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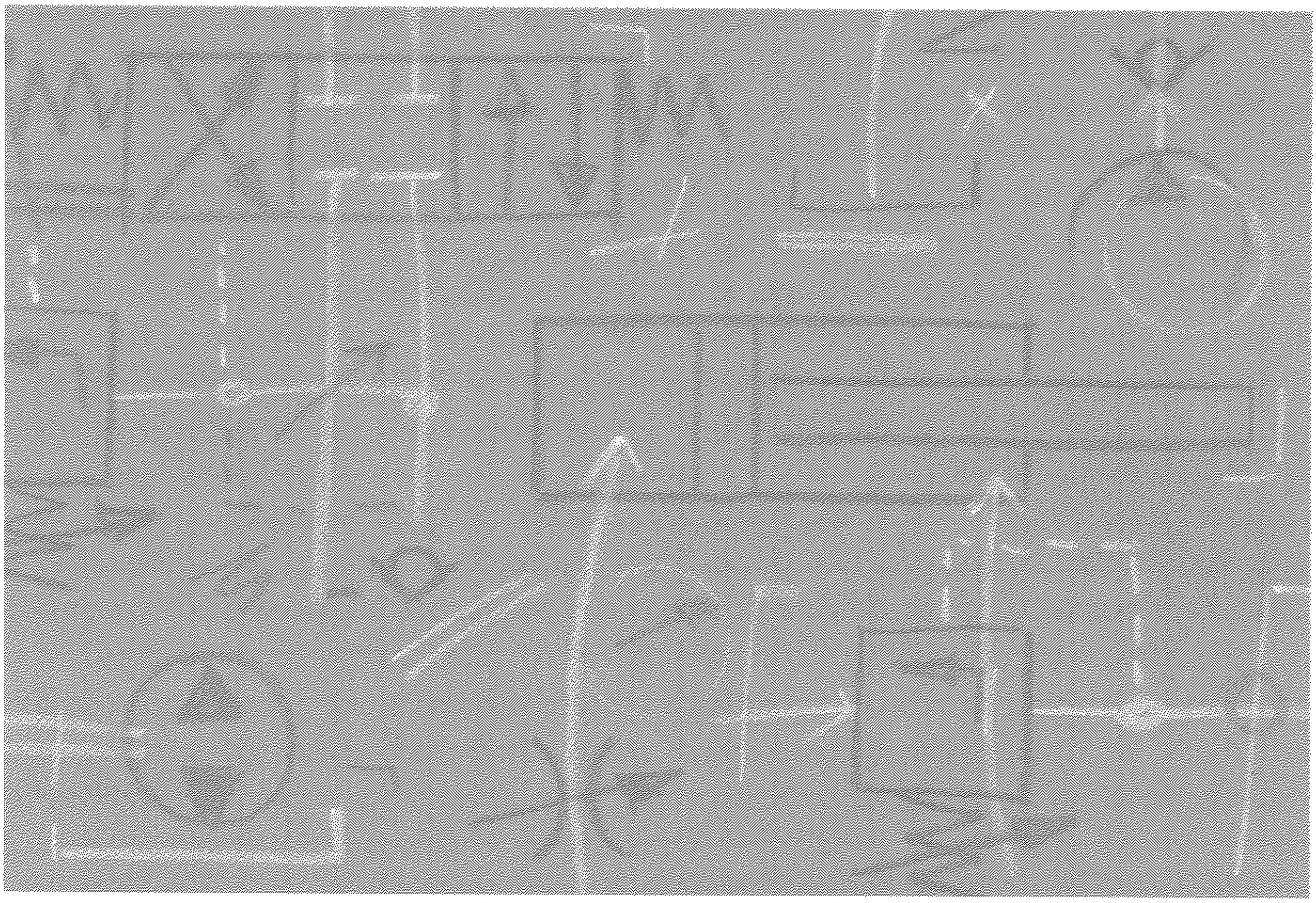

Planning and Design of Hydraulic Power Systems

Hydraulic Trainer, Volume 3

Rexroth Hydraulics

The Hydraulic Trainer
Volume 3

Planning and Design of Hydraulic Power Systems

A training manual for
the planning and design of
hydraulic power systems

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Foreword

Modern hydraulic drives and control systems have achieved wide-ranging significance. Together with mechanical, electrical and pneumatic drives they are “state-of-the-art” equipment and have some important advantages which make them outstanding.

Since hydraulic systems are being used in almost every branch of industry there has naturally been a growth in the demand for good, useful information on the subject. This explains the numerous publications dealing with the components of hydraulic systems and their applications.

It has become apparent, however, that it is no longer sufficient to study individual components in isolation. The correct interplay of the various parts of a system is more and more the principal, overriding factor. Users expect, and quite rightly so, that firms who market hydraulic systems should also be accountable for those systems.

Of course, this presupposes full competence in the technology. And, increasingly, the users of hydraulic systems are having to acquire the same technical competence.

A large number of excellent books have already been written on the subject of hydraulics but none of them deal specifically with hydraulic “systems”; most of them confine themselves to the components.

This manual *Planning and Design of Hydraulic Power Systems* attempts to fill the gap.

The manual deals with the way in which the components of hydraulic systems work together. The authors describe exactly what should be involved in the planning, design, manufacture and execution of hydraulic systems. There are numerous tables, diagrams and illustrations to clarify the functional relationships and interdependent aspects of the systems. They are a very useful aid in day-to-day working. Practical examples and the principal relevant standards will be found at the end of each chapter.

The manual is not only intended for users, it can also be very useful to those undergoing initial training or retraining.

The framework of professional training for hydraulic drive and control technology is constantly expanding. This manual will prove a valuable aid to those interested in keeping their knowledge of the subject up-to-date.

The manual is the result of a team effort and thanks are due to all the authors for their contributions. Special thanks must go to Messrs. Hans H. Faatz and Rudi A. Lang who undertook the general coordination of the project.

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1 Introduction

Basically, hydraulics is concerned with the transmission of force and power by means of the static pressure of a fluid. The tasks are performed by hydraulic systems which, in the market place, are in competition with mechanical, electrical and pneumatic systems. Hydraulic systems have a number of advantages over the latter but there are also a few disadvantages.

Advantages of hydraulic systems

- transmission of high forces within a small space
- high energy density
- energy storage capability
- stepless variation in motive quantities, such as speeds, forces and torques
- easy monitoring of forces
- rapid reversal due to low component masses (low inertia)
- fast operating response
- uniform motion (free from shock and chatter)
- wide transmission ratio
- simple conversion from rotary to linear motion or vice versa
- design freedom in the arrangement of components
- physical separation of drive input and output by pipes or hoses
- automatic control of all types of motion by pilot valves and electric signals
- easy usage of standard components and sub–assemblies
- overload protection
- minimum wear rates because hydraulic components are lubricated by the operating medium
- long service life
- energy recovery capability

Disadvantages of hydraulic systems

- pressure and flow losses in pipes and control devices (fluid friction)
- fluid viscosity sensitive to temperature and pressure
- leakage problems
- compressibility of the hydraulic fluid

The basic design of a hydraulic system and the power flow within it is shown in *Fig. 1*.

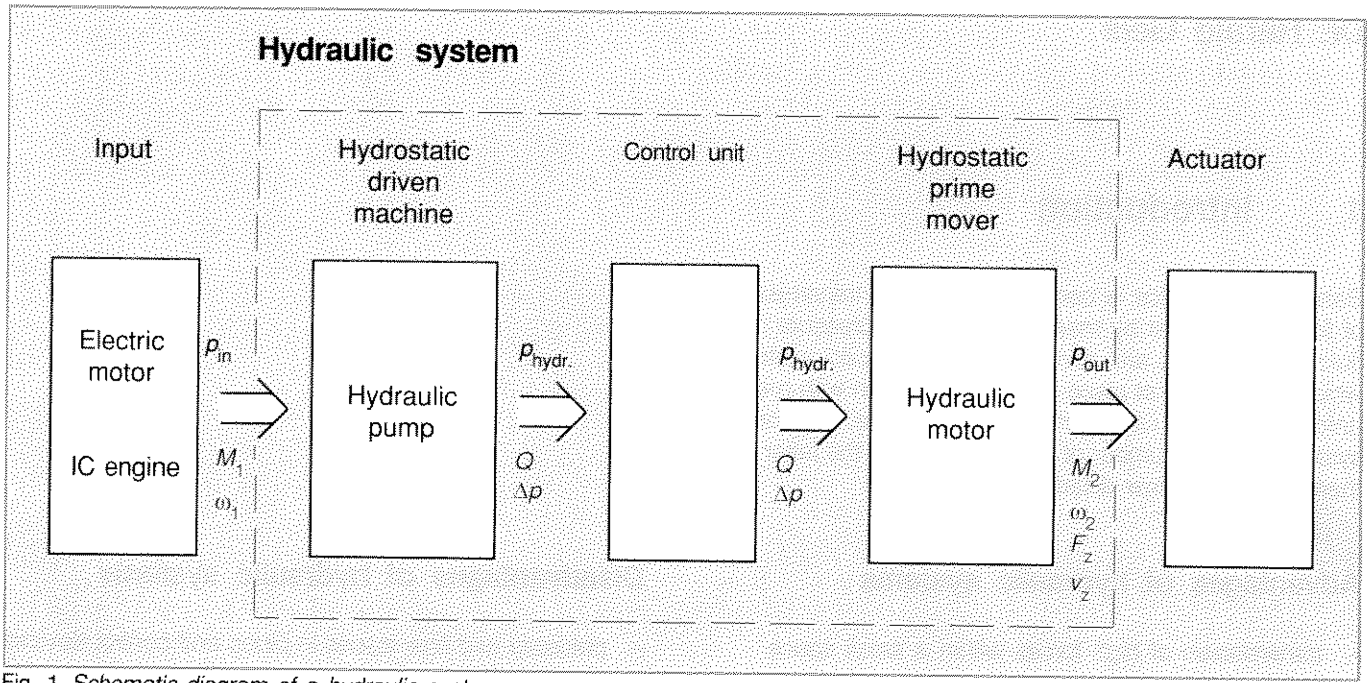


Fig. 1 *Schematic diagram of a hydraulic system*

In a hydrostatically driven machine, powered by an electric motor or internal combustion engine, mechanical energy (M_1, ω_1) is converted into hydraulic energy (Q, Δp). The control unit directs the hydraulic energy to the hydrostatic prime mover through various devices which regulate the pressure, direction and magnitude of the flow of fluid. The hydrostatic prime mover converts the hydraulic energy back into mechanical energy; it can be either rotating (M_2, ω_2), linear or reciprocating (F_C, v_C) depending on what the output requires. In addition there are such items as pipes, filters, heat exchangers, hydraulic accumulators, etc. that are not shown in *Fig. 1*.

The specific needs of hydraulics must also be taken into account so that the stated advantages of hydraulics compared with other forms of control can be fully utilized.

2 Planning procedure

The most important prerequisite for achieving a satisfactory answer to any problem in hydraulics is the adoption of a systematic procedure for the planning and execution of the hydraulic system. *Fig. 2* shows the sequence of the planning procedure in the form of a flowchart.

3 A description of the flowchart

Fig. 2 shows how, before the actual solution to the problem is produced, there are a large number of individual thoughts and ideas, i.e. the past experience of the planning engineer, which first have to be arranged in the proper sequence. This demands a careful approach otherwise the satisfactory functioning and economic efficiency of the installation can be compromised.

The impetus to plan and design a hydraulic system can come from any of the following sources

- request from sales
- customer's problems and enquiries
- competitors' advances
- market analysis
- trend studies
- own ideas
- patents

3.1 The task and its formulation

An important starting point for successful planning is a clear and detailed statement of the task involved.

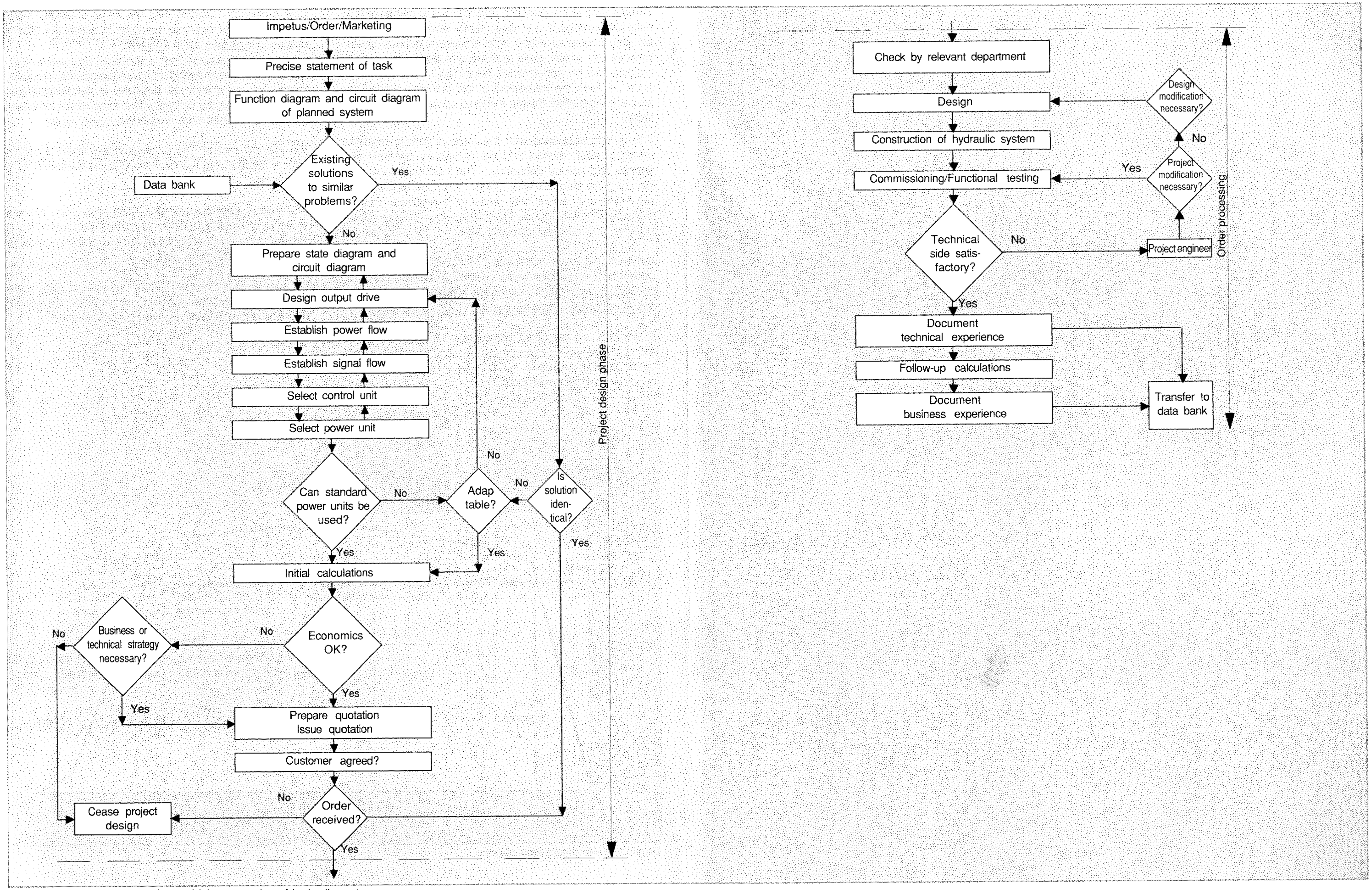


Fig. 2 Flowchart for the planning and job progressing of hydraulic systems

The first step of the planning procedure is to gather all the data and arrange it in a clear, easily–accessible way. A sensible course of action is to prepare a general questionnaire to which extra questions relating to specific projects can be added when necessary. This questionnaire will form the framework of the planning procedure and, amongst other things, it should contain the following facts:

The motion sequence with the force or torque requirements of each motion and the necessary dynamic response and natural frequency. The load sequence, i.e. including the intervals when there is no fluid or pressure requirement or where only pressure is required. This is particularly advantageous for optimum design when considering hydraulic accumulator systems, for example.

A motion sequence expressed in words is often difficult to understand, incomplete and, primarily, unclear. This is particularly true of difficult sequences involving several actuators whose motions overlap.

A diagram, on the other hand, provides both user and manufacturer with a common, simple and clear means of communication with brief notes inserted where necessary for even better understanding.

Take a plastics moulding machine as an example. Here we use a movement–time diagram in which the motion sequence is shown as in *Diagram 1*.

The ram moves in rapid traverse up to the moulding charge, i.e. as quickly as possible, is decelerated and then compresses the charge with a force which increases in a predetermined time sequence.

The forward motion has to be stopped when a certain force is reached but the force must to be sustained for a short time.

After curing comes controlled decompression followed by the ram travelling back to its starting position. This is followed by a fixed interval for ejection and the introduction of a new charge of plastic.

