Load pressure calculations			
Parameter	**Formula**	**Value**	**Unit of Measure**
Maximum dynamic load pushing	+Fd	78	kN
Maximum dynamic load pulling	-Fd	36	kN
Bottom pressure at 0-Load	$p_{L,0} = \frac{A_a}{A_b} \cdot p_s$ (8.38)	107×10^5	N/m²
Bottom pressure at +Load	$p_{L,+} = p_{L,0} + \frac{F_{d,+}}{A_b}$ (8.39)	176×10^5	N/m²
Bottom pressure at –Load	$p_{L,+} = p_{L,0} + \frac{F_{d,-}}{A_b}$ (8.40)	75×10^5	N/m²

Table 8.4.2.B Calculation of the load pressure (pressure induced by the load)

Explanation of the table:

The load pressure at the bottom side varies between 75 bar and 176 bar. This means that there is a sufficiently high pressure drop available for the servo valve that controls the cylinder: 245 – 176 = 69 bar for the direction of the volume flow to the cylinder and 75 – 0 = 75 bar for the volume flow from the cylinder to the return pipe.

On the basis of this load pressure calculation we can conclude that the bore and the rod diameter of the cylinder could be reduced This means that the volume flows to and from the cylinder will then reduce relative to the square of the diameter reduction. What the calculation doesn't show is what the result for the minimum rod diameter is as a result of the buckling load of the cylinder. The buckling load calculation is part of the confidential information of this project. For the purpose of this example it suffices to say that the chosen rod diameter was not allowed to be chosen smaller.

8.4.3 Back-up accumulator

One of the conditions for the control system is that it must be able to continue functioning for a while (about 20 s) if one of the two diesel driven pump sets fails. It is clear that, apart from this, the volume flow to a hydraulic actuator is not continuous but that it has a sinusoidal pattern. This means that the volume flow to the six cylinders also has a sinusoidal pattern . The movements that need to be compensated also depend on the movements of the ship for a certain state of the sea. A small supplier type ship will behave differently from a large crane ship. The heading of the ship relative to the wave movement is an important parameter in this.

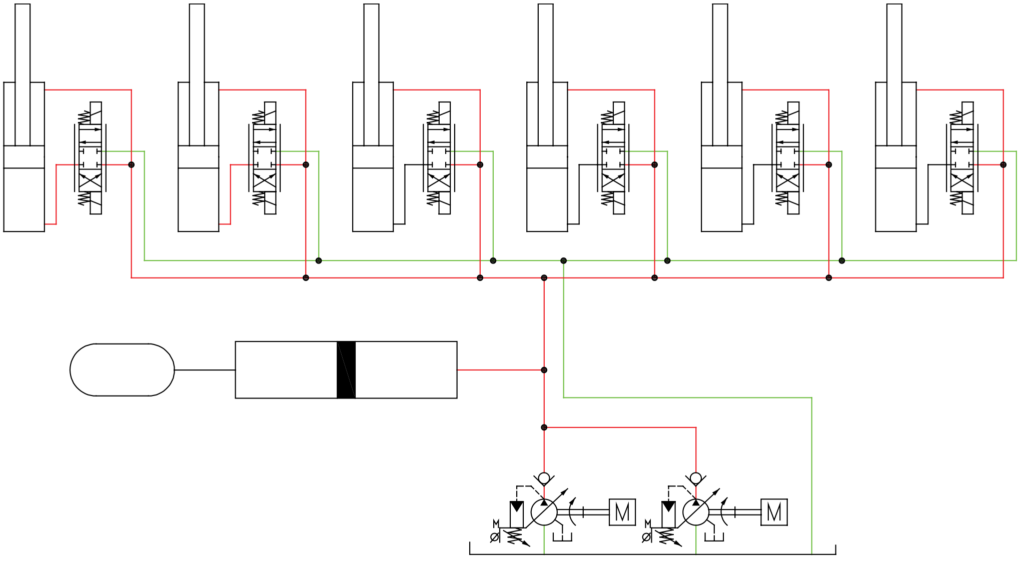

Fig 8.4.3.A Basic hydraulic diagram of the Ampelmann platform with a piston type accumulator as backup system

The simulator models and software programs currently available provide information about the movements that a ship makes in a particular state of the sea in a particular location on the world-wide oceans (Response Amplitude Operators or RAO). Using these parameters in software that translates the ship's movements into the linear movements of the six actuators, provides time based information about the total volume flow that the system requires.

With these volume flow data it is possible to make a simulation model of the two pump sets, the six hydraulic actuators linked to them and a hydraulic accumulator with back-up gas bottles. In the model it is possible to process the detailed dynamic characteristics of a pump with pressure control and of the behaviour of an accumulator and the gas bottles linked to it. Such a dynamic model gives instant insight into the system pressure, the parameters for the accumulator station and ultimately if the system will be able to operate after one of the pump sets has failed.

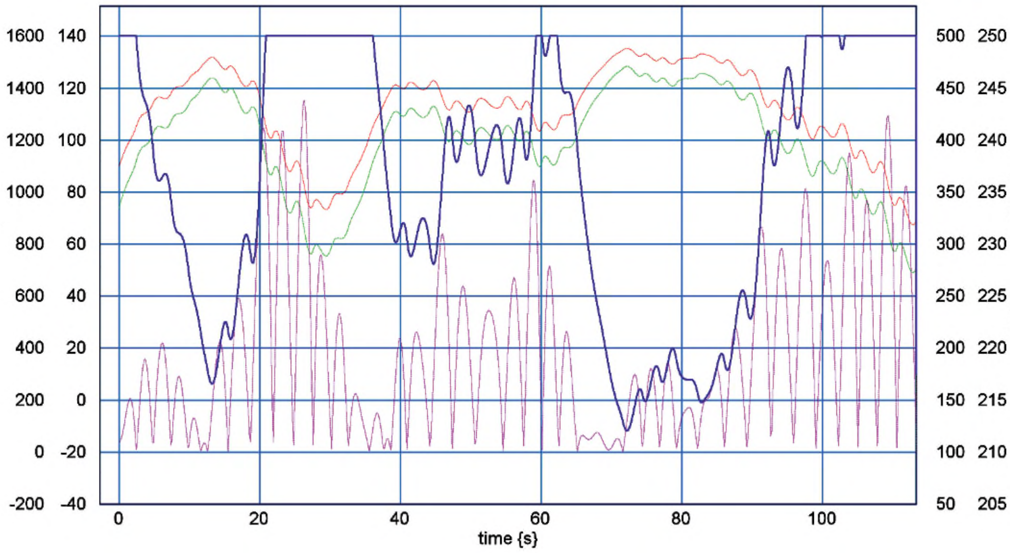

Fig 8.4.3.B Results of the simulation model for the behavior of the accumulator together with the two variable pumps

The graph shows the following parameters:

- Blue continuous line : Nett pump flow (120-500 lpm)
- Purple dash-dash line : Total flow for all six actuators (0-1350 lpm)
- Green dash-dot line : Actual system pressure (225-247 bar)
- Red dotted line : Accumulator hydraulic volume (70-138 dm^3)

In these results it is easy to recognize that the pumps will reach their maximum capacity if the volume flow to the actuators is high and that in that case the accumulator will deliver power to the system for a short period of time. When the volume flow to the actuators subsequently drops, the pressure controlled pumps will maintain their maximum output for a while to provide the accumulator with new energy. When the accumulator delivers energy the system pressure drops and the pressure will rise again when the output from the pumps is higher than the uptake from the six actuators.

8.4.4 Design detail

Just as with the previous application of an active control mechanism for a J-Lay tower, the safety aspects for this type of application are very high too. The system is after all only used for the transfer of people between a moving ship to a static platform. The end users of this type of machinery are the oil related industries for whom the safety aspects are more important than the operational availability. In these circumstances an FMEA is a very good way to investigate the failure characteristics of a system. To guarantee safe operation of the system, two servo valves and two individual position sensors are applied, one valve will be the active valve, whilst the other operates as a backup.A number of important functions of the active valve is controlled, which makes it possible to activate the backup valve as soon as a fault has been detected.

High demands are put on the position control system. This can immediately be translated into demand for an accurate servo valve with an internal position feedback of the main piston. Special attention is paid to the dimensions of the piston in the neutral position of the valve.

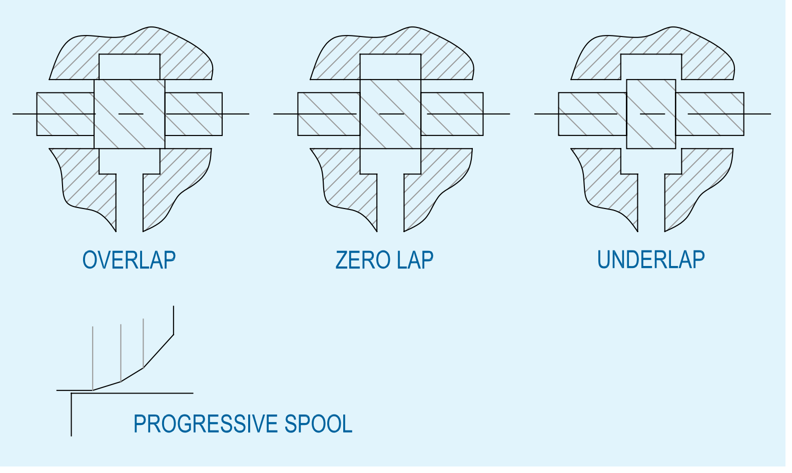

Fig 8.4.4 Different spool dimensions for different control functions

There are three different types of spools.

A zero lap piston will close all the control ports when it is in the exact neutral position and functions best, in control technology terms, in a servo system. In terms of production technology it is difficult to manufacture a spool that achieves this zero lap condition for all control ports.
The overlap spool is the most common spool. It is clear that it needs to be moved a little bit before it is possible to generate a volume flow. This means that this type of valve has a 'dead zone'. In terms of control technology this a disaster.

An underlap valve is an excellent alternative for a servo system and is, in terms of control technology, the equivalent of the zero lap spool. The disadvantage is that there will be a flow from the P-port to the A-port and from the B-port to the T-port in the neutral position. For the necessary valve capacity for this project an extra flow of 50 lpm for each valve could be a normal value. That means that a total of: 6 servo valves x 2 for each manifold x 50 lpm = 600 lpm flow would be lost as leakage.
A very good alternative for the overlap piston is the piston in the drawing with a progressive spool. The volume 'amplification' of this valve increases in three steps as the control system moves the valve from the neutral position. Applying this type of valve provides a 'rest' movement of only 1,5 cm for a typical wave height of H_s = 1,5 m when applied to the moving platform.

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Chapter 9

Heave compensation

Motion Control in Offshore and Dredging

Chapter 9

Heave compensation

In the offshore and dredging industries activities are carried out during which machinery that is put overboard comes into contact with the seabed whilst the ships or platforms are moving. For dredgers this could be the suction pipe of a hopper dredger. For offshore installations the examples would include the drilling of a well, installing heavy objects on the seabed, or the laying of pipes with roll and pitch compensated j-lay towers. The movements of the ship resulting from the sea swell and wind waves mean that large variations occur in the forces that are exerted on the suction pipe, the drill pipe, the lifting cable or in the pipelines that are being installed on the seabed. Heave compensation systems are used to compensate, as much as possible, for these movements. A distinction is made between linear and rotating compensation systems. In this chapter we will discuss linear compensation systems.

P. Albers, *Motion Control in Offshore and Dredging*, DOI 10.1007/978-90-481-8803-1_9,

9.1 List of symbols

A_b	=	bottom area of cylinder	m^2
A_{load}	=	effective area of submerged structure	m^2
A_w	=	effective area of cable	m^2
C_d	=	friction coefficient	
C_{HC}	=	stiffness of compensator	N/m
F	=	force	N, kN
H_v	=	heave	m
L_w	=	length of cable	m
M	=	mass	kg
M_a	=	added mass	kg
M_s	=	submerged mass	kg
MW	=	molecular weight	
R_g	=	specific gas constant	$m^2/s^2.°K$
T	=	temperature	°K
V	=	volume	m^3
b	=	critical pressure ratio	
d_w	=	cable diameter	m
m_{cr}	=	critical mass flow	kg/s
m_w	=	mass of cable	kg
p	=	pressure	Pa
t	=	time	s
x	=	heave of vessel	m
y_1	=	vertical displacement of passive cylinder	m
y_2	=	vertical displacement of structure	m
κ	=	adiabatic gas constant	
ρ	=	density	kg/m^3
ρ_w	=	submerged density of solid bar	kg/m^3
ρ_{water}	=	density of seawater	kg/m^3

9.2 Passive heave compensation, quasi-static behaviour

The simplest form of heave compensation is the passive compensator.

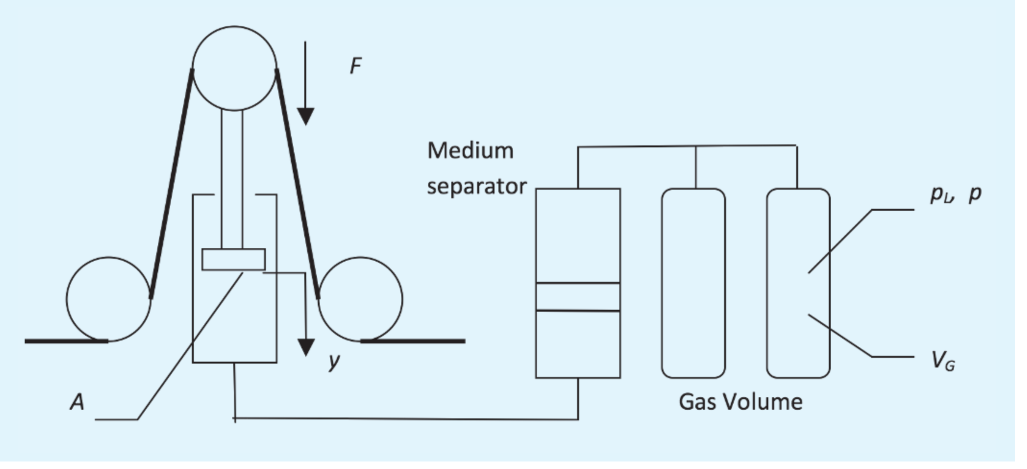

Fig 9.2.A Basic principle of passive heave compensation.

In this type of system a cable is guided over a sheave that is supported by a hydraulic cylinder. The hydraulic part of the cylinder is separated from a connecting gas volume by a medium separating piston type accumulator. The combined cylinder/gas system forms a spring mechanism of which both the pre-tension (the gas pressure in the system) and the stiffness (the volume of the linked gas container) can be set. It is not strictly necessary to use a piston type accumulator. Bladder types are also being used. Piston types have the advantage that larger sizes are available.

The load pressure p_L is calculated by:

$$p_L = \frac{F}{A} \tag{9.1}$$

The gas pressure can be calculated with the ideal gas equation:

$$p \cdot V^{\kappa} = \text{constant} \tag{9.2}$$

Where:

V	=	actual gas volume	m^3	κ (kappa)	=	adiabatic gas constant
p	=	gas pressure	N/m^2			

The period of the wave movement is often in the range of 6 to 12 s, which means that the gas will behave adiabatically.
The value of kappa varies with the temperature and the pressure. The value for kappa is between 1,4 and 1,8 for operating pressures of 500 bar and temperatures between -10 °C and +50 °C (see paragraph 4.4.4 for more details).

The stiffness of the spring, expressed as

$$C = \frac{\Delta F}{\Delta y} \tag{9.3}$$

can be found by calculating the derived function *dP/dV* around a certain operating point. It is fairly meaningless to use the derivative for the complete mechanical range of the compensator , because the gas does not behave in a linear way and because the gas volume varies a lot during the movement. If the stiffness is to be used in simulation programs the best approach is to define the operating limits and linearise the stiffness in that region.

A more practical approach is to tabulate the values of the different parameters at the extreme operational positions of the cylinder and to find the stiffness from these data.

Suppose that the medium separator has a total net volume of V_M and that the pipes connecting the medium separator to the cylinder and the gas volume have a volume of V_L and the total gas volume is V_G.

Note: the volume of the medium separator is always ca 10% larger than the maximum displacement volume of the bottom end of the cylinder. This is to prevent the piston inside the medium separator from hitting its end positions. A separating piston is not used in the medium separator for dredging applications. There the gas is in direct contact with the hydraulic fluid.

Also suppose that the medium separator operates around its mid volume. Then the volume of the gas in that mid operating position is:

$$V_1 = 0.5 \cdot V_M + V_L + V_G \tag{9.4}$$

Fig 9.2.B A magnificent picture of a jack up vessel at work in an offshore wind farm. A similar vessel from A2SEA , the Sea Jack (formal Jumping Jack) can be lowered and lifted with fast acting winches. When the legs are lowered to the seabed, or the barge is being lifted, the lifting winches are applied with passive heave compensator systems to limit the force variations due to the heave of the barge. (Picture by Courtesy of A2SEA)

And the gas volume changes with:

$$\Delta V = \pm \frac{1}{4}.H_V.A \qquad (9.5)$$

Where:

A	=	bottom area of the cylinder	m²	V	=	volume	m³
H_v	=	heave	m				

If we neglect the mechanical and fluid friction in the cylinder and the friction in the mechanical sheaves then the gas pressures (and hydraulic pressures) in the compensator behave according to the adiabatic gas law and can be written as:

$$p_{MAX} = p_L \cdot \left(\frac{V_1}{V_1 - 0.25xH_VxA} \right)^{\kappa} \qquad (9.6)$$

And: $$F_{MAX} = \frac{p_{MAX}.A}{2} \qquad (9.7)$$

And: $$F_{MAX} = \frac{p_{MAX}.A}{2} \qquad (9.8)$$

And: $$F_{MIN} = \frac{p_{MIN}.A}{2} \qquad (9.9)$$

If the vessel moves with a total vertical heave of H_V, then the stiffness of the compensator is expressed as:

$$C_{HC} = \frac{F_{MAX} - F_{MIN}}{H_V} \qquad (9.10)$$

Or: $$C_{HC} = p_L.A \frac{\left(\left(\frac{V_1}{V_1 - 0,25xH_VxA} \right)^{\kappa} - \left(\frac{V_1}{V_1 + 0,25xH_VxA} \right)^{\kappa} \right)}{2.H_V} \qquad (9.11)$$

For example:

Specifications of heave compensator			
Parameter	**Variable**	**Value**	**Unit of measure**
Bore diameter cylinder		0,360	m
Maximum expected heave	H_V	2,5	m
Average external Load	F	900	kN
Volume medium separator		0,15	m³
Piping volume		0,01	m³
Connected gas volume		0,4	m³
Adiabatic constant	see 4.4.4	1,7	
Water depth		500	m
Calculation of parameters			
Bottom area	A	0,0102	m²
Average gas pressure	p_L	177x10⁵	N/m²
Average gas volume	V_1	0,485	m³
Maximum gas pressure	p_{MAX}	225x10⁵	N/m²
Minimum gas pressure	p_{MIN}	143x10⁵	N/m²
Compensator stiffness	C_{HC}	1.65x10⁵	N/m

Table 9.2 Calculation results for the parameters of a passive heave compensator

If a compensation system is applied to the suction pipe on a dredger the pipe will partially rest on the bottom. The stiffness C_{HC} of the compensator will be set dependent on the sea/riverbed conditions. A high stiffness is required if the bottom is soft so that the suction pipe is kept at a more or less constant suction depth.

The stiffness C_{HC} is directly proportional to the average pressure p_L, as can be seen from the formula. This means that if we want to achieve the same stiffness for the heave compensation system for any depth of water, then we will need to link more gas bottles to the passive system when the water is deep, because there is a larger cable weight under the water.

Fig 9.2.C A set of gas bottles, total volume 21600 dm³, with design pressure of 300 bar for a riser tensioner system (Courtesy of Huisman)

The gas pressure must be in accordance with the actual work load before a passive system can be used. To achieve this, a main valve is placed in the pipe between the passive cylinder and the medium separator. This valve can only be opened when the pressure on both sides of the valve is more or less the same. This valve can in fact only be opened automatically because the load due to the heave movement varies continuously.

Imagine that it was possible to open the valve manually. Then, if the gas pressure were to be too low or too high, the passive cylinder could move immediately to the end position, which could happen at very high speeds. Large impact damage could result for either the cylinder or the compensator.

In passive heave compensation systems for crane ships the maximum speed of the compensation system is determined by the maximum vertical speed of the tip of the crane. Speeds of 1,25 m/s are not unusual which means that the passive cylinder must be able to move at a speed of 0,625 m/s. For the passive cylinder dimensions shown above this would mean a maximum operational volume flow of 3825 lpm. In order to keep the friction in the passive system to a minimum, it will be necessary to make the dimensions of the pipe work and the main valve very large. Only cartridge type valves of the size type NG100 to NG150 can be considered for the main valve.

9.3 Passive heave compensation dynamics

The calculation of the quasi static characteristics gives a good insight into the workings of a passive heave compensator. To understand the various aspects of the operational use of the system it will be necessary to assess the dynamic characteristics.

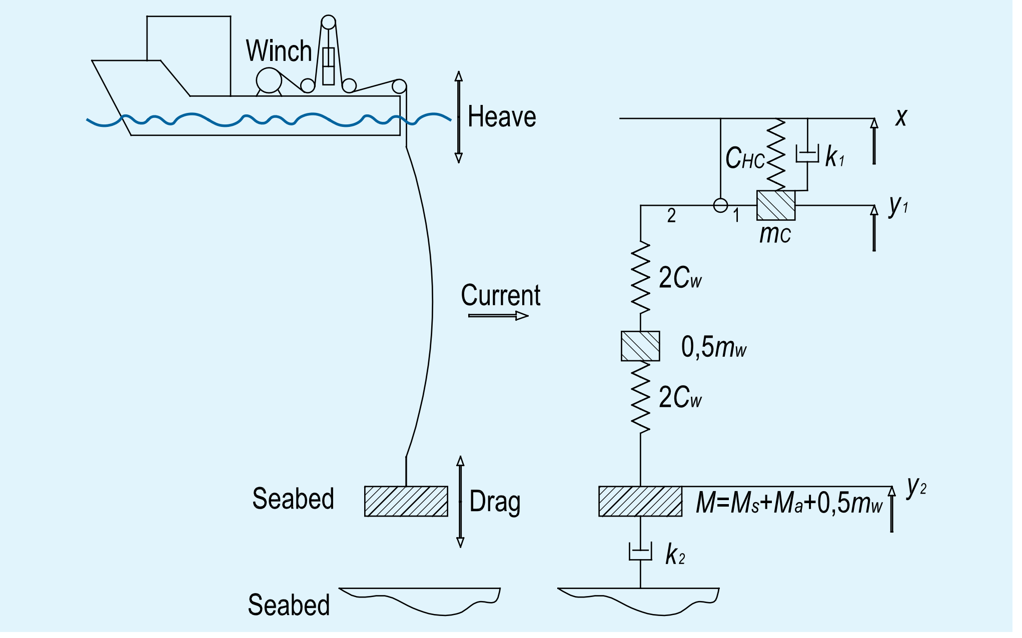

Fig 9.3.A Mass-spring model of a passive heave compensating system

The drawing shows a situation where a load is put on the seabed from a moving ship. The heave is represented by x and the movement of the structure is represented by y_2. The calculation in paragraph 9.1 has already shown that the passive heave compensator can be represented by a spring mechanism with stiffness C_{HC}. The cable is guided across a sheave on top of the passive cylinder. In the figure this is represented by the mass m_c and the lever with ratio 1:2. The mass m_C, representing the cylinder rod plus the guiding sheave, has values of 4000 to 6000 kg. The ratio 1:2 comes from the fact that the cable is guided by the sheave where the cable achieves the double speed of the passive heave cylinder speed. As result of this lever also the load forces result in a double force for the passive heave cylinder. Note that when the cable has a larger reeve ratio this has to be implemented in the block diagram. As with every mass-spring system a damper is present with friction k_1.
The friction k_1 of the passive heave cylinder is very low. This due to the very low flow losses in the large diameter piping in between the medium separator and the passive cylinder and because of the large size of the open/close valve in this pipe connection. For the cylinder with parameters of table 9.2 the friction k_1 is estimated at a value of 5800 N.s/m.

The total mass of the structure is made up of its own submerged mass M_s and the added mass M_a. The submerged mass is the structure's own mass minus the mass of the water displaced by the submersion, the well known law of Archimedes. For a structure that is being moved under water, it is also necessary to add an extra mass that is dependent of the shape of the structure. This does in fact concern the extra water mass that needs to be dragged along. In practice the added mass can be between 1 and 10 times the own mass of the structure.

$$M = M_s + M_a \tag{9.12}$$

The lifting cable forms an extra mechanical spring. The mass of the cable itself is, in case of large cable forces and depth of water, also high relative to the net mass of the structure.
The formula for the surface area A_w of the cable with a diameter d_w is:

$$A_w = 70\%.\frac{\pi}{4}.d_w^2 \qquad (9.13)$$

The factor of 70% is there because the cable is not contiguous due to the spaces between the strands.
The mass of the cable is:

$$m_w = \rho_w.A_w.L_w \qquad (9.14)$$

Where:

ρ_w	=	submerged density of a solid bar (about 6800 kg/m³)	L_w	=	length of the cable	m

The cable thickness is 90 mm for a crane with a single cable lifting capacity of 200 mTon (metric Ton). The submerged own mass of the cable is, according to the above formula, 30 kg/m. A lifting cable does in fact need to be modeled as being made up of a large number of shorter cables, each with its own part mass and stiffness. In a simulator model it is relatively easy to achieve this type of split.

If the cable is split in more and more pieces, more higher order harmonic movement frequencies will occur. This will slow down the simulator model. In practice it suffices to use a model where the cable is split into two parts and where, for the dynamic characteristics, half the mass of the cable is added to the mass of the load. In figure 9.3.A half of the cable mass is added to the structure mass.
The stiffness of a cable is:

$$C_w = \frac{A_w.E_w}{L_w} \qquad (9.15)$$

Where:

E_w	=	modulus of elasticity ≈ 100×10^9	N/m²	C_w	=	cable stiffness	N/m

If the water depth = cable length of 500 m then the stiffness C_w of the cable is $0{,}89 \times 10^6$ N/m. The stiffness of each section of 250 m is $1{,}78 \times 10^6$ N/m.

The cable also moves in a horizontal direction under the influence of the current and, in that way, creates a spring stiffness and damping for the system. The stiffness and damping resulting from this movement can be resolved into a horizontal and vertical component. Only the vertical components are of interest for the dynamic assessment of the structure. Research has found that the effects of currents on the vertical stiffness of the lifting cable can be ignored for loads of up to 200 mTon.

The resistance force created by the speed with which the load moves through the water is equal to:

$$F_d = -\frac{1}{2}.\rho_{water}.A_{load}.C_d.\left|\dot{y}\right|.\dot{y} \ \text{[N]} \qquad (9.16)$$

Where:

A_{load}	= cross section of the load	m²	C_d	=	friction coefficient	

The friction coefficient for a square object is in the range of 0,5 to 0,6.

As the drag function is non-linear we cannot use it in differential equations etc. If we make the function linear for a vertical speed range of ≈ 1 m/s then we get an approximation:

$$F_d = -\frac{1}{2}.\rho_{water}.A_{load}.C_d.2\,\dot{y}\cdot sign\left(\dot{y}\right) \tag{9.17}$$

or

$$F_d = -k_2\cdot\dot{y}\cdot sign\left(\dot{y}\right) \tag{9.18}$$

with

sign(y) = sign value of dy/dt	k_2	=	friction	N.s/m

This means we can represent the drag forces by a linear damper with friction k_2.
To gain insight into the dynamics of a passive system in a relatively easy way, the situation will be assessed for shallow water where the higher order effects of the stiffness and own mass of the lifting cable can be ignored.

The block diagram for a passive heave compensator can be represented by the following model:

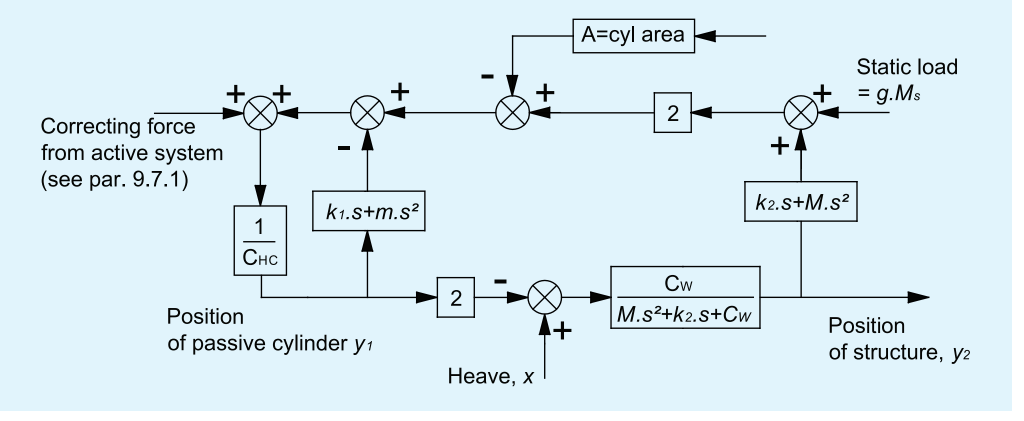

Fig 9.3.B Presentation of the block diagram for a passive compensator

For reference see figure 9.3.A. For simplicity the mass of the cable is being neglected. Although the effect of the cable mass is large on the total mass *M*, the effect of the additional spring mechanism in the system is of a secondary order. With s being the Laplace operator the block $k_1.s+m.s^2$ represents the friction and mass/spring forces on the passive cylinder. The blocks with the gain of 2 are for the sheave arrangement as explained before.

The external load forces are made up of static and dynamic forces. The static external force is cancelled out by the static force of the passive cylinder as long as the gas pressure in the passive part corresponds to the static load. The dynamic forces are retrieved from the position of the structure y_2, multiplied with the block $k_2.s+M.s^2$. If an active system would be present, the location where its correcting forces adapt tot eh system are also shown. More about active systems is specified in paragraph 9.7.1.

The total transfer function of the block diagram of figure 9.3.B is a 6th order function. It is very difficult to find the natural frequencies of this system in the frequency domain. Effects like coulomb friction from cylinder seals or cable sheaves are also not yet implemented. As coulomb friction introduces non-linear effects, frequency domain analysis would not even be possible anymore.

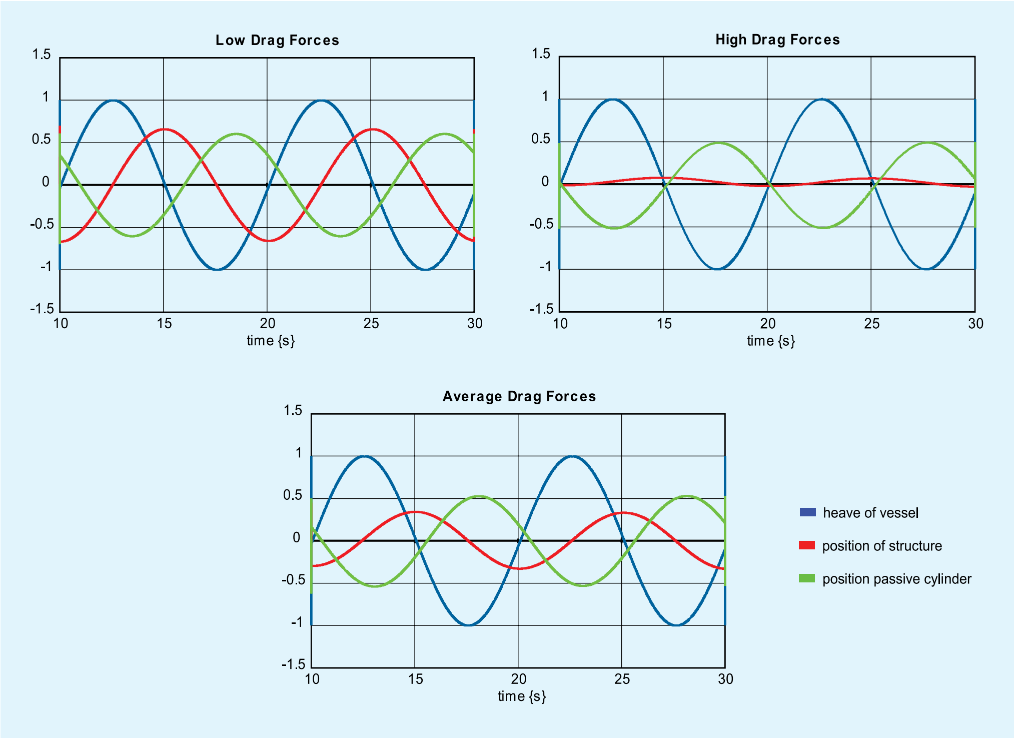

Fig. 9.3.C *Time response on a regular wave with a period of 10 s for the model of a passive Heave Compensator for different drag forces of the submerged structure.*

The regular wave is simulated with a sinusoidal that is equal to the typical wave period. In practice this match is fairly good but we must remember that higher order periods are also present. The simulation shows that the amplitude of the movement of the structure is dependent on the drag force or damping of the submerged structure.

The mass of the structure and the stiffness of the cable together form a mass spring system with its own characteristic natural frequency. If the frequency of the wave movement corresponds with the natural frequency of this mass spring system, then a resonance rise of the movement may take place during which the forces on the cable can easily become a factor 2 higher and it is even possible that a 'slack wire' situation will occur.
Because of the unknown factor caused by the water added mass, it is difficult to predict precisely at what depth of water this resonance rise will occur, but let us suppose that the added mass is 100% of the submerged mass. For the system in our example in table 9.2 the cable stiffness varies from $2C_w = 3{,}56 \times 10^6$ N/m at a depth of 250 meter to $2C_w = 0{,}46 \times 10^6$ N/m at a depth of 2000 meter. In this case the natural frequency of the mass of 90 mTon with an added mass of 90 mTon varies from approximately 0,7 Hz to a value of 0,23 Hz where the total mass M has to be increased with 50% of the additional cable mass of 52500 kg .
It is certain that a resonance rise may take place and that it will take place at some stage during the lowering of the structure. It depends on the drag force or damping of the system to what extent the forces on the cable will vary when this occurs. In practice the operator will lower the load as fast as possible when variations in the load on the cable are observed and a resonance rise is apparently taking place because the natural frequency of the cable has been reached. The resulting effect is that extra drag forces will develop as a result of the large lowering speed which in turn will result in a higher damping factor.

9.4 Offshore applications

For the applications we will discuss in this paragraph, the load has either been placed on or is in partial contact with the seabed. In those cases the heave compensation is only required to compensate for the forces resulting from contact with a moving ship. The variations in the forces in the compensator are then solely determined by the absolute vertical movement of the ship and the stiffness of the compensator.

The drill pipe connects the rig surface equipment with the bottom hole assembly and the drill bit. Applying a steady weight onto the drill bit is essential for efficient drilling, but unfortunately upward heave of the vessel can lift the drill bit clear of the bottom. Similarly, downward heave movement can result in an excessive weight on the drill bit. To compensate for the rig movements a drill pipe compensator is used. The most conventional type is the in-line type. Over time, the drilling vessels became larger with better motion characteristics, and as the drilling depths increased, the compensator became bigger and was placed at the top of the derrick, known as the crown block. They can carry loads of up to 450 mTon with a compensating stroke of 8,3 m. The connected gas volume is rather large as the load variation on the drill bit is typically less than 4-5 %.

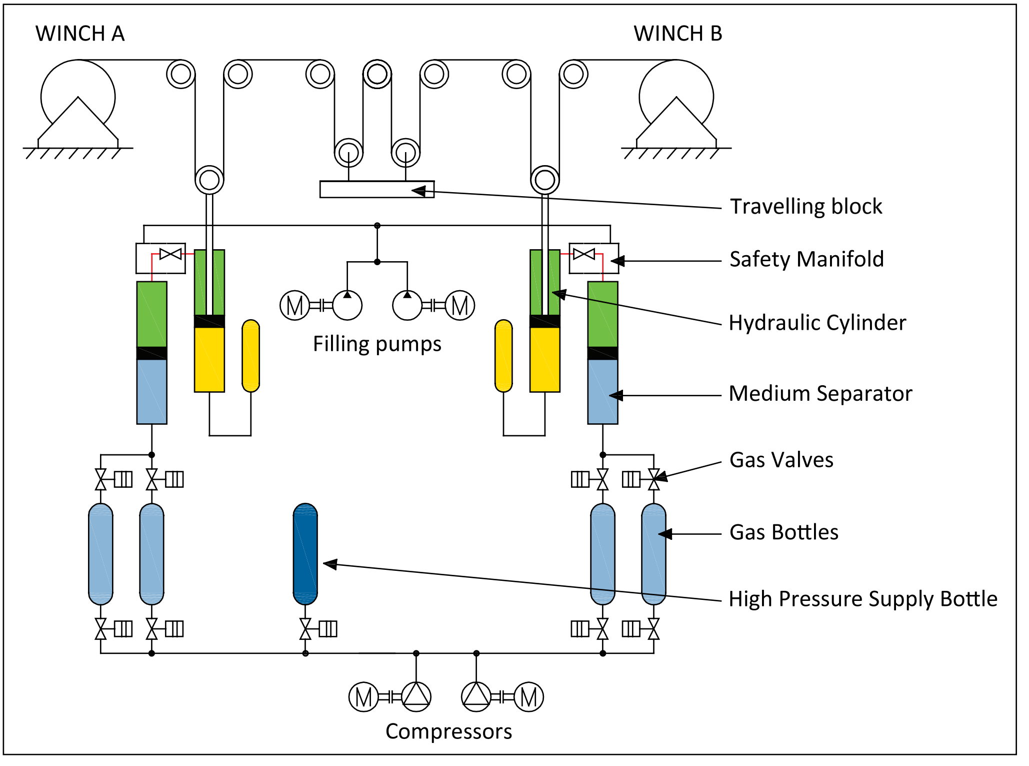

Fig 9.4.A Drill string compensator, existing of 2 heave compensators in parallel

A modern version of the crown block compensator is achieved by applying two sets of compensators, fitted in parallel. If there is a malfunction in one of the compensators, then the other compensator takes over its functionality immediately.
Marine riser tensioner

The marine riser is a pipe that extends from the drilling platform down to the seafloor. Drilling mud and cuttings from the borehole are returned to the surface through the riser. The top of the riser is attached to the drillship while its bottom is secured at seafloor level. Due to the vessels heave the same effects

Fig 9.4.B Wire line tensioner (Courtesy of Bosch Rexroth)

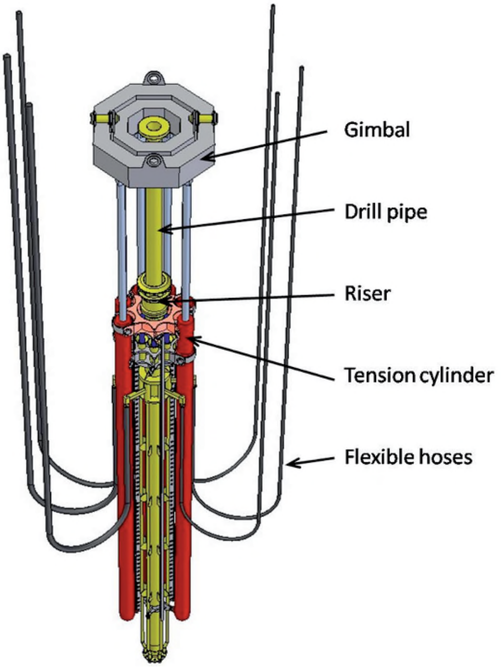

Fig 9.4.C Direct riser tensioner, existing off 6 individual cylinders (Courtesy of Huisman

occur here with the riser as with the drill pipe. To prevent the riser from buckling a riser tensioner system is provided. Two main types exist; the wire line tensioner and the direct acting tensioner.

Wire line tensioners are, as the word says, used to keep wire lines tensioned. In the Offshore industry many wire lines are used to hold or hoist a specific load. Relative movements can cause a wire line to become slack and therefore causing dangerous shock loads. Wire line tensioners are used in drilling packages as well as in transport applications such as offshore cranes. The wire line tensioners are hydraulic/ pneumatic springs made-up from a combination of hydraulic cylinders, piston accumulators, pressure vessels and hydraulic and pneumatic valves. The cables run over sheaves.

Wire line riser tensioners maintain tensioning on the marine riser with the help of wire lines. The direct riser tensioner is mechanically connected directly to the riser tension ring by a shackle or a similar connection. The direct riser cylinders are long stroke (15 m or more) and pull the load, which means that they carry the load of the riser. For each project optional extras like accumulators and specific installation and safety valves can be added. These would be specialist designs for the project. The piston rod coating is critical, as the cylinder is acting under extreme environmental conditions.

Production riser tensioners are used for deep water production platforms such as Tension Leg Platforms (TLP), deepwater Self Elevating Platforms (SEP) and Floating Production Storage Offloading vessels (FPSO). On each platform a number of smaller production risers need to be tensioned to cope with the relatively small rig movements. Production riser tensioners are normally grouped in a cassette frame carrying four or more tensioner cylinders with their related equipment. The system has built in redundancy, which means that it can stay operational if one tensioner in a cassette is not functioning.

Tension, system stiffness, stroke, etc. are project specific as the rig specific movements during normal working conditions, storm, hundred year storm, hurricane, etc. will be set in the project specific parameters.

9.5 Type of gas

In heave compensation systems a combination of high pressure liquid and compressed gas is used. In the dredging industry a combination of mineral oil and compressed air is always used. The pressure that can be applied and the compression ratio that can be used is very limited in these applications. The highest operational pressure in a heave compensation system for drag hopper dredgers is approximately 80 bar. At high compression ratios for gas it is possible that the fluid will combust spontaneously.

This is why a combination of non-combustible fluid and compressed air is used for traditional offshore applications such as the drill pipe and riser compensators. For heave compensation system on crane vessels a combination of mineral oil and nitrogen is used.

9.5.1 Combustion of a fluid

Combustion of a fluid can only happen if three important conditions are met. There must be a combustible material, oxygen (from the compressed air) and a combustion source.

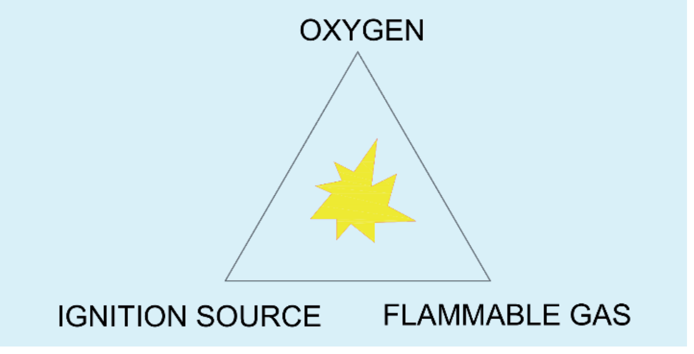

Fig 9.5.1 Three necessary conditions for an ignition

If any one of these conditions is not present, then combustion cannot take place. More specifically, the combustion source must reach a minimum temperature before a fluid can ignite. This temperature is defined as the “flash point” or “ignition temperature” of a fluid. The ignition can also only occur if a sufficient quantity of fluid vapor is present. This quantity is indicated by the “flammable or explosion limit”.
The lowest “flash point” of mineral oils is 160 °C. More details about this temperature will be given by manufacturers in their ‘material safety data sheets’.

The presence of a medium separator, like a piston or a membrane, is not considered as a method of of ignition prevention. After all, a piston will always show some ‘leakage’ (or transport) of the fluid past the seals. This means that it is always possible that a combustible material is present in the gas section of the cylinder or accumulator, even if this is only very little. In the same way it is possible that there is a leak in the membrane of for example an accumulator, which means that the combustible fluid comes into contact with the oxygen in the gas.

The temperature of the hydraulic fluid and/or the mixture of vaporised fluid and gas can rise for the following reasons:
- Gas compression
- Operating temperature
- Static electricity
- Mechanical sparks

Gas compression

Based on the ideal gas law, the temperature rise from a starting temperature T_1 to a final temperature T_2 resulting from a 100% adiabatic compression from pressure p_1 to pressure p_2 can be calculated as follows:

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{\kappa-1}{\kappa}} \quad \text{or} \quad \frac{p_2}{p_1} = \left(\frac{T_2}{T_1}\right)^{\frac{\kappa}{\kappa-1}} \tag{9.19}$$

with

T	=	temperature	K	p	=	pressure	N/m²

The adiabatic constant κ, is dependent of the gas temperature and pressure. The theoretical maximum value for $\kappa = 1{,}4$. In practice higher values occur, see figure 4.4.4.B and the table below:

Characteristic of compressed air		
Temperature °C	Pressure bar	κ
50	100	1,5388
50	150	1,5942
50	200	1,6355
50	250	1,6628
50	300	1,6785
50	350	1,6854

Table 9.5.1 Values for κ for different pressures

The larger value of the polytropic constant, the larger temperature rise. That is why the value of κ is set at 1,69 here.

It is clear from the formula that the end temperature T_2 does not only depend on the rise in pressure but also on the starting temperature T_1. Assuming a starting temperature of 60 °C (= 333 °K), a flashpoint temperature of 177 °C and a safety margin of 60 °C relative to the flashpoint temperature, the maximum allowable temperature would be $T_2 = 177 - 60 = 117$ °C = 390 °K.

This means the maximum permissible pressure increase will be:

$$\frac{p_2}{p_1} = \left(\frac{390}{333}\right)^{\frac{1{,}69}{1{,}69-1}} = 1{,}47 \tag{9.20}$$

This means that the pressure ratio for the gas may not be higher than 1,47 to limit the temperature rise to a value of 117 **°C.** The absolute value for the starting pressure is thus not significant.

The dredging industry has used a combination of mineral oil and compressed air in the heave compensator for the suction pipe for many years. The maximum working pressure for this type of system is 80 bar at the moment. Do remember that it isn't the starting pressure that's significant but the increase in pressure that can cause the high rise in temperature. The other point is that these are often passive systems which usually have a starting temperature around a maximum of 45 °C. Because of this lower starting temperature and the lower value for κ the allowable compression ratio p_2/p_1 is now 1.80 for an ISO-VG46 oil. This means that the 'natural' safety margin is already much bigger.
In these heave compensation systems no physical separation has been fitted between the fluid and the gas. The gas will be in direct contact with the fluid. A layer consisting of a mixture of fluid and gas is not formed under these conditions.

9.5.2 Static electricity

As a result of the internal friction between the fluid particles combined with the poor electrical conductivity of the mineral oil, it is possible that an electrical field is formed between for example the piston and the cylinder housing. The seal between the piston and the cylinder housing (in an accumulator) will need to dissipate the difference in voltage that may occur. Most bearing rings are nowadays made from a bronze or carbon alloy, both good conductors. The dissipation of the static electricity present in the cylinder is usually guaranteed via the piston rod.

There are no references in the literature about the resistance in ohms between the piston and the cylinder housing of a piston accumulator. It is however very unlikely that static electricity will build up to such an extent that a spark could be formed.

9.5.3 Sparks caused by mechanical friction

A spark can be created when metal parts collide. In an accumulator or a cylinder, metal components can, in practice, only collide if there is a malfunction in the installation that could, for example, lead to very large speeds in which case the piston could slam into the end plate of the cylinder at a high speed.

This type of situation could, for example, develop if a lifting cable broke whilst there was no provision to switch off the gas pressure on time. Such a situation will however always lead to a drop in gas pressure and thus a reduction in temperature of the flammable mixture that may be present. In these situations, a spark could lead to a local ignition.

9.6 Using nitrogen

If a combination of mineral oil and nitrogen is used another important safety factor needs to be taken into account. The potential danger consists of the development of a low oxygen content in enclosed spaces when a leak occurs in the nitrogen storage system. The normal oxygen content under atmospheric conditions is 20.9 % by volume. The safety data in the following table come from a safety bulletin from the U.S. Chemical safety and Hazard Investigation Board and gives details of the risks for lower oxygen percentages:_

Volume % Oxygen	Possible effect on the human body
20,9	Normal
19,0	Several sub-conscious disadvantageous psychological side effects.
16,0	Increased heart rate and shortness of breath, slower mental response times and attention span, limited co-ordination.
14,0	Unusual fatigue, emotional confusion, bad co-ordination, impaired judgement.
12,5	Very badly impaired judgement and co-ordination, bad respiration and possible permanent damage to the heart.
<10	Unconsciousness, palpitations and death.

Table 9.6.A Available oxygen % in an enclosed space and its effect on the human body

A bottom limit for the oxygen percentage in enclosed spaces is usually set at 18 %.

The effects of a possible nitrogen leak depend, among other things, on: the size of the leak, the pressure in the nitrogen container and the volume of the space in which the nitrogen container is stored as well as the possibly available ventilation in that space.

In a normal condition the confined volume of a ships section V_s contains 20,9 % of oxygen, or 79,1% of nitrogen. To calculate the effect of a nitrogen loss which is mostly expressed in the unit kg/s we have to apply the mass contents of the nitrogen in that ships section in order to be able to calculate the time that it takes to end up in a dangerous condition.

$$t = V_s \cdot \frac{\left(\frac{82}{100} - \frac{79{,}1}{100}\right) \cdot \rho_{N2}}{L} \tag{9.21}$$

With:

V_s	=	ships section volume	m³	t	=	time	s
L	=	nitrogen leakage	kg/s	ρ	=	density nitrogen	kg/m³ (1,25)

As an example the time has been calculated in which a low oxygen percentage of 18% is being reached when the Nitrogen content of a gas bottle (typical size 1000 dm³ and charged at 270 bar) is released into that area.

Typical leaks are (by example at 200 bar pressure difference):
- Leak of a gas safety valve, 0,05 kg/s
- Leak of a DN16 ball valve with a specific leak size of 3 mm diameter , 0,5 kg/s
- Leak of a DN50 ball valve with a specific leak size of 5 mm diameter , 1,2 kg/s

The gas release has been calculated for confined areas with volumes of : 100, 500, 1500 en 3000 m³

Possible Gas leak	Mass flow kg/hr	Time in s to reach 18% Oxygen volume Confined area: volume, m³			
		100	500	1500	3000
Safety Valve	0,05	73	363	1088	***
DN16 Valves, seal, typical 3 mm diam	0,5	7	36	109	***
DN50 Valves, seal, typical 5 mm diam	1,2	3	15	45	***

***=18% is never been reached

Table 9.6.B The necessary time to reach an 18% oxygen volume in different areas

The following table provides the final oxygen percentage if the gas bottle of 1000 dm3 at 270 bar Nitrogen is completely released into the confined area.

Final Oxygen percentage			
Confined area volume, m³			
100	500	1500	3000
6.6	**14.6**	18.3	19.5

Table 9.6.C Final oxygen percentages if pressurized nitrogen bottle releases all the gas into the confined area

Red values indicate a dangerous condition. Note that in these examples no ventilation has been taken into account. In the areas of 1500 and 3000 m3 a low oxygen percentage of 18% will never be reached.

Several measures need to be taken to reduce the dangers of nitrogen as much as possible.

- Ventilation of the enclosed space
- Design of escape routes
- Continuous monitoring
- Personal protection measures
- Instruction, training and information

Concerning ventilation

If there is sufficient ventilation, then it is possible to keep the percentage of the oxygen content above the lower limit of 18%. A good ventilation system can only be designed if the amount of nitrogen leakage that could occur is known. For example, if a pressure control valve fails and the value for C_v is known, then the amount of gas that could flow out at a certain nitrogen pressure is known.

Design of escape routes

Escape routes that are available to the user if there were to be a nitrogen escape need to signposted for each different set of operational circumstances. Since nitrogen is marginally lighter than air (approximately 2%) it will have a tendency to concentrate in spaces in higher locations. This means too that the oxygen percentages will be lower in those locations. This means that it is recommended that escape routes are not planned through those spaces.

- In many locations it can be very difficult to determine escape routes. This is often caused by the presence of cage ladders and or obstacles along the escape route. In these situations it will be necessary to provide the user with means for personal protection at all times.
- The design of escape routes is not limited to spaces on ships alone. Production hangars are often used for the manufacture and testing of the equipment. These too need to be treated as enclosed spaces.
- Escape routes will need to be signposted in accordance with the locally applicable standards (pictograms, colours, illuminated signs etc).

Continuous monitoring

Before users enter an enclosed space it must be possible to send out a warning signal if the oxygen percentage in the atmosphere in the space is at a dangerous level. Signal generators with a 4-20 mA output signal exist for this purpose. These measurements are connected to the central alarm system onboard ship.

If the volume percentage drops below the recommended lower limit of 18% then the users of the space concerned need to be warned. This also applies to users or rescue workers that want to enter the space concerned.

Please note: this type of monitoring equipment will also require twice yearly calibration.

Personal means of protection

Figure 9.6.A Example of a personal monitor, the picture shows the instrument for H_2S. The same monitor can be ordered for other gases like CO and O_2 (Courtesy of BW Technlogies)

If the safety of users cannot be guaranteed then it is recommended that all users that may be in the vicinity of equipment with nitrogen are issued with personal monitoring units. These are portable measuring instruments which are carried on a belt. The person carrying it would be warned immediately if the oxygen percentage drops below a certain value (of < 19%). This will pre-warn the person concerned, which means that she/he will be able to choose an escape route as early as possible. The use of this monitoring equipment can be limited to the enclosed space(s) concerned.

Instruction, training and information

- To make sure that the user is fully aware of any other dangers in the use of nitrogen a number of extra measures need to be taken:
- Safety instruction in the safety handbook.
- Ongoing training of personnel.
- The use of pictograms on the installation.

9.7 Active heave compensation

A passive heave compensation system reacts to a change in the externally acting forces like acceleration or damping forces from an object below the water surface. In those circumstances the load will always retain a rest movement relative to the seabed. This can cause a large impact on the load as it is being lowered onto the seabed and thus cause substantial damage.

9.7.1 Method of construction

In order to reduce the rest movement even further, active heave compensation systems have been developed. In these cases one or more extra cylinders are mechanically fitted parallel to the passive cylinder. Figure 9.6.1.A shows two possible methods. In the system on the left two active single action cylinders are linked to the passive cylinder. The sum of the two piston rod areas of both active cylinders is equal to the ring surface of the passive cylinder. This has the effect of effectively creating one active cylinder with a virtual continuous piston rod in the way it has been drawn on the right hand side. The rod surfaces of the symmetrical cylinder are then equal to each other.

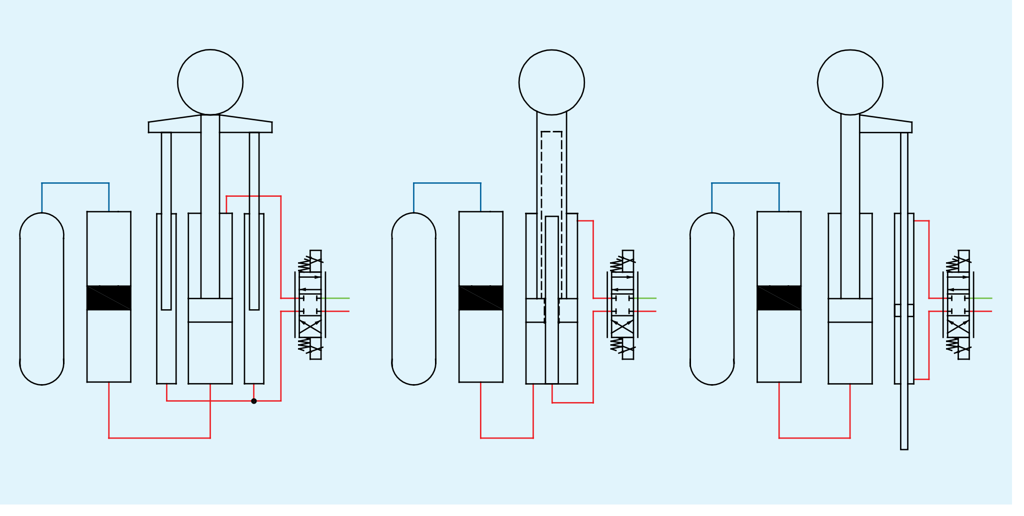

Fig 9.7.1.A Two basic arrangements for an active heave compensating system. At the left two active cylinders are installed in parallel to the passive cylinder. The principle on the right with one separate active cylinder is combined with the passive cylinder in the principle shown in the middle

The method shown in the middle drawing uses a methodology that has been used in flight simulators for many years. In this case the rod of the passive cylinder is hollow. A rod that is fixed to the bottom side of the passive cylinder, combined with the piston, forms an extra cylinder surface for the active cylinder. In practice this gives the same solution as the drawing on the right.

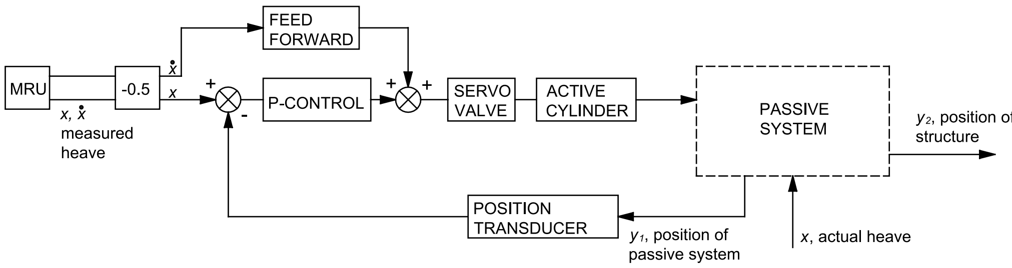

Fig 9.7.1.B Block diagram for the active heave compensating system

The active cylinders are controlled by a servo valve in a position control circuit, supplemented with a feed-forward control mechanism, see paragraph 8.2 for further details. In this arrangement, the most important movements of the ship, 'heave' + 'roll' + 'pitch' are recorded by a Motion Reference Unit (MRU). From this signal it is possible to calculate the exact movement of the tip of the crane relative to the seabed. The movement of the load under water can now be completely compensated for by ensuring that the active cylinders make the same but opposite movement with the help of a position control circuit. It is possible to achieve a rest-movement relative to the seabed of 6% for a crane tip movement of approximately 5 meters. The active cylinders do not need to able to carry the full load since the passive cylinders will already have taken care of 80-90% of that load. The maximum load for the active cylinders is often designed at 10-15% of the maximum load on the cable.

The detailed block diagram for the passive system is shown in figure 9.3.B. In the block diagram you will need to take careful note of the definitions of the different masses. If the active system works well, then the load no longer moves in an absolute sense whilst the dynamic part of the external forces will be zero. This means that there will be no dynamic load from the mass M that is underwater on the active part of this drive mechanism. The static load will be completely compensated for if the gas pressure in the passive part is set properly.

Whether the load no longer moves in an absolute sense depends on the ability of the active system to position the sheave in a 100% opposite movement. From the combined block diagrams of figure 9.3.B and 9.7.1.B it can be seen that the active system (cylinders + servovalve) is subjected to external forces. In chapter 6 we introduced the hydraulic stiffness C_H of a servo system, a parameter that is not to be mixed up with the stiffness C_{HC} of the gas system of the passive heave compensator. The hydraulic stiffness C_H and ω_o of the active cylinders determine to a great extent this ability of the active system.

In the formulas that have been used so far the static or coulomb friction of the seals and of mechanical sheaves has not been incorporated. For the block diagram and simulation software they introduce non-linear effects. For a pure passive system static friction introduces additional position errors. In fact the passive system cannot react on load changes that are induced by the moving vessel if these load changes are within the value of the total static friction.

For active controlled systems a position feedback system is being applied, which means that every position error including those as result of static friction are being corrected. It is clear that lower static friction forces result in better position accuracies. The actual static friction in modern heave compensating systems has values of 1,5-3% of the maximum load capacity.

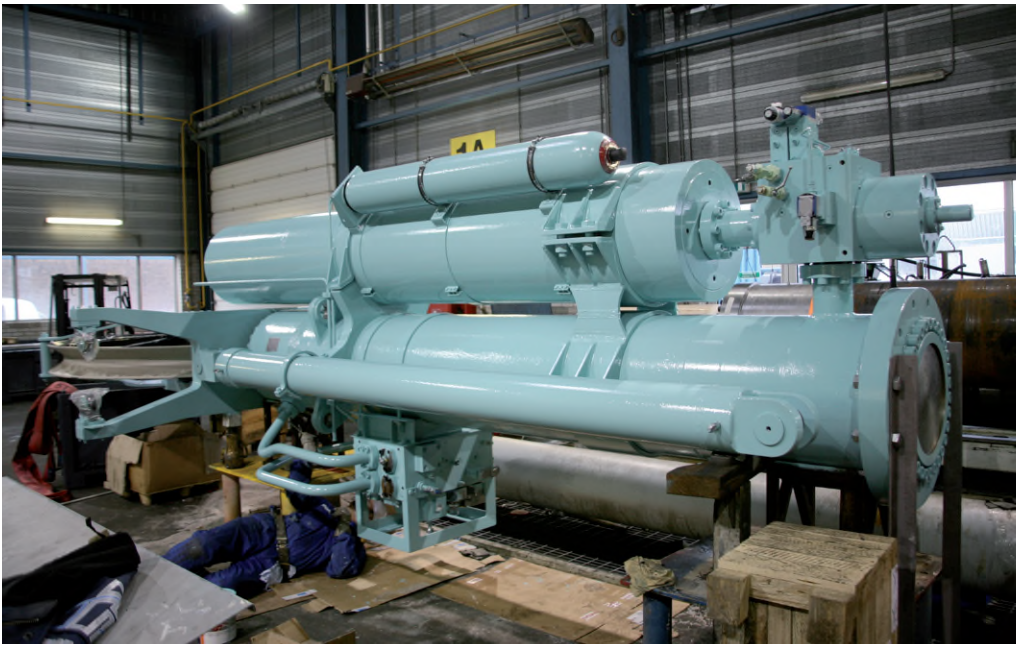

Fig 9.7.1.C Assembly of the main cylinder, two small active cylinders (one is shown at the front) and the medium separator on top, connected via a main manifold with cartridge valve (Courtesy of Huisman)

The conclusion that may be drawn is that the forces in the active system, if the gas pressure p_L has been set properly, are mainly determined by the dynamic movement of the mass of the active part itself (the order of magnitude is 50-80 kN) and by the stiffness of the passive part.
Because it is possible that the piston speed can be quite high, up to approximately 0,75 m/s, it will also be possible that high volume flows will occur in the active part of the mechanism. Fluid flows of up to approximately 2000 lpm are possible for a supply pressure in the active part of 300 bar. The hydraulic power for the active part needs to be generated by a system of constant pressure pumps. Maximum pump output can be slightly reduced by installing accumulators in the pressure pipes. The hydraulic power installed can nevertheless reach values of 800 kW.

9.7.2 Deployment/retrieval of the load

The active system will be operational when the load is eventually set down onto the seabed. Setting down of the load is achieved by releasing the winch. Because the static load on the system will reduce due to the reduction in the load caused by the weight of the load, it will be necessary to reduce the gas pressure in the passive system proportionally. A static imbalance will develop if the gas pressure is not adjusted. This imbalance will have to be compensated for by the active system. The active system will be able to compensate for part of this imbalance, dependent on the heave movement at the time and the stiffness of the passive part of the mechanism.
High pressure gas is allowed to flow from the passive system to a number of gas bottles that have been installed and for which a lower pre-pressure has been set so that a lower gas pressure can be achieved. After all, if nitrogen is used, it is not normal practice to vent the gas off into the atmosphere.
In shallow waters an empty hook means that the force on the cable is low (50-100 kN). When the water depth increases to depths of 1500 m or more, there will be a rest load from the cable's own weight when the hook is empty. For that sort of depth and a winch capacity of 200 mTon the weight of the cable is approximately 40-45 mTon.

Fig 9.7.2 The active Heave Compensator in operation (Courtesy of Huisman)

If a load needs to be retrieved from the seabed, then the reverse load process will take place. In this case it will be necessary to slowly increase the gas pressure in the passive part. For this purpose a number of gas bottles will be brought to a higher pressure of approximately 300 [bar]. The pressure in the passive part can be increased in steps from these reserve gas bottles during the initial lift of the load.

A compressor is needed to increase the gas pressure. The gas pressure at the inlet port can be very low, in the order of 10 – 15 bar, whilst the output pressure can be very high, in the order of 300 bar. A 3-stage compressor with cooling between the steps is required to reach that sort of pressure increase.
The control mechanism of an Active Heave Control (AHC) consists of much more than just the position control mechanism that we have already mentioned. If an AHC is used to install or retrieve a load it is necessary to monitor all process conditions on a continuous basis whilst at the same time it will be necessary to ensure that all components of the drive mechanism remain in good working order so that a safe and optimized system can be guaranteed. If, for example, the gas pressure in the passive system doesn't correspond to the static part of the load, then the load on the active system will be too high. It is necessary that the operator of the system has this information to hand on a continuous basis so that he can slow down the lowering or lifting of the load as and when necessary.

If the load on the active part of the system is too high, then the active part of the software in the PLC will be switched off automatically. The passive part will remain operational to avoid a situation where a very large load would develop on the cable, which could lead to damage to the object that needs to be put down or picked up. To make sure that the passive system is able to function, a large bypass valve needs to be fitted between the A and B pipe of the servo valve. In order to avoid cavitation in the active part as well, it will be necessary to keep the pressure on both sides of the active piston above a minimum value. It is these types of technical measures that make sure that an active heave compensation system can function in a reliable way.

9.8 Design of the gas system

The gas system of passive systems operates at high pressure of as much as 300 [bar]. The formulas that can be used for pneumatic systems cannot be applied to these systems. In this section we discuss the important design rules for high pressure gas systems in more detail.

9.8.1 Pressure equaliser function

Large gas volumes at high pressures are required for heave compensation systems. Compressors are used to create these high gas pressures. The pressure in a gas bottle that is connected to the system is lowered by letting the gas flow from the bottle to a different bottle where the pressure is lower or, if air is used as the gas medium, by releasing the gas to the atmosphere.

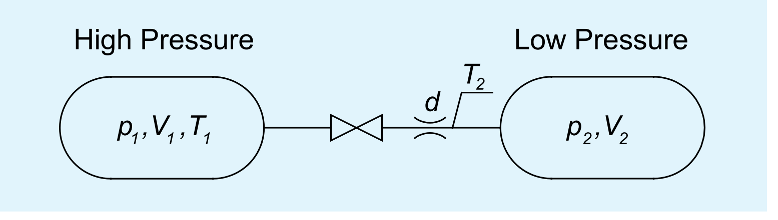

Fig 9.8.1.A Releasing gas from high to low pressure

When a valve is opened gas will start to flow from the high pressure container to the low pressure container via the exit port. The speed of the gas flow through the outlet port can increase to the point where it reaches the speed of sound. If the speed of sound were to be reached, a very high internal restriction would develop in the gas itself. This restriction itself ensures that the speed of the gas flow cannot go higher than the speed of sound. In the calculation below it is assumed that the connecting pipe work between the two gas volumes doesn't introduce an extra pressure drop.