It is easy to see that the diagram provides a good overview of this procedure, especially when there are several similar and overlapping sequences in a system.

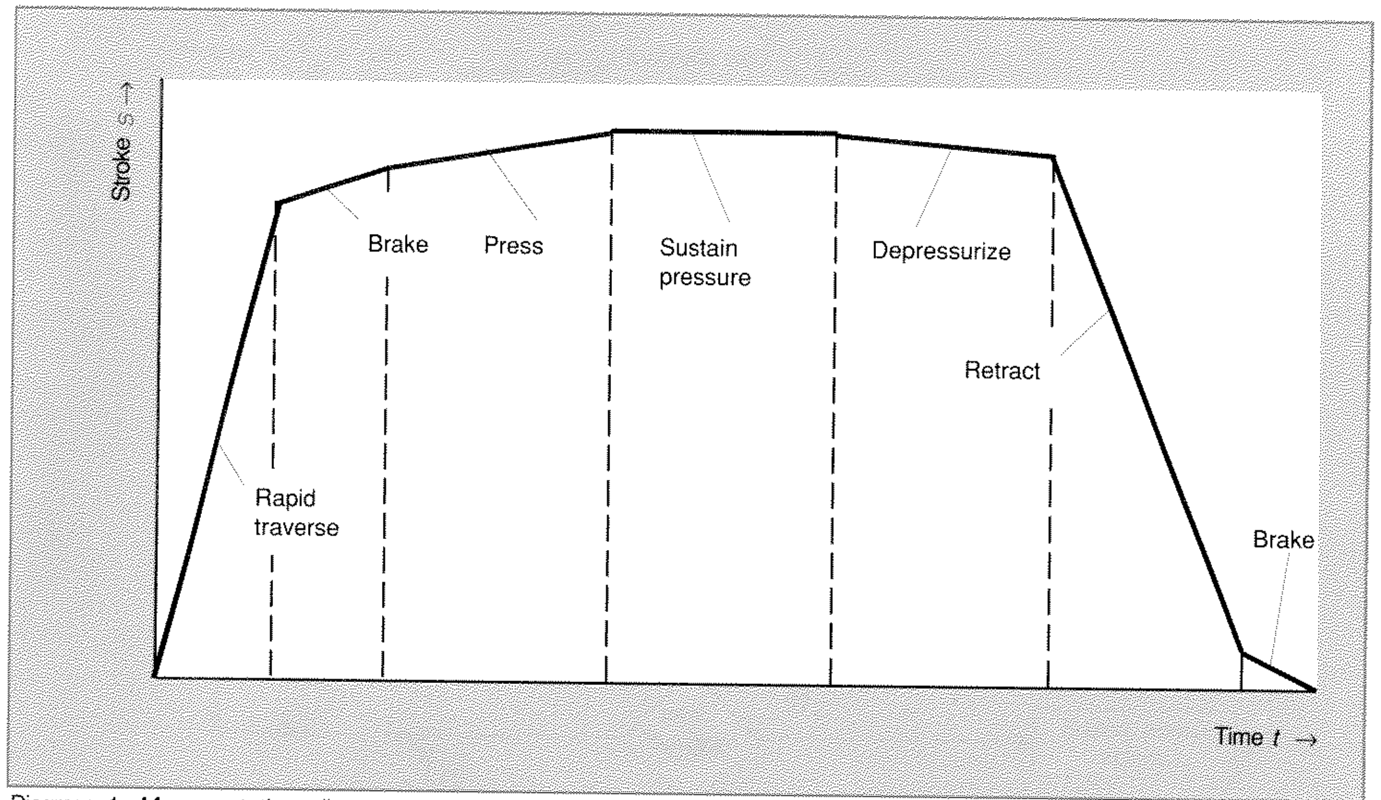


Diagram 1: *Movement–time diagram*

There are basically 2 types of function diagram [1].

a Movement diagram

This shows the interplay of the various elements. It is suitable for simple sequences, draft designs and tender sketches.

b State diagram

This shows the sequence of functions of the elements involved as a motion diagram with its interlocking controls.

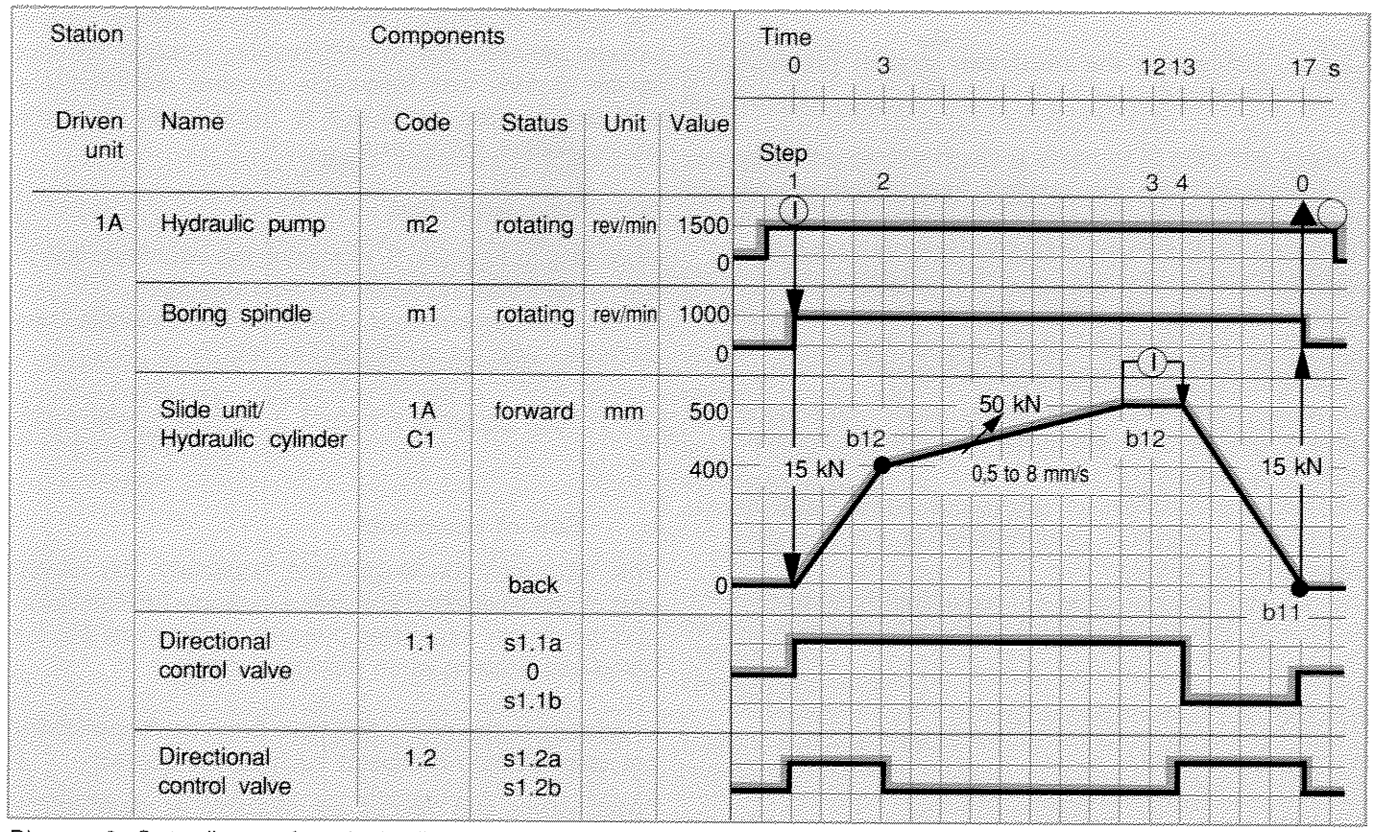


Diagram 2: *State diagram for a hydraulic cylinder [2]*

The actual design of the hydraulic system can begin once the list of requirements and function diagram have been established.

3.2 Designing the output drive

Deciding on the value of operating pressure is very important in selecting a suitable hydraulic actuator, i.e. linear or rotary. Together with the forces and speeds required for the particular task it helps determine the size of the machine and therefore its cost.
In selecting a maximum operating pressure it must be remembered that it must be at least equal to or greater than the sum of the rated pressure of the system and its losses.

$$p_{Nmax.} > p_{max.} \geq p_{installation} + p_{inst.\ losses} \qquad (1)$$

3.2.1 Estimating the maximum system pressure required

A useful first step is to make a rough estimation of the rated pressure from the nominal forces required. In practice, a correction based on experience and depending on the particular type of system (open or closed loop) would then be added to this value. It is, in fact, an estimated pressure drop. A more accurate determination of $p_{inst.\ losses}$ can only be made when the type of equipment to be used in the system has been decided.
Determining the system pressure needs experience. *Table 1* can be used for guidance for some common types of system.
The following example of a cylinder shows how an approximate theoretical determination of the operating pressure is performed:
The required nominal force F_N neglecting the losses in the cylinder, can be calculated from the following formula:

$$F_N \approx p \cdot A \qquad (2)$$

for continuity of flow:

$$Q \approx v \cdot A \rightarrow A \approx Q/v \qquad (3)$$

and substituting Equation 3 in 2 gives:

$$F_N \approx p \cdot \frac{Q}{v} \qquad (4)$$

This simple example shows how, when certain forces are required for example, the level of operating pressure influences both the volumetric flow (adapting the displacement or delivery to the particular task) and the size of equipment. Also, maximum values of operating pressure are laid down for certain fluids. Let us examine the individual parameters and their influences.

a Pressure level

According to Equation 2, increasing the pressure level allows smaller (i.e. cheaper) equipment, smaller bore pipes and improved specific pressure loss efficiencies

$$\eta_n = \frac{p_{inst.\ losses}}{p_{max}}$$

On the other hand, the cooling properties are poorer because the volumetric flow is less and therefore the tank volume is made less (less tank surface area).

In addition, leakage is greater, there is more wear due to friction and erosion (undissolved gas bubbles trapped in the sealing clearances) therefore shorter service life and volume variation due to compression, reduced stiffness in the system, less favourable dynamic characteristics and higher noise levels due to higher peak pressures during control movements.

b Volumetric flow

According to Equation 3, increasing the volumetric flow in the system also increases the flow velocity.
It must be remembered, however, that pressure loss increases with the square of the velocity.

$$p_{inst.\ losses} = \rho \cdot \frac{\bar{v}^2}{2} \cdot \left(\sum_{i=1}^{n} \lambda \frac{l}{D} + \sum_{i=1}^{n} \xi\right) \qquad (5)$$

c Size of equipment

This influences the weight and, especially, the capital cost of an installation. The space available for installation and the directly associated pressure losses must also be taken into account when making a choice.

In addition to the points that have been mentioned there are other economic aspects such as the question:

"What valves and devices are available as standard?"

In the case of an installation that is operated continuously, for example, the need for high efficiency is much more important than for a system that is used only occasionally. In actual practice, certain pressure ranges have become the norm for frequently–used systems which closely approach the optimum in terms of function and economic efficiency. The examples in *Table 1* show some of the most common pressure ranges employed in various industries.

Application	Subdivision into market areas	Hydraulic systems in	Operating pressure range p_{Op} in bar
Industrial hydraulics	Foundries and rolling mills	Walking beam conveyors, handling systems, roll frames	160 to 180 315 to 420
	Machine tools	Planers, slotters, drills, lathes and grinders, hydraulic clamps	50 to 100 50 to 300
		International automobile industry	16 to 120
	Presses	General presses, special-purpose presses, high-pressure presses	250 to 315 400 to 600 to 1000
	Plastics machinery	Injection-moulding and blow-moulding machines, special-purpose machines	150 to 210 250 to 315 300 to 450
	Test stands test beds	Materials test stands Simulators	250 to 290
Hydraulics in steelwork construction, civil engineering and power station construction	Steelwork construction and civil engineering, theatre engineering	Theatre engineering, moving stages, winches, curtains, etc. reactor engineering, air locks steam turbine governors, weir systems, locks, moving bridges, ropeways and lifts	100 to 150 50 to 100 120 to 250 100 to 220 160 to 250
Hydraulics in mining	Mining and water hydraulics	Hydraulic ropeway drives, loaders, cutters, roof supports, tunnel driving machines	200 to 250 200 to 280 320 to 420
Mobile hydraulics	Agricultural engineering	Tractors, combine harvesters, harvesting machines	to 100
	Mobile hydraulics	Transport systems, cranes, fork-lift trucks, bulldozers, excavators, tunnelling machines	160 to 250 350 to 420
Special-purpose hydraulics	Special applications	Transmissions, aircraft rudder actuators	150 to 400
	Piston pumps/ Hydrostatic transmissions	Special-purpose machines, rotating drives for – industry – mobile machines – testbeds secondary control systems winches	to 315 to 315 to 420 to 300 to 300 to 200
Marine hydraulics	Piston pumps/ Hydrostatic transmission	Steering gear, deck cranes, bow doors, bulkhead valves	150 to 250 150 to 300 to 200 to 200

Table 1: *Normal operating pressures in some typical hydraulic systems and installations*

3.2.2 Selecting output devices

When the operating pressure level has been estimated, the next step is to begin deciding on the type of output devices required. In general, the output unit comprises either hydraulic cylinders or hydraulic motors. Both will now be described in some detail.

3.2.2.1 Hydraulic cylinders (linear motion)

There are 3 basic requirement criteria

a Velocity
b Stroke (position)
c Force

It is possible for them to occur either singly or in different combinations.
As the basis for selecting a hydraulic cylinder the following performance details are taken from the statement of task, such as:

– Required force sequence

It must first of all be remembered that in addition to the required force on the piston various other forces are present. These are:
$F_{Fr.}$ = frictional force between piston, guides and seals
F_a = acceleration forces
F_{oil} = the force required to eject the oil out of the other end of the cylinder(at the required cylinder speed)
F_W = force due weight

These are additive:

$$F_{piston} = F_N + F_R + F_a + F_{oil} + F_W \qquad (6)$$

The calculation of the piston area takes into account the approximate supplementary force $F_{Fr.}$ through the hydraulic–mechanical efficiency. Flow losses within the hydraulic cylinder are not taken into account because of their insignificant magnitude provided the sizes of the connections are appropriate.

Within the rated pressure range and when a single–rod cylinder is extending, a guide value for the hydro–mechanical friction is η_{mh}= 0.95 and for the return stroke, η_{mh} = 0.85 to 0.9. The specific value of depends on the tolerances at the piston and piston–rod guides and the seal type.

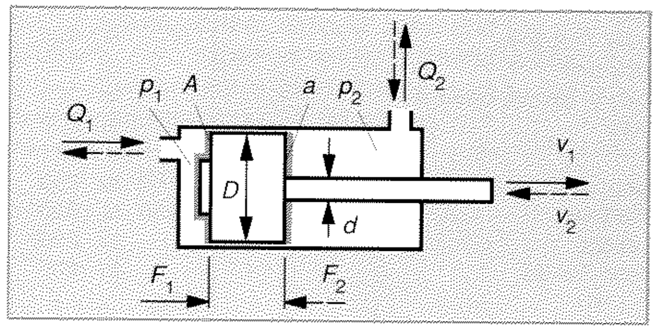


Fig. 3 *Forces acting on piston and piston rod*

Case 1: Extending hydraulic cylinder

$$F_{1Res} = p_1 \cdot A \cdot \eta_{hm1} - \frac{p_2 \cdot a}{\eta_{hm2}} \qquad (7)$$

$$A = D^2 \cdot \frac{\pi}{4}$$

$$a = A - \frac{d^2 \cdot \pi}{4}$$

p_2 can be neglected if it is equal to atmospheric pressure.

Case 2: Piston return (stroke)

$$F_{2\,Res} = p_2 \cdot a \cdot \eta_{hm2} - \frac{p_1 \cdot A}{\eta_{hm1}} \qquad (8)$$

In this case it must be pointed out that, with single–rod hydraulic cylinders, there are design reasons why the area ratio must not be less than a certain figure [3].

$$\varphi_{limit} = \frac{1}{\eta_{hm}{}^2}$$

– Speed of rapid traverse, working and return strokes

The following relationships are applicable to the two cases mentioned above – Case 1 (extending) and Case 2 (retracting):

$$v_1 = \frac{Q}{A} \qquad (9)$$

$$v_2 = \frac{Q}{a} \qquad (10)$$

which give:

$$\frac{v_2}{v_1} = \frac{A}{a} = \varphi \qquad (11)$$

where φ is the area ratio.

– Piston travel, stroke time, dwell time and operating pressure

These depend on the size of the equipment and are taken into account in the design of the hydraulic cylinder [4]. For very strict demands on accuracy of positioning there are three different possible solutions

a Servo cylinder
b Electro–hydraulic linear amplifier
c Multi–position cylinder

A more detailed description of the construction and mode of operation of these items will be found in [2] and [5].