Both compressed air and nitrogen behave very much in accordance with the perfect gas law:

$$p = \rho . R . T \quad (9.22) \qquad \text{and} \qquad R = \frac{8314}{MW} \quad (9.23)$$

Where:

p	=	pressure	N/m^2		MW	=	molecular weight of the gas
R_g	=	the specific gas constant	$m^2/s^2.K$	For air	MW	=	28.97
ρ	=	density	kg/m^3	For Nitrogen	MW	=	28.02
T	=	temperature	K				

Which means that:

R_{air} = 287

And: R_{N_2} = 297

Since the speed of the gas flow is a function of the pressure drop across the port, there is in fact a critical pressure ratio $b = p_1/p_2$ for which the speed of sound will occur.

The pressure $p_{2,cr}$ at which the speed of sound will occur is:

$$p_{2,cr} = p_1 \cdot \left(\frac{2}{\kappa + 1} \right)^{\left(\frac{\kappa}{\kappa - 1}\right)} \tag{9.24}$$

Where:

κ	=	adiabatic gas constant of the gas	p_1	=	inlet pressure of orifice [bar]

If for example:

	κ =	1,6	
And:	p_1 =	200	[bar]
Then:	$p_{2,cr}$ =	99,2	[bar]
And:	b =	0,496	

Another interesting parameter is the temperature of the gas. Due to the decompression the throat of the port will reach a minimum temperature of:

$$T_{cr} = T_1 \cdot \frac{2}{\kappa + 1} \tag{9.25}$$

Suppose that the temperature of the gas at the high pressure side was T_1 = 293 °K, then T_{cr} = 225 °K or –48 °C. This is why a restriction orifice has been installed in series with the valve shown in the drawing. The pressure drop is created across the orifice and not across the valve. If only a small valve would be present it is more than likely that the seals of the valve will not be able to withstand such a low temperature.

If the pressure ratio is above the critical value then the amount of mass $\dot{m}_{cr}$ kg/s flowing through the port will be:

$$\dot{m}_{cr} = \frac{A.p_1}{\sqrt{R_g.T_1}} . \kappa^{0,5} . \left(\frac{2}{\kappa + 1} \right)^{\left(\frac{0,5\kappa + 0,5}{\kappa - 1}\right)} \tag{9.26}$$

Below the critical pressure ratio the amount of mass $\dot{m}$ flowing through the port will be:

$$\dot{m} = \frac{A.p_1}{\sqrt{R_g.T_1}} . \sqrt{\frac{2\kappa}{\kappa - 1} . \left(\frac{p_2}{p_1} \right)^{\frac{2}{\kappa}} \left[1 - \left(\frac{p_2}{p_1} \right)^{\frac{\kappa - 1}{k}} \right]} \tag{9.27}$$

Where: (9.28)

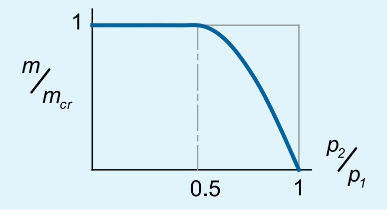

Fig 9.8.1.B Mass flow as function of pressure ratio p_2/p_1

The amount of mass flowing through the orifice as a function of the pressure difference between the two gas volumes can be represented by the graph in the figure above. The graph is strongly non- linear from the critical point onwards. For an empirical approach to the change in pressure in a gas system a linear graph is assumed between the critical point and the point at which the pressures are the same ($p_2 = p_1$).

A large flow of mass will develop as soon as a gas volume at high pressure is linked to a gas volume at low pressure, which will result in a reduction of the mass of the gas in volume V_1 and at the same time an increase in the mass of the gas in volume V_2. The gas pressure of V_1 and V_2 will of course change accordingly. It is relatively easy to calculate the change of the gas pressures with the help of a computer program. You will need to bear in mind that the adiabatic gas constant κ is dependent on the gas pressure P_1 and the temperature T_1.

If you set the variables $\kappa = 1{,}6$, $R_g = 297$ and $T_1 = 293$ °K and the pressure p in bar instead of N/m² then it is possible to rewrite formulas 6.26 and 6.28 together to create a more user friendly formula that will follow approximate the curve of the change in the gas pressures.

$$\dot{m}_{cr} = 1.91x10^{-4}.d^2.p_1 \tag{9.29}$$

In cases where there is an initial critical flow the pressure in vessel V_1 will reduce in a linear way, whilst the pressure in vessel V_2 will increase in a linear way. This rule will apply until gas pressure $p_2 = 0.5 \times p_1$. As the gas pressure in a volume is proportional with the amount of gas in kg in that volume, for a gas volume V_1 it would mean that the pressure p_1 would decrease with a pressure gradient of:

$$\frac{dp_1}{dt} \approx -1.66x10^{-4}.\frac{d^2.p_1}{V_1} \tag{9.30}$$

Where:

d	=	diameter of orifice	mm	V_1	=	volume	m³
p_1	=	pressure	bar				

And for the volume V_2 the pressure will increase with a gradient:

$$\frac{dP_2}{dt} \approx 1.66x10^{-4}.\frac{d^2.p_1}{V_2} \tag{9.31}$$

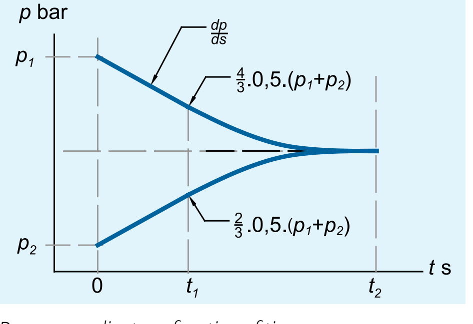

Fig 9.8.1.C Pressure gradients as function of time

Because the pressure gradient and the pressure difference that has to be abridged in the critical phase are known, it is in practice also possible to calculate the time t_1 it will take to reach the point where $p_2 = 0{,}5 \times p_1$. The one condition for this approach is that the gas volumes V_1 and V_2 must be equal.

$$t_1 \approx \frac{2030.\left(\frac{p_1 - 2.p_2}{p_1}\right).V_1}{d^2} \qquad (9.32)$$

again with:

d	=	diameter of orifice	mm	p_2	=	pressure	bar
p_1	=	pressure	bar	V_1	=	volume	m^3

The pressure gradient for both p_1 and p_2 will no longer be linear but polynomial from the critical point to the point where the two pressures are totally leveled out. A quadratic form of this polynomial provides accurate results. The boundary condition at the critical point must be that the gradient of the curve at that point must be equal to the linear gradient. This means that the mathematical parameters for the quadratic curve can be worked out.

This means that the time t_2 can be calculated with:

$$t_2 - t_1 \approx \frac{2030.\left(\frac{p_1 + p_2}{p_1}\right).V_1}{d^2} \qquad (9.33)$$

The formulas approximate the true situation and have proven in practical application that they give a reliable value with a maximum margin of error of approximately 15%.

As shown before, the temperature in the throat of the orifice drops to the critical value T_{cr}. Even if the pressure drop across the valve is not critical, the temperature in the high pressure (HP) container will always drop as the gas decompresses. A lower temperature in the HP bottle of figure 9.8.1.A will also result in a lower pressure than calculated with the above mentioned formula. However due to the heat exchange with the warmer room temperature outside the gas bottle the temperature of the gas in the HP container will also increase slowly.

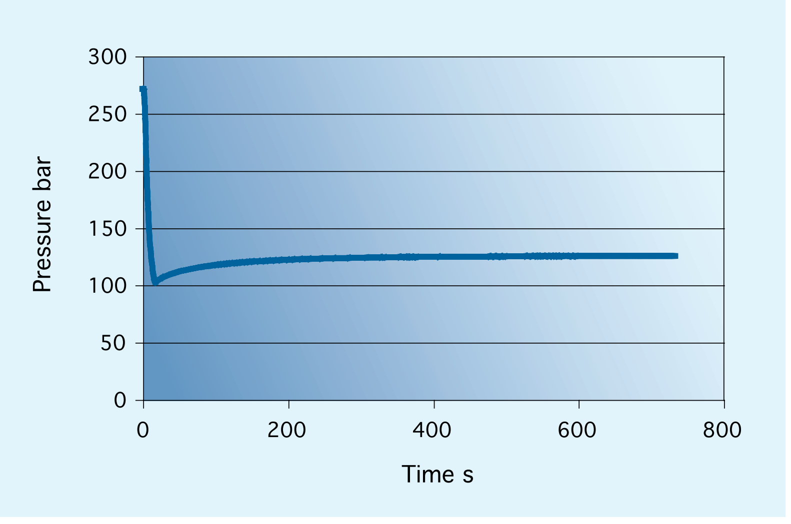

Fig 9.8.1.D Pressure as function of time. Fast pressure drop in a HP container due to gas release to another gas container and slow pressure recovery due to the heat exchange with the warmer room temperature.

In figure 9.8.1.D results of a test are shown where during the first 15 seconds gas from the HP container is released to another gas container. It is clear that the pressure drops also fast. After these 15 seconds the gas release is stopped. We see a gradual recovery of the pressure. This is due to the effect that the gas in the HP container exchanges heat with the warmer room temperature. This test was done with bladder type accumulators, so with a presumable low heat transfer because of the low heat transfer coefficient of the rubber blade. For piston type accumulators and for steel gas bottles the heat exchange will be higher resulting in a faster pressure recovery.

9.8.2 Pressure drop in the pipe work

One important parameter in the design of pipe systems for gas is the maximum allowable speed of the gas, in other words the diameter of the pipe work and the size of the valves for a particular volume flow. For hydraulic systems for example, the advisable maximum flow speed of the fluid is 5 – 6 m/s in the pressure pipe. For gas systems this design parameter is significantly higher.

The same basic design rules apply to pipe work designed for both gas and fluids. As for the fluid design, the first question that needs to be answered is whether the flow is laminar or turbulent.

For air under atmospheric conditions, a flow speed of 5 m/s and a pipe diameter of D = 25 mm the following rule leads to:

$$\mathrm{Re} = \frac{\rho \cdot v.D}{\mu} = \frac{1,24 \cdot 5 \cdot 0,025}{1,8x10^{-5}} = 8611 \tag{9.34}$$

with:

D	=	pipe diameter	m	ρ	=	density of the gas	kg/m³
v	=	fluid velocity	m/s				

The dynamic viscosity ☜ has a strong dependency on the temperature and a weak dependency on the pressure, as can be found from the available literature. A viscosity that is 50% higher is very feasible for higher pressures of approximately 300 bar.

The correlation between the density ρ_p and the pressure is more or less linear, the following approximation applies:

$$\rho_p \approx 1,15 \cdot 10^{-5} \cdot p \tag{9.35}$$

with:

ρ_p	=	density at pressure *p*	kg/m³	p	=	pressure	N/m²

From this formula it is clear that the value of R_e for higher pressures will result in an even higher value than the one shown above. This leads to the conclusion that the flow of high pressure gas will always be turbulent.

The friction resistance for a smooth high pressure pipe can be described as:

$$\Delta p = \lambda . \frac{L}{D} . \frac{1}{2} . \rho . v^2 \tag{9.36}$$

The following formula can be used for the friction coefficient λ:

$$\frac{1}{\sqrt{\lambda}} \approx -1{,}8 \log \left[\frac{6{,}9}{\text{Re}} + \left(\frac{\varepsilon}{D} \right)^{1{,}11} \right] \tag{9.37}$$

Where:

ε = roughness of pipe (0,03 - 0,05 mm)

The friction coefficient λ in this formula seems to be nearly constant at a value of 0,018 for a pressure range of 10-300 bar and gas flow speeds of 5-50 m/s.

Apart from the pressure drop in straight pipes there is an extra pressure drop resulting from the resistance of bends, pipe entries and exits, shut-off valves etc. The resistance coefficients for the different forms of pipe work are mentioned in chapter 3.2. The total effect of these extra resistances for the average design of gas system is a factor in the order of 2 -4 times the velocity pressure: $0{,}5 x \rho x V^2$. In the graph below the velocity pressure has also been calculated for the same design fluid speeds and gas pressures. The pressure has been calculated with formula 9.35

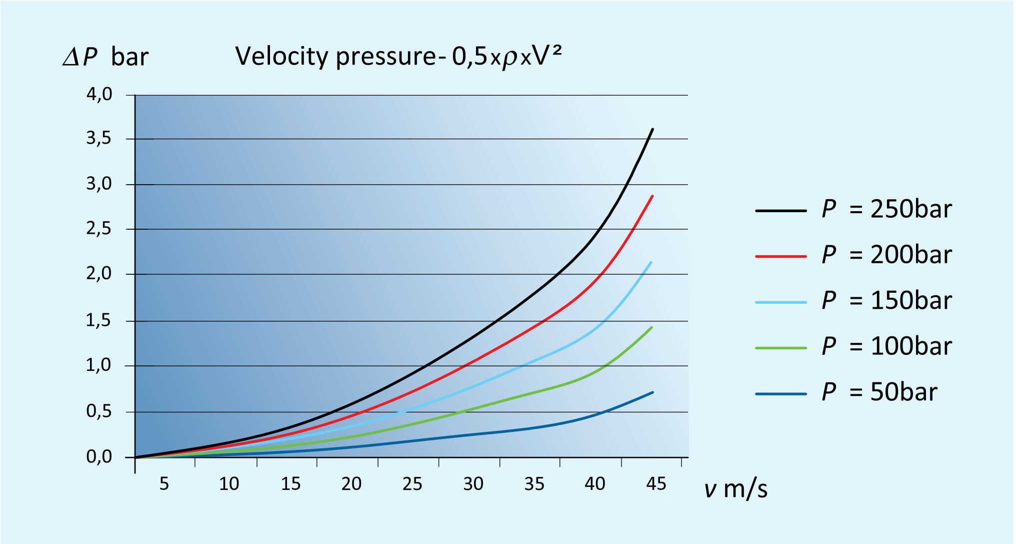

Fig 9.8.2 The pressure drop (velocity pressure) in the pipe as function of the gas flow velocity

The designer can get an impression of the pressure drop that can be expected in a gas pipe system. The designer can choose the designed gas speed dependent on the application. For passive heave compensation systems an extra drop in pressure also means extra variations in forces. The extra variations in forces will have a negative side effect on the ability to control the position in an active system.

In practice a maximum gas speed of 25 – 30 m/s is used for the pipe work of a passive system.

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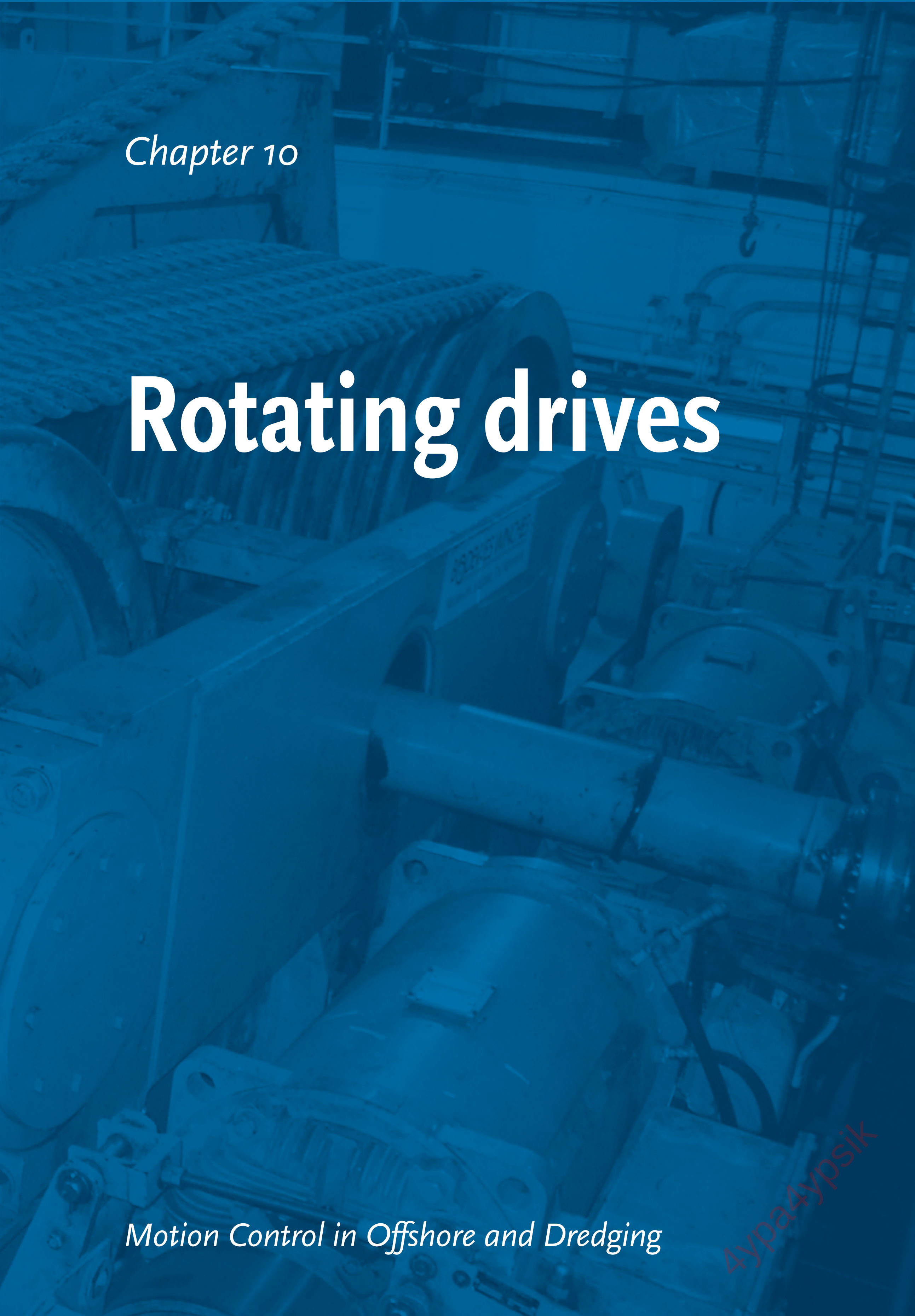

Chapter 10

Rotating drives

Motion Control in Offshore and Dredging

Chapter 10

Rotating drives

In chapter 3 we discussed the basic design rules and principles of rotating hydraulic drives. In chapter 5 we discussed the general characteristics of rotating electrical drives. In this chapter we will make more specific use of these design rules for the design of a number of specific rotating drives in the offshore and dredging industries. We will also provide some practical information about the final design.

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10.1 List of symbols

A_b	=	bottom area of cylinder	m^2
C_o	=	oil stiffness	Nm/rad
E	=	compressibility factor	N/m^2
I	=	inertia	$kg.m^2$
P	=	hydraulic power	kW
Q	=	flow	m^3/s
T	=	torque	Nm
V_{pipe}	=	line volume	m^3
V_{st}	=	stroke volume	m^3/rad
i	=	gearbox ratio	
I_m	=	motor current	A
I_n	=	nominal current	A
n	=	rotational speed	rad/s
p	=	pressure	Pa
t	=	time	s
β	=	damping ratio	
η_{gb}	=	mechanical efficiency gearbox	
η_{vol}	=	volumetric efficiency hydraulic motor	
$\eta_{p\text{-}tot}$	=	total efficiency pump	
$\eta_{p\text{-}v}$	=	volumetric efficiency pump	
η_{wd}	=	mechanical efficiency of winch drum	
ω	=	angular speed	rad/s

10.2 Primary and secondary hydraulic drives

For rotating hydraulic drives we can make a distinction between primary and secondary drives.
We talk about a primary drive when the hydraulic pump or proportional directional valve , see figure 10.2.A, provides a specific flow. The load pressure, meaning the pressure drop across the hydraulic motor, is determined by the mechanical torque that the hydraulic motor must generate so that the load can be rotated.

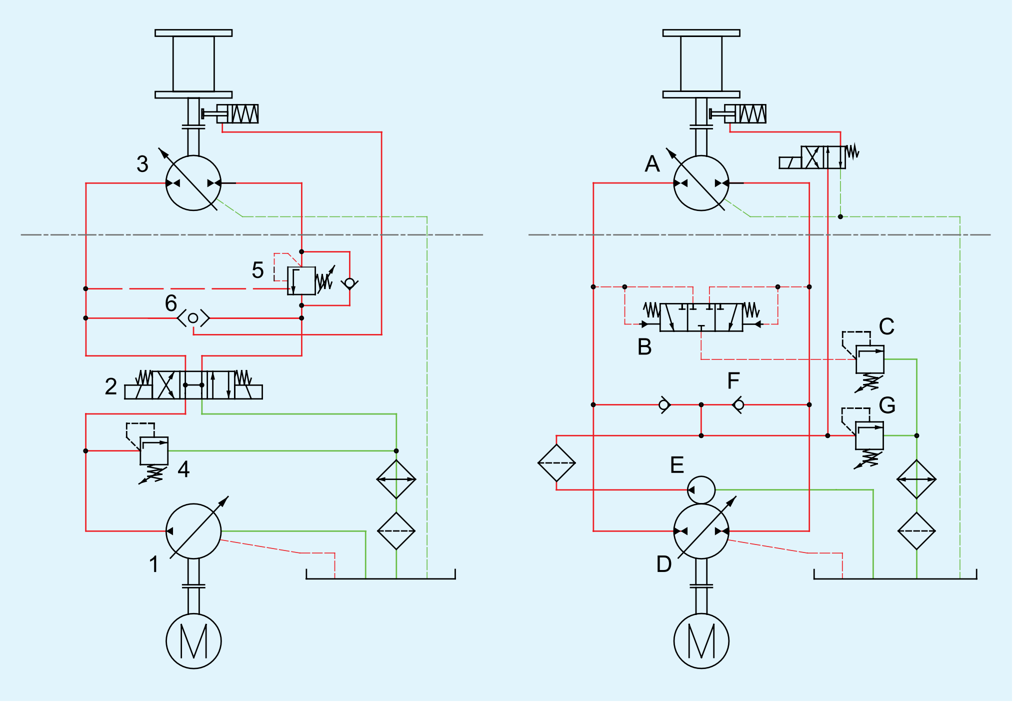

Fig 10.2.A Standard for Open (left) and Closed hydraulic systems (right)

The primary drive can be realised with an open system or with a closed or circulating system.

The open system

The flow to the hydraulic motor (3) in an open system is created by the combination of a variable hydraulic pump (1) and a directional valve (2). The pump in the drawing is a variable pump but can also be designed as a pump with a fixed stroke volume. In that case the valve (2) needs to be a proportional valve. The return connection from the hydraulic motor goes, via the brake valve (5) and the directional valve back to the reservoir. The maximum pressure is limited by the pressure relief valve (4). The hydraulic operated brake is controlled with the highest available pressure in the system, taken from the system with the shuttle valve (6).

The hydraulic motor in this example is also a variable one but can also be used as a motor with a fixed stroke volume. The number of revolutions per minute that can be reached is higher for a motor with a variable stroke volume than for a motor with a fixed stroke volume. If a motor with a variable stroke volume is used, then a small stroke volume can only be set if the torque that can be generated by the hydraulic motor is still large enough to drive the load. If the application is used for a lifting winch, this is extremely important. If the torque cannot drive the load, then it is possible that the load will drive the hydraulic motor, which would create a dangerous situation and which would almost certainly cause damage to the hydraulic motor too. In this application the stroke volume can be set between 100% and a minimum, often between 25 and 30%.

The closed system

In a closed system both ports of the hydraulic motor (A) will be connected to a pump (D) with variable flow output. The pump is able to create a flow in both directions. Depending on the flow direction the outlet and inlet port of the hydraulic pump and of the hydraulic motor change in function. In the circuit, one pipe will become a high pressure (HP) pipe and the other a low pressure (LP) also known as the boost pressure pipe. The hydraulic motor can be driven by the load, for example with a releasing winch, where the hydraulic motor will drive the pump and the power will be fed back into the electrical circuit.

The oil in the closed circuit needs to be refreshed continuously to stop it from being heated up and to prevent it getting contaminated. An extra feed or booster pump (E) that will suck the cooled oil from the reservoir and feed it into the then LP side via a filter and two non-return valves (F) are always fitted for this purpose. A special type flushing valve (B) has been fitted so that sufficient oil can be discharged from the circuit. The flushing valve will always be in the position that allows oil to be discharged to the reservoir from the LP pipe via a pressure control valve (C). A pressure control valve (G) has also been fitted to the outlet pipe of the pump (E). The pressure setting of valve (G) is usually about 5 bar higher than the pressure setting of valve (C), which is usually set between 16 and 24 bar. The setting of valve (C) is also the minimum pressure in the LP pipe and must also be available to generate sufficient input (boost) pressure for the pump in case the pump drives the hydraulic motor or in case the hydraulic motor drives the pump. In later examples, see chapter 12, we will show how too low a boost pressure can lead to severe technical problems.

The system response (= the response when the speed of the hydraulic motor has to change), is often lower in a closed system than in an open system. This is usually caused by a longer set of pipes and the resulting lower oil stiffness of the drive mechanism. The system response in an open system is also dependent on the length of piping between the proportional element and the hydraulic motor. In case of the use of a proportional valve, that valve can be installed close to the hydraulic motor, which results in a higher oil stiffness.

The secondary drive

We talk about a secondary drive, see figure 10.2.B, when the hydraulic pump provides a constant hydraulic pressure and the stroke volume of a variable hydraulic motor is varied. The drive behaves like a torque generator, dependent on the stroke volume of the hydraulic motor. The speed of the drive will be determined by a step less acceleration or deceleration of the drive which is developed from the equilibrium in the torque between the hydraulic actuator and the torque that the load demands.

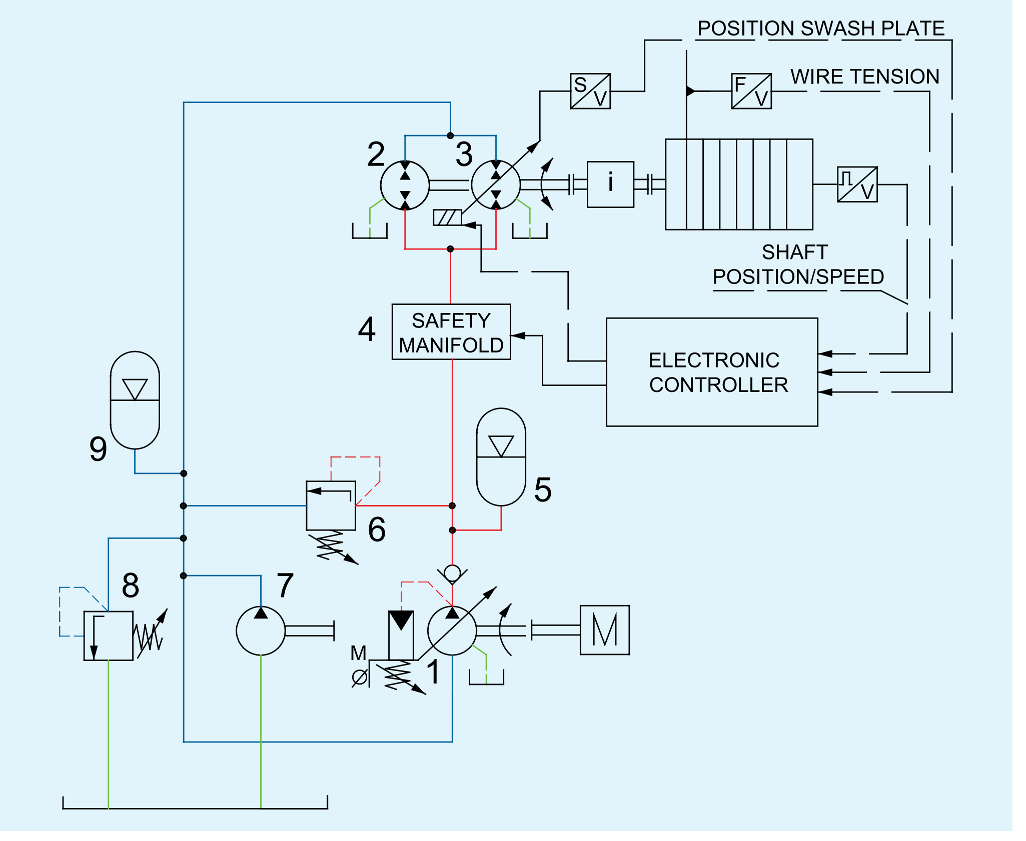

Fig 10.2.B Special closed loop circuit for the principle of a secondary drive

For the secondary drive, a special form of the closed circuit drive system is required. A variable motor has to be chosen for this type of application. The flow from the pump (1) is always in the same direction. A check valve prevents a volume flow in a different direction. The pump has also been fitted with a pressure control system which ensures that there is a constant pressure in the HP pipe. The hydraulic motor will always either be a motor with a variable stroke volume or a combination of a motor with a fixed stroke volume (2) together with a motor with a variable stroke volume (3). The reason for choosing the combination of a motor with a fixed stroke volume (say 250 cc/rev) together with one with a variable stroke volume (500 cc/rev) is the lower price of the combination compared to the price for a single motor with a variable stroke volume of the combined capacity of 0-750 cc/rev. In this case the stroke volume of the shown combination can be adjusted steplessly between 0 and 100% (0-750 cc/rev).
A safety manifold (4) is required. In principle it consists of large sized cartridge valve that can close off the hydraulic pressure in case of failure in the system. A system relief valve (6) limits the maximum pressure. The accumulator (5) can be added to provide more hydraulic power in case of a sudden high flow requirement. The accumulator (9) will then be necessary to accept a sudden high return flow from the hydraulic motor. The boost pump (7) is required to flush the return line with cooled and filtered fluid.
The torque delivered by the hydraulic motor drive system is based on the basic formula that was already discussed in paragraph 1.7.3:

$$T = \Delta p \cdot V_{st} \cdot \eta_{hm} \tag{10.1}$$

Where:

V_{st} =	Stroke volume	m^3/rad	T =	Torque	Nm
Δp =	Pressure drop across motor	Pa	η_{mech} =	mechanical efficiency of motor	0,8 - 0,98

We can now assume that the pressure difference Δp in the above formula is constant. This means that the output torque from the hydraulic motor, apart from a possible variation in the mechanical efficiency, is directly proportional to the set stroke volume. The torque that will be delivered by the hydraulic motor minus the externally working torque of the drive mechanism will cause the drive system to accelerate with the reduced moment of inertia I_{red} in accordance with the simplified block diagram shown below.

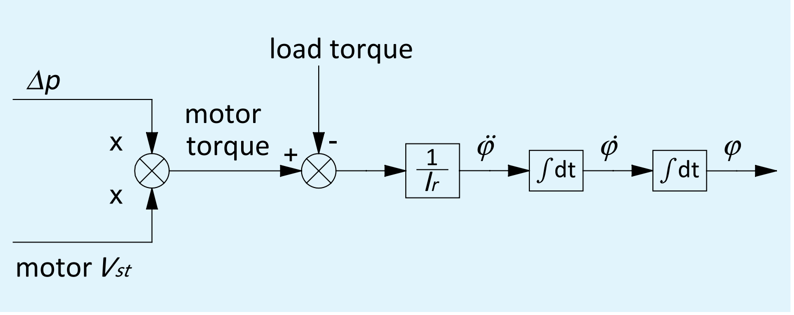

Fig 10.2.C The load with reduced moment of inertia I_r is accelerated with the resulting torque

The result is an angular acceleration of . After integration this will give the angular speed and an angular rotation of. If the swash plate of the hydraulic motor has a step less adjustment then, dependent on the external load, the motor axle will accelerate and decelerate steplessly too. In case of a winch drive the speed and the angular position of the driven axle are being used to creating a feedback loop, see figure 10.2.D. In addition the linear position of the swash plate of the variable motor is used in a feedback loop. This type of control mechanism is, in principle, unstable because two integrators will have been put in series. The flow to a hydraulic motor is obtained from the actual speed and the stroke volume.
For special applications instead of the angular speed also the cable tension may be used as a feedback signal. The drive is then able to function as a constant tension system. This principle has been applied at the project to cut (saw) the sunken transport ship Tricolor in 9 pieces, see also paragraph 10.7

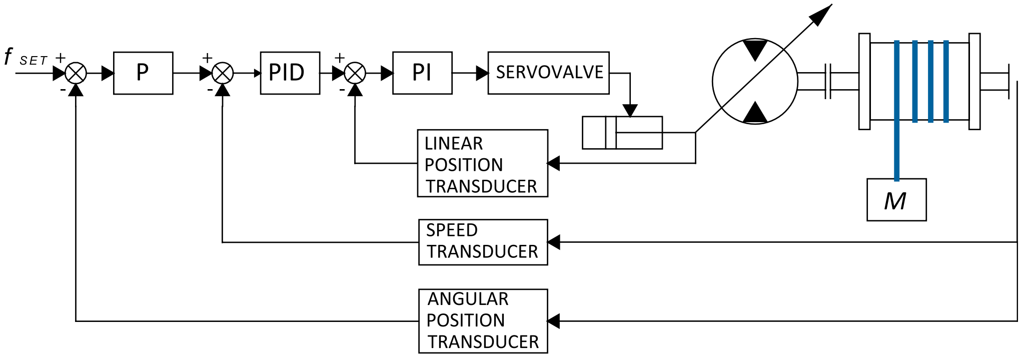

Fig 10.2.D Block diagram for the total secondary hydraulic drive

Hydraulic accumulators are often included in the HP and LP pipes of a secondary system so that it will be possible to generate large volume outputs from the hydraulic motor for short periods of time. In those cases it is also possible that the load drives the hydraulic motor which will start to operate like a pump from the LP to the HP pipe. This means that mechanical energy is converted into hydraulic energy, which will be accumulated by the hydraulic batteries. The accumulators in the LP pipe are necessary to help the booster pump maintain the minimum necessary booster pressure, even in situations where the hydraulic motor starts to function as a pump and will also suck oil from the booster pipe.