– Buckling

If a piston rod is relatively thin and there is a high thrust acting on it, it is advisable to calculate the buckling [6]. This is particularly important in the case of "strength" hydraulic cylinders with long strokes which are installed at or near the horizontal because they are then subject to additional lateral forces due to bending (as shown in *Fig. 4*).

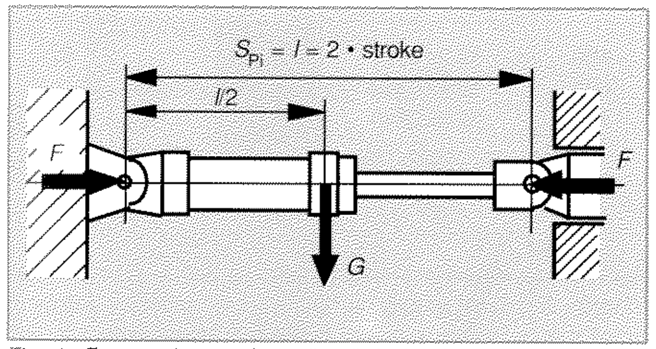


Fig. 4 *Forces acting on a hydraulic cylinder installed horizontally*

3.2.2.2 Hydraulic motors (rotary motion)

When selecting hydraulic motors it is necessary to take into account performance quantities such as

- speed and speed ranges
- torque and power, both peak and continuous
- type of load (constant or fluctuating)
- type of duty
- constant or variable displacement
- angular acceleration
- swivel times of control systems
- magnitude of starting torque
- steady–state motion in the low speed range
- type of operating medium

With these operating parameters in mind, the project engineer can select a suitable hydraulic motor from the maker's catalogue [5]. As an aid to selection, *Table 2* lists the basic design features and principal characteristics of hydraulic motors. It is advisable to discuss the planned application with the manufacturer.

When selecting a hydraulic motor it must be remembered that the losses which occur depend both on the type of construction and the size of the motor. This point is particularly important with large installed powers and long working hours, i.e. when there is high utilization of the motor.

The causes of losses are the same with all hydraulic machines, motors and pumps alike. Leakage losses increase the effective displacement whereas hydraulic–mechanical losses effectively reduce the output torque below the theoretical value. Remember that the maximum pressures listed in *Table 2* are above the permitted continuous operating pressures of the systems. The pressures are quoted in accordance with DIN 24312.

As high system pressures are being used it is essential, in order to avoid high volumetric losses, for the displacement element to move within its surrounding walls with minimal clearance. At the same time, however, the gap between the displacement element and the surrounding walls, e.g. a piston in a cylinder bore, must not be too small otherwise the friction losses will be excessive, i.e. those from the hydraulic–mechanical source.

These facts are applicable to all types of displacement machines and it can therefore be concluded that the volumetric and hydraulic–mechanical losses are the main factors governing the efficiency and operating characteristics of hydrostatic machines and installations.

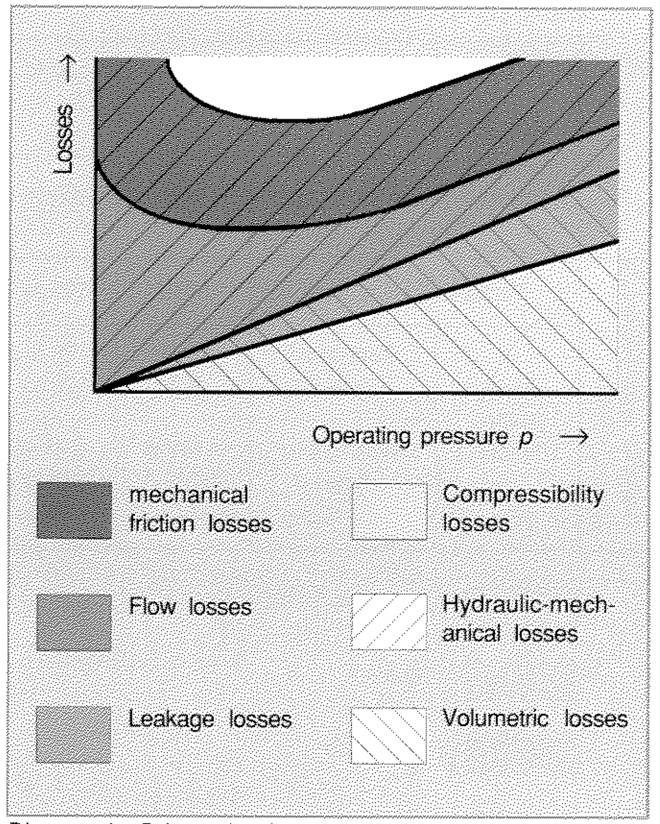


Diagram 3: *Schematic of the losses in a hydrostatic machine (motor)*

We will now examine the main types of losses.

Type of motor	Rexroth designation	Nominal size, i.e. displacement in cm^3	p_{max} > p_{rated} in bar	Speed range according to size in rev/min	Starting characteristics	Noise level	Efficiency $\eta_{t\ max}$
Fixed displacement motors – Gear motors	G2 G3	6 to 38	250	500 to 3000			85
Fixed displacement motors – axial piston motors – high speed – Bent-axis types	A2FM A2FE A2F/BR5	10 to 250 28 to 180 355 to 1000	450 450 400	50 to 6000 50 to 4750 50 to 2240			92
Fixed displacement motors – axial piston motors – high speed – Swash plate types	A4FM	22 and 28	450	30 to 4000			91
Fixed displacement motors – axial piston motors – Slow-speed types	MCS MC(4) MC(6)	200 to 1500	250	5 to 500			90
Fixed displacement motors – Orbit-type motors	MZA MZD MZF MZK	60 to 270	225	10 to 1000			85
Fixed displacement motors – Radial piston motors – Cam-ring type	MCR	500 to 3000	450	3 to 250			91
Fixed displacement motors – Radial piston motors – Eccentric type	MR	190 to 7000	420	1 to 500			92
Variable displacement motors – Radial-piston motors	MRV	190 to 7000	420	1 to 500			92
Variable displacement motors – Axial piston motors – Bent-axis type	A6VM A6V	28 to 355 28 to 107	450	50 to 8000 **			92
Variable displacement motors – Axial piston motors – Swash-plate type	A10VM A4VS * A10VSO*	45 40 to 250 28 to 71	315 400 315	30 to 3600 ** 6 to 4900 ** 40 to 3600 **			91

good

better

very good

excellent

* Suitable for use in load matching (secondary control) circuits.
** Values only applicable when untilted.

Table 2: *Comparison of different types of motor from their principal characteristics. The data is taken from catalogue data [5] and refers to mineral oil as the operating medium*

Volumetric losses

These arise from the flow through clearances when there is a pressure differential present and other forms of leakage.

This gives:

$$Q_e = Q_{th} + Q_{Cl} \qquad (12)$$

Q_e = Effective volumetric flow

Q_{sp} = Clearance flow

and the volumetric efficiency becomes:

$$\eta_{VM} = \frac{Q_{th}}{Q_e} = \frac{Q_e - Q_{Cl}}{Q_e} = 1 - \frac{Q_{Cl}}{Q_e}$$

Friction in narrow clearances between components moving relative to each other cause hydraulic–mechanical losses, with the result that the motor output torque M_{out} is less than the theoretical motor torque M_{th} based on the pump pressure and the theoretical displacement volume of the motor.

$$M_{th} = p \cdot Q_{th}$$

$$M_{out} = M_{th} - M_{hm} \quad \rightarrow \quad \eta_{hm\ M} = \frac{M_{out}}{M_{th}} \qquad (13)$$

Thus, the overall efficiency becomes

$$\eta_{tot\ M} = \eta_{hm\ M} \cdot \eta_{VM} = \frac{M_{out} \cdot \omega}{\Delta p \cdot Q_e} \qquad (14)$$

where $\omega = 2 \cdot \pi \cdot n$

This means that the output power is:

$$P_{out} = M_{out} \cdot \omega = \Delta p \cdot Q_e \cdot \eta_{tot\ M} \qquad (15)$$

Flow losses

These arise due to changes in flow direction and at throttling points and variations in cross section. The flow of fluid loses some of its energy in overcoming these resistances.
The actual amount must be determined by experiment and depends on the geometry of the component, the velocity and the specific gravity of the fluid. Because of the close relationship between frictional losses from internal components and changes in direction, vortices, etc., in practice, designers use a loss index ξ.

$$\xi = \Sigma\ \lambda \cdot \frac{l}{d} + \Sigma\ \zeta \qquad (16)$$

where λ = Pipe friction coefficient
l = Pipe length
d = Pipe diameter
ζ = Loss index of individual components

The pressure loss can then be determined from:

$$p_{pipe\text{-}loss} = \xi \cdot \frac{\rho}{2} \cdot \bar{v}^2 \qquad (17)$$

where $\bar{v}$ is the mean velocity of the fluid.

3.3 Selecting the control unit

3.3.1 Task of the control unit

The task of the control unit is to link the function of the input drive system with that of the output drive system according to predetermined rules.

It is necessary to distinguish between

a Power flow

or the flow of the fluid in direction, magnitude and pressure level from the hydraulic tank to the actuator, including any necessary devices such as valves, and

b Signal flow

the task of which is to collect and process all external data necessary for controlling the power flow according the function to be performed. The signal flow thus transfers information smoothly into power flow.

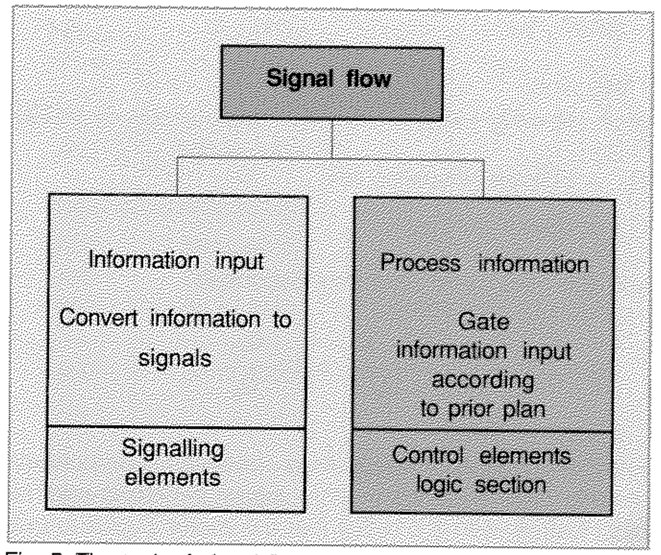


Fig. 5 *The task of signal flow*

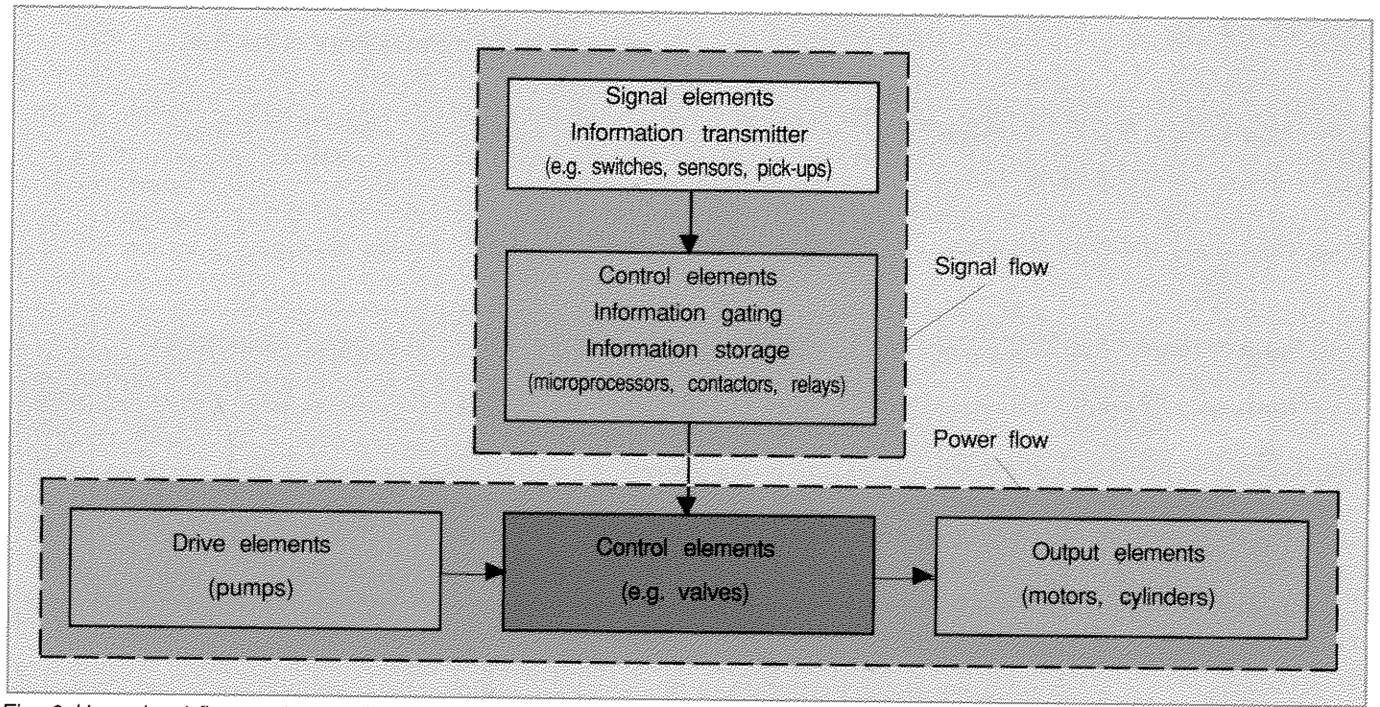


Fig. 6 *How signal flow works together with power flow*

Once the circuit diagram and required output power are known, the nominal sizes of the individual actuators and control devices are also fixed. A free choice of valve actuation remains. This can be mechanical (lever or cam), electrical, electronic, hydraulic or pneumatic or, for optimum operation, a combination of one or more of these.

3.3.2 Determining the signal flow

The rules for determining the signal flow are laid down in the function diagram to VDI 3260 (see *Section 3.1*) and in the statement of task for the installation. In many cases the planner of the hydraulics does not have an overview of the complete signalling system, so close co-operation with other experienced control specialists, i.e. electronic and electrical engineers, is necessary.

3.3.3 Determination of the control of the power flow

At this stage of the planning it is sensible to decide on the control of the power flow – which is naturally closely linked to the signal flow – i.e. to select all the valves and fittings necessary for proper functioning of the hydraulic system.

The basis for this is once again the function diagram and the peripheral factors of the statement of task which govern the type and size of the valves.

It is also necessary to decide where, in the hydraulic system, the individual valves must be fitted in order to perform their tasks.

This is particularly true of pressure valves, flow valves and check valves, and it also has an effect on the accessibility of the installation and its flow characteristics.

Mode of operation and technical data of the individual devices are dealt with in [5], [6] and [7].

When selecting the different control elements, there are certain common features that can be formulated:

- mode of operation (binary, digital, analog, proportional)
- static and dynamic characteristics
- efficiency
- space required
- cost
- reliability
- safety features
- maintenance costs.

Devices which take their function signals from the signal flow must be coordinated with it so that no uncontrolled reactions can take place.

3.3.4 Selecting a method of measurement

In order to ascertain the effect of the signal flow on the power flow it is necessary to measure various operating states, primarily hydraulic variables such as pressure, flow and temperature. In addition, it may be necessary to measure mechanical variables such as travel, force, speed, velocity and acceleration.

A measuring system is selected according to the following key factors:

- the measuring accuracy required (resolution, hysteresis)
- measuring range
- upper and lower frequency limits
- permitted temperature range
- permitted pressure range
- electrical and mechanical disturbances
- distance between sensor and signal processor
- reaction to environmental factors
- availability of suitable equipment
- cost

Generally speaking, measuring sensors should be simple, cheap and universal in application. They should also be interchangeable. But their most important characteristic is stability in order to ensure that the absolute measured–values are reproduced reliably according to the criteria specific to the installation.

3.4 Selecting the power unit

Once the output actuator and control unit have been selected, and hence the required power output and losses in the hydraulic system are known, attention can be turned to selecting the required power unit which comprises a pump, tank and accessories such as filter and cooler.

3.4.1 A preview of the different power systems

The first step is to clarify what type of circuit is to be employed. Table 3 lists the various factors which help in this decision.

		Type of circuit		
		open (throttle control)	open (pump control)	closed/ semi-closed (make-up system)
Duty	continuous		X	X
	intermittent	X		
High power				X
Operating pressure	low (up to 160 bar)	X		
	medium (up to 250 bar)	X	X	X
	high (up to 450 bar)		X	X
System	simple	X	X	
	complex		X	X
Fast control response		X	X[1]	X[1]
Compact (fluid tank)				X
Type of output	rotary			X
	linear	X	X	

[1] Only when pipe runs are short or when used in secondary control circuits

Table 3: *Criteria for selecting the type of circuit*

In this connection it must be pointed out that, in actual practice, the decision on the type of circuit ultimately depends on the particular application of the system and on the experience of the planning engineer because it is usual for there to be a mixture of the various factors listed in *Table 3*.

Apart from selecting the type of circuit, the type of power system to be employed is also important. The two basic forms are the individual power unit and the centralized fluid supply system, the latter being used primarily for larger hydraulic systems.

3.4.1.1 Individual drives

These are normally used when it is necessary to prevent any mutual interference between different motions. Different types of pump can be used depending on the particular duty.

Fixed displacement pumps

The advantages are the relatively low price and the constant delivery volume per revolution. If the hydraulic system is working considerably below the maximum delivery a reduction in the efficiency must be anticipated. The reason for this is the need to return the unwanted delivery back to the tank via a branch line (the pressure relief valve).

$$P_{losses} = (Q_{pump} - Q_{actuator}) \cdot p_N \quad (18)$$

Where p_N is the maximum continuous operating pressure setting.

Variable displacement pumps

By using a variable displacement pump it is possible to match the delivery to the needs of the actuator. Its disadvantage is its greater initial cost.

Combination of pumps

Another method of obtaining different volumetric flow rates is to use several fixed displacement pumps connected in parallel. With this arrangement the drive power can be matched to the required hydraulic power in steps, so there is no major excess drive power.

Yet another economic alternative is a combination of constant displacement pump and hydraulic accumulator which can be used in cases where there is a high flow demand for a short period of time.

Q_{max} within the cycle time (0.5 to 0.66)

Another factor is whether the hydraulic system is to be stationary (industrial) or mobile.

Unlike an industrial installation a mobile hydraulic system can only employ a certain number of pumps because of the limitations of the engine available. The problem is optimizing the utilization of the power provided by the common engine. This can only be achieved economically with variable displacement pumps [8] to [14] since fixed displacement pumps are much less suitable.

3.4.1.2 Centralized fluid supply system

This type of system is used when

- there are several similar actuators in use which are not all needed simultaneously
- several actuators with very different requirements are to be combined
- the demand for fluid pressure is small compared with the time of a work cycle.

The best course of action is to divide the total power between a number of pumps which should be identical if possible. Depending on the requirements, the centralized fluid supply system can be arranged as follows.

Pressure fluid system with fixed displacement pump and accumulators

The task of the accumulators is to provide the system with sufficient energy at times of peak power demand.

The fixed displacement pump or pumps is/are started and stopped according to the state of charge of the accumulators. This type of arrangement is good when there are a number of actuators connected to the system and the work cycles are long, e.g. on machine tools.

Other advantages of such accumulator installations are:

- savings on pumps, drive motors and tanks
- lower installed power
- less space required
- lower noise level.

Disadvantages are:

- similar power losses to constant pressure systems (throttling control)
- hydraulic accumulators are subject to special approval regulations.

The design and mode of operation of accumulators and the criteria for their selection are dealt with in detail in the chapter on hydraulic accumulators. The pumps must be rated so that the volume of fluid taken from the accumulator can be replaced after the work cycle.

Pressure fluid system with variable displacement pumps

With this type of system the pressure in the common line is maintained by variable displacement pumps. It should be noted that, when large quantities of pressure fluid are taken from the common line, the control time of this type of pump means that dips in the pressure can occur.

Pressure fluid system with variable displacement pump and accumulators

Peak demands are covered by the hydraulic accumulators so that the installed pump power can be reduced compared with a system employing only variable displacement pumps and the recovery times of the pumps can be faster.

3.4.1.3 Power matching (or saving)

Hydraulic systems convert mechanical power into hydraulic power, which is easy to transmit, control and distribute, and which is subsequently converted back into mechanical power by hydraulic cylinders or motors. If the pump has to supply several valve–controlled actuators and the operating conditions are unfavourable, there may be a substantial power loss due to throttling which also heats up the fluid. Such operating conditions arise in the part–load range when the pump is delivering more fluid than needed by the actuators or when the pump pressure has been set above the pressure required by the actuators. Consequently, it is energy–efficient if the drive power (i.e. pump delivery and pressure) can be matched to the demand (i.e. load sensing, load matching control).

In addition to hydraulic concepts for matching input power to output power there are also electrically–based systems. A comparison of electrical and hydrostatic drives has been carried out by Metzner [15].

He lists the following advantages of load matching control (secondary control) over electrical speed control:

- better dynamic response
- more compact
- lighter
- low inertia
- lower system cost above 50 kW.

Fig. 7 gives an overview of the capabilities of hydrostatic drive systems.

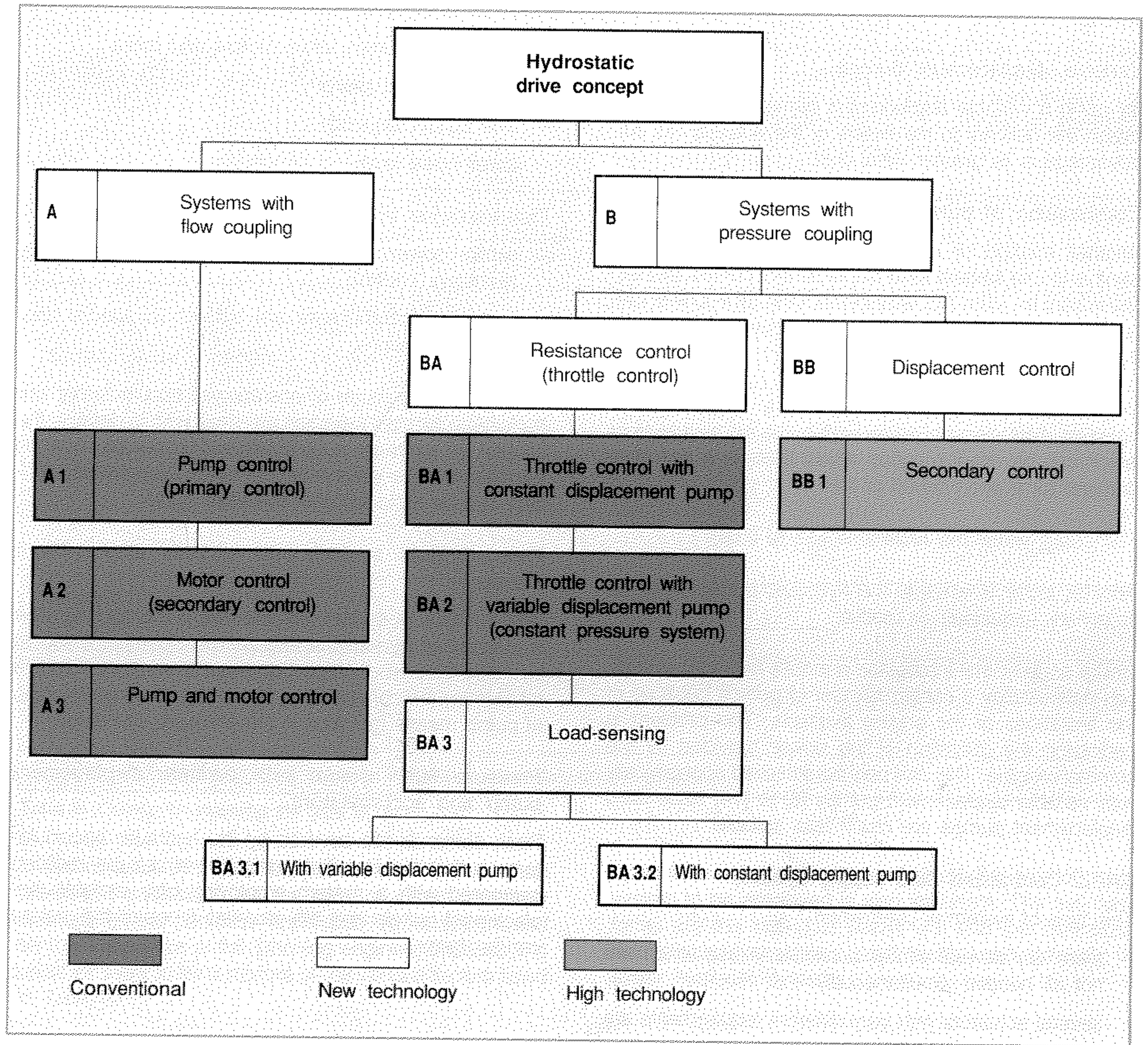


Fig. 7: *Hydrostatic power systems*

A Systems with flow coupling

This system, known in practical circles as hydrostatic transmission, employs the principle of a pump (the primary unit) drawing power from the energy source and transmitting it hydraulically to one or more hydraulic motors (the secondary units). At that point, the hydraulic energy is re–converted into mechanical power, usually in the form of a rotary motion.

Depending on the particular application and the controllability required, there are further variations in the systems:

A1 Systems with variable displacement pump and fixed displacement motor (primary control)

Open–loop circuit – open suction *(Fig. 8)*

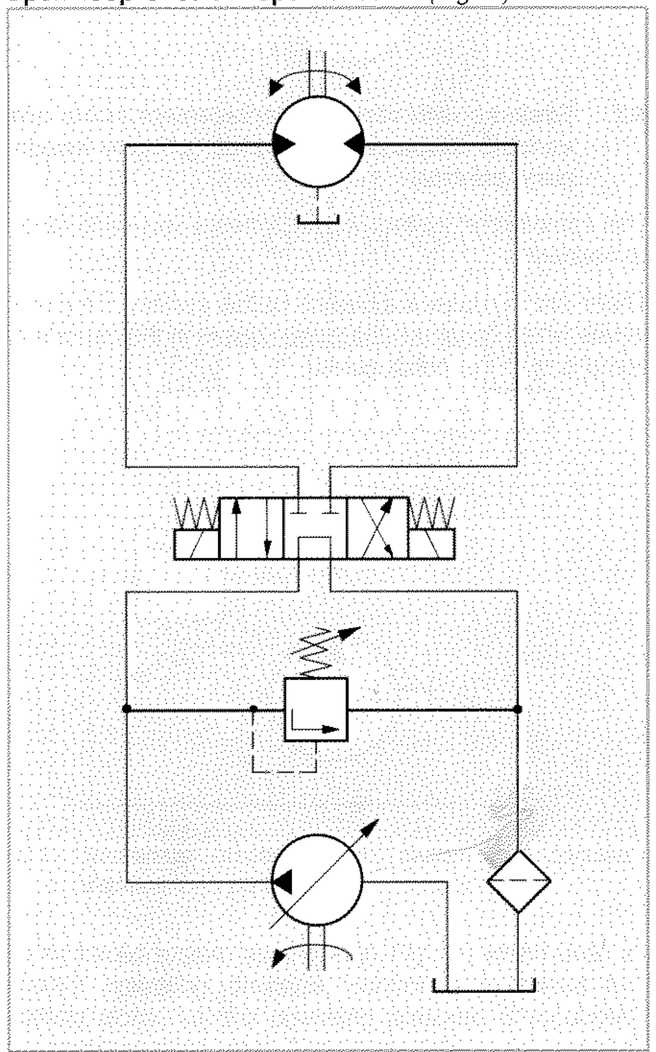


Fig. 8: *Pump control in an open–loop circuit – suction circuit*

With an open–loop circuit the fluid flows from the tank to the hydraulic pump from whence it is delivered to the hydraulic motor. From the hydraulic motor the fluid flows back to the tank at low pressure and is subsequently drawn into the pump suction again.

The output direction of the hydraulic motor can be changed by interposing a directional control valve.

A pressure relief valve protects the hydrostatic drive from overloads. Filtration usually takes place in the return line.

Closed–loop circuit – boosted circuit *(Fig. 9)*

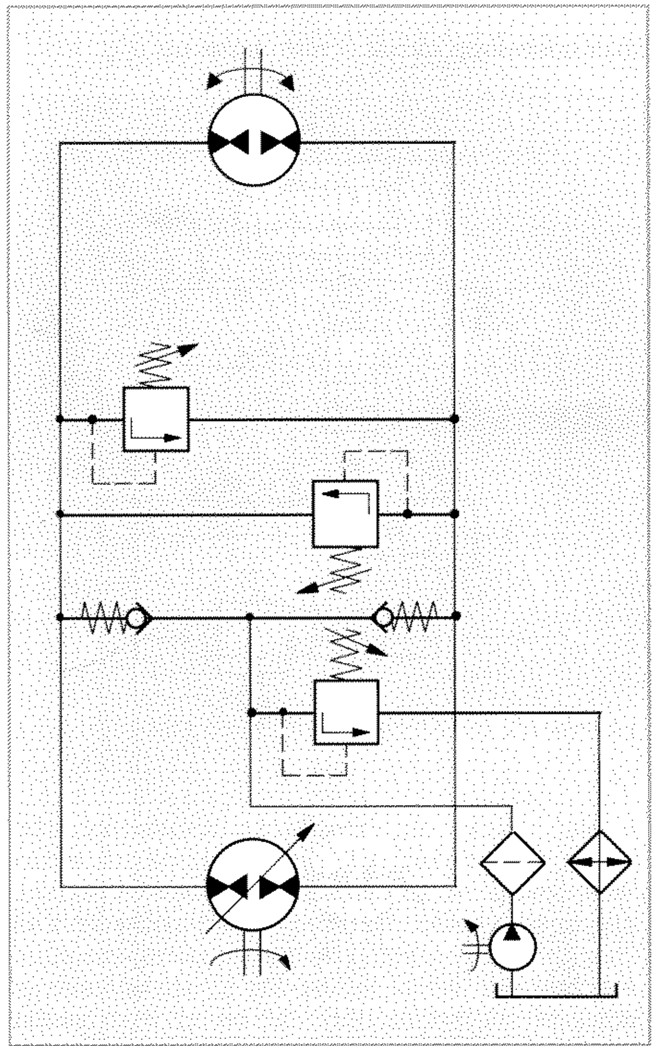


Fig. 9: *Pump control in a closed–loop circuit – boosted circuit*

In the closed–loop circuit the fluid discharged from the hydraulic motor is returned directly to the hydraulic pump. For filling the circuit initially and for topping up to replace the unavoidable leakage there is a boost pump which must usually have a delivery of about 15% of the main pump delivery. When there are rapid changes in load it can sometimes be necessary to boost up to 100% of the main pump delivery, which can be done from hydraulic accumulators. Two separate adjustable pressure relief valves protect the hydrostatic transmission against overload. Filtering of the fluid takes place in the return line of the flushing valve or in the boost delivery line itself.

The closed–loop circuit allows pump and motor to interchange functions so that torques and forces at the output can be transmitted to the prime mover through the hydraulic pump. This reversal of the energy flow provides almost entirely loss–free braking.

However, this is only true if the pump and motor are of approximately the same size.

Semi open–loop circuit – with make–up *(Fig. 10)*

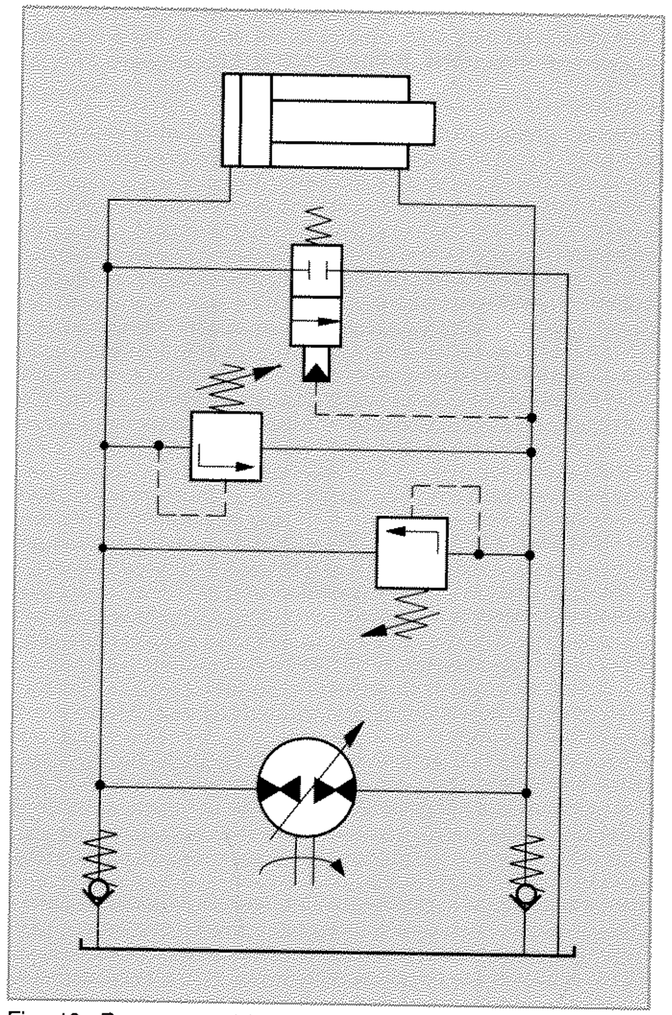


Fig. 10: *Pump control in a semi open–loop circuit – with make–up*

With a semi open–loop circuit there is an insufficient flow of fluid in one direction from the hydraulic motor back to the pump.