The dynamic system response of a secondary control mechanism is very high. This is because a constant pressure is kept in the HP pipe, which means that there will be no compressibility of the oil volume in it. This in turn means that the length of the pipes between the pump and the hydraulic motor doesn't have any influence on the response time of the drive mechanism. As a result it is possible to adjust the swash plate of the hydraulic motor in milliseconds which in turn gives a very high responsiveness.

10.3 Comparison of a hydraulic and an electrical drive

In this paragraph we use a winch drive to make the comparison between the different forms of hydraulic drive mechanisms and those of a frequency controlled induction motor.

The example concerns the drive mechanism for a lifting winch for which the specification includes a large lifting capacity with a low cable speed as well as a high cable speed for a lower hoisting load. The winch has a total of 5 cable windings or 5 layers. One special demand is the need for an extra operational option for a constant tension (CT) function where the force on the cable needs to be kept as constant as possible.

Parameter	Value		Unit of Measure
Drum diameter	740		mm
Wire diameter	40		mm
difference in layer diameter	74		mm
1st layer diameter	781		mm
3rd layer diameter	929		mm
5th layer diameter	1077		mm
	Low Speed	High speed	
Maximum tension	250	62,5	kN
Contingency tension 10%	275	68,8	kN
Nominal wire speed, 3rd layer	0,33		m/s
Maximum wire speed, 3rd layer		1,33	m/min
Wire power, nominal	91,7	91,7	kW
Mechanical efficiency of the gear drum, η_{gd}	0,98	0,98	
Maximum torque 5th layer, T_m	137	34	kNm
Peak Torque 5th layer (incl+10%), T_p	151	38	kNm
Maximum drum speed, 3rd layer	6,9	27,4	rpm
Mechanical power	98,6	98,6	kW

Table 10.3 Mechanical parameters that need to be achieved in the 'Low speed' and 'High speed' mode

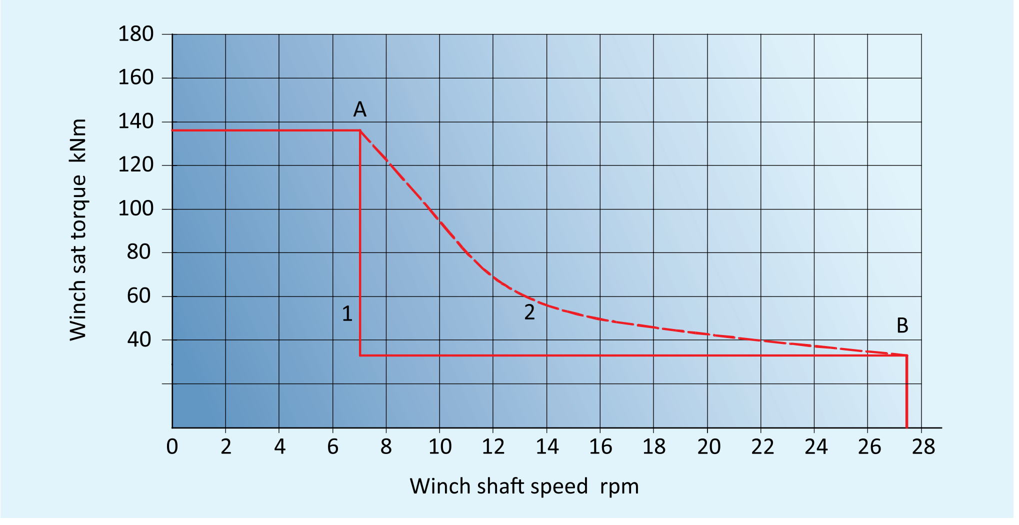

Fig 10.3 Load characteristic of the winch

According to the graph, the torque required for the winch drum corresponds with line no 1. The torque is 137 kNm (A) up to 6,9 revolutions per minute and 34 kNm (B) from this speed up to 27,4 revolutions per minute. The required power for the drive is 98,6 kW for these two different, but quite typical, sets of the number of revolutions per minute. The user would obtain a more optimal efficiency from a drive that follows line 2. This is a line of constant power. We will return to this point when we discuss the different types of drive mechanism.

10.3.1 Hydraulic drive

We start with the calculation of two different hydraulic drive mechanisms, both with a closed loop control system. In the first of the two example drives a hydraulic motor with a large stroke volume capable of driving the winch drum directly because of its large torque capacity has been chosen. A high speed hydraulic motor that drives the winch via a gearbox has been chosen for the second example. The gear ratio of the gearbox needs to be such that the hydraulic motor is able to deliver the necessary torque but also so that the maximum number of revolutions per minute for the hydraulic motor and the input shaft of the gearbox are not exceeded for a given maximum stroke volume and pressure difference. By choosing different combinations of the gear ratio of the gearbox and stroke volumes of the high speed hydraulic motor it is often possible to find several possible options. These options allow the designer to check the cost price and delivery times for the different options which can help in the choice of the final solution.

For this example we have chosen a 3-stage gearbox with a ratio of 117,5 and a mechanical efficiency of 0,985 for each stage. The total mechanical efficiency of the gearbox is then

$$\eta_{gb} = 0,985^3 = 0,956 \qquad (10.2)$$

Note: The specified gearbox efficiency η_{gb} is valid for the full load. Be alert that for lower loads the efficiency is much worse.

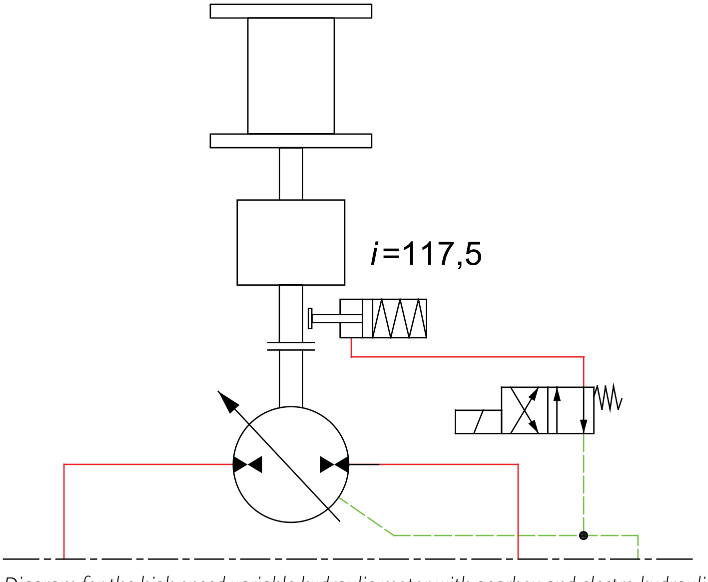

Fig 10.3.1 Diagram for the high speed variable hydraulic motor with gearbox and electro hydraulic operated disc brake

An important difference between the two chosen hydraulic motors is that the high torque motor has a maximum working pressure of 250 bar whilst the fast running hydraulic motor has a working pressure of 350 bar. The limit of 250 bar is not generally applicable to this type of hydraulic motor but does apply to the type of hydraulic motor chosen in this case. For high torque hydraulic motors with a lower stroke volume working pressures of up to 350 bar are also allowed.

If the results of the two calculations are put next to each other the effects of the different choices become much easier to understand. More details about the formulas that have been used can be found in paragraphs 1.5 and 1.7. 3. In the calculation sometimes the applied formula has been used.

		High torque motor		High speed motor		
		Low speed	High speed	Low speed	High speed	Unit
Stroke volume of the selected motor	V_{st}	38.000	12.667	250	60	cc/rev
Maximum pressure	p	250	250	350	350	bar
Specific torque	$= \dfrac{V_{st}}{20.\pi}$ (10.3)	604,8	201,6	3,98	0,96	Nm/bar
Mechanical efficiency of the motor	η_{hm}	0,98	0,98	0,93	0,93	
Maximum operating pressure drop	$p_m = \dfrac{20 \cdot \pi \cdot T_m}{V_{st} \cdot \eta_{hm}}$ (10.4)	232	173	315	328	bar
Peak maximum pressure drop (incl+10%)	$p_p = \dfrac{20 \cdot \pi \cdot T_p}{V_{st} \cdot \eta_{hm}}$ (10.5)	255	191	347	361	bar
Boost pressure closed loop	p_b	14	14	14	14	bar
Maximum operating pressure	$\Delta p_m = p_m + p_b$ (10.6)	246	187	329	342	bar
Peak maximum pressure (incl+10%)	$\Delta p_p = p_p + p_b$ (10.7)	269	205	361	375	bar
Motor speed	n	6,9	27,4	844,9	3379,8	rpm
Volumetric efficiency	η_{vol}	0,94	0,94	0,94	0,94	
Volume flow of the motor	$Q = \dfrac{V_{st} \cdot n}{1000.\eta_{vol}}$ (10.8)	277,0	369,4	224,7	215,7	lpm
Hydraulic power	$P = \dfrac{p.Q}{600}$ (10.9)	117,7	117,5	123,3	123,0	kW
Pump volumetric efficiency	$\eta_{p\text{-}V}$	0,95	0,95	0,95	0,95	
Pump total efficiency	$\eta_{p\text{-}tot}$	0,88	0,88	0,88	0,88	
Pump speed	η_p	1760	1760	1760	1760	rpm
Maximum displacement volume of the pump	$V_{st-p} = \dfrac{Q \cdot 1000}{\eta_{p-V} \cdot n_p}$ (10.10)	165,7	220,9	134,4	129,0	cc/rev
Prime mover power	$P_{in} = \dfrac{P}{\eta_{p-tot}} + P_{boost}$ (10.11)	135,3	135,1	141,4	141,2	kW
CT variation	$= 2.\left(1 - \eta_{gd}.\eta_{gb}.\eta_{hm}\right).100$ (10.12)	7,92	7,92	25,8	25,8	%

Table 10.3.1.A Results of the comparison of a High Torque and a High speed motor for the winch drive, for input values see also table 10.3

Comments on the results:

- The mechanical efficiency of 0,98 of the high torque hydraulic motor is considerably higher than the 0,93 of the fast running hydraulic motor. This has knock-on effects on the working pressure and on the total efficiency of the drive mechanism.
- The stroke volumes for both hydraulic drive systems have been chosen in such a way that the maximum occurring working pressure in the HP pipe is just lower than the maximum allowable working pressure for the motor concerned.

- Both types of hydraulic motor have a variable stroke volume. A higher motor speed can be achieved by steering the motor to a smaller stroke volume. In those cases it is also possible to realise a lower torque. The high torque motor can only be switched from the maximum to one lower stroke volume. This means that for this motor only line 1 in the graph in figure 10.3 is possible. Switching from the large stroke volume to the small stroke volume can easily be achieved by the use of a directional 4/2 valve. The fast running hydraulic motor can be controlled steplessly from the maximum to the smaller stroke volume. This hydraulic motor can thus achieve the 'constant power' line 2. To be able to create a step less adjustment for the hydraulic motor it will be necessary to fit a number of extra proportional valves as well as an electronic control mechanism for which information about the actual load pressure of the drive mechanism needs to be provided.
- The minimum boost pressure of 14 bar in the table has been used as a standard. Some manufacturers do however specify higher values.
- The necessary volume flow for the fast running hydraulic motor is lower than that for the high torque hydraulic motor because it is able to work at a higher working pressure.
- The necessary stroke volume of the variable pump for a high torque hydraulic motor is considerably higher than that for a fast running hydraulic motor. This is only the case because a lower maximum working pressure needs to be applied for the high torque motor concerned.
- The total mechanical efficiency, as calculated from a particular hydraulic input pressure to the force on the cable is determined by the mechanical efficiency of the winch drum ηwd , the hydraulic motor ηmh and, in the case of the fast running hydraulic motor, also that of the gearbox η_{gb} . The mechanical loss can be expressed as the percentage Y%. For a constant tension control mechanism it is not just the winch that is being driven but it is also possible that the winch drives the hydraulic motor. In that case all the mechanical efficiencies would be reversed. To determine the variation in the force on the lifting cable it will be necessary to use a value of 2.Y%.
- The power from the prime mover, the electric motor, is determined by the drive power of the main pump, supplemented with sufficient power to drive the boost pump. The flow from the booster pump is in practice 15% of the output from the main pump. If a boost pressure of 14 bar is required and if the efficiency of the booster pump is somewhat lower at 80% then the extra power required for the booster pump is:

$$P_{boost} = \frac{0,15 \cdot Q \cdot 14}{600 \cdot 0,8} \tag{10.13}$$

  This result is 1,6 kW for the high torque application and 1,3 kW for the high speed application.
- The variable pump is driven by an electromotor. In order to fully assess the total power losses it is necessary to take the efficiency of the electrical drive into account. An efficiency of 95,6% is given for an electromotor with this type of power. This means that the electrical power for the different hydraulic solutions and the total efficiencies for the different solutions will be:

	High Torque drive	High speed drive	Unit
Mechanical power at winch shaft	98,6	98,6	kW
Mechanical power at pump shaft	135,3	141,4	kW
Electrical power	141,5	147,9	kVA
Total efficiency	69,7%	66,7%	

Table 10.3.1.B Comparison of efficiencies for a High torque and a High speed motor

10.3.2 Electrical drive

For the design of an electrical drive mechanism it is necessary to assess the thermal loadability . An induction motor has its own ventilation which has been tuned to the heat given off at nominal speed. If the speed of the motor reduces then the cooling capacity and thus the torque of the motor reduces too.

A higher torque can be delivered at lower revolutions per minute if the motor has been designed with a higher cooling capacity. In that case curve 2 can be followed instead of curve 1 in the constant flow band. Because the winch must be able to operate at a low number of revolutions per minute for longer periods of time, an electric motor with increased cooling has been applied. In some designs it is also possible to have an increased water cooling capacity.

The curves for the load caused by heat for a standard induction motor (1) and a motor with increased cooling (2) look as follows:

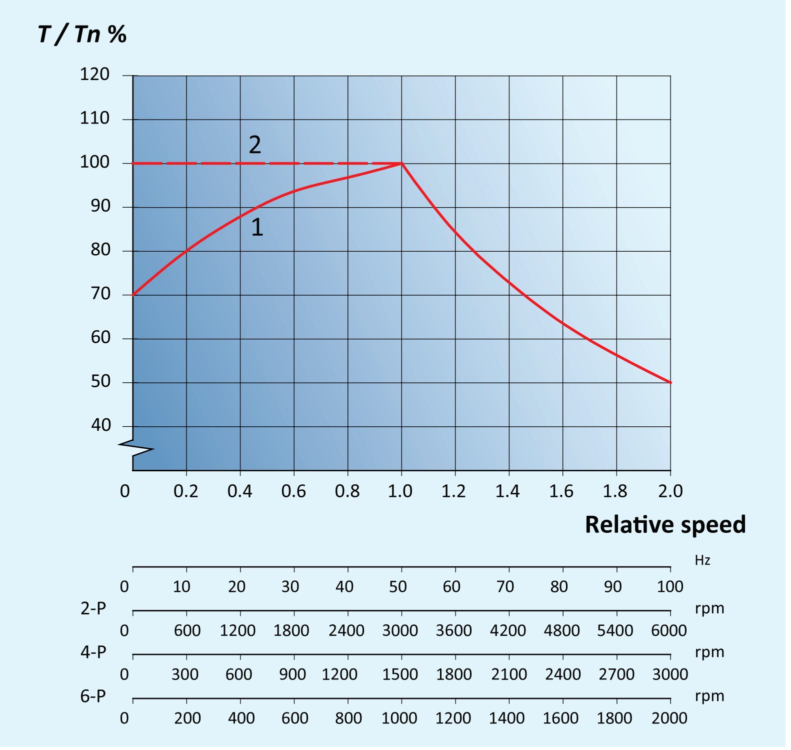

Fig 10.3.2.A Loadability of an AC induction motor

The upper horizontal scale shows the relative speed (relative to the nominal speed). The other axis shows the motor frequencies and the number of revolutions per minute for the motor in the case of a motor with 2, 4 or 6 poles.

An AC induction motor can be overloaded up to the maximum torque of T_{max} for a short period of time. The motor needs to be selected on the basis of the torque that will need to be delivered in the drive process. The cooling capacity of the motor needs to be increased so that the required torque can be delivered at a low number of revolutions per minute. In some situations where it may be necessary to deliver a high torque at low revolutions per minute over longer periods of time water cooling will be applied. In those cases it is even possible to deliver a torque of up to 140% of T_n. An important advantage of an electrical drive for this winch is that it will be possible to achieve the ideal load as per line 2 if frequency control is installed. The ideal line can be followed as closely as possible by controlling the number of revolutions per minute of the step less drive which means that the torque can be controlled by controlling the flux.

The thermal load of a frequency converter is generally more critical than that of an induction motor. The converter needs to be chosen carefully, together with the supplier, on the basis of the parameters that have been obtained for the current converter.

To adjust the torque and the number of revolutions per minute demanded by the winch to that of the electric motor a gearbox has been applied in this case too, in this case with a ratio of i=94. This too is a three stage gearbox with the same mechanical efficiency of 0,956. This means that it is theoretically still possible to choose between a 2 and a 4 pole motor.

The two characteristic points A and B of the load curve in figure 10.3 are typified by:

Gear ratio =96	A	B	Unit
Winch shaft torque	137	34	kNm
Winch shaft speed	6,9	27,4	rpm
Motor shaft torque	1480	370	Nm
Motor shaft speed	662	2650	rpm
Motor shaft power	103	103	kW

Table 10.3.2.A Typified characteristic points A and B of figure 10.3

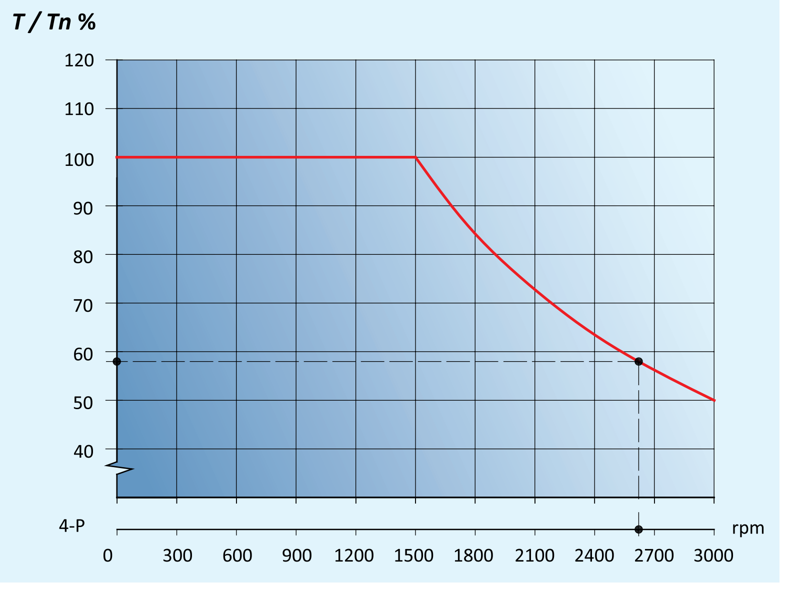

Fig 10.3.2.B Load ability for a 4-pole motor in the characteristic load conditions

At first we determine the electrical parameters for a 4-pole motor. Because of the increased cooling the following applies for the torque T_n of the motor:

The allowable load at 662 rpm is 100% with a required torque of T_n = 1480 Nm, at 2650 rpm it is approximately 58% for q required torque of T_n = 637 Nm. This means that we need an electric motor with a nominal minimal torque of 1480 Nm at a nominal 1489 rpm. The applicable power is 230,7 kW.

The choice is for a motor with the following nominal characteristics: 250 kW (400 V, 1602 Nm, 434 A, 50 Hz and 1489 rpm).

The motor current is:

$$i_m = \frac{T_{load}}{T_n} \cdot i_n = \frac{1480}{1603} \cdot 434 = 400,7 \qquad (10.14)$$

For a 2-pole motor we will need to choose a gearbox with a different ratio if we want to be able to utilise the full range of the number of revolutions per minute. In this case we have chosen a ratio of 185,2. This too is a 3-stage gearbox, again with an efficiency of 0,956.

The two typical points A and B of the load curve are then characterised by:

Gear ratio =185.2	A	B	Unit
Winch shaft torque	137	34	kNm
Winch shaft speed	6.9	27.4	rpm
Motor shaft torque	768	192	Nm
Motor shaft speed	1269	5074*	rpm
Motor shaft power	103	103	kW

Table 10.3.2.B Characterisation of typical points A and B

Note: the 5074* rpm value in this table is high. It needs to be checked if the gearbox can withstand such high speeds. Also the brake system needs to be designed to be able to brake such a high speed in short time.

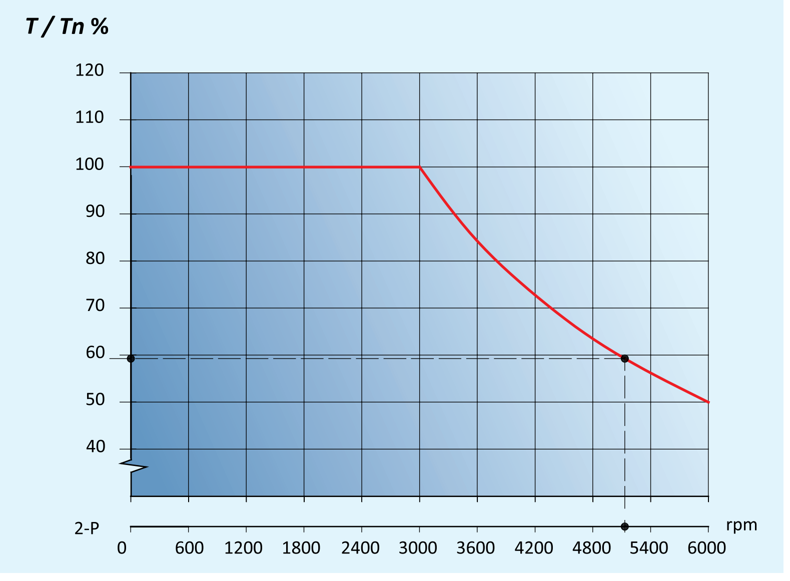

Fig 10.3.2.C Loadability for a 2-pole motor in the characteristic load conditions

At 1269 rpm the allowable load is 100% with a required torque of T_n = 768 Nm and at 5074 rpm approximately 59% with a necessary torque of T_n = 325 Nm.

In this case a motor with the following nominal characteristics has been chosen: 250 kW, (400 V, 800 Nm, 422 A, 50 Hz and 2983 rpm).

The motor current is:

$$i_m = \frac{T_{load}}{T_n} \cdot i_n = \frac{768}{800} \cdot 422 = 405 \quad (10.15)$$

It was to be expected that both electric motors would require about the same amount of electrical power. Their load patterns would be more or less the same. The 2-pole motor is a lot cheaper than the 4-pole motor.

10.4 Realisation of the drive

Here we will give the necessary extra information necessary for the realisation of the design to make a distinction between the different types of drive mechanism.

10.4.1 The high torque motor solution

There are now designs available for a high torque hydraulic motor where the motor housing rotates and the drive axle is static. Fixing the housing directly onto the drum of a winch creates a very compact design.

The band brake can be placed directly onto the circular part of the rotating motor housing.

Fig 10.4.1.A Compact design of a high torque motor and winch (Courtesy of Hägglunds Drives)

By changing the hydraulic motor from a high stroke volume to a lower stroke volume, the hydraulic motor can rotate faster whilst maintaining the same maximum volume flow for the pump. The output torque of the motor will however reduce proportionally with the reduced stroke volume. The adjustment for this type of motor, needs to be made when the motor has stopped. There are however other high torque motors that can be adjusted whilst the motor is operating.

To determine at which stroke volume the motor needs to be set, it is necessary to measure the load on the cable with the help of a load cell. The smaller stroke volume will be chosen when the load on the cable is less than a certain value, in this example < 62.5 kN. If the load on the cable is higher, the stroke volume has to be set at a higher volume too. If the load on the cable changes from low to high during the process, then the winch will need to be stopped so that the stroke volume can be adjusted. If the load on the cable changes from high to low during the lifting process, then it is not strictly necessary to stop the winch and it will be possible to continue working at a high stroke volume and a lower number of revolutions per minute.

These conditions require the use of a computer or a simple PLC. The decision to switch from a high stroke volume to a lower one cannot be left solely to the operator because an error of judgment can lead to the loss of the pulling force of the winch.

The braking power of this installation is required for the lowering of a load. The load does after all drive the drum and thus the hydraulic motor in this situation. In this situation the motor works as a hydraulic pump and drives the variable pump in the closed circuit. The pump axle exerts a driving torque on the electric motor which means that the motor will start functioning as a generator for the electrical network that would normally be used as electrical feed for the electric motor. The electric motor will start to turn a bit faster at the nominal speed + the slip. The release speed of the winch is now also determined by the flow that the adjustable variable pump can take from the hydraulic motor.

The net winch power can be determined from the load on and the speed of the cable, in this case 250 kN and 20 m/min (or 0,33 m/sec), which gives a power of 83,3 kW. Because all the working mechanical efficiencies will be reversed and the cumulative efficiency is 69,7%, the winch will deliver a maximum power of 83,3 x 69,7% = 58,1 kW to the electrical network at maximum load on the winch.

The requested constant tension CT control system will be used if a load has been hooked up and the platform on which the winch is situated moves relative to the position of the load. This happens if a supply ship delivers goods to a crane vessel or if the winch is connected to an underwater load that is positioned on the sea or riverbed (for example the suction tube of a dredger).

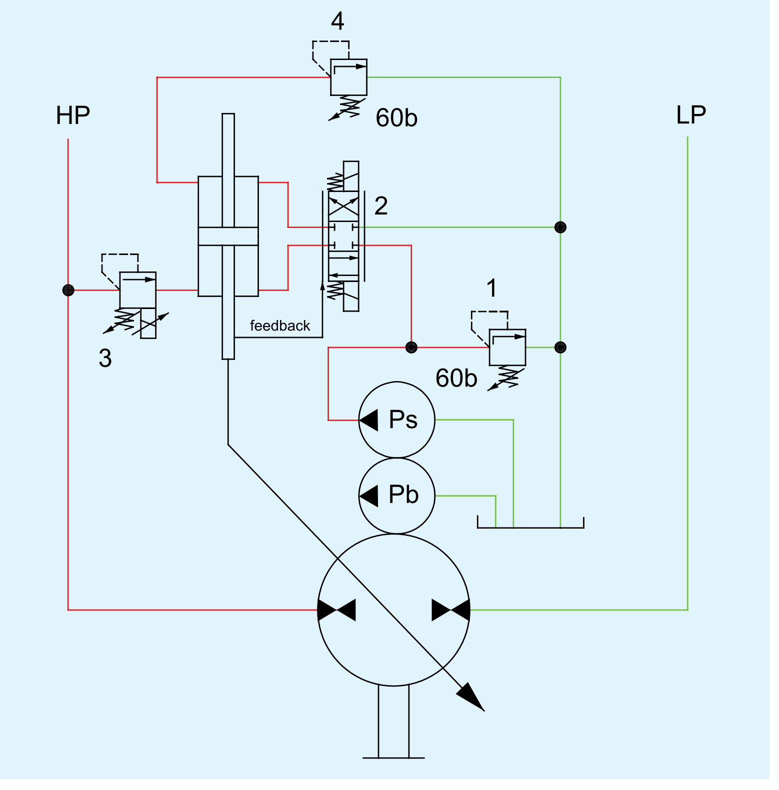

Fig 10.4.1.B Principle of CT via swash plate control of the variable pump

The CT is influenced by the torque delivered by the hydraulic motor, the number of windings on the drum and the mechanical efficiency of the drum and any other cable sheaves in the installation. The most accurate way to achieve constant tension on the cable is achieved if the signal from the load cell can be fed back to the control mechanism of the drive system that will then regulate the pressure drop across the hydraulic motor. This will require an extra control loop with a computer or PLC. The required accuracy of the discussed applications is however not very high. The main requirement is usually that one wants to maintain a minimum cable load of for example 40 kN plus or minus 20 kN.

It is also possible to achieve CT in a simpler way by keeping the load pressure constant in a closed loop system. The load variation that does still occur is then determined by the mechanical efficiency of the hydraulic motor (98%) and the winch drum (also 98%). In paragraph 10.3.1 we had already calculated that the total variation in the force will be a maximum of 7,9%, which is more than sufficiently accurate for the cable load specified above.

It is also possible to build an extra servo pump P_s alongside the boost pump P_b of the closed loop system described previously. This pump delivers the control power for proportional valve 2 that controls the stroke volume of the pump in a stepless way. Either a mechanical or an electronic feedback mechanism is often applied to optimize the accuracy of the swash plate position. For CT controls the variable pump adjustment is controlled in such a way that the pump sends its maximum output to the HP pipe.

The variable flow from the pump is normally achieved by controlling the proportional valve (2). Servo pressure is used to control the position of the swash plate. The proportional sequence valve (3) is fitted in a branch of the HP pipe. This sequence valve will open as soon as the pressure in the HP pipe is higher than the pressure set for this proportional valve. The position of the swash plate will then be adjusted in the direction that will reduce the stroke volume and the pressure in the HP pipe. The oil that is displaced in the opposite section of the control valve will be discharged via pressure control valve (4). Because of the higher pressure level of the sequence valve (3) relative to the servo pressure, the CT controls will overrule the proportional output controls.

10.4.2 The high speed motor solution

In order to adjust the torque and the number of revolutions per minute of the winch drum to the available torque and number of revolutions of the hydraulic motor, it is necessary to use a gearbox. In this case a planetary gearbox has been chosen because of the high gearing ratio. There are designs available where a drum is fitted to the output axle of the gearbox which can act as the carrying element for the winch drum, in the same way as with the high torque motor design.

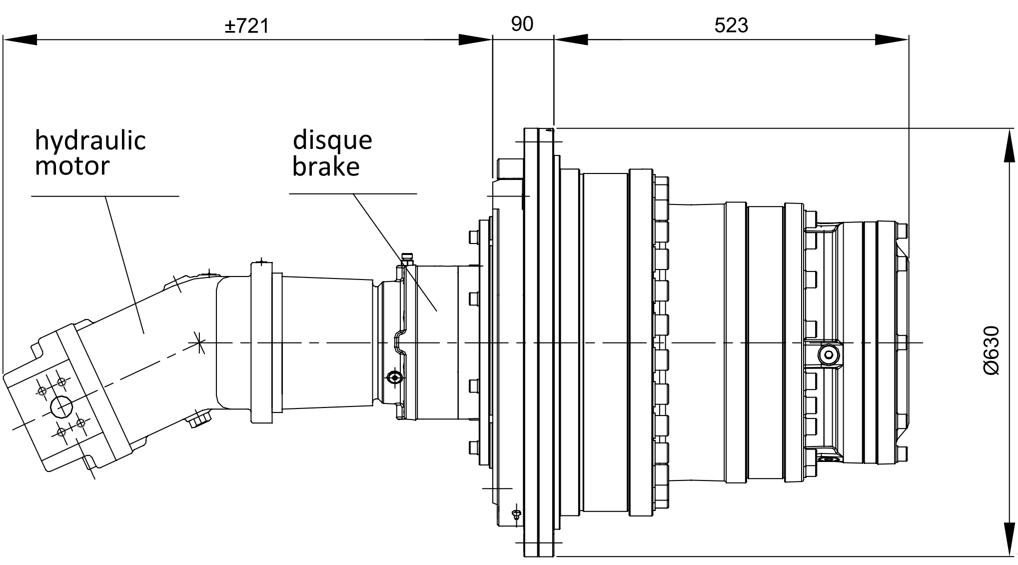

Fig 10.4.2.A Planetary gearbox as construction part of the winch drum with lamellar brake and variable hydraulic motor (Courtesy of Brevini)

The input axle of the gearbox has been fitted with a lamellar brake. This type of brake can only be used as a parking brake, i.e.: only after the drive mechanism has come to a stop. The brake can be used as an emergency brake but will then have to be inspected for possible internal damage or wear.

A different type of gearbox is the one where the output axle is fitted as a pinion that can drive a gearwheel which has been fitted to the winch drum. In this case the designer will be able to choose to apply several drive units for one single gearwheel, which allows for the distribution of the winch's drive power over several drive motors. This design is often used for slewing drives for cranes or for thruster drives. To compensate for the forces on the gearwheel teeth, the exit axle of the gearbox will then be fitted with extra radial bearings. The gearbox ratio in these solutions will of course be different, see figure 10.4.2.B, because there is already a gearing ratio for the open drive between the pinion and the gearwheel on the winch drum (usually in the order of $i = 5$ to $i = 9$).

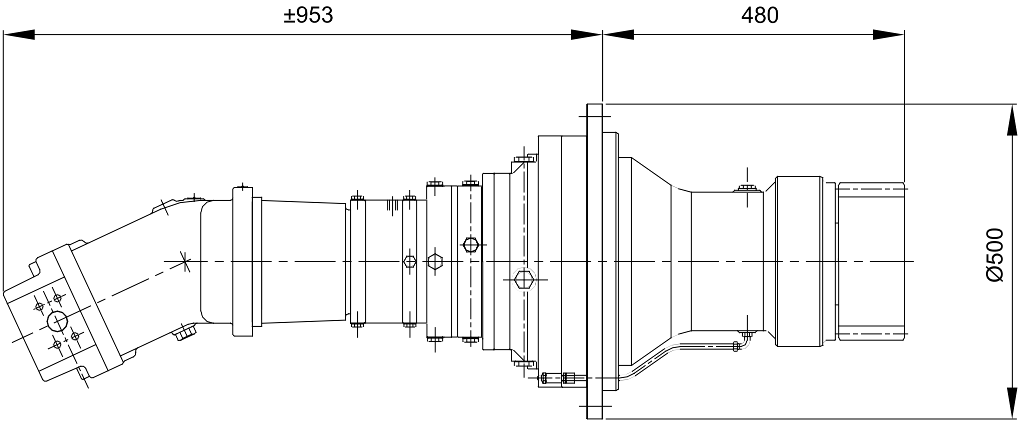

Fig 10.4.2.B Another compact design of a hydraulic motor with a disque brake and gearbox with a pinion as output shat. This construction is often used for slewing drives of cranes. (Courtesy of Brevini)

The hydraulic motor with variable stroke volume can be switched from the maximum to the minimum position with the help of a 4/2 magnetic valve, just like the high torque motor, The stroke volume of a fast running hydraulic motor however can always be adjusted whilst the motor axle is turning. This is an important advantage for the winch drive under consideration because it means that it will not be necessary to stop the winch when a change of load occurs. Measuring the cable load will however remain a necessity.