The quantity lacking is made up from the tank through make–up valves.

The output drive element is usually a single–rod cylinder so that more fluid is returned in the other direction to the hydraulic pump than it can deal with. The difference is diverted back to the tank through a directional valve. Two pressure relief valves protect the hydrostatic transmission against overload.

Pump control is used mainly for individual drives, high powers and continuous operation. The main difference between the open–loop circuit and the closed–loop circuit is the reversibility. A disadvantage of both systems is that their dynamic response decreases with increasing length of connecting pipework.

A2 Systems with constant displacement pump and variable displacement motor (secondary control)

This version has fewer practical applications. Its disadvantage compared with the **A1** version is that a minimum speed must be maintained.

$$n_{\text{min motor}} = n_{\text{pump}} \cdot \frac{V_{\text{pump}}}{V_{\text{max motor}}}$$

A3 Systems with variable displacement pump and variable displacement motor (primary–secondary control)

This combination system is used when a very wide range is needed.

There are no losses inherent in the system. The only losses which occur are the power losses in the motor and pump and in the boost pump.

B Systems with pressure coupling

BA Resistance (throttling) control

This is often used for intermittent operation and for multiple drives, e.g. machine tools and mobile installations. *Figs. 11 to 14* show various options with resistance control whereby the *Fig. 11* version is preferred for lower powers and the *Fig. 13* version for higher powers.

BA1 Resistance control with constant displacement pump *(Fig. 11)*

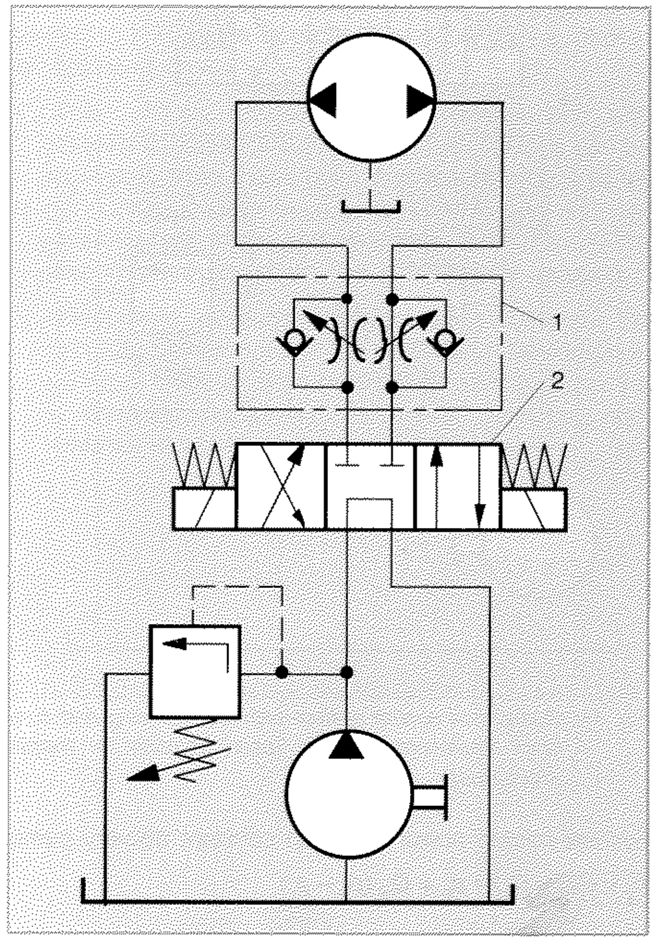


Fig. 11: *Resistance (throttling)control system with constant displacement pump*

In a hydraulic system as shown in *Fig. 11*, which uses a constant operating pressure, control is exercised by throttling alone, i.e. the operating pressure is regulated by a pressure relief valve. The pump delivery not needed by the actuating device is converted into heat at the pressure relief valve and so goes down as a loss in the energy balance. The pressure differential ($p_p - p_L$) not needed by the actuating device is dissipated at valves 1 and 2 and also goes down in the energy balance.

System–related power loss

$$P_V = (Q_P - Q_M) \cdot p_P + Q_M (p_P - p_M)$$

BA2 Resistance control with variable displacement pump*(Fig. 12)*

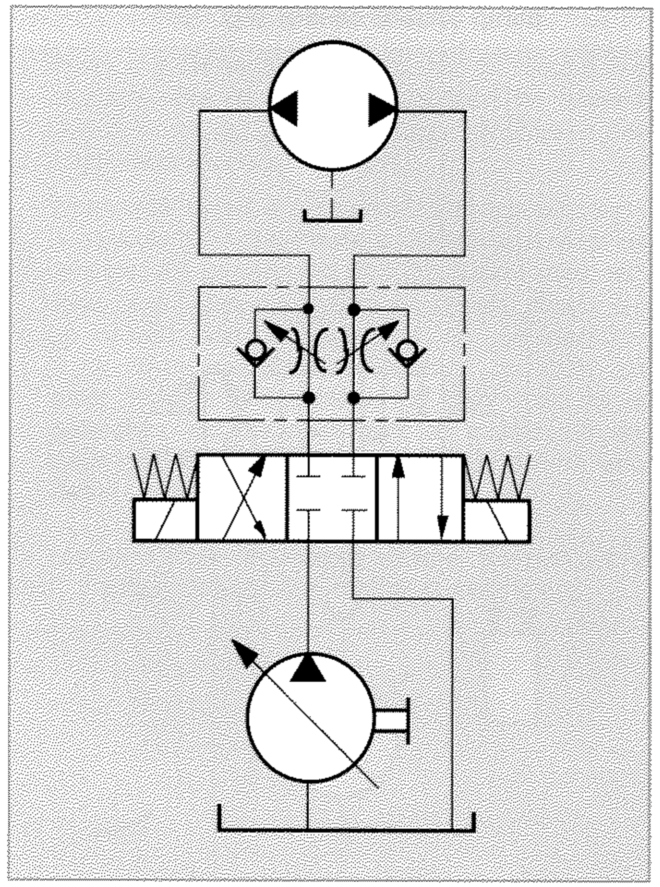


Fig. 12: *Resistance control system with variable displacement pump*

Unlike conventional installations, this system uses a variable displacement pump. The swivel angle of the pump, i.e. its delivery, is regulated by the system pressure in the pump line. The pump only delivers the precise quantity needed by the actuating device at any moment. The disadvantage of this system is that the response pressure must be set very high so that the actuating device can continue to be supplied when under high load.

System–related power loss

$$P_V = (p_P - p_M) \cdot Q_M$$

BA3 Load sensing

BA3.1 Load sensing with variable displacement pump (closed centre) *(Fig. 13)*

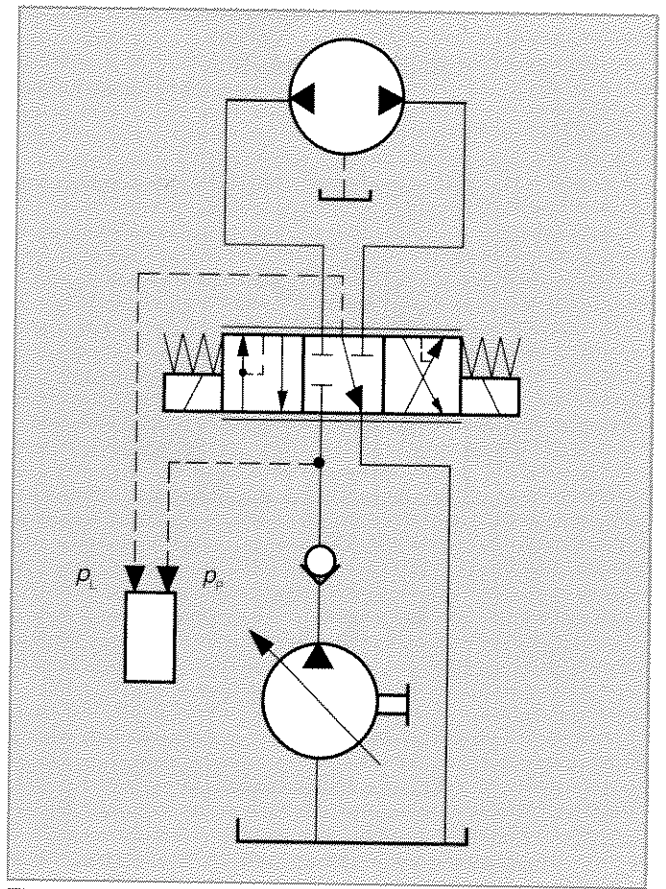


Fig. 13: *Resistance control system with load sensing and variable displacement pump (used primarily for mobile machines)*

By means of a load pressure feedback system the pressure and flow can be matched to the needs of the actuator. Only slightly more hydraulic power is provided than is needed by the various, individual actuators; thus, energy is saved. This type of control is very sensitive and is almost independent of the load pressure; meaning easy operating control.

System–related power loss with variable displacement pump

$\mathbf{P_V = (p_P - p_L) \cdot Q_M}$ where $\mathbf{p_P - p_L \approx}$ **constant** $\approx$ **15 bar**

BA3.2 Load sensing with constant displacement pump (open centre) *(Fig. 14)*

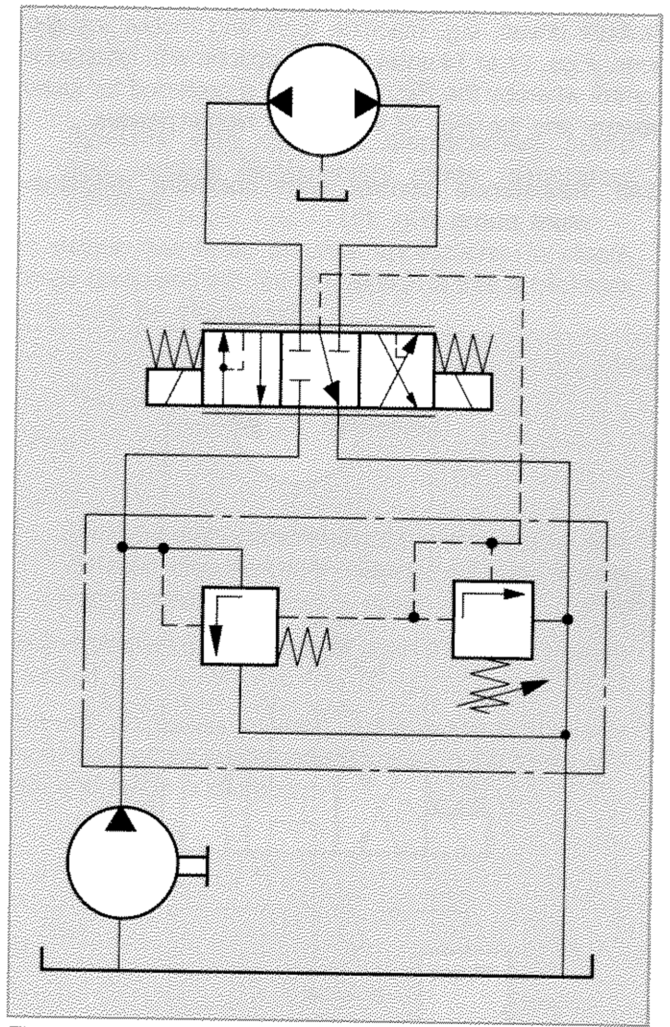


Fig. 14: *Resistance control system with load sensing and constant displacement pump (used primarily for mobile machines)*

System–related power loss with constant delivery pump

$\mathbf{P_V = (Q_P - Q_M) \cdot p_P + Q_M\,(p_P - p_L)}$

$P_V = (Q_P - Q_M) \cdot (p_L + 15 \cdot 10^5\ \text{N/m}^2) + Q_M \cdot 15 \cdot 10^5\ \text{N/m}^2$

where $p_P - p_L \approx$ constant $\approx 15 \cdot 10^5\ \text{N/m}^2 = 15$ bar

BB Displacement control (load matching systems [with secondary control])

BB1 Load matching – Secondary control *(Fig. 15)*

Either of these terms means control of the displacement of secondary unit, i.e. the actuator. A good example is speed control of a hydraulic motor. When, as in certain cases, external energy is fed into the system, the motor works as a pump in order to keep the velocity or speed constant. This is why we speak of the unit as a "secondary unit" and not of a hydraulic pump or hydraulic motor.

The system is described in more detail in [10], [15] and [16]. As with the systems described in Section **A**, there are no system–related losses. An interesting fact, according to [17], is the application of secondary control in the following cases:

- When various actuators work in parallel and with different sequences (e.g. on ships) and energy can be recovered from those units that are braking (i.e. regenerating) in order to be used for other units that are motoring (e.g. in lifts and elevators). It means that the total installed power can be substantially reduced.
- When there are considerable distances between the hydraulic power unit and the actuators. The compressibility of the fluid plays no part here because control is exercised directly with the motor. Therefore, high dynamic response control is possible.
- When a good power–to–weight ratio is needed, e.g. on ships, vehicles, machine tools, etc. The hydraulic units are considerably more compact and lighter than comparable electrical units [15].
- When work cycles are repeated systematically and it is possible to recover energy. Typical examples are vehicles, e.g. town buses, fork–lift trucks, ships' winches, centrifuges, etc., where the braking energy can be stored and utilized later for acceleration. Track–driven vehicles are another possibility as, when driving round a bend, the inner units act as generators and the outer units as motors with the result that, for a specific speed, energy can be recovered.

Of course, it must be debated whether the success of such a system is worth the cost of its installation.

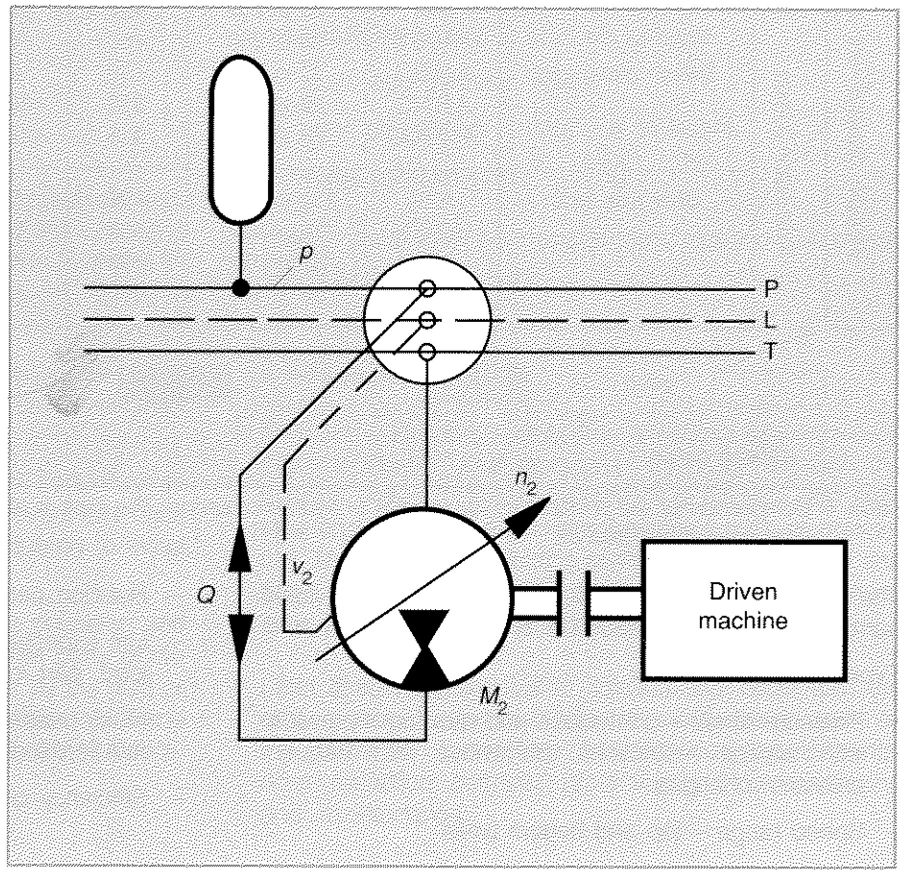


Fig. 15: *System with pressure coupling (secondary control)*

3.4.2 Selecting the pump

From what has been said so far, and having determined the operating pressure, we are now in a position to consider the type of pump to be employed. *Table 4* lists the different types and nominal sizes of hydraulic pumps and permits them to be compared.