Various manufacturers are able to deliver a variable stroke hydraulic motor with an automatic stroke volume control mechanism. An important advantage of this arrangement is that the stroke volume will be adjusted in a stepless way, which means that the characteristics of line 2 can be followed completely. The other advantage is that this type of control mechanism doesn't require additional electronic control circuits.

Shown is an example of a variable hydraulic motor with automatic, operating pressure dependent, adjustment of the motor's output volume. The control unit measures the operating pressure internally at A or B and, when the pressure reaches the set pressure value, the controller swivels the motor with increasing operating pressure from $V_{g\text{-}min}$ to $V_{g\text{-}max}$. For safety reasons it is not allowed to start the control at $V_{g\text{-}min}$. When an external pilot pressure is applied to the X-port the variable motor swivels to the maximum angle.
The CT control for a fast running hydraulic motor functions in exactly the same way as those for the high torque motor mentioned earlier. An important difference with the high torque motor is the total mechani-

cal efficiency of the drive mechanism. An axial piston motor has a lower mechanical efficiency than a high torque motor. The planetary gearbox brings an additional mechanical efficiency of approximately 98,5% for each axle pin. This means that the total cable load variation for a constant pressure control mechanism is approximately 25,8% (see also paragraph 10.3).

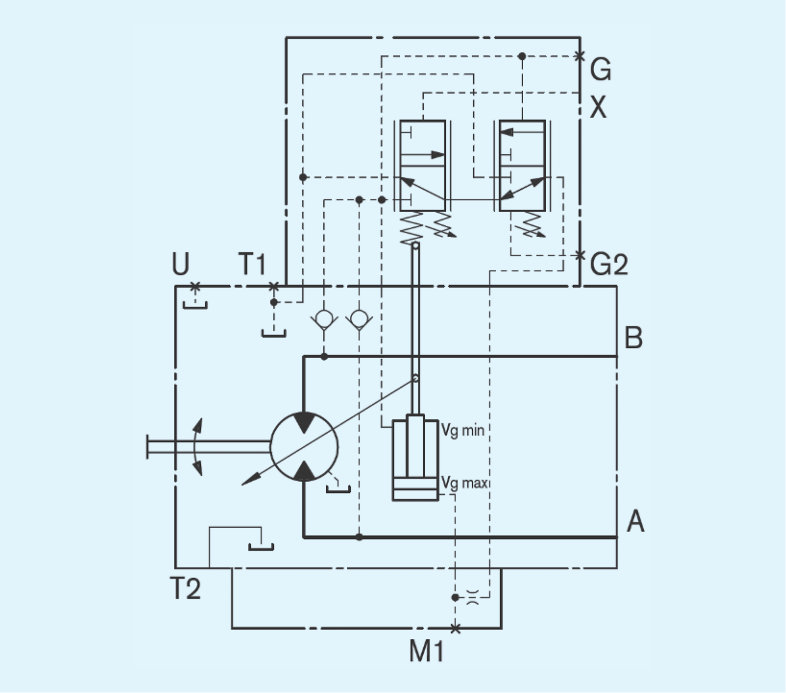

Fig 10.4.2.C Hydraulic diagram of a variable hydraulic motor with automatic, operating pressure dependent, adjustment of the motor's displacement volume (Courtesy of Bosch Rexroth).

The mechanical efficiency of some types of motor at the starting position is much worse than in the running condition. Mechanical efficiencies of only 60% at start-up are quite common. The supplier/manufacturer of the hydraulic motor will be able to give you this value.

10.4.3 The electric motor solution

The advantage of an electrical drive is that, as with the variable fast running hydraulic motor with automatic swash plate control, the whole range of the motor's revolutions per minute can be reached in a stepless way without having to stop the drive. A disc brake has been fitted between the electric motor and the gearbox. The brake can be activated electrically or hydraulically.

The drive now consists of a gearbox connected to the winch, an AC induction motor and a frequency controller which regulates the number of revolutions per minute across the whole range of the torque in such a way that the net delivered power follows a constant power curve. Detailed attention needs to be paid to the maximum current that needs to be delivered to the motor, also at low frequencies, for the calculation of the specification of the frequency regulator.

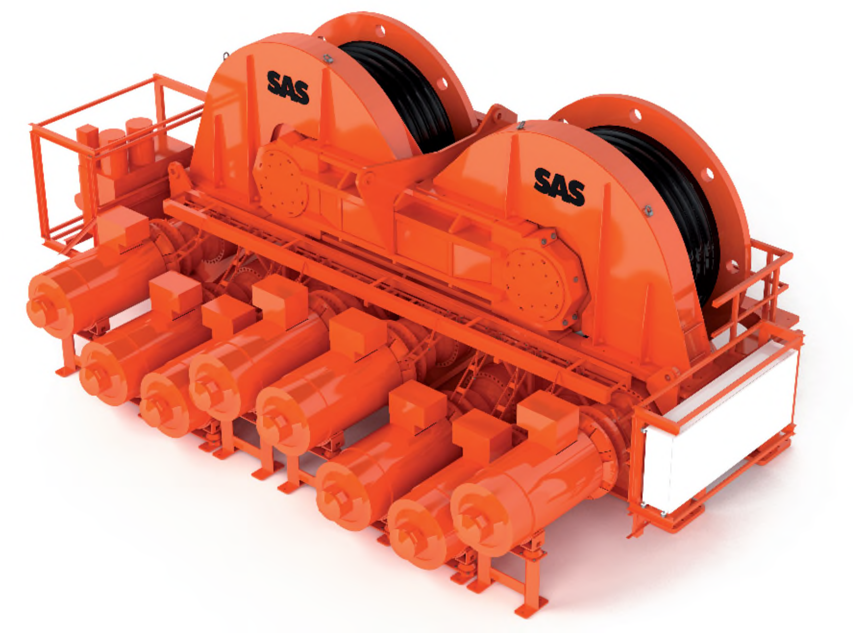

Fig 10.4.3.A 4000 kN traction winch with 8 electrical drives and gearboxes (Courtesy of SAS)

10.4.4 Design choice

In the previous paragraphs we have done the calculations for the most important parameters for the three possible drive mechanisms. The designer needs to give consideration to several aspects of the design before an optimum choice can be made. Because there are significant differences between the different operational circumstances we will not go into further detail about the final choice for the winch. For all the other cases, the following design considerations need to be taken into consideration:

- The dimensions of the design relative to the available space., do we have more space in diameter or in length
- The extra space required for an HPU, the electrical switch box and the frequency drive
- The total weight of the design
- The dynamic considerations that are of significance
- Whether or not the drive is in continuous or intermittent operation
- The available cooling medium (air or water)
- Environmental conditions such as temperature (maximum and minimum), air humidity, level of pollution, ship's movements, explosion risks (Atex)
- The available expertise of operational and maintenance personnel
- The vulnerability of the components and/or the whole installation, including the operational availability of the drive mechanism
- Technical requirements posed by legislation, guidelines, application of relevant standards etc.
- Certification requirements of the registration organisation, approval authorities etc.
- Costs of the design, including purchase and installation
- Maintenance costs

After all the questions about the above aspects of the design have been answered it will be possible to make the correct choice for the rotating drive. In practice it happens more and more that the electrical drive with a frequency controlled drive motor is the preferred choice for rotating drives of > 200 kW for on board offshore installations and dredging vessels.

10.5 Hydraulic drive in open loop

It is not always necessary to use a hydraulic motor in a closed loop system. That would mean a separate variable pump would be needed for each hydraulic motor. A hydraulic motor can also be applied to a so-called open circuit where a pressure connection P, a return connection T, and in the case of a hydraulic motor also always a drain connection DR, are required. The example is about the drive mechanism for the longitudinal movement of a pipe handler in the line-up station of an S-lay barge. The hydraulic motor drives a roller mechanism with the new single joint that needs to be welded to the main pipe on it. The main pipe will always move in a longitudinal direction as a result of the movement of the ship relative to the main pipe during the welding process. There needs to be continuous contact between the single joint and the main pipe during the welding process whilst the maximum force on the beveled edge cannot be exceeded because the beveled edge may get damaged. In technical term this is called the endo movement.

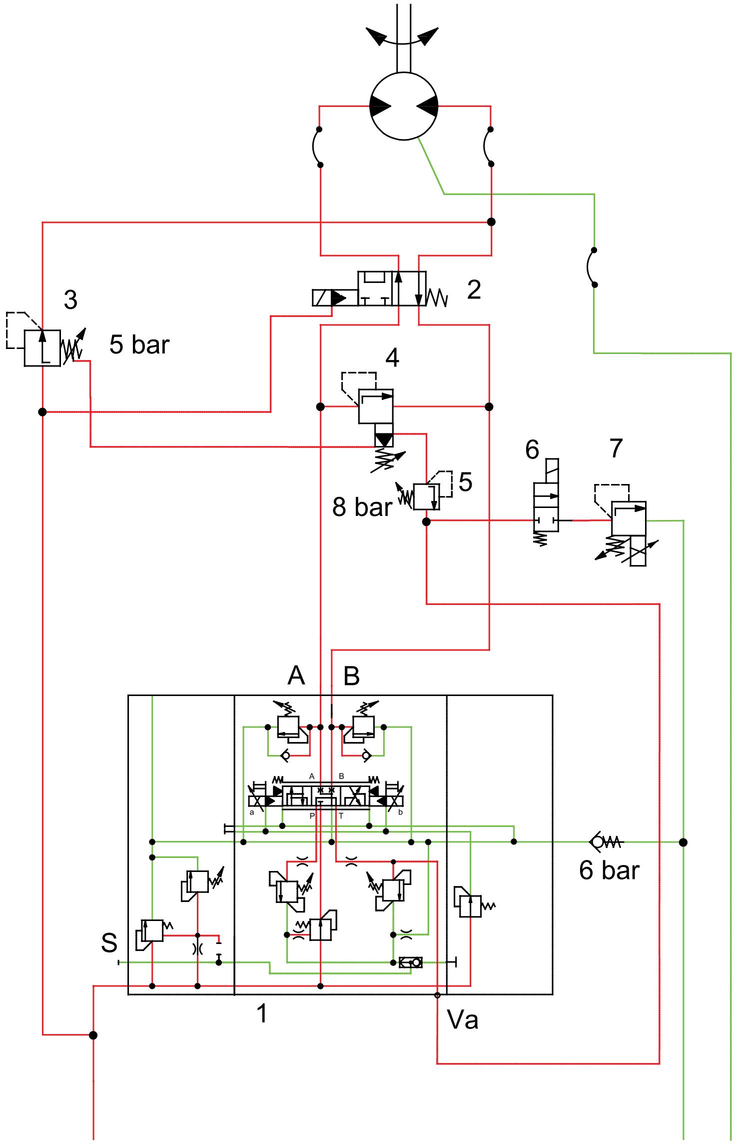

Fig 10.5.A Endo movement in S-Lay barge line up station with CT control

The speed of the hydraulic motor can be varied steplessly in both rotating directions with the help of proportional valve (1). This valve has specifically been chosen because it is possible to externally control the 2-way pressure compensator valve fitted in it via the V_a port. A 4/2 directional pilot operated valve position (2) is fitted so that the hydraulic motor can be switched to a freewheeling position at the moment that the single joint has been welded to the main pipe and the main pipe is pulled through for the next single joint. The 6 bar back pressure valve in the return pipe as well as the pressure reduction valve in position (3) with a setting of 5 bar make sure that there will be sufficient boost pressure in the hydraulic motor to prevent cavitation.

For the CT function the main proportional valve in position (1) will be opened fully so that there will be flow in the A-port. The electrically controlled directional 2/2 valve in position (6) will be activated at the same time. The pressure in the A-port is then limited by the pressure control valve position (4). The pressure setting of this valve is proportional with the use of pilot control valve (7). The pilot pressure of proportional valve (7) is also measured at the connecting port V_a of the main proportional valve, which in turn will limit the maximum pressure from the main proportional valve to the A-port.

Because of the extra pressure control valve in position 5 which is set at 8 bar, the pressure in the A-port will be 8 bar higher when the hydraulic motor is driven by the pipe and the volume flow through pressure control valve in position (5) is generated by the hydraulic motor.

These CT controls are built out of standard components and can be applied to any type of hydraulic motor. The other advantage is that no electronic controls are required. The possible disadvantage is an extra hysteresis of 8 bar (the pressure setting of the valve in position (5)). The hysteresis is of course also caused by the reversal of all the mechanical efficiencies of the drive mechanism. Quite a lot can be gained by making a good choice of for example a high torque motor instead of a fast running motor with gearbox.

The hysteresis will also increase as a result of the extra pressure drop that will develop across the pilot pressure control valve in position (4) in case of a high volume flow. It is therefore important that the dimensions for the volume capacity of that valve are chosen carefully.

10.6 Dynamics of rotational drives

The drive torque of an actuator is determined by the sum of a number of parameters. The assumption is that the mass of inertia of the drive doesn't change. For a winch for example, that would only apply if the load doesn't change during the process. The torque necessary to overcome the friction of a drive as well the eventual torque required to control the process can be constant or can vary over time.

$$T_a = (I_a + \frac{I_L}{i^2}).\frac{d\omega}{dt} + \frac{T_{fr} + T_L}{i} \qquad (10.16)$$

where:

T_a	=	torque of the actuator	Nm	I_L	=	inertia of the load	kg.m²
T_{fr}	=	torque due to friction	Nm	i	=	gearbox ratio	
T_L	=	static torque of the load	Nm	ω	=	angular speed of actuator	rad/s
I_a	=	inertia of the actuator	kg.m²				

The moment of inertia or in short inertia (resistance to a change in rotational motion) of the load I_l is reduced by a factor i^2 because of the gearbox. This is called the reduced inertia. The inertia caused by the load is determined by the inertia of just the load and the inertia of the drum together with the cable windings on it. Each step of the gearbox itself also has an inertia, which should be calculated through to a reduced inertia on the actuator axle.

In our example of the winch drive of paragraph 10.3 the static load, and thus the static torque T_l, is constant if we assume a load caused by the mass only. To get a better picture of the theoretically possible rotational acceleration dω/dt at the actuator, the formula above has been applied to the four drive mechanisms from paragraph 10.2.

The maximum possible rotational acceleration at the drum can then be calculated with:

$$\frac{d\omega_{dr}}{dt} = \frac{1}{i} \cdot \frac{d\omega}{dt} \qquad (10.17)$$

with:

ω_{dr}	=	angular speed at the drum	rad/s

A higher peak torque can be delivered by all four drives, high torque motor, fast running motor and electric motor (4-pole and 2-pole type), for a short period of time. For the hydraulic motors a higher working pressure is required, which can easily be realised in a closed circuit, as long as the pressure relief valves in the circuit have been set at a high enough value.

The rule for the electric motor is that it can deliver a torque of T_{max} over a short period of time.

Parameter	Value	Unit			
Drum diameter	1077	mm			
Mass of the load	15000	Kg			
Inertia of the external load	4350	kg.m²			
Inertia of the drum	1100	kg.m²			
Total inertia of the load	5450	kg.m²			
Peak Torque 5th layer	82.200	Nm			
High torque motor	High torque motor	High speed motor	Electric motor 4-pole	Electric motor 2-pole	Unit
Gearbox ratio	1	117,5	96	185,2	
Intermittent peak pressure (allowed)	320	400			bar
Intermittent peak torque	193.600	1592	4168	2080	Nm
Inertia of the actuator	152	0,061	5,7	2,7	kg.m²
Maximum acceleration at the drum ω_{dr}	19,8	16,6	5,5	3,1	rad/s²

Table 10.6.A Calculation of the maximum possible acceleration at the drum for the four possible drives.

This result is based purely on the torque that can be delivered by the actuator for a short period of time. The hydraulic drive design with the high torque hydraulic motor stands out on account of the 'reserve' power that's available to accelerate the load.

The actuator together with its drive components also has a limiting dynamic factor. For many drives the dynamic factor doesn't or hardly plays a role, because the system will not be subjected to large speed variations. Increased dynamic characteristics do however apply to heave compensation systems. This means that for this type of system the dynamic characteristics of the drive system must be determined in detail. A simulation model will be set up for this, using the formulas discussed in this book. The following approach will suffice for an initial rough approximation of the dynamic reaction time of a rotating hydraulic drive mechanism.

In paragraphs 6.7 and 6.8 we have already shown that a rotating hydraulic drive system behaves like a mass spring system that will have its own natural frequency ω_o and a dynamic transfer function:

$$H_s = \frac{\omega}{Q_p} = \frac{1}{\frac{s^2}{\omega_0^2} + 2\cdot\beta\cdot\frac{s}{\omega_0} + 1} \qquad (10.18)$$

and:

$$C_o = \frac{2\cdot V_{st}^2\cdot E}{\pi\cdot V_{st} + 2\cdot V_p} \qquad (10.19)$$

with:

$$\omega_o = \sqrt{\frac{C_o}{(I_a + \frac{I_L}{i^2})}} \qquad (10.20)$$

Let's assume that we have a closed circuit where the variable pump has been placed approximately five meters away. This means that internal diameter needs to be 32 mm for a planned flow of 280 lpm and a maximum flow speed of 6 m/s. That would result in a pipe volume V_p of 0,004 m³. The modulus of elasticity of the oil $E = 10^9$ N/m². This means that we can calculate the natural frequency of the two hydraulic drive systems.

Parameter	Symbol	Value	Unit	
Drum diameter		1,077	mm	
Load inertia	I_L	5,450	kg.m²	
Line volume	V_p	0,004	m³	
Elasticity of oil, E	E	1,00 x 10⁹	N/m²	
		High torque motor	High speed motor	Unit
Gearbox ratio	i	1	117,5	
Actuator inertia	I_a	152	0,061	kg.m²
Displacement volume		38000	250	cc/rev
Displacement volume	V_r	6,05 x 10⁻³	3,98 x 10⁻⁵	m³/rad
Stiffness of drive	C_o	2,71 x 10⁷	3,90 x 10³	Nm/rad
Natural frequency	ω_o	69,6	92,5	rad/s

Table 10.6.B Calculated parameters for the dynamic behaviour of the 2 hydraulic motor solutions in a closed loop system

Comment on the table:

The natural frequency of the fast running hydraulic motor is higher than that of the high torque motor. This is strongly influenced by the volume in the pipe relative to the stroke volume of the hydraulic motor.

The dynamic transfer function H_s can be rewritten as:

$$H(s) = \frac{\omega_0^2}{\left(s + \beta.\omega_o\right)^2 + \omega_o^2\left(1-\beta^2\right)} \tag{10.21}$$

This means that we can now also determine the response of the transfer function to a step change:

if: $$\beta.\omega_o = \sigma \tag{10.22}$$

and: $$\omega_d^2 = \omega_o^2 \cdot \left(1-\beta^2\right) \tag{10.23}$$

then: $$h(t) = 1 - e^{-\sigma t}\left(\cos\omega_d \cdot t + \frac{\beta}{\sqrt{1-\beta^2}} \cdot \sin\omega_d \cdot t\right) \tag{10.24}$$

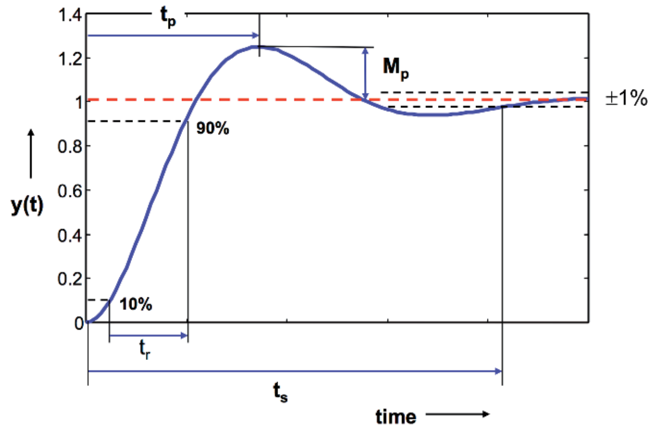

Fig 10.6 Step change response for a 2nd order system with $\beta << 1$

The graph shows the response of this 2nd order system to a step change. The following applies to the various parameters in the graph:

$$t_r = \frac{1,8}{\omega_o} \tag{10.25}$$

$$t_p = \frac{\pi}{\omega_o \sqrt{1-\beta^2}} \tag{10.26}$$

$$t_s = \frac{4,6}{\beta.\omega_o} \tag{10.27}$$

The result of the dynamic calculations is that the drive with the fast running hydraulic motor will provide a faster response than that of the high torque motor in these operating conditions. For the winch drive itself such a fast reaction time is usualy not required. Winch drives normally have a dedicated ramp generator being built into the electronic control circuit that , on purpose, limits the most fastest reaction time to a minimum of 3-5 seconds. A standard variable pump also has a reaction speed of approximately 0,5 s from zero to the maximum output.

For an electrical drive another dynamic factor is important. The inverter delivers the frequency, current and the voltage to the AC induction motor. A magnetic field is built up on the basis of the flux = *V/f*. There is little known information in the literature about the dynamic characteristics or reaction speeds necessary to build a magnetic field. Experience tells us that the magnetic field and thus the ability of the electric motor to generate the necessary torque can be established in 1- 2 msec. This delaying factor can be ignored when compared with the delaying factor of a hydraulic drive.

10.7 Secondary drive for sawing mechanism

In January 2003 the car transporter ship Tricolor sank after it collided with a tanker on the North Sea just off the Belgian coast. The wreck needed to be sawn into 9 pieces first before it could be removed. The saw consisted of a long cable equipped with a number of sawing bushes positioned over a length of about 20 meters, creating something like a string of beads. The sawing cable, approximately 42 mm thick, was wound around two winches, positioned on pontoons on either side of the wreck. The sawing function was created by hauling the sawing cable over part of the sunken wreck at the high speed of 60 m/min, with a nominal tension of 700 kN and a maximum peak force of 1080 kN. The important factor during the sawing was that the pulling cable was controlled for speed, whilst it had to be possible to set a braking force for the releasing cable.

Fig 10.7.A Picture of the saw for the tricolor, taken during tests

It proved possible to build 2 new winches of the necessary power output as well as 2 diesel hydraulic-pump units within the required delivery period of 2 months. A hydraulic drive similar to the one described in paragraph 10.1 was chosen. A secondary hydraulic drive offered many advantages.

A winch with a secondary drive can easily be switched from a speed control for the pulling winch function to load control for the releasing winch function. The reaction speed of the secondary drive is so fast that the force from the releasing winch can be constant when there are strong fluctuations in the sawing load.

For each winch, three diesel powered variable axial piston pumps have been fitted in parallel to form the pressure network for the secondary drive unit. When the winch performs the releasing function, the secondary motor works like a pump. The energy release from the winch cable is then transformed into hydraulic energy for the constant pressure network.

If a set of pumps is driven electrically then the hydraulic energy can be 'pumped back' into the electrical network via the pumps. This would not be possible if the pumps were driven by diesel engines. This is why a non-return valve has been fitted to the pressure line of the pumps together with an extra, electrically controlled, pressure limiter valve so that the recovered energy from the cable can be released (converted into heat). For more details, see figure 10.2.B.

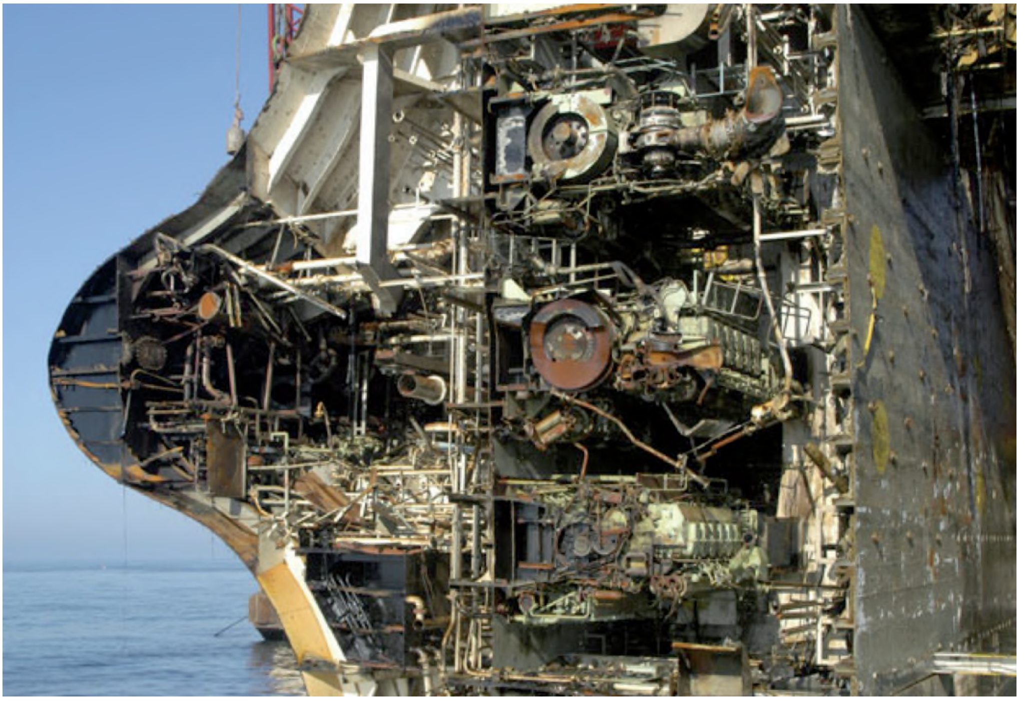

Fig 10.7.B The stern section of the wreck, just sawn by the hydraulic driven winches

The photo shows clearly that the Tricolor was successfully sawn into pieces by using two hydraulically driven winches with a secondary drive mechanism. This is the first sawn off section in which the main propeller shaft was also cut into pieces. A total of 8 sawing cuts were made to divide the Tricolor into 9 pieces that were small enough to be lifted from the seabed.

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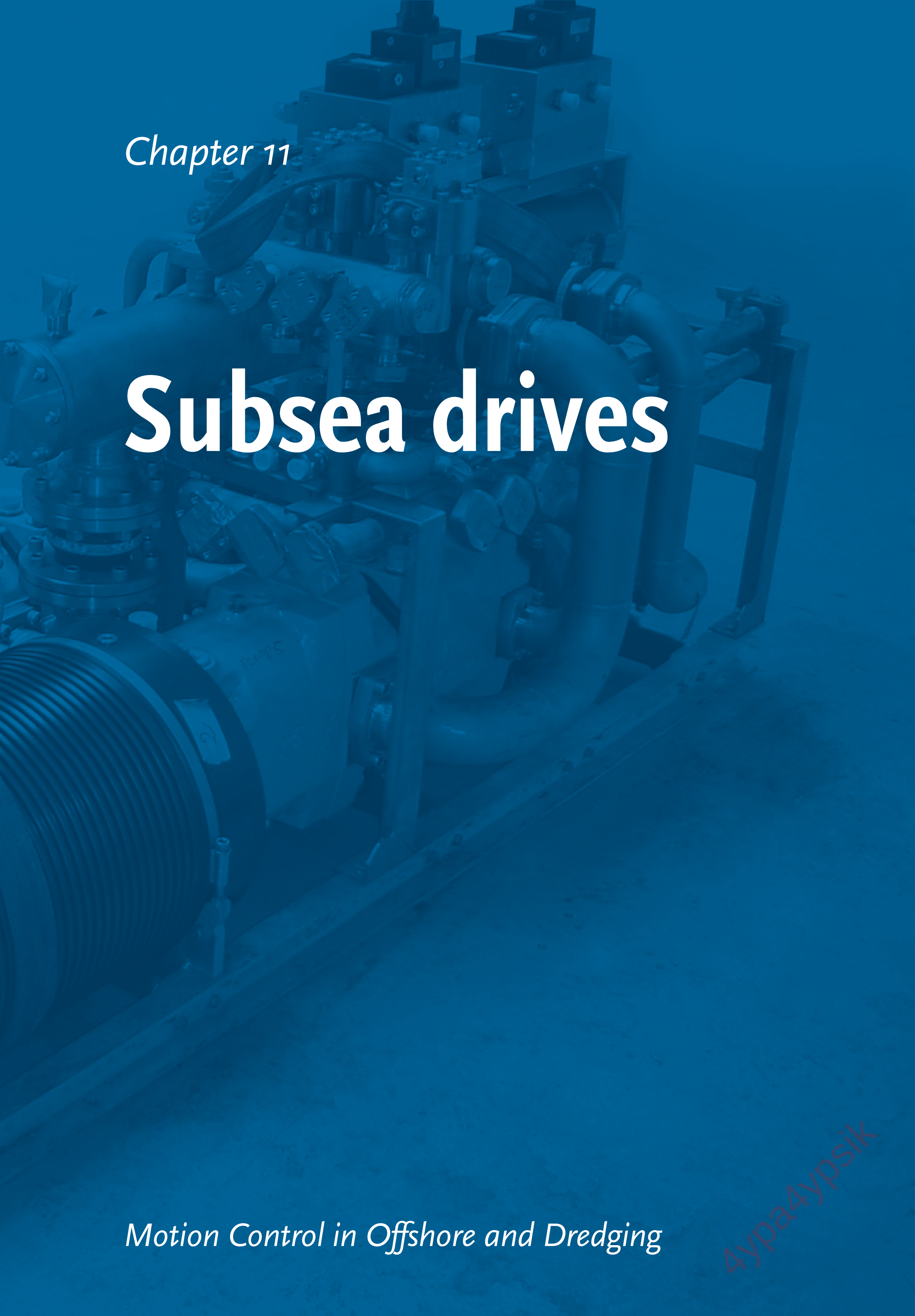


Chapter 11

Subsea drives

Motion Control in Offshore and Dredging

Chapter 11

Subsea drives

Because of the large number of activities in the offshore and dredging industries that take places under water, drives have been developed that function in that environment. In this chapter we will discuss the most important aspects of the hydraulic drives that have been developed for that environment.

In the early days the primary drives were installed above water level whilst the actuator would be placed below the water line. In those days, the mainly hydraulic energy would be supplied to the actuator via hydraulic hoses. This method is still used for underwater piling hammers that work at depths of up to 1200 meters. Because the activities are taking place at ever greater depths, hydraulic drive units that get their energy via electrical cables have been developed that can operate completely below sea level. The electrical units have been developed to the extent that they can now operate at depths of up to 3000 meters with power output in the order of 6,5 MW. These units are used for process equipment like water separation, compression units and the pumping of oil and gas.

The control mechanism on a Remote Operated Vehicle (ROV) consists of a drive unit for the thrusters and the manipulators as well as the camera and the tilting function. The actuators that are used in these circumstances are nearly always hydraulic. The hydraulic energy is generated in the ROV itself, whilst the electrical feed is supplied via an umbilical. The ROV is connected to the docking station via the umbilical, which can be several hundred meters long.

P. Albers, *Motion Control in Offshore and Dredging*, DOI 10.1007/978-90-481-8803-1_11,

11.1 List of symbols

A_b	=	bottom area of cylinder	m^2
H_w	=	water depth	m
P_w	=	hydrostatic pressure	N/m^2
g	=	specific gravity	m/s^2
ρ	=	density of seawater	kg/m^3

11.2 Subsea hydraulic drives

If an actuator is placed under water then, first and foremost, it needs to be resistant to seawater and all the phenomena such as the growth on the outside as well as the penetration of salts that occur in that environment. The housing is often made from Stainless Steel or Duplex materials and special coatings can also be applied (eg. as specified by Norsok). Apart from that, the piston rod is also fitted with a coating, which is nowadays often ceramic. The piston rod seal is probably one of the most important elements of the construction, since it will be in contact with both the sea water and the hydraulic liquid.

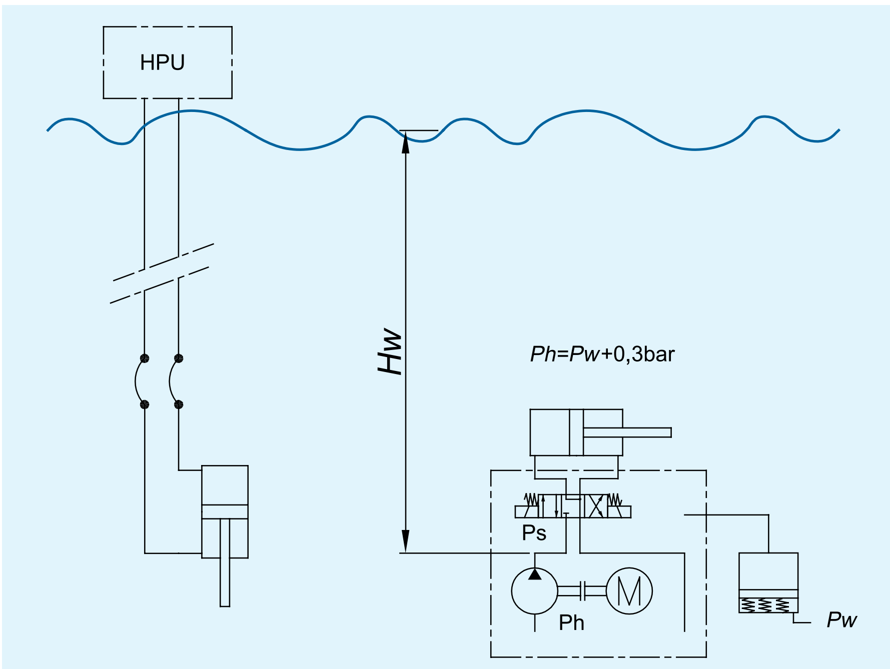


Fig 11.2.A Hydraulic systems for subsea with HPU installed above and below water

The formula for the hydrostatic pressure at a certain depth is:

$$P_w = \rho . g . H_w \qquad (11.1)$$

Where:

ρ	=	density of seawater 1004 – 1008	kg/m³	H_w	=	water depth	m
g	=	specific gravity	m/s²				

The hydraulic pressure in the actuator that has been placed below sea level is determined by the external load and the resistance of the pipe work but also by the pressure of the hydraulic liquid column, which is determined by the above formula but this time with a lower density for the liquid of approximately 900 kg/m³. This means that the hydraulic pressure for an underwater actuator that isn't always running is approximately 10% lower than the surrounding water pressure. The piston rod seal will be able to resist a water pressure that is 2-5 bar higher but not the 10-15 bar that will exist at depths of 1.000 to 1.500 meters. The second problem develops with the hydraulic hoses, which will have made up of shorter pieces of 40 meters. It has been discovered that the hose itself will develop a rather high transverse oscillation if the water pressure is higher than the absolute pressure of the hydraulic liquid. This has been particularly noticeable where piling hammers are concerned.