Type of pump			Rexroth designation	Nominal size i.e. delivery in cm^3	$p_{max} > p_{rated}$ in bar	Speed range according to size in rev/min	Pulsation characteristics	Noise level	Efficiency $\eta_{t\,max}$
Constant displacement pumps	Gear-type pumps	external teeth	G2 G3 G4	3 to 100	250 250 250	500 to 5000 500 to 5000 500 to 5000			85 bis 90
		internal teeth	GM	25 to 50	210	900 to 1800			90
	Vane-type pumps		V2	10 to 36	175	900 to 3000			86
	Radial piston pumps		R4	0,4 to 20	700	1000 to 3400			90
	Axial piston pumps (bent axis type)		A2FO A2F KFA2FO	10 to 250 200 to 1000 45 and 63	450 400 350	1500 to 3150 950 to 1800 2000 to 2240			92 92 92
Variable displacement pumps	Vane-type pumps		V3 V4 V5	12 to 63 20 to 125 8 to 63	100 160 70	1000 to 1800 750 to 2000 900 to 1800			85 85 85
	Axial piston pumps	Swash plate type	A4V A4VSO A4VSG A10VO A10VSO	28 to 250 28 to 250 28 to 250 28 to 100 28 to 100	450 450 450 315 315	500 to 4250 * 500 to 4250 * 500 to 4250 * 1000 to 3000 1000 to 3000			91 91 91 91 91
		Bent axis type	A2V A7VO A7V A8VO	250 to 1000 20 to 1000 20 to 1000 28 to 107	400 400 400 400	500 to 2500 * 500 to 4100 500 to 4100 500 to 3150			92 92 92 92

good | better | very good | excellent

* Closed circuit

Table 4: *Comparison of the principal characteristics of hydraulic pumps. The data is taken from the various data sheets [5] with the units used on mineral oil.*

The requirements contained in "the statement of task" narrow down the type and size of pump even further.

Typical key factors are:

- Pressure range – continuous and maximum pressure with service restrictions, behaviour with frequent pressure changes
- Speed range – top and bottom limits
- Relation between pressure and speed
- Life expectancy with various load profiles (continuous load, part load, overload, pressure peaks)
- Fluid used – purity and temperature requirements (NAS, viscosity); see chapter on Hydraulic Fluids
- Efficiency characteristics for different types of load

- Suitability for uniform or sharply fluctuating demand
- Operating reliability and noise
- Variable displacement pumps – control rate and range, internal leakage losses, choice of control devices
- Ease of maintenance and repair

Let us examine one or two of these points in detail.

Efficiency η

This is primarily of interest at high powers and high utilization factors = (operating hours/total hours).

A lower efficiency or a higher power loss heats up the fluid and necessitates more cooling.

The utilization factor also plays an important role in this connection.

$$\eta_A = \frac{\bar{V}}{V_{max}} \qquad (19)$$

η_A	using
> 80 % high →	Constant displacement pumps
30 < 80 % medium →	Variable displacement pumps
< 30 % low →	Constant displacement pump with accumulator

Table 5: *Guide values for V > 0)*

Service life

This is very dependent on the operating pressure (e.g. the life of ball bearings is reduced by 7/8 if the operating pressure is doubled) and on the purity of the fluid and its temperature.

Noise level

All positive displacement machines work on similar principles.

Their chambers are filled in a continuous sequence as they rotate, being closed afterwards to prevent any back flow and then being opened again subsequently to expel the contents.

This intermittent process causes pressure pulsation and hence vibration which is transmitted to other parts of the installation in the form noise, carried either by the fluid or the mechanical structure. Pumps also radiate air–borne noise from their outer surfaces as a result of their operating principle.

The usual methods of reducing such noise are:

- to de–couple the structure–borne noise (by mounting on anti–vibration mountings, etc.)
- by restricting the transmission of fluid and air–borne noise (with accumulators and silencers)
- by avoiding additional sources of vibration.

More detailed information on this subject will be found in the chapter on Noise Reduction and in [18].

Fluid

If it is intended to use fluids other than mineral oil it is advisable to consult the manufacturer of the hydraulic equipment first.

3.4.3 The tank or reservoir

Another important item in the power unit is the tank in which the operating fluid is stored. Apart from storing the fluid used in the system, its tasks are also to assist in removing contamination and water, air separation, to act as a heat exchanger and to allow the return fluid to settle.

In the majority of applications it should also be able to support the hydraulic components, namely pumps, valves and other accessories such as accumulators, filters, etc. In the case of large installations with a centralized fluid supply system this is not always necessary. All the points that have been mentioned make certain demands on the construction and size of the tank. They are formulated in VDI 3230 as follows:

- static and dynamic rigidity
- good dissipation of heat due to losses
- easy maintenance.

Further details will be found in the chapter on Steelwork Design for Power Units.

3.4.4 Accessories

Grouped under this subheading are those devices which perform filtration and storage of the fluid, regulate the thermal status of the installation (e.g. coolers and heaters) and indicate the operating variables such as pressures. The criteria for selecting such items, their design and construction features will not be dealt with here because later chapters cover them comprehensively.

Later chapters also deal in considerable detail with the other principal aspects of planning and design such as the pipework connecting the individual components, noise reduction, corrosion protection and commissioning and maintenance.

4 Documentation of practical experience

In this connection, documentation means the systematic collection of information, its analysis and its preparation into a convenient form for use. It should be remembered that even the best documentation is worthless if it is neglected.

The main source of information, apart from magazine articles, patents, etc., is the practical experience gained from the design and execution of other hydraulic installations. The information gathered has an exclusively practical slant, which means that it should be used and not merely gather dust!!

Basically, documentation covers two aspects:

1 Indicating to the planning engineer the current state–of–the–art

2 Indicating possible and practical methods and solutions which have already proved successful and are available. The economic aspect is an integral part of this.

In general, engineers solve the problems that confront them with expertise and the greatest possible economy in mind.

In the search for the correct solution, many decisions must be taken the value of which is in direct proportion to the nearer the approximate the ideal solution and the quicker they can be obtained. Such decisions are easier to achieve if the necessary information is available at the same time, i.e. when planning begins. Good documentation is ,therefore, one of the best ways of arriving at good solutions to future new problems, i.e. saving, storing and imparting the knowledge of experience gained.

5 Economics

In order to have a clear view of the economics of an installation it is essential to break down the costs into a more easily assimilated pattern.

From *Fig. 16* it will be clear that it is impossible for the planner to forecast the costs at the beginning of a project. Consequently, any initial calculations make use of past experience from comparable projects. The extent to which the tender prices agree with the final prices is also part of the tenderer's know–how. It has been established that there is always a cost advantage for the user in purchasing a complete system and, at the same time, he is assured of good service. Another plus point is the reduced number of possible sources of error, i.e. practically no interface problems, and optimum coordination of the electrics and electronics with the hydraulics. Since only the project engineer bears responsibility for the planning and execution, the user's risk is also minimized. Another worthwhile exercise is a comparison of purchase price with operating costs. Operators often go for low tenders without giving due consideration to the very important after–costs associated with commissioning, maintenance and energy.

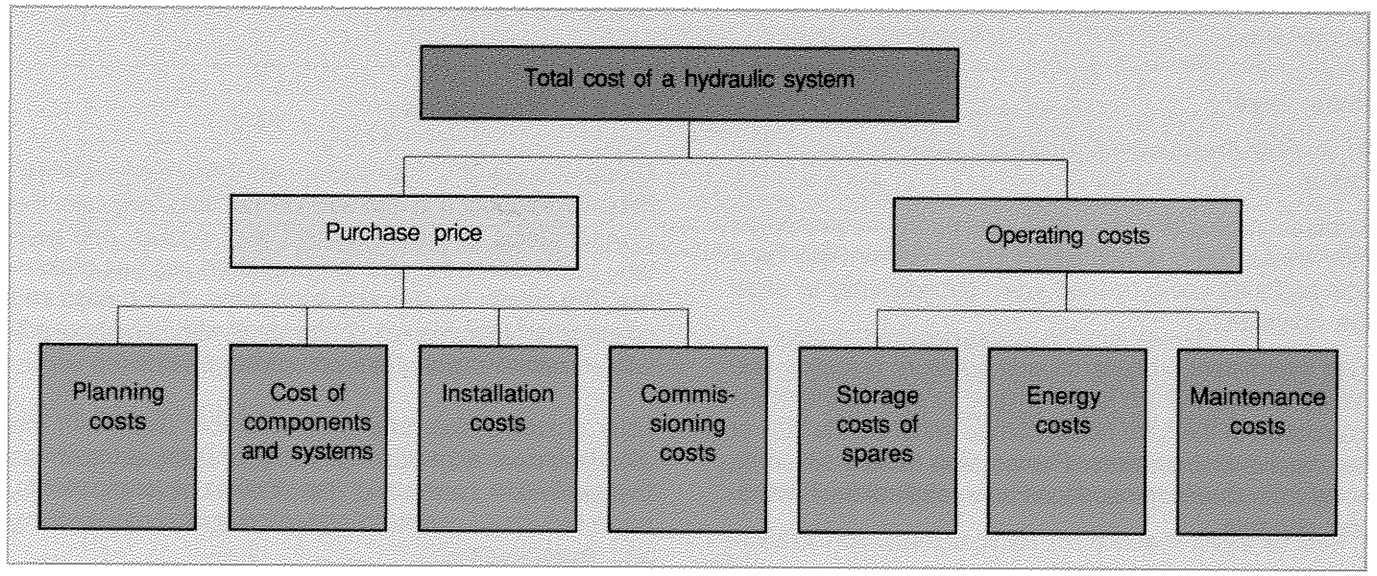


Fig. 16: *Breakdown of the total cost of a hydraulic system*

6 A guide to project design

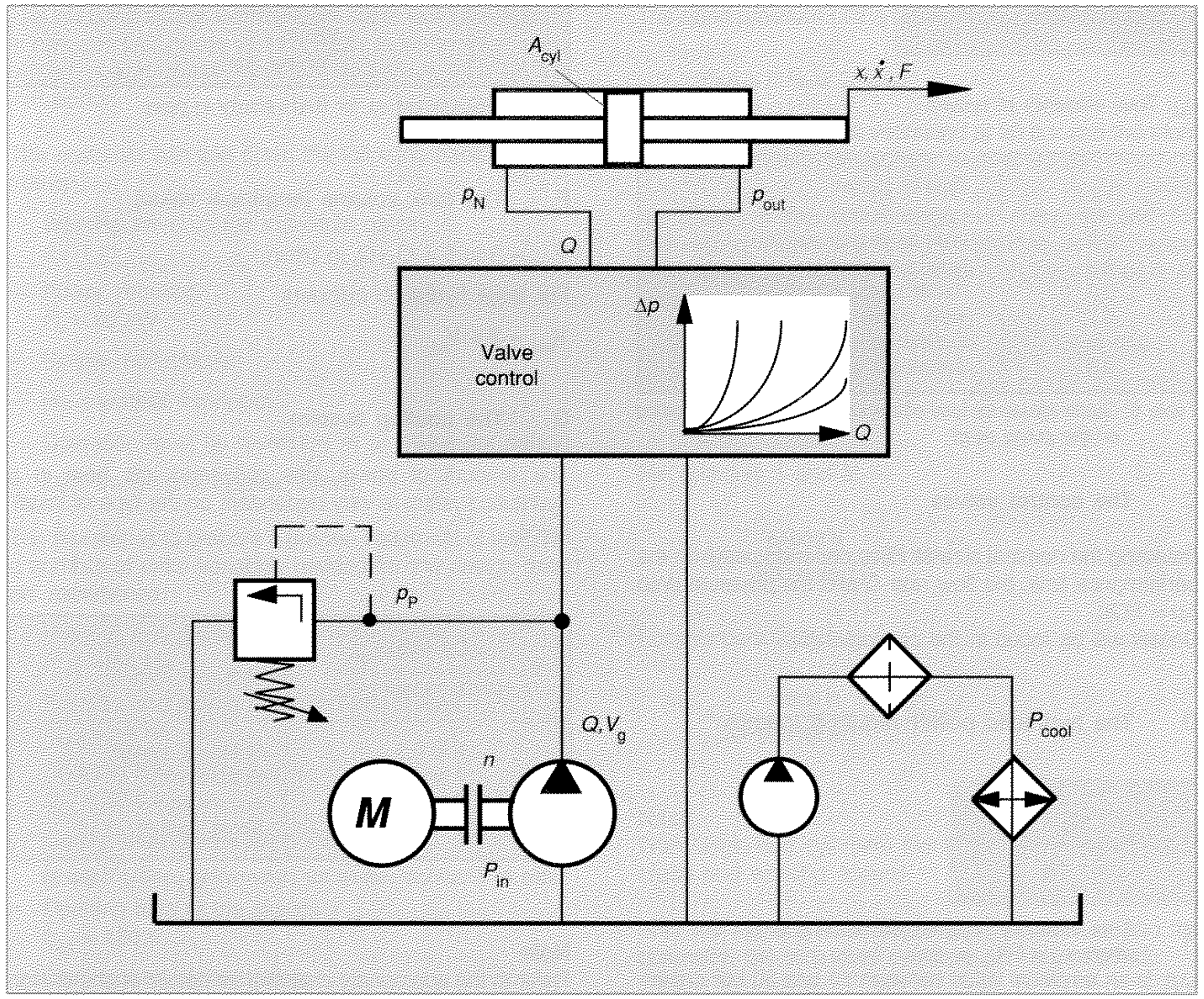


Fig. 17: *Diagrammatic layout of a hydraulic installation;* $x, \dot{x}, F$ *are the output performance values required by the user*

The procedure for designing a system

A The hydraulic cylinder

$$F = p_N^* \cdot A \quad \Rightarrow \quad A = F/p_N^*$$

p_N^*: Provisional system pressure (see *Table 1* for empirical values)

The size of a suitable hydraulic cylinder can be taken from the catalogue [5] ($A_{cyl} > A$).

B The power unit

$$Q_p \geq Q_{cyl} \qquad Q_{cyl} = A_{cyl} \cdot \dot{x} \qquad Q_p = V_g \cdot n \cdot \eta_v$$

V_g: Pump displacement (geometric)
n: Speed
η_v: Volumetric efficiency, see catalogue [5]

A suitable combination of hydraulic pump (V_g) and electric motor (n) must be selected to satisfy the requirements.

$$P_{in} \geq \frac{P_{out}}{\eta_{tot}} = \frac{F \cdot \dot{x}}{\eta_{tot}}$$

η_{tot}: Total efficiency of system

η_{total}	Type of pump
0,8	Internal gear pump
0,8	Axial piston pump
0,8	Radial piston pump
0,7	Vane pump
0,65 to 0,7	External gear pump
0,5	Centrifugal pump

Table 6: *Guide values for total efficiency of a system depending on the pump used*

C The control valves

Selecting the nominal size of valve depends partly on the performance limit, the flow rate Q and the pressure drop $\Delta p = f(Q)$.
Details will be found in the catalogue [5].

D The hydraulic power required (pump)

$$P_P = \frac{Q_P \cdot p_P}{\eta_t}$$

η_t: Total efficiency of pump, see catalogue [5]
where $p_P = F/A_{cyl} + \Delta p_v$
Δp_v is an empirical value between 10 and 50 bar depending on the type of installation. It is related to the flow velocity and to the type and length of pipe.

$$P_P < P_{in}$$

E Pipe sizes (empirical values)

$$d_{Pipe} = \sqrt{\frac{4 \cdot Q_P}{\pi \cdot v_{max}}}$$

v_{max}: This depends on the pipe length, operating pressure and installation details

v_{max} in m/s	for
0,5 to 1,0	Suction line
3	Return line
5	Delivery line

Table 7: *Guide values for fluid velocity in pipes*

F The tank volume (empirical value)

$V_{tank} = 3$ to $5 \times Q_p$ (Q_p in L/min)
For large hydraulic cylinders
$V_{tank} = 3 \times V_{cyl}$

G The cooling system

The amount of power loss converted into heat in a hydraulic system is very dependent on the type of installation.

$x = \dfrac{\text{Cooling } P_{cool}}{\text{pump capacity } P_P}$	Hydraulic system
0,1 to 0,3	Standard system
0,2 to 0,5	System with accumulator
0,5 to 0,9	Servo system

Table 8: *Empirical values for determining cooling capacity*

Note
Refer to the appropriate chapters for details on the design of hydraulic accumulators and the fitting and filtration rating of filters.

7 An example of project design

The following simple example of a canal lock gate system is intended to illustrate the procedure to be adopted in the project design of the hydraulic system in such an the installation.

The customer has laid down the following basic requirements:

- Working travel of the individual gates
  $s = 1$ m
- Power required
  $P_N = 80$ kW
- Simple construction
- No electrical interlocks
- Operation must be reversible at any time (for safety)
- Lock transit time approx. 5 min.

The first step is to prepare a precise statement of task, i.e. a number of extra variables needed for the design must be ascertained:

- Pushing and pulling forces on the gates
  $F_{N1,3} = 64$ kN
- Pushing and pulling forces on the sluices
  $F_{N2,4} = 14$ kN
- Opening time of the gates
  $t_{A1,3} = 30$ s
- Closing time of the gates
  $t_{Z1,3} = 60$ s
- Opening time of the sluices
  $t_{A2,4} = 180$ s
- Closing time of the sluices
  $t_{Z2,4} = 30$ s

If there is no background from previous similar installations it will be essential to draw up a circuit diagram *(Figs. 18 and 19)* and a function diagram *(Fig. 20)*.