To avoid an internal liquid pressure that is too low, an adjustable back pressure valve is fitted in the return lines for piling hammer applications that ensures an increased pressure of 2-6 bar in the actuator, which means that the internal pressure in the hose is always higher than that of the seawater.

If the whole hydraulic system is completely under water, then there is no hydraulic connection that will create a hydrostatic column as it would do with an actuator that had been placed below the water line where the pressure in the reservoir is equal to the water pressure itself, which would be the equivalent of 10,5 bar for every 100 m depth below water! In less deep water (up to 50 meters) the valve set is sometimes placed in a pressure resistant housing that is able to withstand that level of pressure.

A much better solution is the installation of a compensator. This is an 'open' piston mechanism in which the piston has been pre-pressurised with a set of springs. The bottom side of the compensator has an open connection to the sea water, whilst the top side is connected to the hydraulic pressure in the reservoir. The piston in the compensator will come to an equilibrium position once the hydraulic installation has been filled with liquid. This means that the static pressure in the hydraulic liquid will always be higher than the water pressure, irrespective of the depth of the water in which the system operates. In most compensators the pressure in the set of springs is approximately 0,3 – 0,5 bar.

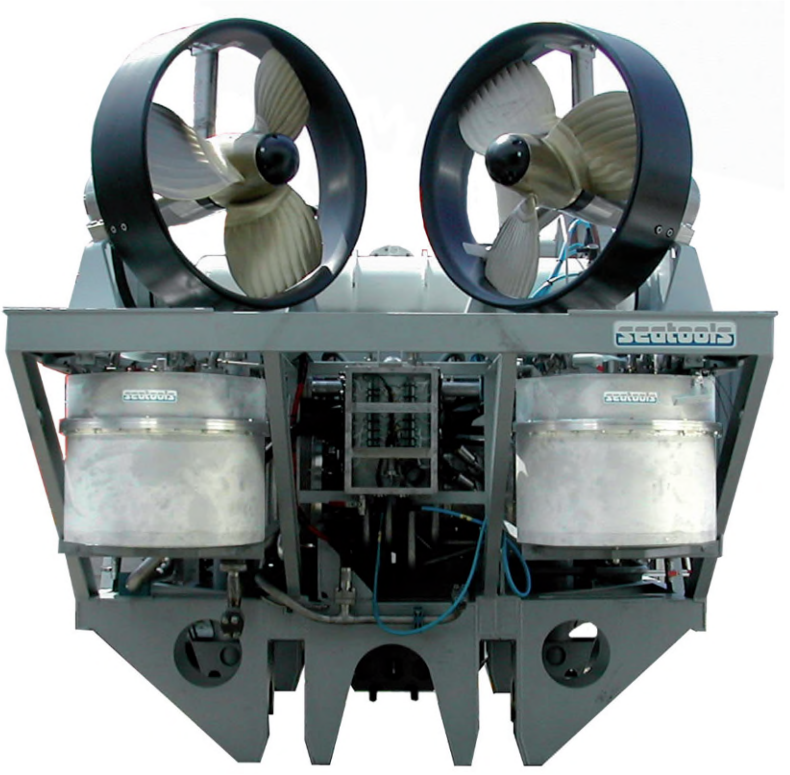


Fig 11.2.B Two 1000 litre compensators in stainless steel material for use subsea (Courtesy of Seatools)

In these sub-sea systems there is effectively no longer a reservoir at all. The volume of the compensator will have to be calculated in such a way that the compensator can compensate for the volume variations in the hydraulic liquid resulting from higher pressure (approximately 1% per 100 bar), possible temperature variations (approximately 1% per 16°C) and resulting from the varying actuator volumes, and volume differences between the bottom and rod side of the cylinders. Compensators have now been developed for a net ΔV of 0,5 – 1000 ltr.

The system pressure P_s of the system that has been placed under water will also be increased by the value P_h if the pressure in the reservoir (the compensator) is increased. This means that the absolute pressure of the system can reach values of up to 650 bar. As a result of these very high absolute pressures there will also be a higher internal tension for the materials that are being used for, for example pump housings, pump components, valve housings and valve components. It turns out that most components designed for 'above water' use at pressures of up to 350 bar can be used under water, without further adaptations, at depths of up to 3000 meters.

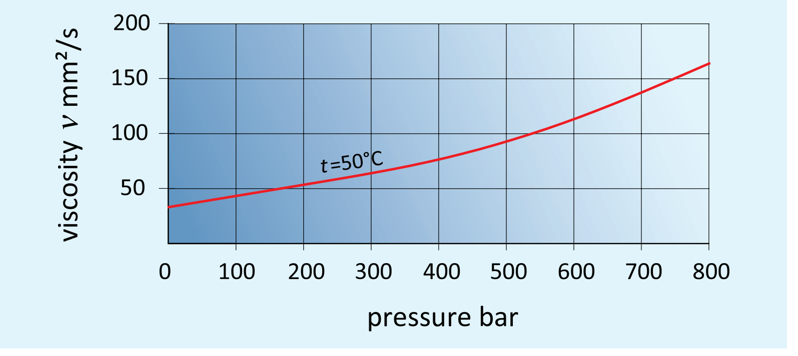


Fig 11.2.C Viscosity dependence on pressure

The viscosity of the oil is dependent on the pressure and the temperature (see also paragraph 3.3). That the pressure plays an important part can be seen from the graph.

Special attention needs to be paid to the possibility that air pockets can develop in the system, for example with pump sets that have been installed in a vertical position, in which case the axle, and thus the axle bearing, has been mounted at the highest point. The air will be able to escape easily if an extra oil canal is drilled into the bearing housing and the bearing will also be submerged in the lubricating liquid.

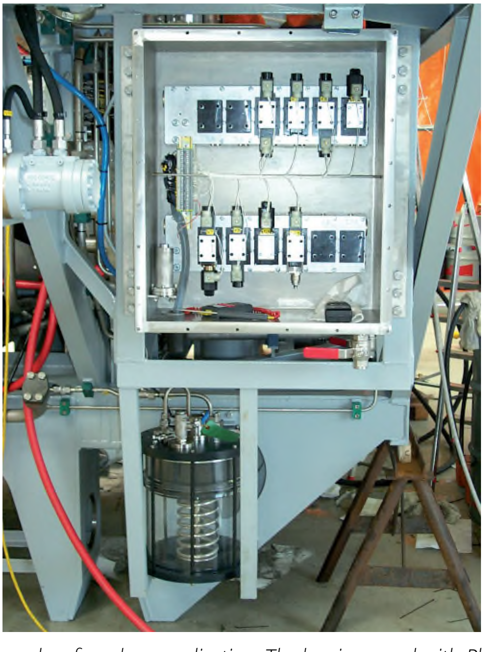


Fig 11.2.D Valve box with servovalves for subsea application. The box is covered with Plexiglas to enable easy inspection when the ROV is above water. Normally the box is internally filled with hydraulic fluid. (Courtesy of Seatools)

The hydraulic function valves will not be exposed to the water from the working environment but are instead placed in a box filled with oil. This box will then be subject to the pressure P_h which means that the same absolute pressure as for the pump will also apply to the valves. A standard industrial valve can also be applied at an increased absolute pressure. In these cases solenoids will need to be built as wet-pin solenoids. There is an exception for proportional valves. Many of these valves are fitted with a circuit board with 'on board' electronics for the feedback of the position of the main piston. Such electronics are often not suitable for submersion in hydraulic liquid, let alone work at high absolute pressures.

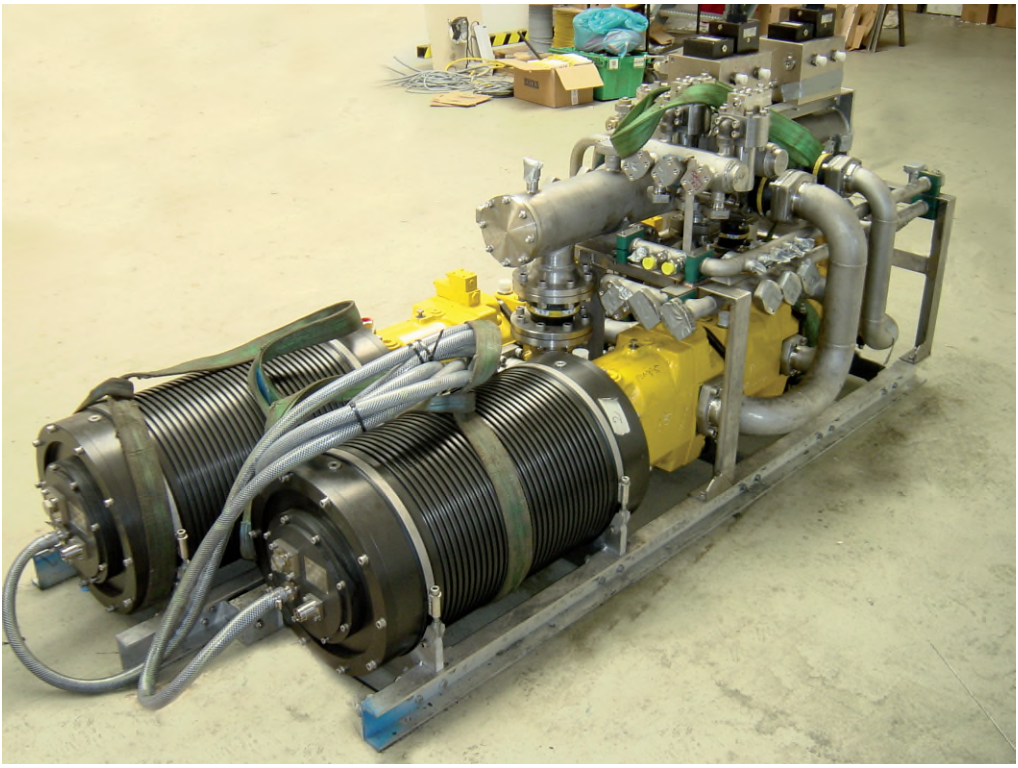


Fig 11.2.E Electromotor-pumpset for subsea application. The electromotor is designed for a voltage of 4 kV (Courtesy of Seatools)

The electric motor for a subsea HPU will also be submerged in the seawater. To make this possible, special AC induction motors have been developed where the stator housing is seawater resistant and where the necessary cooling is often achieved by using the hydraulic liquid from the hydraulic installation as the cooling medium. The stator housing has simply been fitted with an extra supply and exhaust port connection for the cooling liquid. In order to keep the current losses in the supply cable to a minimum, high voltage motors with a voltage of approximately 4 kV have been developed.

11.3 Subsea hammering

Driving piles into the ground/seabed can, for example, be necessary to establish a foundation for a platform or to create a solid anchor point for the anchor chains that are often used. The driving of the pile happens in the same way as on solid ground. The stroke energy is obtained from the impact of a large weight that is dropped on the pile from a great height, often supported by an extra pre-tension mechanism, in our case a hydraulic accumulator. The housing for the hammer is filled with compressed air to get around the water resistance of the pile hammer.

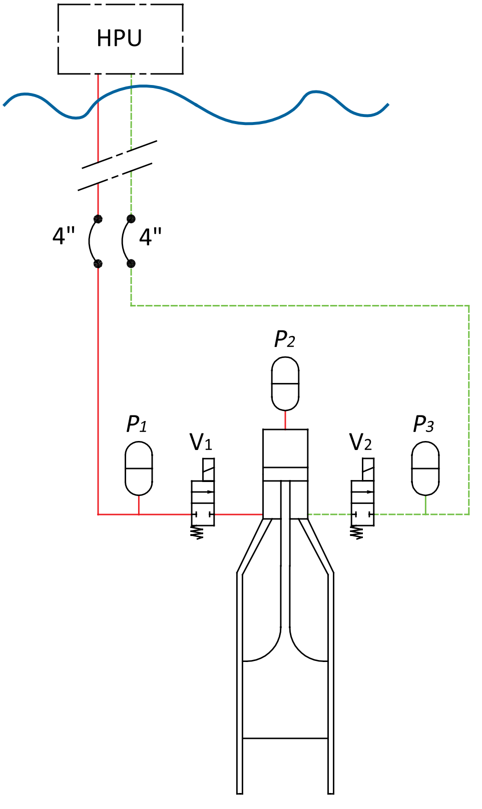


Fig 11.3.A Hydraulic diagram for a subsea hammer with HPU above water. Note the several hydraulic accumulators that are being used.

The HPU provides a continuous volume flow in the direction of the underwater hammer. The accumulator with the preset gas pressure P_1 operates as a buffer for the pressure peaks that can occur in the supply line as a result of the water hammer. After valve V_1 has been opened, the hydraulic cylinder is lifted up together with the hammer whilst the accumulator that is connected to the bottom side with a preset gas pressure P_2 is brought to a high hydraulic pressure at the same time. After valve V_1 has been closed and valve V_2 has been opened, the hydraulic cylinder will, as a result of gravity and the hydraulic pre-tension, get a high downwards speed. The accumulator with preset gas pressure P_3 is there to limit the peak pressure in the return pipe that results from the momentarily high volume flow from the hammer cylinder. All three accumulators have different preset gas pressures, each of which is also dependent on the depth of the water where the work is taking place.

The underwater hammers that are used in conjunction with an HPU above water can at the moment work with volume flows of up to 4.000 lpm and working pressures up to 300 bar which means that the total hydraulic power they require is approximately 2000 kW.

Fig 11.3.B *Heerema's largest hammer from Menck with a hydraulic power unit built onto and around the hammer. The electrical power and control of the hammer is run via an umbilical. (courtesy of Heerema)*

Because of the desire to drive piles at greater depth and because of the increasing weight of the pile hammer ever greater hydraulic power needs to be supplied and dissipated through the hoses. Apart from the hydraulic energy, ever greater quantities of compressed air are needed for the piling process. To get around the inconvenience of larger hydraulic and air hoses, HPU's that are attached to the hammer unit and are transferred to water depths of approximately 2.400 meter have been developed for this environment too. The electrical energy for the electric motors and the communication and control cables are connected to the hammer by umbilical cable.

11.4 Salvage of the Kursk submarine

On 12th August 2000 all 118 crew members on board perished when the Russian nuclear submarine sank in the Barents Sea. A joint venture of two Dutch companies, Mammoet and Smit International received the order to salvage the Kursk in May 2001. The first task was to saw the nose off because otherwise there would be a great risk of instability if the nose broke off during salvage operation.

Sawing ships into sections was not new for Smit International. Normally a ship would be ´sawn´ by dragging a heavy chain to and fro across the ship´s hull. It would be more realistic to talk about nibbling rather than sawing. The material of the Kursk´s hull was however much stronger than that of normal surface ships. Additionally there was no previous experience with a hull thickness of 50 mm. Smit developed a new type of saw in the run-up to the Kursk project. The saw is now made up by threading the sawing cable through a number of sawing bushes. These sawing bushes are made up of a steel core with a thick layer of hard solder with a large number of pieces of cemented hard carbide embedded within it. If the 'sawing thread' is moved across the hull of the ship at the right speed, of up to approximately 1 m/sec, and with the right force then the material is eaten away. A total pulling force on the pulling side of up to approximately 450 kN and a breaking force of 100 – 250 kN on the releasing side together with the sawing angle determine the transverse force with which the sawing bushes are pushed against the ships hull.

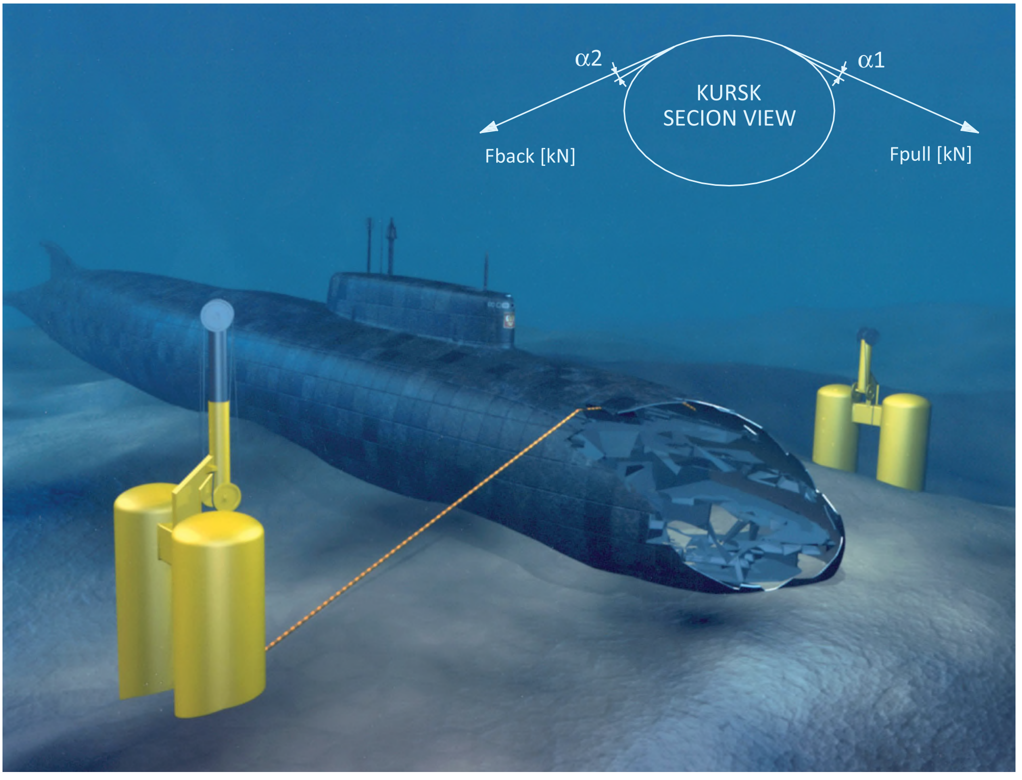


Fig 11.4.A and Fig 11.4.B Combined action of cable forces and saw angles and the way it was applied subsea (Courtesy of Smit)

Initially it was hoped that it would be possible to move the sawing cable to and fro with two winches on board ships. The expected swell of 2,5 meter in the Barents Sea and the relatively high elasticity of the winch cables made that type of approach impossible.
One so-called suction anchor was placed on each side of the Kursk to create two fixed points so that the sawing cable could be hauled over the nose of the submarine at a depth of 100 meters at the required

speed and with the necessary force. By sucking a vacuum underneath after the anchors had been put on the soft seabed, they would, so to speak, sink into the seabed of their own accord, just like a ship's anchor. One double working hydraulic cylinder was placed on each anchor. The sawing cable was run over a system of sheaves to create a quadruple fall. This created a maximum cylinder force of 1.800 kN at a piston speed of 0,25 m/s.

Time was of the essence for the Kursk salvage. It was necessary to use existing hydraulic apparatus. The cylinders with a bore of 420 mm were in fact spare cylinders from the top mechanism of mobile cranes owned by Mammoet. The preference was for diesel powered pump units. Units of this type with a capacity of 475 kW each and with pumps for a closed loop system were also in use to drive Mammoet's platform trailers. This meant that the basis of the drive system was in essence already there.

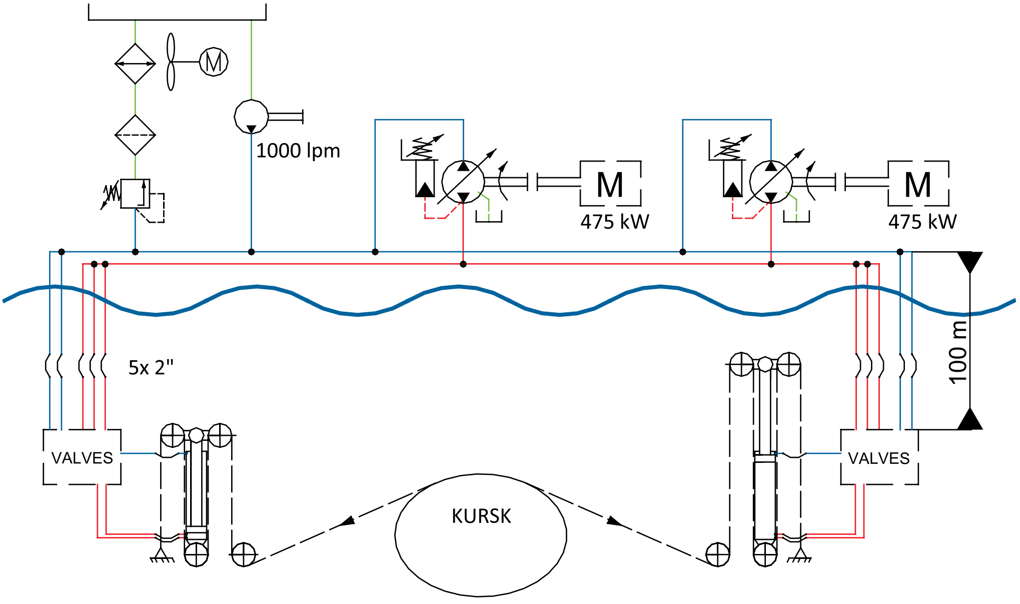


Fig 11.4.C Basic hydraulic diagram of the Kursk sawing method

This determined the choice of a closed loop system for the hydraulic drive mechanism. The pumps were built into a constant pressure control mechanism to get around the effects of the compressibility in the 120 meter long pressure hose. Because one cylinder is always moving out whilst the other cylinder is being pulled in by the saw it is possible to put the cylinder into a closed loop. The largest size hose available on the market was 2". The pressure drop in the hoses meant that 5 hoses had to be installed for each drive cylinder. This made the total length of hose for the two cylinders 1.200 meter!! The pressure drop in the supply and discharge hoses would still be 2 x 25 bar, a total power loss of 133 kW!

An extra diesel powered boost pump with pressure limiter, filter and cooling functions was installed to create a real closed loop system. Immediately, a large pump (1.000 lpm) was chosen so that the hoses on board could be flushed with the boost unit too. An ISO cleanlyness of 13/10 was achieved with a 10μ pressure filter.

This meant that the control of the sawing speed and the reversal of the sawing motion needed to take place under water in a hydraulic valve block.

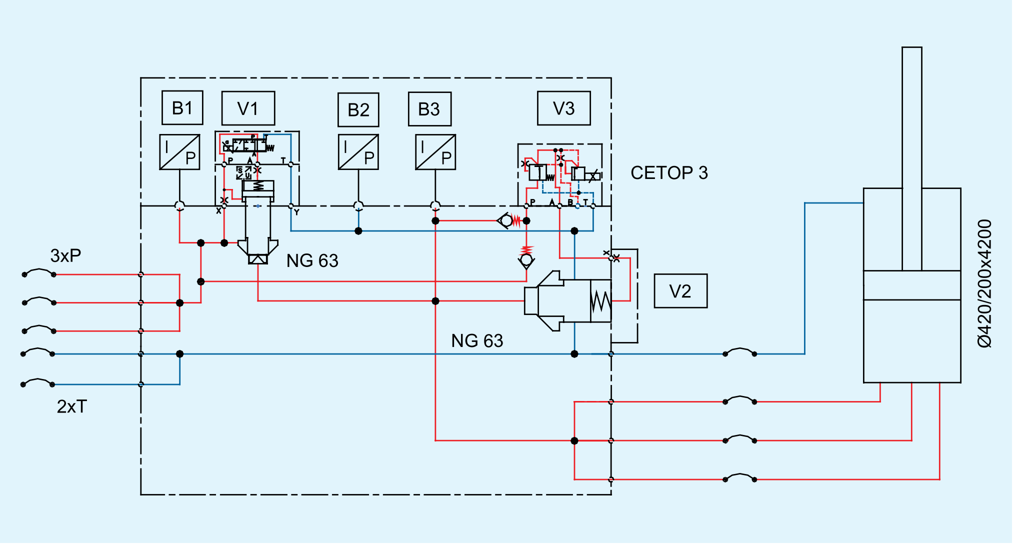


Fig 11.4.D Detailed hydraulic diagram of one of the subsea installed control valve manifolds

The piston side of both cylinders was always connected to the boost side of the closed circuit. The bottom side of the 'pulling' saw cylinder was connected to the high pressure side of the closed loop system via an electro-hydraulic proportional flow controller V_1 to create the outwards movement. This meant that the saw speed could be controlled proportionally. The bottom side of the 'releasing' cylinder was connected to the boost side of the circuit via the cartridge valve V_2. The back pressure could be set with the help of a proportional pressure control valve V_3.

It was necessary to keep the pressure drop across the 1.200 meters of pipe work to a minimum. Hence the choice of an oil that complied with ISO VG22 and had a minimum operating temperature of 30 °C. The expected temperature of the surrounding sea water was 10 °C. The calculations showed that the heat loss for the 1.200 meters of hoses with a total cooling surface area of 245 m² and an oil temperature of 30 °C would come to approximately 104 kW. Heat losses would also occur in the cylinders that had been placed under water. Given the necessary parameters these losses came to approximately 20 kW. The expected hydraulic losses, especially in the pumps varied between 140 and 175 kW. This meant that a cooling capacity of 50 kW would suffice for the oil cooler that used air as the cooling medium.

If the sea water temperature was too low, then it would not be possible to achieve the necessary operating temperature of 30 °C and the viscosity would become too high, resulting in a higher than expected pressure drop in the pipes. Hence the presence of an extra pump unit so that the oil could be warmed up if necessary.

The control block had to be designed, drilled and tested in 3,5 weeks. All electrical valves and transmitters have been designed into the top surface of the block. The mounting holes are clearly visible in the photo of figure 11.4.E. A pressure resistant and waterproof cover with waterproof cable guides was placed on top of this surface.

If 120 meter of hose is hanging overboard without any extra measures being taken, then there is a large chance that the sleeves of the hose will break free as a result of the high weight of the hose. For each 40 meters (the maximum commercially available length) of hose, steel cables are attached to the steel joint flanges for vertical stress relief. An umbilical cord made up of five parallel hoses with a total length of 120 meter for each sawing cylinder. For planning reasons, the flushing of these hoses as well as the pressure testing of the completely assembled pipe system has been carried out on board too.

It is clear that, in the end, Smit International managed to cut off the nose. Because the sawing cable broke the nose was in fact cut off three times. Each time the sawing had to restart at the top of the cut because the cut was clogged up with metal bits etc. It was a great success as far as the sawing technique was concerned. The cutting capacity of the bushes was extremely high. The water test installation in figure 11.4.F shows how the cuttings are literally flying off. To prevent the solder layer from melting, lots of water was used as a cooling medium.

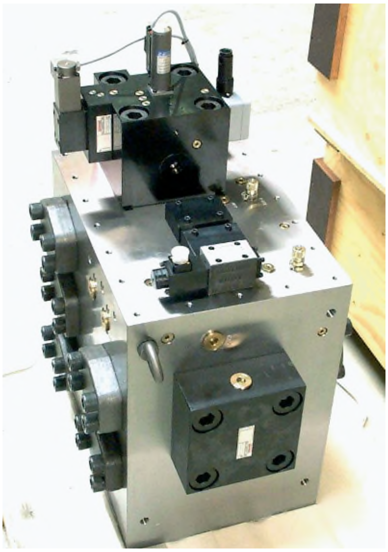


Fig 11.4.E The hydraulic control manifold as it was installed subsea, on the top a stainless steel box was placed to cover all electrical operated valves and instruments (Courtesy of Hydroplus)

Fig 11.4.F Test of the hydraulic drives saw mechanism in Kirkenesk Norway, water is required to cool the saw. Under water sufficient cooling would be available of course.

11.5 A hydraulic drive for a subsea grab

The White Rose offshore oil field is positioned some 350 kilometers east of St. John's, Newfoundland, Canada. The South White Rose oil pool covers approximately 40 square kilometers and contains an estimated 200-250 million barrels of recoverable oil.
The area is subject to frequent iceberg immigration, which forms a serious hazard for offshore installations and has to be taken into account in the design of offshore oil and gas facilities. To protect the subsea wellheads and manifolds from iceberg scouring they are lowered into the seabed in so-called glory holes. Bos Kalis from The Netherlands got the order to excavate the holes with a large grab deployed from a vessel.

Fig 11.5.A The hydraulic driven grab during sea trials in 2001 (Courtesy of Seatools)

To maintain the required position accuracy of ± 5 cm an Remote Operated Vehicle was placed on top of the grab structure. The mechanical design and controls of the ROV were designed by Seatools. The hydraulic drive for the ROV and grab cylinders was designed by Ingenieursbureau Albers bv.

The two main functions of the electro hydraulic system are to provide the linear movement, by means of two hydraulic cylinders, for the opening and closing of the grab as well as the rotating drive for the four independently operating thrusters. For a number of reasons a closed loop system was chosen for both systems:

- The volume flow can be controlled continuously through stepless variation of the pump output.
- A number of functional pressure control valves and non-return valves are built into the variable pump as standard. Valves that would otherwise have to be included in the liquid filled box.
- If boost pumps are added to the system it is possible to drive the cylinder with a closed loop system.
- Pumps for a closed loop system are very compact which means that the dimensions for the sub-sea power pack can be kept to a minimum.
- The closed loop system has a very high total efficiency which means that the heat losses are minimal because there are no volume control valves with their associated pressure drop.

Because of the need to drive the four thrusters independently and because both cylinders need to be operated independently of each other too, a total of six variable pump sets is required. Additional pumps are required for:

- Boost pressure. The variable pumps already have an internal boost pump. In order to compensate for the difference in volume of the two cylinders an extra two boost pumps are required.
- The housing of the electric motors is lubricated and cooled with a separate circuit, based on the same mineral oil, but totally independent of the main circuit. This requires two extra cooling pumps.
- A number of other support functions, like the ability to open the grab in case of a complete power failure, has been built into the complete systems If this wasn't the case, the grab could get fixed to the seabed. Two extra auxiliary pumps with hydraulic accumulator sets have been built in for this purpose.

All the pumps mentioned above can be delivered as standard as so called bolt-on versions. The maximum number of pumps that can be connected is dependent on the power of the built on pump and is often limited to three to four pumps per pump-set. For that reason it was decided to split the electro hydraulic power over two identical pump sets, each consisting of one electric motor with a through axle and a triple pump set at each axle end.

The propellers are each driven by a variable piston pump in a standard closed loop system. The pumps are provided with electronic feedback of their swash plates for a better accuracy of their flow control features and thus better control of the propeller thrush.
Figure 11.5.B shows the drive mechanism for one of the two halves of the grab. Because of the weight of each half of the grab, the cylinder needs to be kept in the open position with the help of a brake valve V_1. For the closing movement of the grab, the controller needs to open the brake valve through the pilot pressure of the 4/3 valve V_5. Because of the surface ratio of the cylinder a shortage of oil will develop on the rod side which will be supplemented from the booster system via non-return valve V_4. When the grab opens, a surplus will occur at the bottom side. This oil will be discharged to the booster circuit via the pilot operated non-return valve V_2.

The booster circuit consists of a total of four pumps (two of which cannot be seen in the diagram). The combined output of these pumps goes to the closed circuit of the cylinder via the special subsea filter Z_1. The booster pressure is controlled with the pilot operated pressure control valve V_3.

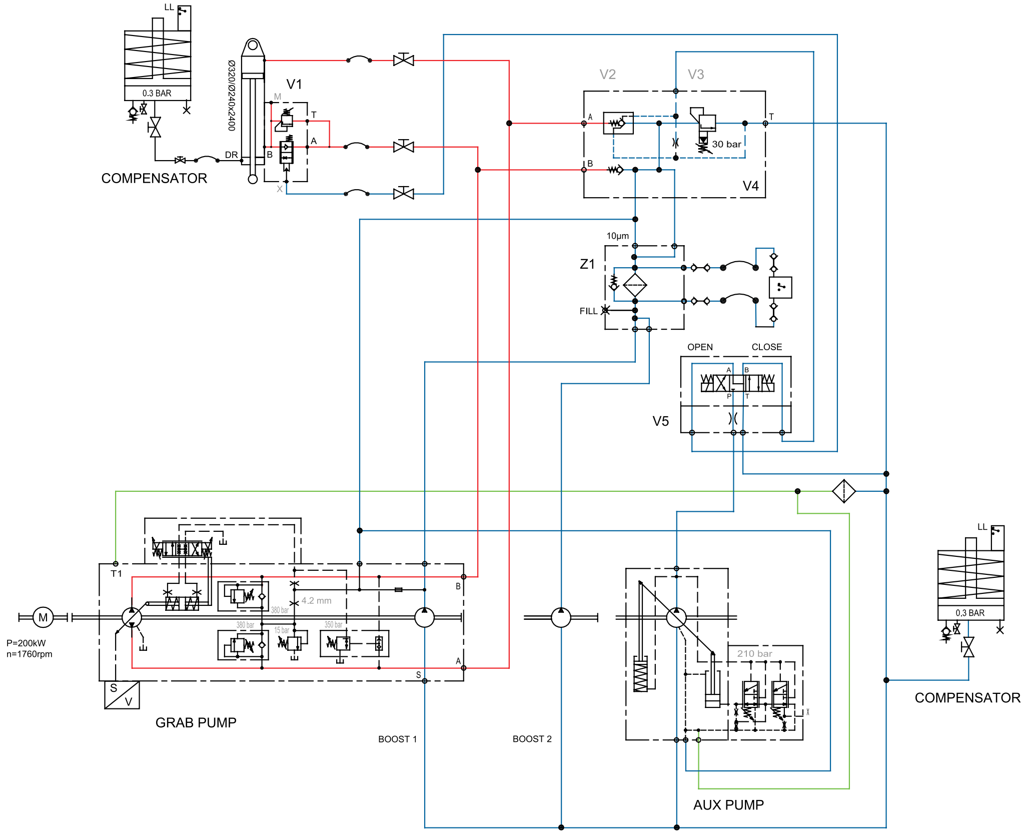


Fig 11.5.B Hydraulic diagram of the grab function

A compensator is connected to the suction pipes of the pumps. This compensator has sufficient capacity to compensate for the difference between the bottom and the annular side of the cylinder as well as for the temperature and pressure compensation of the oil. It is worth paying attention to the extra compensator that is connected to the seal chamber of the cylinder. The rod seal is fitted with an extra seal set which creates a chamber that could be contaminated with sea water.

Special attention needs to be paid to the cooling of the electric motors and the hydraulic equipment in subsea conditions. The convection of heat via the housing of, amongst other things, the electric motor, the pipe work and the cylinder housing for subsea systems is much higher than it is for systems that operate above water. This is because the heat transfer coefficient from metal to water is much higher. The total heat transfer coefficient from the oil to the water can be calculated if the material of the housing and the wall thickness are known. A total heat transfer coefficient of 71,5 kW/m^2.K has been calculated for this system.

By calculating the total heat transferring surface area of the system: pump and electric motor housing, pipe work, filter housing and cylinder surface it is possible to calculate the system temperature that will be required to be able to dissipate the heat for a given combination of the amount of heat generated and the seawater temperature.

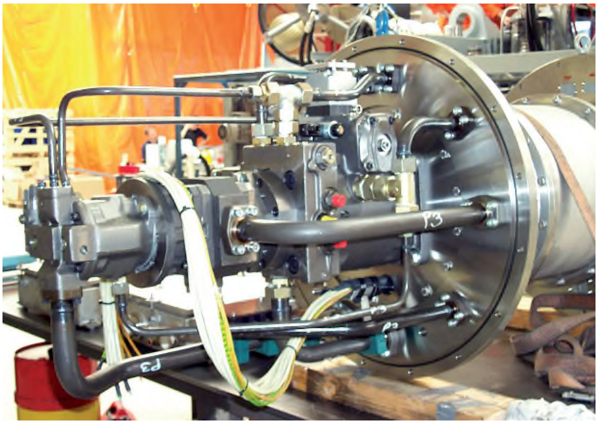
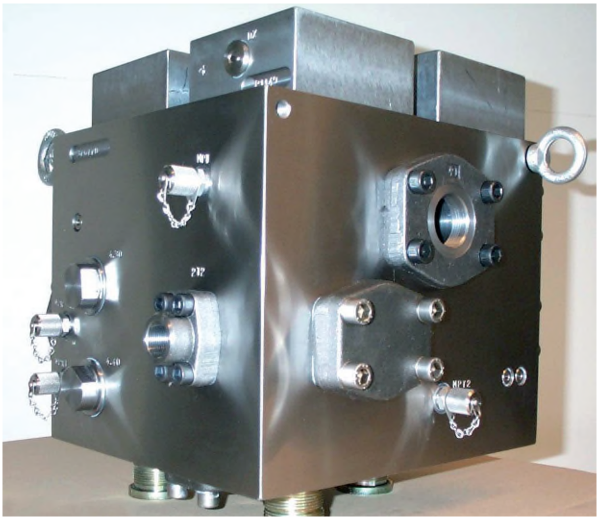
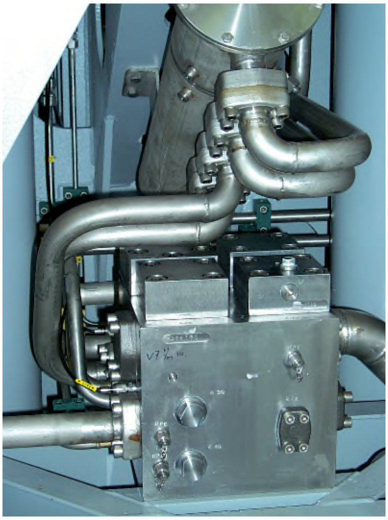


Fig 11.5.C Left above: Triple pump with closed loop pump, boost pump and auxiliary pump. Left below: Stainless steel manifold for the boost circuit. Right: Stainless steel boost filter, boost manifold and hydraulic piping (Courtesy of seatools)





Below are some of the temperature parameters of this system (for 1 electromotor pump set):

Heat loss of the electric motor (average value):	25,9	kW
Heat loss for all boost pumps (1 HPU):	16	kW
Heat loss for the main pumps (1 HPU):	16,6	kW
Heat loss coefficient:	71,5	$W/m^2.K$
Heat dissipating area for motor pot:	9,5	m^2
Heat dissipating area for hydraulic unit:	11,5	m^2
Necessary temperature difference for motorpot:	38	K
Necessary temperature difference for HPU:	40	K

11.6 Subsea clamp systems

The piles that are necessary for anchoring systems are driven into the seabed with the help of hammers. These piles will of course need to be put into the correct location by a crane first before the piling hammer can do its job. Specialist hydraulic tools have been developed to make the handling of the pile easier.

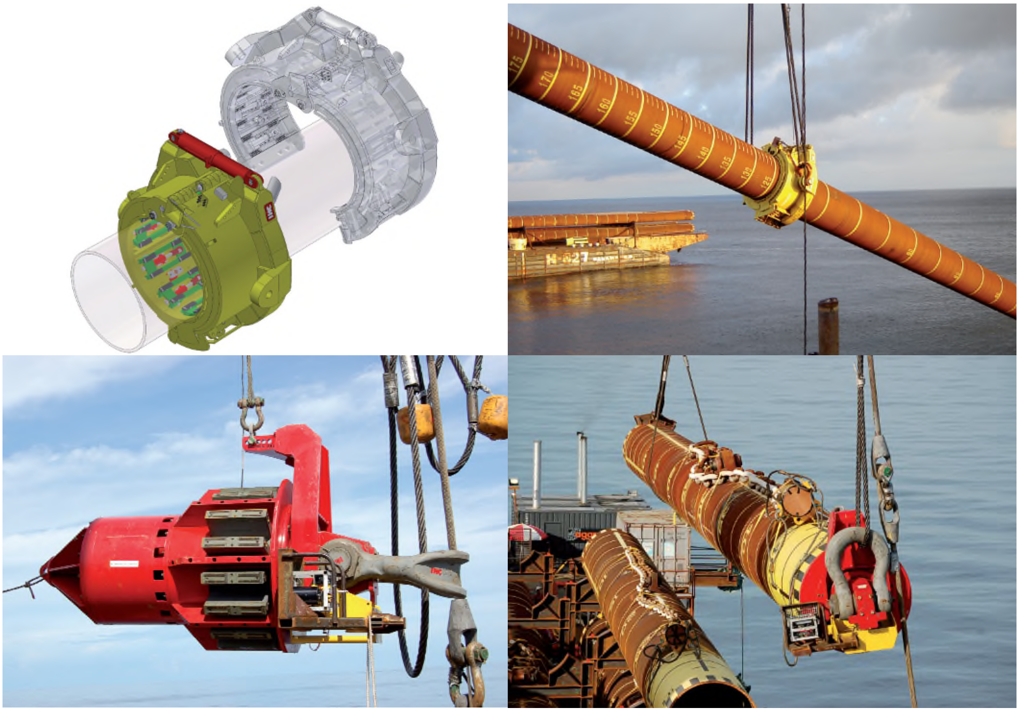
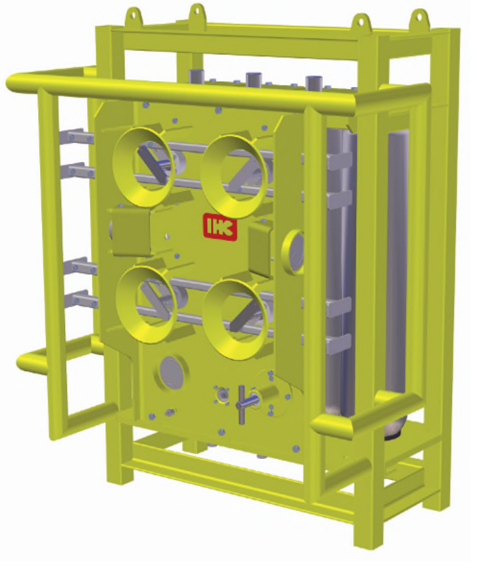


Fig 11.6.A Left above: Principle of an external lifting tool ELT, Right above: An ELT in use to handle the pile, Left below: Clamping can also be provided internally with the ILT, Right below: The ILT in use with an accupack for hydraulic power supply. (Courtesy of IHC)



The pipe is clamped by means of hydraulic pressure. The pressure is maintained by an accumulator pack that can be connected to the tool. Dependent on the necessary switching volume these packs can be made up of several parallel accumulators. The clamping function can be switched on and off by pilot pressure connected to the lifting tool via an umbilical or, especially in deeper waters, by controlling shut-off valves in the accumulator pack. These shut-off valves can be operated by a manipulator on an ROV.

Fig 11.6.B Computer 3D model of an accupack. The control of the valves is done by an ROV (Courtesy of IHC)

11.7 Subsea valve control

In the offshore oil and gas industry, there are many examples of shut-off valves that have been placed under water. Examples can be found in:

- Christmas Tree valves, an assembly of valves, spools, pressure gauges and chokes fitted to the wellhead of a completed well to control production. Christmas trees are available in a wide range of sizes and configurations, such as low- or high-pressure capacity and single- or multiple-completion capacity.
- Subsea manifolds, an arrangement of piping and/or valves designed to control, distribute and often monitor fluid flow. Manifolds are often configured for specific functions, such as a choke manifold used in well-control operations and a squeeze manifold used in squeeze-cementing work. In each case, the functional requirements of the operation have been addressed in the configuration of the manifold and the degree of control and instrumentation required.

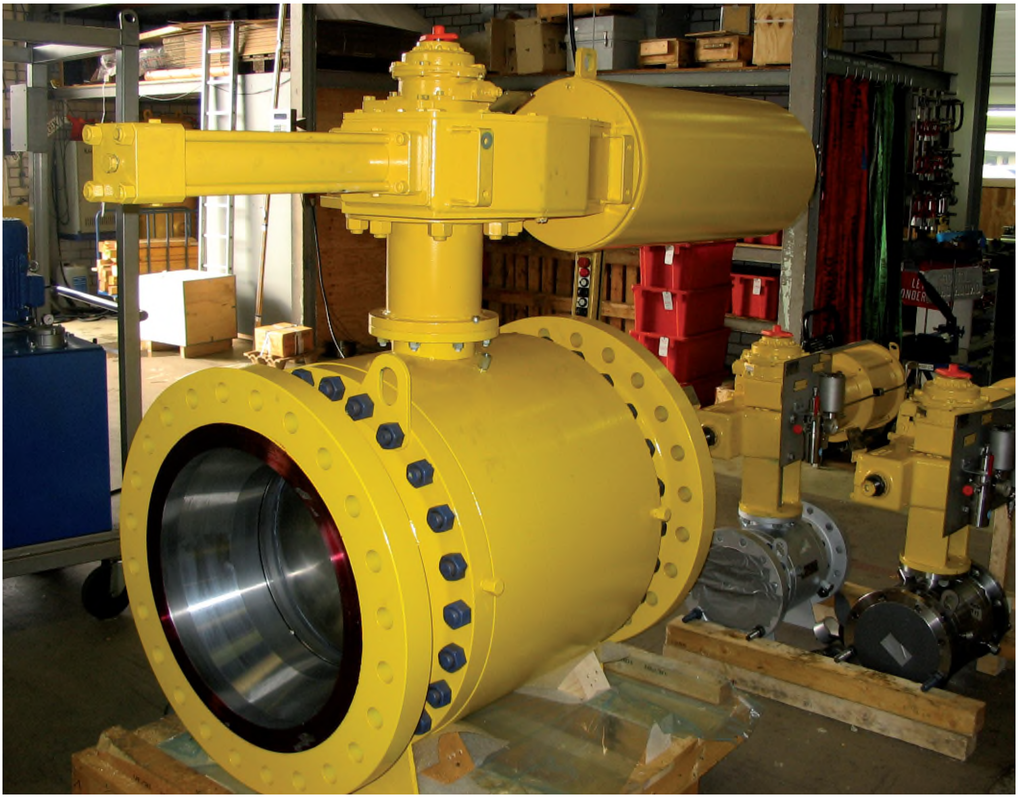


Fig 11.7.A *Subsea ball valve with hydraulic actuator at the front and spring actuation at the opposite end for the fail safe position (Courtesy of Frames)*

Shut-off valves can be controlled with handles for ROV operations, with electrical actuators or with hydraulic shut-off valves. The hydraulic energy is often stored in control stations where the operation of the shut-off valves takes place through control pipes in an umbilical.
In most cases the control stations are installed on a ship or a platform.

The number of hydraulic accumulators and back-up gas bottles are determined by the total volume and pressure that must be available when the valves are operated. The applications also show a great difference in the decompression time of the accumulators. Times can range from two seconds to several minutes. It is clear that an adiabatic process takes place in the case of the two seconds decompression time.

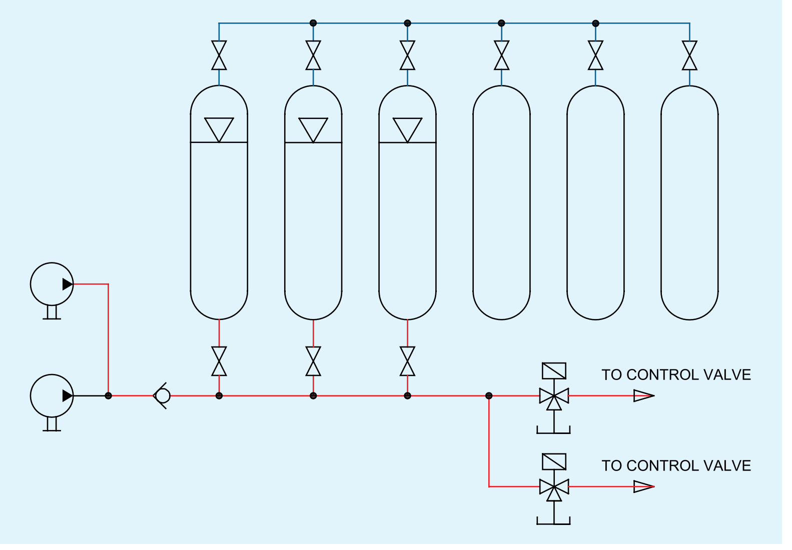


Fig 11.7.B *Hydraulic diagram for the subsea valves control station*

The dual pump arrangement allows a slow recharge of the hydraulic accumulators. The dual arrangement is chosen as a system backup. A pressure switch detects when the recharge pressure is reached and stops the pump(s). Although the recharging process, which lasts several minutes, is slow it cannot be calculated as being 100% isothermal. Due to the compression of the gas the gas temperature rises. After a while (one-two minutes) the gas temperature gradually drops and the hydraulic pressure decreases as a consequence. The pumps have to be re-started two or three times to reach the state where the pre-charge pressure has become stable.

Fig 11.7.C *A number of accumulators in parallel might be necessary to provide the storage of hydraulic power. (Courtesy of Frames)*

11.8 Subsea electrical drives

Recoverable oil and gas reserves are located at ever greater depths, now as deep as 3000 meters, and at ever increasing distances from each other. The well pressures during the exploitation process are constantly reducing too which makes it, in some cases, ever more difficult to extract the oils with higher API grades from the reservoir. Compressors and pumps have been developed that can be placed at these depths with their drive systems so that more product can be extracted from the reservoirs.

A different application is the positioning of a complete water separator systems where the water injection pumps need to be placed under water too.

All drive mechanisms for this type of process equipment consist of AC induction motors which are sometimes frequency controlled. The design of these drives is such that maintenance work only needs to be carried out after three to five years of continuous operation. The electrical feed for the electric motors takes place through umbilical's which can be as long as 10 km. Voltages up to 6.600 Volt are used to keep the voltage losses across the umbilical to a minimum. The power of the motor drives can now vary up to 6,5 MW.

Chapter 12

Safety design rules

Motion Control in Offshore and Dredging

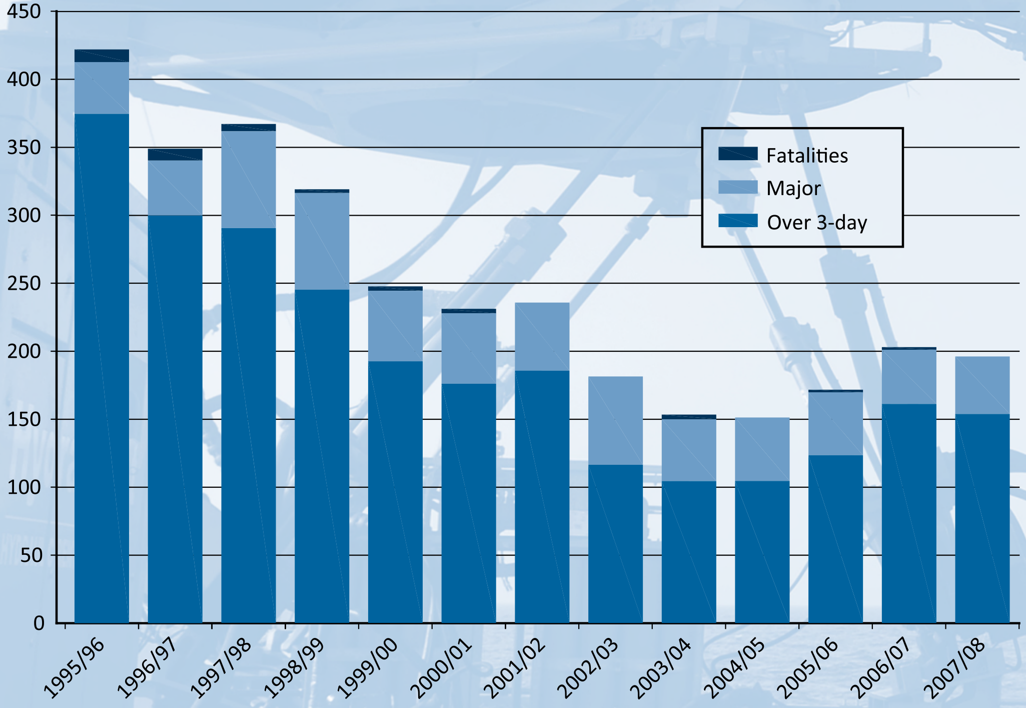


Fig 12.A *Injuries by severity, April 1995-March 2008, as reported to the official HSE (Health and Safety Executive, a UK Government Organization) as occurred in the Offshore industry on the UK shelf.*

Chapter 12

Safety design rules

A machine or system designer always has to take account of design rules. First of all there is the functional specification with which the design has to comply in all circumstances. Then there are, as a rule, several other types of design rules which are dependent on the demands from the client, the country of origin of the producer or the manufacturer. The rules can clearly be divided into the following categories:

- Legal requirements, for example in Europe the different directives that are applicable.
- For ships that will sail under a particular flag the rules from that national registration office may apply, for example ABS (American Bureau of Shipping) for all ships sailing under the American flag.
- Design rules and sometimes also the certification of the installation in accordance with the demands of certification authorities. If the technical demands of a certification authority apply will be determined separately for each project and client.
- International norms and standards
- Manufacturers' standards for commercially agreed technical requirements.

To design a system that can, in the end, be used safely is one of the core objectives for most designers. For a number of applications and/or clients, safety is seen as the number 1 priority. The table for offshore production units on the UK shelf covering the period from 1995 to 2008 shows that accidents which cause physical harm still happen.

During the whole design process a designer continuously has to ask her or himself if danger could occur if a part or a combination of parts in the installation were to fail. It is always necessary to look for solutions that either remove the chance of an accident or reduce it to a minimum if this is the case. This rather subjective assessment can work out very differently for different designers with differing technical knowledge, expertise and judgment dependent on cultural and social factors. This is why several, strictly procedural, assessment techniques have been developed that allow for all aspects of the design to be assessed.

P. Albers, *Motion Control in Offshore and Dredging*, DOI 10.1007/978-90-481-8803-1_12,

12.1 Legal requirements

The installation or drive units that are designed for use on ships or platforms are usually covered by the legislation of the country under whose flag the ship or platform operates. However, the following explanation of the differing legal requirements for countries that are members of the EU will show that these rules do not, in a number of cases, necessarily apply to ships and mobile platforms.
This means that the ship or offshore platform needs to comply with the demands of the national registering office dependent on the flag of the country under which it is registered.

No or only minimal technical specifications exist within the requirements of such national registration offices. The institutions that will need to carry out the certification process for the design with the help of the technical requirements will then, in the absence of their own technical norms, use the technical standards that are linked to the CE mark in Europe.

International organisations such as the IMO (International Maritime Organization) will also set rules regarding the safe design of offshore installations in particular for example for dynamic positioning and helicopter and rescue vessel facilities.

Irrespective of whether the requirements come from a national registration office, an umbrella organisation or for example the EU directives, they all share the basic assumption that a safe installation needs to be created.

Fig 12.1.A CE mark for equipment that complies with the applicable Directive in the EEC

The European guidelines have been created to guarantee so-called free trade traffic between the member states of the EU where one member state is not allowed to set individual technical requirements relating to the execution of the design. A second, not unimportant principle is that the producer or manufacturer is essentially responsible in case of an accident that causes severe economic damage or bodily harm. However, if the design complies with all the requirements of the relevant guideline(s), then the CEO, who is ultimately the responsible person in the company concerned can no longer be prosecuted in that case.

12.1.1 Machinery Directive

The machine guideline 2006/42/EG or Machinery Directive (MD) uses the following definition for a machine: that it consists of an assembly of components of which at least some can move and the movement of which is achieved by mechanical means. The second condition for a machine is that it can perform an independent function. It is clear that a hydraulic actuator will only perform a function once it has been incorporated into an installation and supplied with hydraulic energy fitted with control components. In the case of the supply of only a single actuator one talks about a machine component and the guideline doesn't formally apply.

A manufacturer will have to supply proof that the requirements of the relevant guideline(s) have been met. These requirements are summarised in the formal text of the guideline and are of a very general nature, sometimes rather subjective and open to interpretation. If the rules of good craftsmanship are applied then one has generally complied with the requirements. Real proof that the rules of the MD have been applied correctly can only be supplied if the manufacturer can also show that the design complies with the internationally recognised EN standards, which will often mention requirements at a detailed technical level.

A manufacturer is however never obliged to apply an EN standard. This is done because a new innovative technology could otherwise not be applied because a technical standard wouldn't exist for it. If the manufacturer can show in a different way that a safe design has been created then that would suffice too.

Be careful, for many orders clients will already refer to a number of technical standards in the specification. Compliance with these standards will then not be based on legal requirements but has been contractually agreed.

The MD applies to the operational use of the machinery, not only during maintenance, but also to manufacture, testing and commissioning of the machinery.

The MD does specifically not apply to:

- Pressure vessels (See the separate Pressure Equipment Directive, PED)
- Seagoing vessels and mobile offshore units together with equipment on board such vessels or units.

The MD therefore does not apply to many offshore and dredging applications. It remains possible however that the MD applies during the building, testing and commissioning of the equipment. We have already mentioned that a licensing authority can always test the equipment on the basis of the MD or other directives. This will then be agreed during the contract negotiations between the client and the manufacturer.

You will need to take note of the fact that a manufacturer can deliver part of a machine like a Hydraulic Power Unit (HPU) or the pipe work of a hydraulic installation. It is also possible that the complete machine is covered by the MD because it may be possible to use it on-shore. The producer of the completed installation (the assembly of the individual machine components) will only be able to issue a CE declaration if the parts/components have been designed and built in accordance with a European technical EN norm. In this situation too it concerns an agreed technical requirement rather than a legal requirement.

Many specific EN standards have now been issued specifically for the specialism of hydraulic and/or electrical drive technology. These standards can only be applied for the design of a drive system. Most of these norms are already applied by manufacturers of for example hydraulic valves, pumps, electric motors and control cabinets. To monitor the compliance of the technical details is a time consuming task for the designer of an installation of drive technology. One standard that should be one of the standard tools of the designer of hydraulic drive mechanisms should be EN-ISO 4413 (2008): "Hydraulic Fluid Power, General rules and safety requirements for systems and their components". This standard contains simple and easy to understand rules that apply to the design and manufacturing of hydraulic installations. Compliance with this norm is de facto a must, irrespective of whether a CE mark in accordance with the MD is applicable.

In most cases a drive mechanism that is destined for the offshore or dredging industries does not need to comply with the MD. The procedures, especially for obtaining a safe design, as indicated in the MD, are complied with by many equipment manufacturers. The reason is that they set conditions for the design in a constructive way, eventually resulting in the situation where the manufacturer achieves a certain quality and safety level that is based on an internationally recognised method. This is why the most important steps that need to be followed during the design process follow here:

1. During the design process risk analyses need to be carried out to check if it is possible that a situation can develop that may cause physical harm during the building, testing or running of the machinery. The precise way in which such a risk analysis can be carried out is described in detail in standards EN 14121-1, EN 12100 (2&3) and EN13849. A risk analysis will result in a number for each possible failure of a part or the whole of the machinery. The value of the number is determined by:

a. The nature or severity of the harm, slight; recoverable injury or serious injury.
b. The probability that the situation will occur during the lifetime of the machinery; rarely or often.
c. The possibility for the user to protect her/himself against the possible danger; from not at all to very well.

The worst score is given if an accident would cause a serious physical injury with a high chance that it will occur during the lifetime of the machine and against which the user cannot protect her/himself as compared to a situation where there is a slight chance that a minor injury will occur against which the user can protect her/himself.

After this the designer will have to change the design in such a way that the number representing the level of risk of each failure will be reduced to the lowest possible value or to the point where the risk disappears altogether. The same risk analysis will have to be carried out again after the review has taken place.

2. After the risks of an accident have been reduced as much as possible or have been removed altogether through the design of the machinery protective measures have to be put in place either on the machinery or for the operator.
3. The last procedural step is to warn the operator about any remaining risks of the machinery through means of pictograms and instruction panels on the machinery, not forgetting the ones in the user's manual.

12.2 Pressure Equipment Directive

The Pressure Equipment Directive (PED) and the MD were both developed to allow for free trade between member states of the EU or for trade from outside the EU into countries in the EU, which does not allow individual member states to impose additional technical demands.

As with the MD, the PED too does not officially apply to ships and mobile offshore units as well as apparatus for these ships and offshore units. The technical demands of the PED are nevertheless accepted and used by the large producers in the EU and by the certification authorities that are often involved in the approval of pressure vessels.

Technical conditions are set in the PED relating to the design, manufacture and application of apparatus that is considered to be high pressure apparatus according to that law.

The PED applies to any apparatus where the maximum permissible pressure is higher than 0,5 bar. Definition: the indicated pressure is the pressure relative to atmospheric pressure. For the purpose of this law, the apparatus is divided into different categories (I up to IV), see figure 12.2.1. The PED will indicate which design and manufacturing rules apply once the correct category has been established,

The PED does not apply to the following products:
- Apparatus for use on ships, rockets, aircraft and mobile offshore units.
- Storage vessels for liquids, including hydraulic reservoirs, where the gas pressure does not exceed 0,5 bar.
- Apparatus no higher than “Category I” of this law, as long as the manufacturer of the apparatus applies the MD.

Please note: Although the PED doesn’t officially apply to ships and mobile offshore units, its technical requirements are accepted by many certification authorities from these industries. This means that the PED is a good alternative if there is no specific certification for an installation.

In the following paragraphs we will discuss the division into categories in more detail. At this point it is important to point out that a large proportion of the hydraulic apparatus comes into “Category I”. We can conclude that this new law does not set any new requirements for this type of apparatus since it nearly always falls under the Machinery Directive. The following examples are part of category I:
- Motors
- Control valves, including proportional and servo valves
- Flow control valves
- Pressure control valves, with the exception of a number of pressure limiter valves (safety valves).
- Check valves

The hydraulic apparatus in the following list may fall in a higher category, but it is important to study the division into categories first to make sure that the apparatus does indeed form part of these categories:
- Large hydraulic cylinders
- Hydraulic accumulators
- Large pipe work to and from these apparatus
- Safety relief valves that limit the pressure in the apparatus.

So, the PED divides the apparatus into the different categories I to IV. A number of parameters is considered for this division: is the medium a gas or a liquid; what is the vapour pressure if it is a liquid and last but not least if the gas or the liquid is dangerous or not.

Is the medium a gas or a liquid

This question can be answered quite easily. A hydraulic system uses, as a rule, only liquid as medium to transfer power. The presence of a few percentage point of air in the liquid that isn't dissolved, doesn't change that. This means that we assume that the medium is a liquid.

In a hydraulic accumulator or a pressured reservoir (> 0,5 bar) we are considering a gas as well as a liquid. The separator for the mediums (piston or bellow) has no bearing on the Guideline for Pressurised Apparatus. In those cases the possible danger is determined by the medium that poses the greatest danger. Very often this is the gas that is present in the installation. The gas system of a heave compensator is certainly covered by the PED.

What is the vapour pressure if it is a liquid

Definition: The vapour pressure is the pressure at which the liquid turns into vapour.

The vapour pressure is dependent on the temperature of the liquid. The guideline stipulates that the vapour pressure must be assessed at the operating temperature. The PED uses 0,5 bar as the borderline for separation into the danger category.

The vapour pressure for fluids that are often used in hydraulics is given in the table below. Fewer technical requirements are stipulated for these products because the vapour pressure is < 0,5 bar for all the liquids in the table.

Vapour pressure in [bar] of hydraulic liquids (at 50 [°C])				
Mineral oil HLP	Oil in water HFA	Water Glycol HFC	Synthetic HFD	Water
$1{,}0 \times 10^{-8}$	0,1	0,1 – 0,15	$< 1 \times 10^{-5}$	0,12

Table 12.2.A Vapour pressures of fluids that can be applied in hydraulic systems

BE CAREFUL! There are HFA and HFC liquids that have a vapour pressure > 0,5 bar at an operating temperature of 80 °C. The applications will then come into a higher danger category immediately with the consequence of stricter demands for the apparatus. It can be of great importance to aim for a low operating temperature if this type of liquid is applied.

Is the gas or the liquid dangerous or not

If one of the following characteristics applies then a substance is considered dangerous:

Explosive, very slightly flammable, slightly flammable, flammable, very poisonous, poisonous, oxidizing.

Legal definitions have been established for all the categories. The manufacturer of the substances has to indicate if the substance concerned conforms to one of these categories. This feature will then be included in the so-called 'Product Safety Sheets' of the producer. The user can ask the producer for these Product Safety Sheets. If a product has one of the aforementioned characteristics then it has to be mentioned on the packaging by means of a label.

The usual hydraulic liquids, HLP, HFA, HFC, HFD and Water do not come under these classmarks given above. If in doubt, you will have to ask for the Product Safety Sheets. This will allow you to draw the following conclusion:

Hydraulic liquids are NOT considered dangerous.

Only nitrogen and in a few cases compressed air are used as gasses in hydraulic control technology. Although it seems ridicules, see paragraph 9.6 for the dangers of Nitrogen, both these gases also come under the category of not-dangerous substances.

12.2.1 Division into categories

The guideline uses tables for the allocation of an application to a particular category. Table 12.2.1. provides the detailed table number that needs to be applied based on the use of liquid or gas, on the operating pressure P and on the volume V or diameter D of the apparatus.

Applicable table for CATEGORY I to IV The dimensions to be used are: pressure P in bar, volume V in litre, diameter DN in mm.		
Application	**Conditions**	**TABLE no**
Accumulator	IF V > 1 AND IF PxV > 50 OR IF P > 1.000	TABLE 2
Pressure vessel: cylinder	IF P > 10 bar AND IF PxV > 10.000 OR IF P > 1.000	TABLE 4
Piping: Gas Only from/to gas system Or from/to accumulator	IF DN > 32 mm AND IF PxDN > 1.000	TABLE 7
Piping: Liquid Only from/to pressure vessel Only from/to accumulator	IF P > 10 AND IF DN > 200 OR IF PxDN > 5.000	TABLE 9

Table 12.2.1 Selection of the applicable table 2,4,7 or 9 based on the pressure and size of the apparatus.

For example:
A 200/140 x 1200 cylinder is suitable for a working pressure of 250 bar. The volume V of the cylinder is:

$$\frac{\pi}{4} \times 2^2 \times 12 = 37,7 \text{ litre} \qquad 12.1$$

The cylinder fulfils the condition that P > 10 but not the condition that PxV > 10.000. This means that the PED doesn't set any additional conditions for the design of the cylinder.