As the system will operate intermittently and work at low pressure, as well as having to be simple and cheap, the open–loop circuit is the one to choose according to Table 3.
The operating pressure is decided according to DIN 19704 – the standards referring to hydraulic steel structures – and is set at 120 bar max. because of the emergency and manual operation required.

The rated pressure of the system is calculated as follows:

$$p_{Anl.\,N} = p_{max} - p_{Anl.V},$$

whereby the losses are estimated at 30 bar and therefore the usable system pressure is fixed at 90 bar.

As shown in the circuit diagrams *(Figs. 18 and 19)* hydraulic cylinders are being considered as the output drive elements. The cylinder size can be estimated as follows, neglecting losses:

$$F_N \approx p \cdot A \qquad \text{or}$$

$$A \approx F_N / p$$

For cylinders 1 and 3:

$$A_{1,3} \approx \frac{64000\ \text{N}}{900\ \text{N/cm}^2} \approx 71\ \text{cm}^2$$

For cylinders 2 and 4:

$$A_{2,4} \approx \frac{14000\ \text{N}}{900\ \text{N/cm}^2} \approx 15{,}5\ \text{cm}^2$$

Note:
With single–rod cylinders there are two different piston areas and working speeds.

With the aid of the catalogue [5], the pressure and the geometry, the individual cylinders can be selected:

Cylinder 1	CD 250 B 125/70 x 1030
Cylinder 2	CD 250 B 63/45 x 350
Cylinder 3	CD 250 B 125/70 x 1030
Cylinder 4	CD 250 B 63/45 x 350

Remember that, according to the formula $Q = v \cdot A$, the choice of hydraulic cylinder also has an effect on the size of the pump. Once the operating pressure, volumetric flow and circuit diagram are known, the sizes of the individual components can be ascertained, and hence the components themselves, e.g. the valves.

Knowing the output power and the losses of the individual control devices, and hence the drive power required for the hydraulic system, it is possible to establish the power unit, comprising pump, tank and accessories such as filter and breather. For reasons of cost, standard units should be installed whenever possible With this particular project standard components can be selected and mounted on a special size of tank.

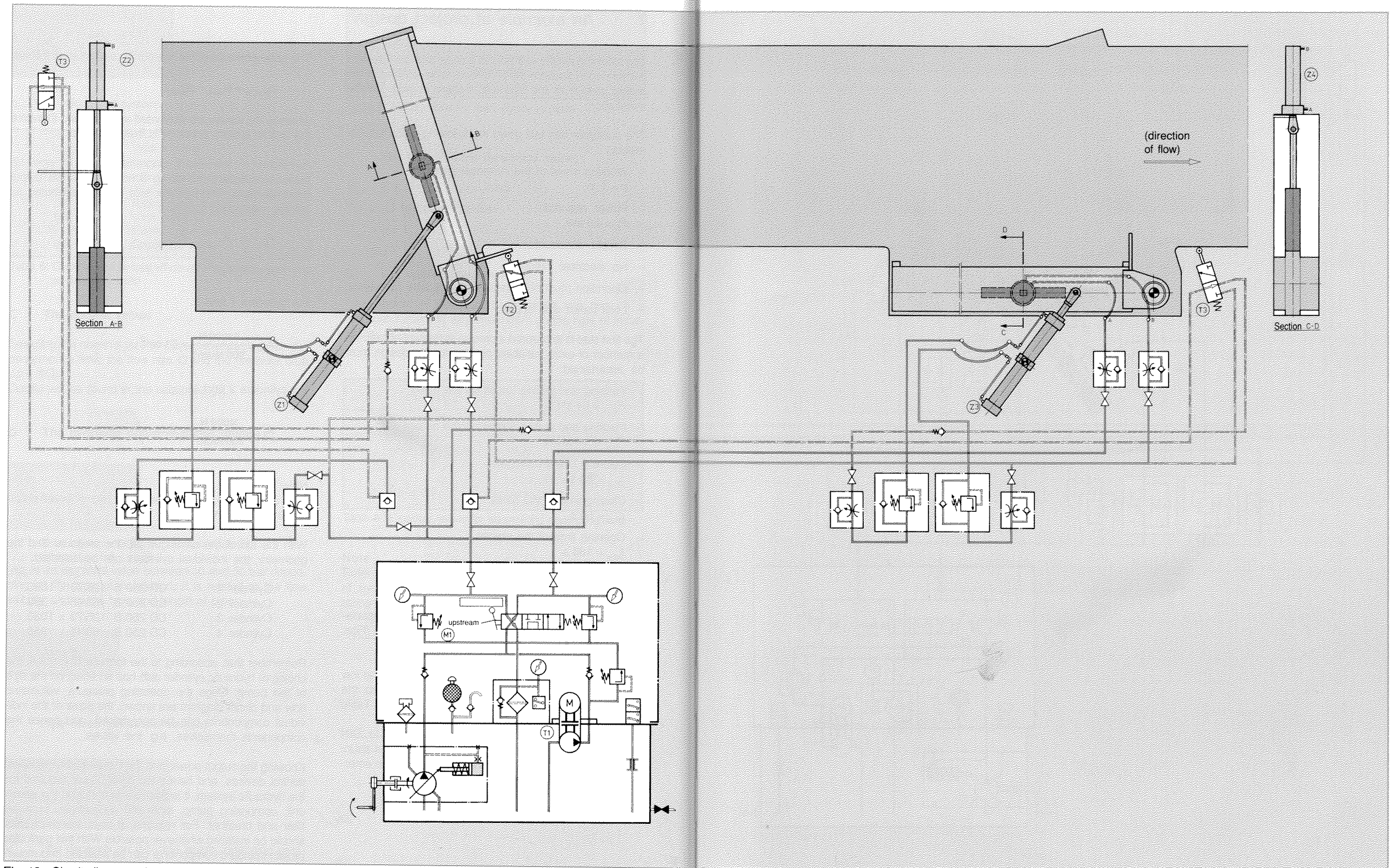


Fig. 18: *Circuit diagram of a canal lock (travelling upstream)*

Planning Hydraulic Systems

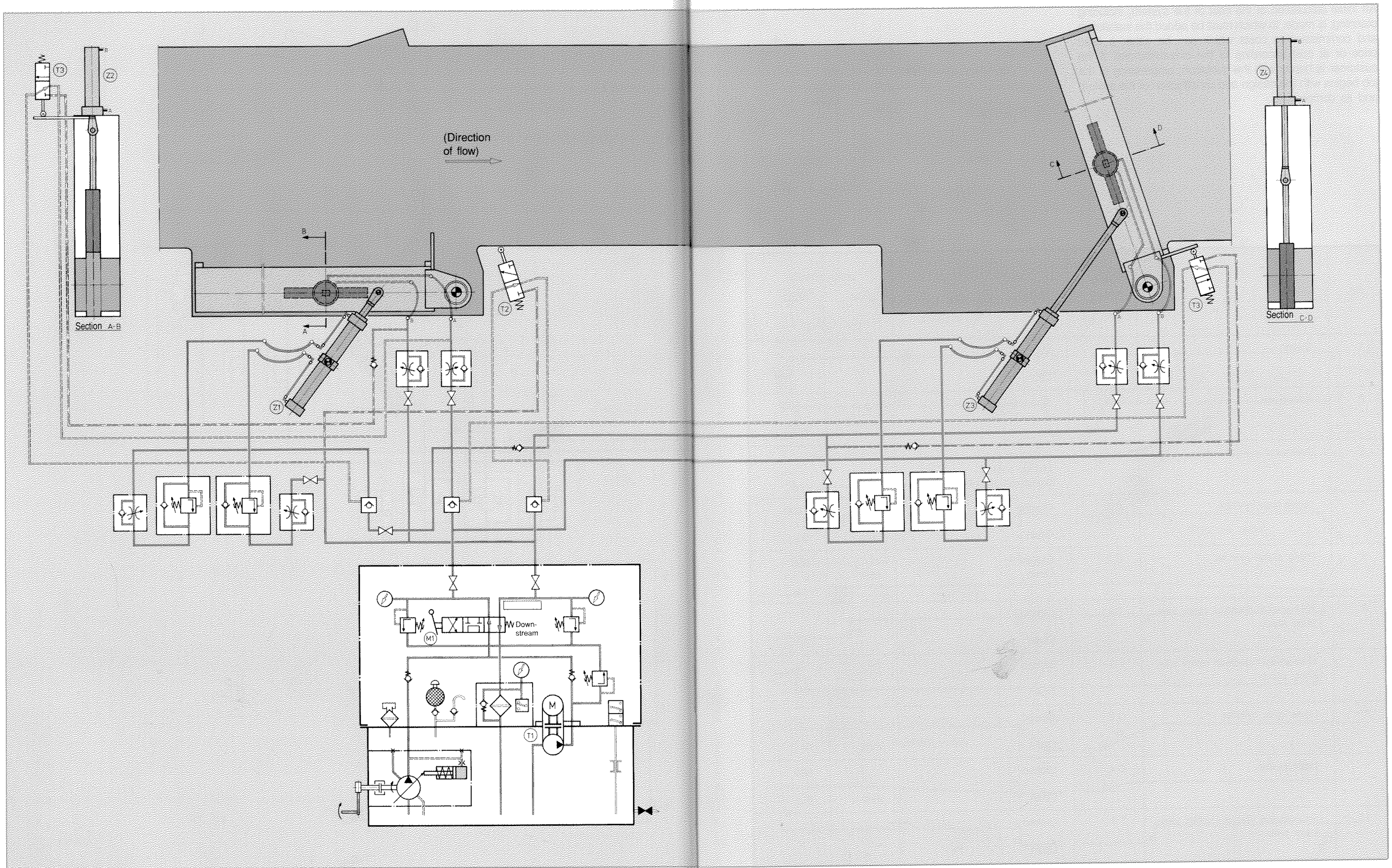


Fig. 19: *Circuit diagram of a canal lock (travelling downstream)*

An initial calculation of the cost of the system, including planning, is made, to which must be added the installation and commissioning costs which can be either a fixed price or at cost according to the circumstances. If the customer is happy with the quotation, progressing of the job begins with the design and construction of the system and its components.

No.	Item	Designation	Status	Units	Value
1	Switch	1.1	On Off		
2	Power unit	T 1	Running	r.p.m.	1500 0
3	Directional control valve	M 1	Upstream Middle position Downstream (manual actuation)		
4	Upper gate cylinder	C 1	Closed Open	mm	1030 0
5	Filling sluice cylinder	C 2	Closed Open	mm	350 0
6	Roller-actuated directional control valve, filling sluice	T 2	Closed Open		
7	Roller-actuated directional control valve, filling sluice	T 3	Closed Open		
8	Lower gate cylinder	C 3	Closed Open	mm	1030 0
9	Drain sluice	C 4	Closed Open	mm	350 0
10	Roller-actuated directional control valve, lower gate	T 3	Closed Open		

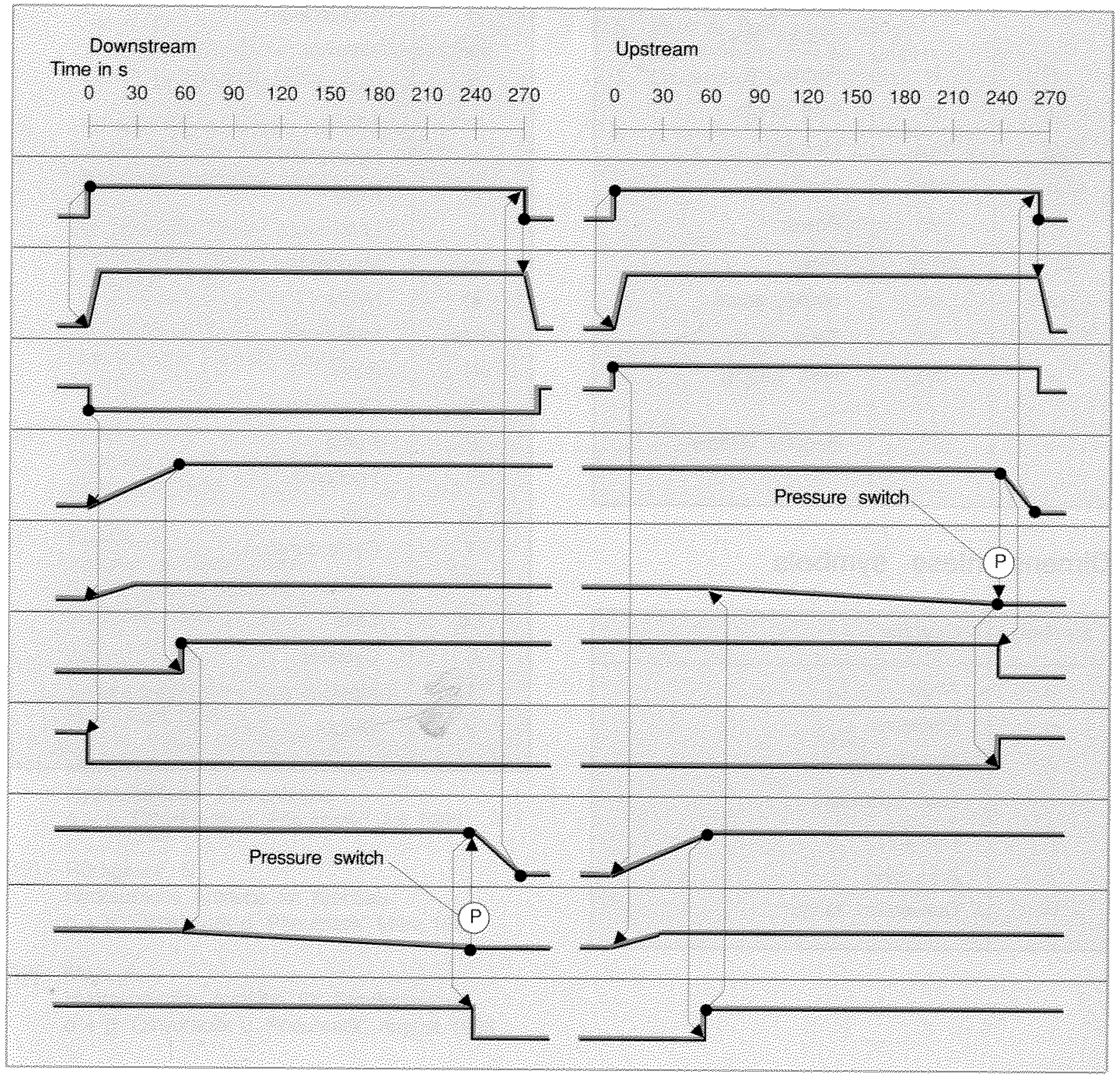


Fig. 20: *State diagram for the canal lock in Fig. 17*

8 Symbols and subscripts

Symbols

Symbols	Units	Quantity
a	mm^2	Opening area, Annulus area
A	mm^2	Area, piston area
d, D	mm	Diameter (piston, pipe)
F	N	Force
l	m	Pipe length
M	Nm	Moment
p	N/m^2	Pressure
P	W	Power
Q, V	m^3/s	Flow
s	m	Stroke, travel
t	s	Time
v	m/s	Flow velocity
ω	s^{-1}	Angular velocity

Dimensionless symbols

Symbols	Quantity
α	Angle
η	Efficiency
λ	Pipe friction
ξ	Loss coefficient
ρ	Density
φ	Area ratio
W	Conversion range

Subscripts

Symbols	Quantity
A	Utilization
out	Drive out
in	Drive in
sys	System
b	Acceleration
e	effective
G	Weight
tot	Total
h	Hydraulic
Pi	Piston
L	Load
m	Mechanical
M	Motor
max	Maximum
N	Rated
Op	Operating
P	Pump
Fr.	Friction
Cl	Gap
T	Cycle duration
th	Theoretical
Lo	Loss
C	Cylinder
1	Input
2	Output

9 References

[1] VDI-Richtlinien VDI 3260
Funktionsdiagramme von Arbeitsmaschinen und Fertigungsanlagen
VDI-Verlag GmbH Düsseldorf 1977

[2] Ebertshäuser, H.
Planung hydraulischer Anlagen
o + p "Ölhydraulik u. Pneumatik" (1984)

[3] Obermeyer, G.
Kräftebetrachtung an einem differentialgeschalteten Druck- und Zugzylinder
o + p "Ölhydraulik u. Pneumatik" Oktober 1966

[4] Krämer, O.
Zylinderentwicklung für Kunststoffmaschinen
Rexroth Fachtagung 1985

[5] Hydraulik Komponenten, RD 00 101
Katalog über das Lieferprogramm der Unternehmensgruppe Mannesmann Rexroth

[6] Der Hydraulik Trainer, Band 1
Mannesmann Rexroth GmbH
2. Auflage 11.80

[7] Der Hydraulik Trainer, Band 2
Mannesmann Rexroth GmbH
1. Auflage 1.86

[8] 7. A.F.K.
Energiesparende Hydraulik
Fluid, April 1986

[9] Roorda, T.
Verluste auf Sparflamme
(Load-Sensing-System)
Fluid, Mai 1986

[10] Nikolaus, H.W.
Sekundärgeregelte Antriebstechnik
Fluid, April 1985

[11] Mayr, A.
Hydrostatische Fahrantriebe
o + p "Ölhydraulik u. Pneumatik" Juli 1985

[12] Rückgauer, N.
Hydraulische Antriebe im Kranbau
o + p "Ölhydraulik u. Pneumatik" März 1986

[13] Mayr. A.
Hydrostatische Antriebe für Straßenwalzen
o + p "Ölhydraulik u. Pneumatik" August 1983

[14] Wiest, W.
Hydrostatische Antriebe für Landmaschinen u. Traktoren
o + p "Ölhydraulik u. Pneumatik" Juli 1983

[15] Metzner, T.
Kennwerte der Dynamik sekundärdrehzahlgeregelter Axialkolbeneinheiten am eingeprägten Drucknetz
o + p "Ölhydraulik u. Pneumatik" 30 (1986) Nr. 3

[16] Kordak, R.
Sekundärgeregelte hydrostatische Antriebe
o + p "Ölhydraulik u. Pneumatik" 29 (1985) Nr. 9

[17] Was ist Sekundärregelung?
Mannesmann Rexroth GmbH
Druckschrift Nr. RD 09481

[18] Auswahlkriterien von Hydropumpen
o + p "Ölhydraulik u. Pneumatik" Sept. 1980

Notes

Hydraulic Fluids

Peter Drexler

1 Introduction

Theoretically, any fluid can be used as the operating medium in hydraulic systems since they all follow Pascal's Law.