In such cases the PED will only set the following conditions for the product:

The product must be fit for purpose, have been designed and produced following the rules of good craftsmanship and have been supplied with a suitable instruction manual. It must also be possible for the name of the manufacturer or its representative identifiable. A comprehensive identification plaque will be sufficient. No CE marks can be put on the product.

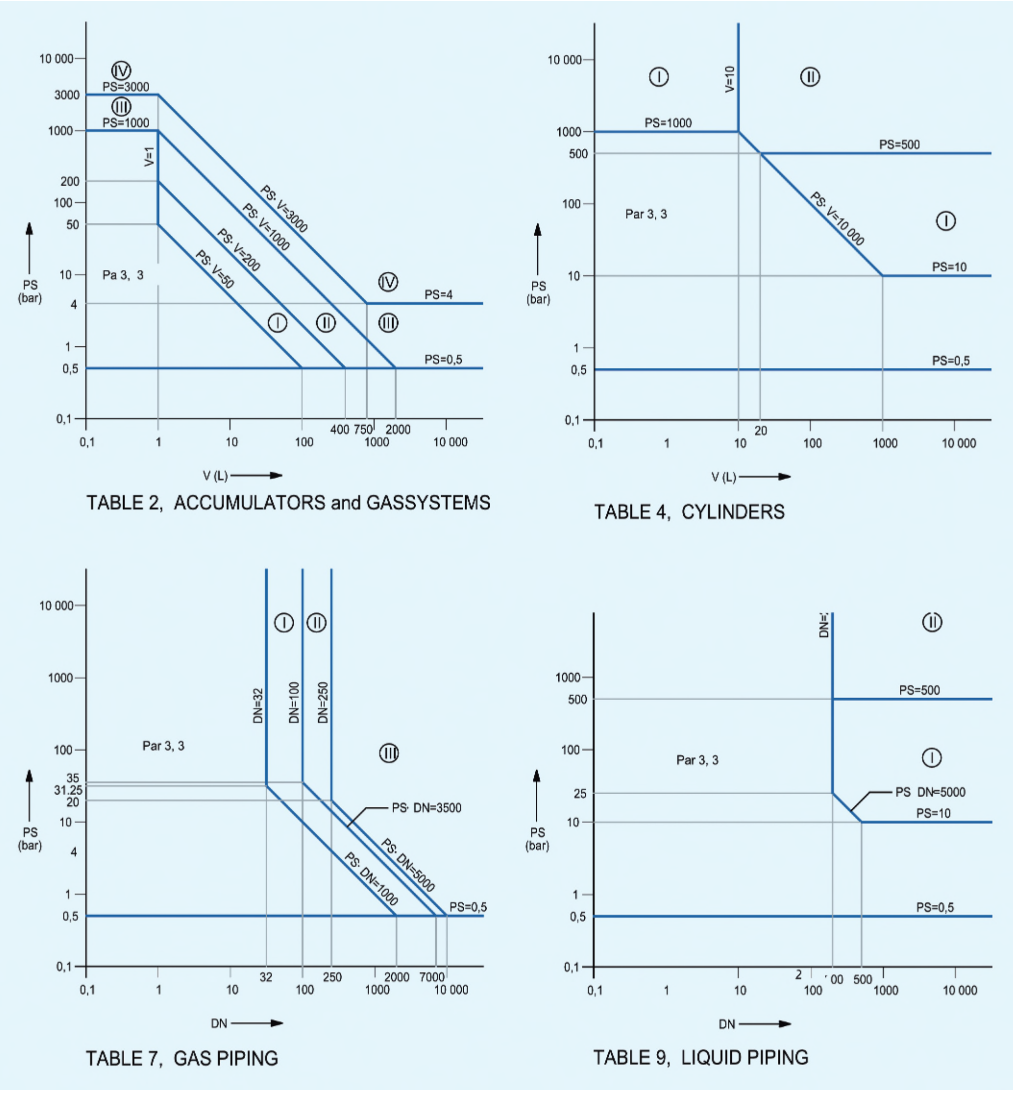


Fig 12.2.1 *The CATEGORY can be established by using the correct table whilst using the pressure, the volume and the possible diameter as the parameters. The category is given as a roman numeral I, II, III or IV.*

12.2.2 Requirements for the product

The tables mentioned above provide the division into categories. We have already mentioned that no special requirements exist if the apparatus is classified as ‘article 3, part 3’.

If the apparatus doesn’t fall into a category higher than I and if the apparatus is built into a machine or is part of a machine then the PED doesn’t stipulate any additional requirements.

Apparatus from category I that isn’t covered by the MD and apparatus from categories II, III and IV have to comply with the requirements of the PED. The PED indicates how a manufacturer has to prove that the design, the manufacture and final quality control have to take place to comply with the legal requirements. The level of the category is an indication of the level of risk of the application. As the product gets into higher categories a so-called ‘Notifying Body’ (recognised certification authority) needs to be involved in the processes covering design, manufacture and testing. We refer you to the guidelines for further details.

Pressure relief valves that are used to limit the pressure in pressure vessels from apparatus in categories II, III or IV need to comply with the same requirements from the PED.

The manufacturer or the authorised representative for the product is obliged to comply with the technical requirements from the guidelines. They can only bring the product onto the EU market if they apply a CE-hallmark to the product and if they issue a CE-declaration with it. This CE-declaration needs to comply with appendix VII of the PED.

12.3 Atmospheres explosive directive

The Atmospheres explosive directive (ATEX) guideline applies to all apparatus and protection systems that are meant for use in places where the risk of explosion exists. The safety, monitoring and control provisions that have been installed outside the area where the risk of explosion exists but which are there to make sure that apparatus and protection systems function properly are also covered by the guideline.

The ATEX itself only provides general definitions and basic conditions. As with the other guidelines there is no legal obligation to comply with the underlying technical EN standards but conversely, if apparatus comply with the relevant EN standard then it conforms automatically with the guideline as well. All manufacturers of the apparatus or producers of systems follow the detailed technical regulations from the many standards in this area faithfully.

Again, as with the MD and the PED, the ATEX doesn't apply to "sea going vessels and mobile offshore units together with equipment on board such vessels or units". It is strange that this exception is made since the danger of explosion can certainly exist on board such ships or offshore units. The reason for this is that differing technical requirements for apparatus are set across the world, like the NEC (National Electric Code) requirements in the USA. Most industrialised countries have, since 2006, declared that they will accept the technical requirements and approvals of each others' guidelines. This means that a designer can, in practice, use apparatus with certificates from one of the internationally recognised certification authorities.

12.3.1 Division into groups and categories

Apparatus and systems for use in an explosion risk area are divided into two different groups:

Group I: Meant for use in underground mines

Group II: Meant for uses in all other environments

The subsequent division into a number of categories is based on the degree of protection offered by the apparatus.

Category 1: Equipment that provides a high level of protection. Equipment from this category can be used in a situation where there is either a continuous danger or a danger over long periods of time or very frequent danger present that an explosion may occur. This means that there is a dangerous mixture of air and gas, vapour or mist present or that there is a similar air/dust mixture.

Category 2: Equipment that provides a high level of protection and that can be used in situations where the mixtures mentioned above "are likely to occur occasionally".

Category 3: Equipment that offers a normal degree of protection and that can be used in situations where the mixtures mentioned above "are unlikely to occur or, if they do occur, are likely to do so only infrequently and for a short period of time".

These are clearly very generalised requirements. The underlying technical standards contain many details and in particular methods that are available to the producers of the apparatus so that the desired degree of protection can be achieved. In paragraph 12.3.3 we will go into more detail about the technically available methods of protection.

12.3.2 Division into zones and division into gas groups

It is necessary to establish first what the chance is that an explosive mixture is present under normal process conditions before it will be possible to establish what category of equipment will be required. This is the so-called division into 'Zones' (in the US these are called divisions) and it needs to be carried out by experts who will indicate, with the help of a 3D model of the ship or the offshore installation, what the chance is that an explosive mixture is present. This requires knowledge of the process and the way in which the installations that will be used work.

Zone 0: An explosive mixture is present during more than 1.000 hr/yr

Zone 1: An explosive mixture is present during more than 10 but less than 1.000 hr/yr

Zone 2: An explosive mixture is present during less than 10 hr/yr but still enough to necessitate the containment of the source of ignition.

In the US the National Electric code is in force. The NEC500 divides a hazardous area into Divisions, with Division I approximating Zone 0 and Zone 1 together. Division 2 is an approximate match for Zone 2.

It will be clear that apparatus from category 1 is linked to applications in a zone 0 area. Category 2 apparatus is allowed in a zone 1 area and category 3 apparatus in a zone 2 area. If there is no chance that a dangerous mixture will occur on part of a ship or offshore unit then that area will be designated as a safe area.

The process experts will, apart from the zones allocation, also need to indicate what type of gas or liquid mixture may escape. An additional specification is established with which the apparatus needs to comply too. A further distinction is made in the table for the division of the groups of gasses, similar to the one mentioned in the ATEX guideline, which is in accordance with the IEC/CENELEC standards and which also shows the matches with the norms that follow NEC500, which are used in the US.

Indication of the classification of several gasses							
IEC/CENELEC	NEC500	T1-450°C	T2-300°C	T3-200°C	T4-135°C	T5-100°C	T6-85°C
I	Mining	Methane					
IIA	Class I / Group D	Ethane Benzene Toluene Methane Propane	Ethanol n-Butane	Benzene Diesel Heating Oils Hydraulic fluid	Acetaldehyde		
IIB	Class I / Group C	Coal gas	Ethylene				
(Group IIB + Hydrogen) IIC	Class I / Group B Class I / Group A	Hydrogen	Acethylene				Carbon disulphide

Table 12.3.2 *Indication of the classification of several gases. The temperature range T1-T6 is the safe temperature below which it is not possible to ignite a mixture of oxygen and the gas.*

12.3.3 Methods of protection

Apparatus that needs to provide protection against ignition can be constructed in several different ways.

The best known methods of protection are:

Ex d
This method keeps an ignited air-gas mixture inside the housing. Hot gasses are led to the outside through long narrow gaps in such a way that they are cooled to below the point where a new ignition takes place outside the housing. This method places specific demands on the construction of the housing, especially on the resistance to internal explosions, dimensioning of the gaps, but also on the use. For instance the equipments may not be opened while energized. The advantage is that there are no particular requirements for the electronics in the housing, only for wiring that is led to the exterior. As a result the power supply will often need to be Ex d or Ex e as well, while signal inputs will be Ex i (see below).

Ex e
This method prevents sparking in passive equipment. In these circumstances this could be caused by disconnecting or short-circuiting wiring. This method attempts to prevent that from happening by placing specific requirements on terminals and connectors. As a result the method is mostly suitable for junction boxes and the like.

Ex i
This method is called intrinsic safety. Here the stored energy in the apparatus is limited to below the ignition energy of the air-gas mixture. The surface temperature is kept below the auto-ignition temperature of the air-gas mixture by limiting the supplied power.

Deviation from the methods in the standards is often necessary. The manufacturer then needs to prove that the alternative measures taken guarantee at least the same level of safety as when the standards would have been followed. This often requires testing by the manufacturer and/or the notified body.

Ex n
A type of protection applied to electrical apparatus so that, in normal operation, it is not capable of igniting a surrounding atmosphere and a fault capable of causing ignition is not likely to occur. This type of protection is allowed in Zone 2 and is often used for electric motors.

12.4 Classifications bureaus

There are a large number of registration bureaus in the world that register ships under different national flags. Amongst other things these bureaus set rules for the design and build of complete ships and off-shore units.

The most important classification bureaus are:

ABS, American Bureau of Shipping,	http://www.eagle.org/
BV, Bureau Veritas,	http://www.veristar.com/
CCS, China Classification Society,	http://www.ccs.org.cn/en/
DNV. Det Norske Veritas,	http://www.dnv.com/
GL, Germanischer Lloyd,	http://www.gl-group.com/en/
KR, Korean Register of Shipping,	http://www.krs.co.kr/eng/
LR, Lloyds register of Shipping,	http://www.lr.org/
NK, Nippon Kaiji Kyokai Japan,	http://www.classnk.or.jp/hp/en/
RINA, Registro Italiano Navale,	http://www.rina.org/
RS, Russian Maritime Register of Shipping,	http://www.rs-head.spb.ru/en/

This does not mean that all deck equipment or drive systems on board have to be designed and built under the registration rules. After consultation the user of the ship or offshore unit will decide for each project if certain installations need to comply with the requirements of the classification bureaus. To make it possible that certification for an installation can be carried out in the future it can be agreed that the installation will be designed in accordance with the technical requirements of a bureau whilst no actual inspection or delivery under these rules will take place.

Different types of classification/inspection exist:

In general an inspection consists of a combination of design approval, manufacturing survey and testing of the final equipment.

Design approval: The aim of the design approval process is to confirm that the design of the object to be classified complies with the class rules. This is to ensure that the final object satisfies the society's requirements and can be entered into a class upon completion of manufacture, testing and commissioning. Design approval includes initial review, issue of comments and review of next document revision to confirm the incorporation of the bureau's comments. Design approval generates approved drawings with approval letters containing approval comments.

Design approval may be carried out for equipment such as:
- Heavy lift cranes
- All necessary equipment for a complete process, like pipe-lay, drilling, dredging, turret handling.

Type approval: This is an approval to have the design of standard produced (off the shelf) items or products approved by a bureau. It is offered as an alternative to case-by-case approval for the whole equipment or parts of it for unlimited and unspecified manufacturing orders within an agreed time window.

Type approval may be carried out for equipment such as:
- Cranes and other lifting appliances.
- Life- and rescue boat launching appliances.
- Anchoring equipment.
- Winches.
- Winch components such as gears and brakes.
- Loose Gear.
- Personnel lifts/Elevators.
- Other specialised equipment for retrieving, hoisting and lowering.

Material certificates:

The material for a construction item has to comply with the specification that has been issued by the manufacturer of that material. Dependent on the application for the material, different requirements are set for the proof (information) that needs to be provided to prove, for example, what the chemical composition of an alloy is or what the notch impact strength, the elasticity limit or the breaking strength of a material is. The method of convincing proof is laid down in different forms of material certificates (certification documents).

In 2005 a new version of standard EN 10204 appeared.

Type of certification document according to EN 10204	Who declares that the material conforms to the specification	On what test results has the certification document been based?
Factory declaration type 2.1	Manufacturer	The results of the inspection are not declared
Factory inspection certificate type 2.2	Manufacturer	See (*)
Factory report 3.1	Hierarchically independent representative of the manufacturer	See (**)
Test report 3.2	Hierarchically independent representative of the manufacturer and authorised representative of certifying agency and appointed by official regulation	See (**)

(*) The inspection has been carried out in accordance with the manufacturer's procedures and the tested product is not necessarily the delivered product
(**) The inspection has been carried out in accordance with the material specification and carried out on the products that are to be delivered or on a test sample that is part of these products.

Table 12.4 Material certification and the required inspections according to the new EN 10204

12.5 Risk analysis

During each design process the designer must at all times take account of the possible dangers that may occur as a result of the failure of all or part of the design, as a result of the use of the machinery or installation or as a result of external circumstances (for example wind and waves).

In paragraph 12.1.1 we mentioned the risk analysis that forms part of the design process under the Machinery Directive. These risk analyses may be valid for installations on seagoing ships and offshore units but will often not suffice. Quite often supplementary risk assessments are requested.

12.5.1 Hazard Identification Study

A Hazard Identification Study or HAZID is a tool for hazard analysis, used early in a project as soon as process flow diagrams, draft heat and mass balances, and plot layouts are available. Existing site infrastructure, weather, and geotechnical data are also required since these are a source of external hazards.

The method is a design-enabling tool, aiming to help organize the HSE (Health Safety Environmental) deliverables in a project. The structured brainstorming technique typically involves designer and client personnel, engineering disciplines, project management, commissioning and operations.

The main major findings and hazard ratings help to deliver HSE compliance, and form part of the Project Risk Register (PRR) required by many licensing authorities.

The HAZID is often carried out at the earliest possible stage of a project by a team of experts, often from the client and the supplier. The HAZID is a more general assessment of the possible dangers when compared with the FMEA that will be mentioned later.

HAZID WORKSHEET						
W=weather, P=people, V=vessel, T= technical						
Hazard Number	Checklist - Guide Word	Potential Hazards & Effects	Threats	Controls (are they in place?)	Mitigation Required	Priority 1, 2 or 3
1	W	Extreme weather operating parameters specification - does it exist?	Wind, Waves	Master's skill & experience. Accurate wind & wave measurement	Adverse weather policy covers most situations so far Weather forecasting is in place Limits on cylinder length reduce effects	3
2	P	Transported workers - Lack of familiarity - motion sickness	Panic, Fatigue	Training & Experience	Confidence acquisition	1
3	T	Power failure of HPU	Mistakes, maintenance, technical failure	Fluid level in diesel drives etc	Backup HPU's including accumulator for 2 minutes back-up	3

Table 12.5.1 Example of a HAZID procedure carried out on the platform system discussed in paragraph 8.3.

12.5.2 Failure modes analysis

The Failure Modes, Effects Analysis (FMEA) and the Failure Modes, Effects and Criticality Analysis (FMECA) procedures are tools that have been adapted in many different ways for many different purposes. They can contribute to improved designs for products and processes, resulting in higher reliability, better quality, increased safety, enhanced customer satisfaction and reduced costs. An FMEA or FMECA is often required to comply with safety and quality requirements, such as ISO 9001, QS 9000, ISO/TS 16949.

FMEA and FMECA are methods that are primarily adapted to the study of material and equipment failures and that can be applied to categories of systems based on different technologies (electrical, mechanical, hydraulic, etc) and combinations of such technologies or it may be specific to particular pieces of equipment or systems or to projects as a whole.

The C of criticality is added to quantify the relative magnitude of each failure effect as an aid to decision making, so that with a combination of criticality and severity, priority for action to mitigate or minimize effect of certain failures may be set.

One of the methods used for the quantitative analysis is the Risk Priority Number (RPN), a subjective measure of severity of the effect and estimate of the expected probability of its occurrence. See NEN-EN-IEC 60812

RPN = S x O x D

Where: S = severity of the failure (injury or economical effect)
O = probability of occurrence
D = likelihood of detection

The scale for these factors is often expressed in numbers between 1 and 10 and originates in the car industry in the USA. Other scales, like 1 to 5 are also used.

Examples for division into scales for the three factors S, O and D are given here:

Rank	Severity of the failure	Meaning (generic marine and offshore/system interpretation)
1	Negligible	At most a single minor injury or unscheduled maintenance required (service and operations can continue)
2,3	Marginal	Possible single or multiple minor injuries and/or minor system damage. Operations interrupted slightly, and restored to its normal operational mode within a short period of time (say less than 2 hours)
4,5,6	Moderate	Possible multiple minor injuries or a single severe injury, moderate system damage. Operations and production interrupted marginally, and restored to its normal operational mode within, say no more than 4 hours.
7,8	Critical	Possible single death, probable multiple severe injuries or major system damage. Operations stopped, platform closed, shuttle tanker's failure to function. High degree of operational interruption due to the nature of the failure such as an inoperable pipe-lay system (e.g. tensioner fails to start, power system failure, mooring system failure) or an inoperable convenience subsystem (e.e. DP)
9,10	Catastrophic	Possible multiple deaths, probable single death or total system loss. Very high severity ranking when a potential failure mode (pipe clamps of j-lay system, fire and explosion) affects safe operation and/or involves non-compliance with governmental regulations

Table 12.5.2.A Example of the scale for Severity of the failure

Rank	Probability of occurrence	Meaning
1	Highly unlikely	Extremely unlikely to exist on the system or during operations
2,3	Unlikely	Unlikely but possible given that the failure rate happens
4	Reasonably unlikely	Likely to exist on rare occasions on the system or during operations
5	Likely	It is likely that consequences happen given that the failure event occurs (a program is not likely to detect a potential design or an operational procedural weakness)
6,7	Reasonably likely	Exists from time to time on the system or during operations, possibly caused by a potential design or operational weakness
8	Highly likely	Often exists somewhere on the system or during operations due to a highly likely potential hazardous situation or a design/procedural drawback
9,10	Definite	Likely to exist repeatedly during operations due to an anticipated potential design and an operational procedural drawback

Table 12.5.2.B Example of the scale for the Probability of the occurrence

Rank	Detection
1	Certain
2	Very high
3	High
4	Moderately high
5	Moderate
6	Low
7	Very low
8	Remote
9	Very remote
10	Absolutely uncertain

Table 12.5.2.C Example of the scale for the possibility of detection

Take note that, in this case, a high chance of detection gives a low score. Imagine the failure of a pressure control valve which is measured with a 3-way pressure sensor as well as a 2 out of 3 alarm system and that a backup pressure control valve has been fitted. A failing pressure control valve is certainly detected and the function of the failing valve will be taken over by the backup valve.

To get the complete picture of the failure rate of the complete design a RPN has to be determined for each individual component. After that a list of priorities is made on the basis of the RPN scores to indicate changes in the design, maintenance or the method of detection that need to take place. The priority is also determined by the severity of the failure. A high severity score gets a low RPN value and falls outside the list of priorities because the probability that it will happen is low or because the chance that it will be detected is low.

After the priorities have been established and the necessary measures have been taken, the PRN will need to be established again to show what effect the measures taken have had.

An FMECA is a very time consuming process for a large group of designers and users. It does however provide a structured insight into the cohesion of the possible measures taken in a technical design, the maintenance schedule or the operational use.

12.6 Examples of real life failures

It is possible to design hydraulic drives as safe drives provided that the conditions and specifications of the components that will be built into the drive system have been adhered to during the design and the build. In the following paragraphs we will discuss two examples of mistakes that were made during the design or build process of a hydraulic drive unit.

12.6.1 Free fall of a winch motor (1)

This case describes the application of a winch drive. See figure 12.6.1. The hydraulic drive consists of a hydraulic motor with hydraulically operated band brake. When the winch is not operational, the band brake is applied. The band brake can be disengaged (lifted) by using the mechanically operated 3/2 valve to apply the inlet pressure P to the hydraulic actuator of the band brake. This mechanically operated 3/2 valve is linked to the manual operated winch control valve.

The winch control valve is a proportional valve. This means that the flow to the hydraulic motor (PAY-In or PAY-Out) is proportional to the manually operated lever of the main spool of this valve.

When the winch control valve is operated, the mechanically operated 3/2 valve is automatically applied and the band brake will be released.

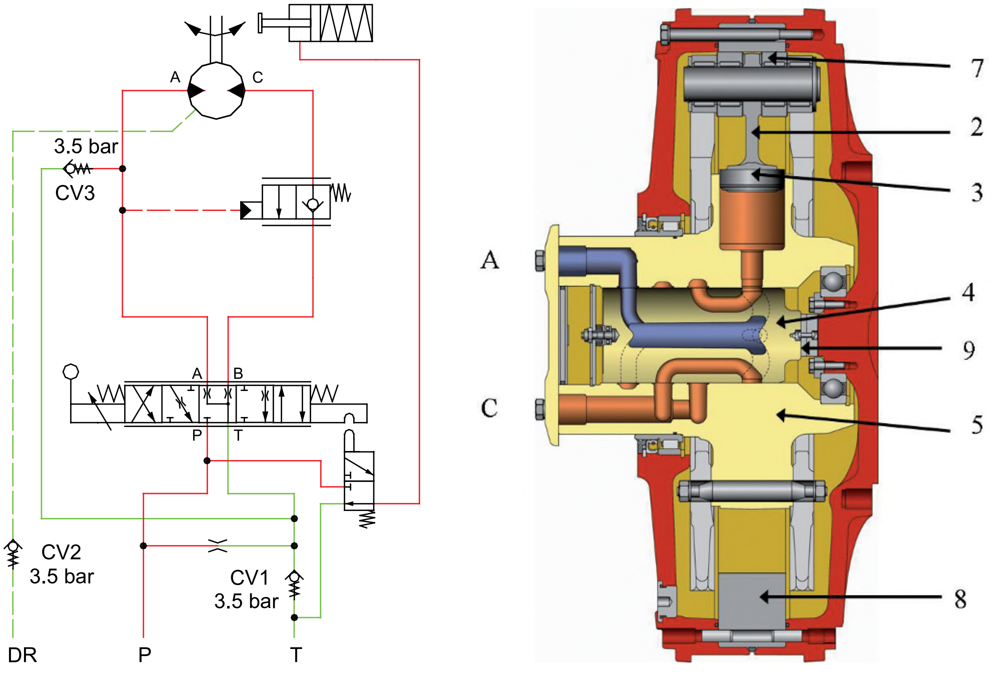


Fig 12.6.1 Hydraulic diagram and sectional drawing of the failing winch control

The hydraulic motor is of the radial piston type with a rotating housing. Mechanical torque from the winch is transferred into hydraulic pressure or vice versa. A rotating distributor inside the hydraulic motor controls the connection of the high pressure port and low pressure port to the radial pistons. The clearances of the distributor and the radial pistons are designed to seal against the high pressure but also show some internal leakage. This internal leakage oil is collected in the casing of the hydraulic motor. There is a drain connection port to drain this oil back to the reservoir.

A back pressure valve CV_2 (check valve) has been installed in the drain line to the reservoir. This check valve ensures that the casing of the hydraulic motor remains filled with oil. Another check valve CV_1 has been installed in the return (T) line to the reservoir. This valve has a cracking pressure of 3,5 bar providing back pressure to the return flow from the hydraulic motor. A small orifice from the inlet pressure line P to the return line ensures that there will always be a back pressure of 3,5 bar. A third check valve CV_2 is present in a line from the return connection of the proportional valve to the A-port line of the hydraulic motor. This check valve ensures that no cavitations can occur in the hydraulic motor if the motor is driven by the load. By mistake the opening pressure of this third check valve was also selected at 3,5 bar instead of 0,5 bar.

What went wrong?

Due to internal corrosion the main control valve did not return to its spring centred position when the manual operated proportional valve was released. That meant that the brake control valve remained in the "brake-disengage" position and the flow via the proportional valve to the low pressure A-port of the hydraulic motor was blocked or strongly restricted.

The hydraulic motor has an even number of radial pistons (3) that are coupled to cam rollers (7). These cam rollers press against the cam ring (8). The "distributor" (4) connects half of the piston chambers to the High pressure port (="C") and the remaining half to the Low pressure port (="A"). If the band brake is lifted, the hydraulic motor is in a condition where the outer cam ring is driven by the external load and the High pressure pistons are driven inside the cylinder block. This pushing inside is supported by the case drain pressure. Oil underneath the piston is pressurised and leaks to the casing of the hydraulic motor. The cam ring (= winch drum) will slowly pay-out. After a while the cam ring has pushed all the High pressure pistons inside.

In a normal situation the Low pressure pistons are pushed out again to the cam ring by the low hydraulic pressure at the "A"-port. Sufficient pressure however was not present due to the high setting of the check valve CV_3.

The hydraulic motor now comes to a situation where none of the pistons/cam rollers is pushed against the cam ring. The rotating part of the motor (case plus cam ring) can rotate now on its main bearing without any circulation of oil, as the cam rollers have lost any contact with the cam ring. As a consequence the winch will be in a free fall situation. Unfortunately a man was standing underneath the winch and was killed.

Lessons to be learned:

- The braking capacity of this winch motor failed due to the existence of a double failure: the continuous lifting of the band brake plus the too high line pressure in the A-port.
- In failure mode analysis normally only one failure at a time has to be taken into account. But only if that particular failure can be detected. If a first failure cannot be detected a second failure may cause the accident. The same in our example, if the presence of a pressure in the brake control line would have been detected the operator would have had the change to correct.
- The accident had nothing to do with the selection of the high torque hydraulic motor. That motor was selected due to its high performance on mechanical efficiency and the ease of constructing the motor onto a winch drum.

- The main reason for the failure was the installation of check valve CV_3 with a back pressure of 3,5 bar. This caused the pressure in the A-line to stay below the case drain pressure. The check valve was designed for a 0,5 bar setting. The mistake was probably made by the installation engineer 'as all other check valves also had a back pressure of 3,5 bar'.
- Proper maintenance for the proportional valve corrosion could have prevented the situation, as the valve than would have returned to its neutral position where the 3/2 valve for the control of the brake would not have been operated.

12.6.2 Free fall of a winch motor (2)

This case too concerns the drive mechanism for a lifting winch, but this time fitted with a fast running hydraulic motor, a gearbox and a closed loop system like the one already described in paragraph 10.2. For this drive mechanism the pressure for the booster valve was set to 15 bar. The hydraulic motor reaches speeds of approximately 2.100 rpm.

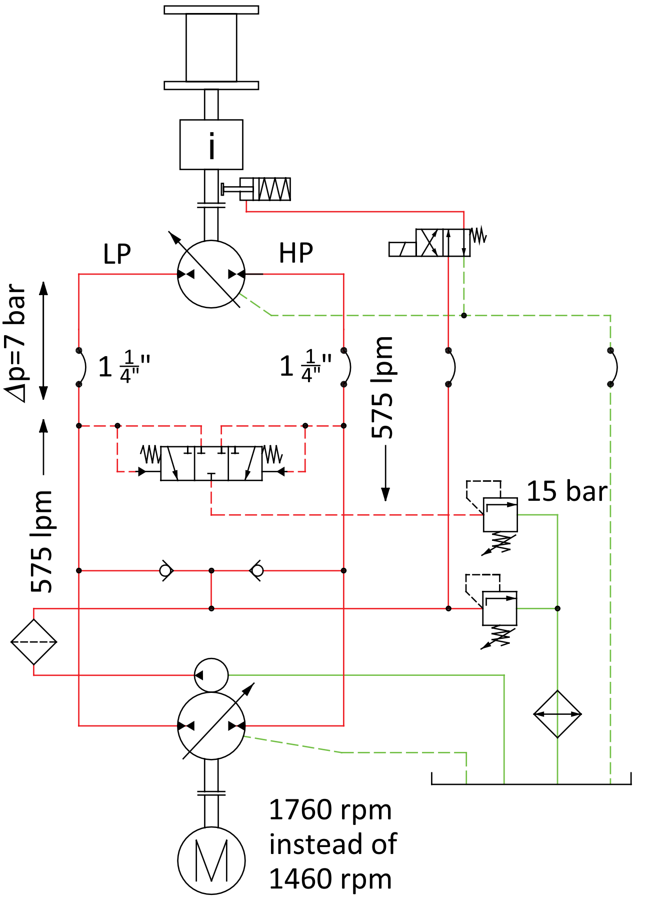


Fig 12.6.2.A Hydraulic diagram of the failure with the high speed motor in a closed loop system

What went wrong?

The pipe work from the variable pump to the hydraulic motor has been carried out with hydraulic hoses. A large volume flow of approximately 575 lpm is generated through the main pipes from the pump to the motor when the winch is released. A pressure drop of 8 bar developed from the pump to the motor because the diameter for the pipe work was designed too tightly. Also the electric motors were running at 60 Hz, so at 1760 rpm, instead of the design speed of 1460 rpm at 50 Hz. This meant that an inlet pressure of only 15-8 = 7 bar remained when the winch reached its maximum release speed. A hose diameter of 1 $^{1}/_{2}$" was specified during the design of the installation. Because this type of hose was not immediately available during the build phase, personnel concerned decided there and then to fit a 1 $^{1}/_{4}$" hose. At the maximum number of 2.100 rpm for the hydraulic motor the inlet pressure of 7 bar is absolutely insufficient to prevent cavitation on the inlet side of the motor.

The result was complete cavitation which meant that the torque characteristics of the hydraulic motor were completely lost. At that moment there was a test load of 150 mTonne in the winch's hook, which came down in a free fall movement. The free fall was stopped by the quick activation of the emergency brake, thus preventing a serious accident. Great mechanical damage was caused to the hydraulic motor and the disc brake. The hydraulic motor completely exploded as a result of the extremely high number of rpm that was reached for a short period of time.

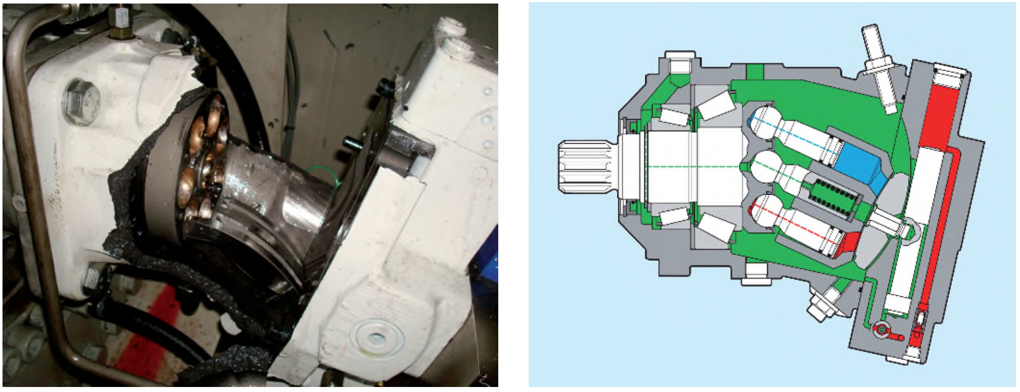


Fig 12.6.2.B Exploded hydraulic motor due to cavitation and the original section view of a new motor

Lessons learned:

The most important condition for a hydraulic motor drive for a lifting winch is that there needs to be sufficient boost pressure, and especially when the winch is releasing, for what is the low-pressure (= suction) side of the hydraulic motor at that point in time. This thorough control needs to take place at the desired operating temperatures and viscosity of the oil. The diameters of the connecting hoses were increased and an extra boost pipe was fitted, directly from the already present boost pump to the low pressure side of the hydraulic motor in the drive system described above.

An improvement of the installation can also be achieved by increasing the pressure setting of the boost pressure control valve (15 bar setting). This may not be possible for all pumps though.

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