Water was used in the early days but since, in its pure form, it does not prevent wear and, in conjunction with oxygen, can cause serious corrosion it is unsuitable for the demands of modern hydraulic power systems.

Mineral oil has come to be the most popular choice primarily because of its good lubricity and excellent protection against corrosion. With the help of special additives the mineral oils have been continually improved over the years.

One of its disadvantages, however, is that it is flammable. Consequently, fire-resistant fluids are often used in hydraulic systems operating in the vicinity of naked flames, molten metal or other high temperature areas.

There is no such thing as the ideal hydraulic fluid. This means that careful selection is absolutely essential in order to find the fluid that will give the best performance for a particular hydraulic system.

2 Requirements of hydraulic fluids

Numerous demands are made on hydraulic fluids, the most important ones being listed below:

- good lubricity
- no corrosion of materials
- good viscosity-temperature characteristics
- good resistance to oxidation and thermal loading
- low compressibility
- minimum foaming
- high specific gravity
- good thermal conductance
- fire resistance for special applications
- non-toxic
- cheap
- wide availability
- low maintenance cost
- easy disposal

3 Properties and selection criteria of mineral oil-based hydraulic fluids

Most hydraulic power systems use hydraulic fluids which have a mineral oil base. Choosing the correct fluid is essential to satisfactory operation of the system, the operating conditions of which must be carefully researched.
The properties of hydraulic fluids depend on:

- the type of base oil
- the amount of refining
- the type and quantity of additives.

Fluids with good low temperature flow characteristics are made from naphtha-based oils. They are used in hydraulic systems with low starting temperatures and maximum fluid temperatures up to 30 °C .

If the low temperature flow characteristics are not the governing factor for selection, it is better to use paraffin-based fluids which possess greater resistance to oxidation and have a better viscosity-temperature characteristic.

A very popular course of action, however, is to blend naphtha-based oils and paraffin-based oils with aromatics in order to obtain the widest possible range of application for the fluids.

Undesirable constituents in the base oil, such as sulphur compounds, are removed by the refining process. Special additives improve the properties of the resultant fluids, such as protection against wear.

The minimum specifications for hydraulic fluids are laid down in DIN 51 524.

3.1 Classes of hydraulic fluids

3.1.1 HL fluids to DIN 51 524, Part 1

HL fluids are oils whose resistance to ageing and corrosion protection have been improved by means of additives.

They are used in hydraulic power systems where temperatures around 50 °C and/or corrosion due to moisture are anticipated.

Since they contain no wear-reducing additives they are of only limited suitability for hydraulic installations. The limitations refer to the actual equipment selected; primarily the pumps and motors and the pressure range. There is no point in making exclusive statements of applicability since equipment is being constantly improved. Manufacturers' literature should be studied in order to obtain up-to-date information.

Hydraulic fluids which attack lead or bearing materials containing lead may not be used even if they satisfy the HL specification to DIN 51 524, Part 1. They are mainly multi-purpose oils, e.g. bedway oils, containing fatty acids or fatty acid esters.

3.1.2 HLP fluids to DIN 51 524, Part 2

HLP fluids have better protection against wear compared with HL fluids. They contain ageing-prevention and corrosion-prevention additives and others for reducing wear under mixed friction conditions, i.e. where excessive wear or seizure can occur between metal surfaces if the lubrication is inadequate.

The wear protection is assessed by means of the tests laid down in DIN 51 354, Part 2 and DIN 51 389, Part 2. The results cannot be compared with each other due to the different conditions of the tests.

Again, HLP fluids must not be used if they attack lead or bearing materials containing lead.

3.1.1 HV fluids

Hydraulic installations exposed to severe fluctuations in temperature or low ambient temperatures, e.g. outdoors, need fluids with a higher viscosity index (VI); they are known as HV fluids. Some of them satisfy the requirements of HLP fluids to DIN 51 524, Part 2 but have additional additives for improving the viscosity-temperature characteristic; these are known as VI improvers. VI improvers can sometimes adversely affect the de-mulsification characteristics and air separation capability and so are only recommended for installations exposed to the temperature conditions mentioned earlier. A new standard DIN 51 524, Part 3 is currently in preparation to lay down the minimum requirements.

In selecting HV fluids a shear loss in the viscosity of up to 30% must be taken into account. This means, for example, that an HV fluid with a viscosity of 36 mm²/s must be used for a pump with a permitted minimum viscosity of 25 mm²/s so that the viscosity does not fall below the permitted minimum when shear losses occur during service.

3.1.4 HLP-D fluids

These fluids contain detergent and dispersion additives. Their purpose is to dissolve any deposits and to hold any impurities (e.g. due to ageing and wear) and moisture in suspension in the fluid.

The impurities have to be filtered out so the filter area has to be increased (design for Δp = 0.2 bar) and the filtration rating reduced by 1 step from, say, 20 to 10µm. This normally doubles the size of the filter, e.g. from Size 330 to Size 660.

The moisture trapped in the fluid can reduce the protection against wear so HLP-D fluids should not be used if the ingress of more moisture than normal is expected, e.g. in wet surroundings.

Some HLP-D fluids contain fatty acids or fatty acid esters which attack lead or bearing materials containing lead so they should not be used.

3.1.5 Anti-pollution fluids

With the growing public awareness of the need for conservation and stricter legislation to protect the environment there is a growing demand for anti-pollution hydraulic fluids, primarily for mobile equipment. Bio-degradable fluids currently available on the market can be divided into two groups:

- fluids with a vegetable base
- fluids with a glycol base.

The materials selected for the equipment must be compatible with the new fluids, e.g. seals, lead content, paintwork.

3.1.6 Multi-purpose oils

Some multi-purpose oils comply with DIN 51 524 and so can be used as hydraulic fluids as well as for such applications as lubricating bedways.

Before using such oils it is advisable to discuss material compatibility with the maker of the equipment.

3.2 Selection

Selecting the correct fluid is just as important to the safe and reliable functioning of a hydraulic system as selecting the correct components.

DIN 51 524, Parts 1 and 2 can be used for choosing the principal data although it should be remembered that these are only minimum requirements. Therefore, it will be necessary to check manufacturers' data concerning such aspects as resistance to ageing, foaming, lead and non-ferrous metal compatibility, purity as-supplied and filtration capacity.

3.2.1 Viscosity

The viscosity of a hydraulic fluid is a measure of its thickness, i.e. the amount of resistance that the fluid particles offer to a thrust exerted on them.

Kinematic viscosity as specified in DIN 51 562 has become the most popular unit of measurement and, in SI units, it is expressed in mm²/s, whereby 1 mm²/s = 1 cSt. If the viscosity is too high the friction and flow losses will also be high as measured by pressure drop and temperature rise. Cold starting of the hydraulic system will be difficult, control will be sluggish and the air separation capacity will be poor.

If the viscosity is too low there will be excessive leakage, more wear and severe overheating of the fluid.

The viscosity falls as the temperature rises. *Diagram 4* shows the limits of the different ISO viscosity classes for liquid lubricants to DIN 51 519.

The change in viscosity with temperature is expressed by the viscosity index which is specified in DIN ISO 2909. The higher the viscosity index the lower the sensitivity of the viscosity to temperature.

This feature is utilized with HV fluids which are designed for large fluctuations in temperature and low ambient temperatures. It means that it is sometimes possible to avoid changing the fluid in a system as the annual seasons change, such as when the system is outdoors.

The viscosity-pressure behaviour of hydraulic fluids is of more importance when operating pressures are higher. Although the increase in viscosity can be low up to a pressure of 200 bar, it can double when the pressure reaches around 400 bar.

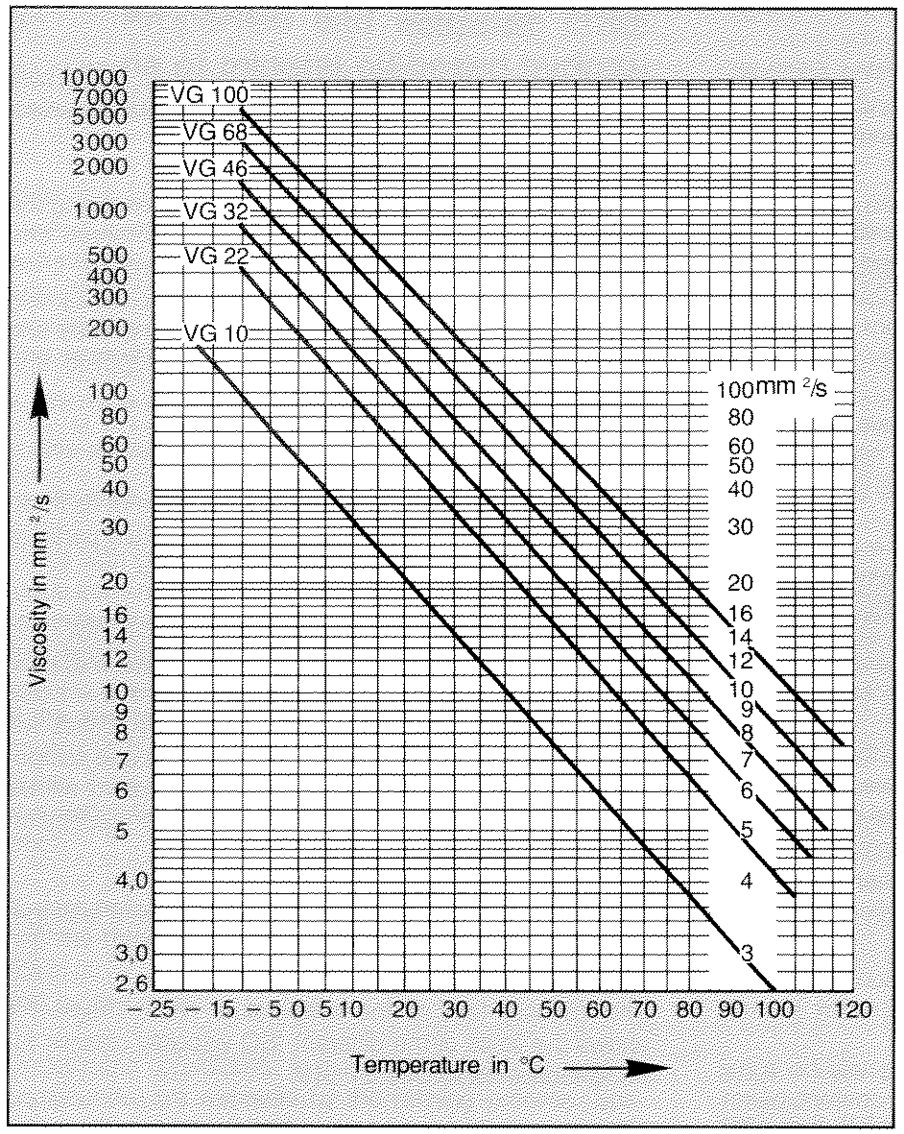


Diagram 4: *Viscosity-temperature diagram*

ISO viscosity class	Mean viscosity at 40,0 °C mm²/s (cSt)	Limits of kinematic viscosity at 40,0 °C mm²/s (cSt) minimum	maximum
ISO VG 10	10	9,0	11,0
ISO VG 22	22	19,8	24,2
ISO VG 32	32	28,8	35,2
ISO VG 46	46	41,1	50,6
ISO VG 68	68	61,2	74,8
ISO VG 100	100	90,0	110,0

Table 9: *ISO viscosity classes*

The viscosity classes for hydraulic fluids are laid down in DIN 51 519 which is based on ISO 3448. ISO viscosity classes VG 10, 22, 32, 46, 68 and 100 have also been transferred to DIN 51 524.

Hydraulic equipment manufacturers' data must be taken into account when selecting viscosity classes.

Example:
Viscosity range for a vane pump:
800 mm²/sec maximum for starting with delivery
200 mm²/sec maximum for starting at zero stroke
16 mm²/sec minimum at maximum operating temperature.

Exceeding the maximum values can cause damage due to insufficient lubricant, whereas values below the minimum can cause excessive wear and leakage.

3.2.2 Pour point

Pour point is the lowest value of temperature at which the fluid will still flow. The method of determining the pour point is described in DIN ISO 3016.

In selecting a hydraulic fluid, remember that the minimum permitted temperature in the hydraulic system must be at least 8°C above the pour point.

3.2.3 Compressibility

Compressibility is the amount by which the volume of a hydraulic fluid changes under pressure.

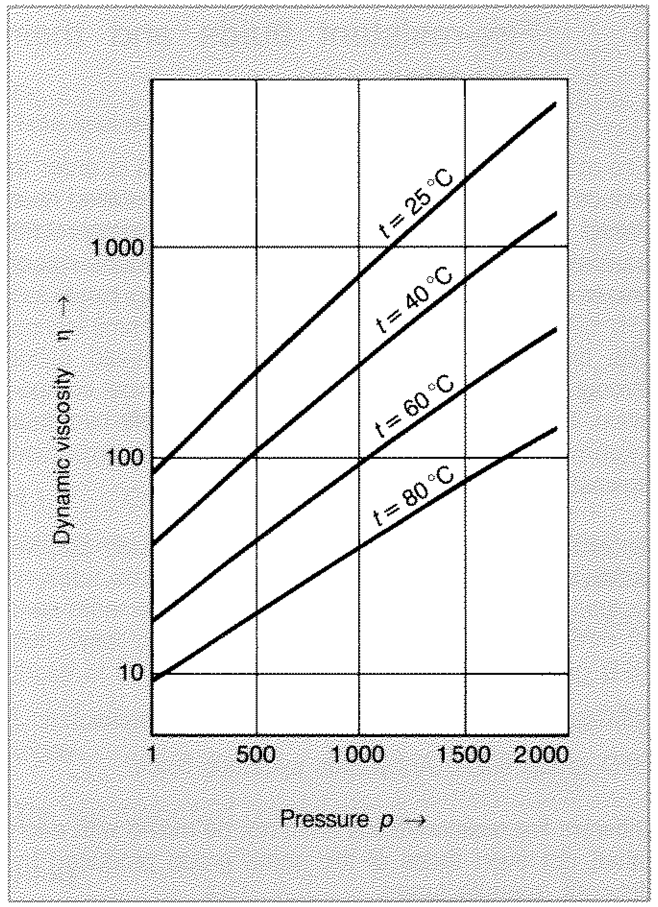


Diagram 5: *Temperature-pressure relationship of viscosity [2]*

In the case of a fluid containing no air bubbles the volume reduces by 0.7% when the pressure is increased by 100 bar.

Up to 150 bar the compressibility can be neglected but beyond that, especially when delivery rates are high, it can have an adverse effect on the functioning of the system. If there is any air trapped in the fluid it will be more compressible and can give rise to disturbances such as noise, vibration and jerky motions at pressures as low as 50 bar.

3.2.4 Air separation capacity

All hydraulic fluids contain dissolved air. If the saturation limit is exceeded during a drop in pressure, e.g. downstream of a throttle, air bubbles will separate out.

Air can also find its way into the fluid from outside; a typical point of entry is at leaks in suction lines. This undissolved air changes the compressibility, reduces the protection against wear and decreases the thermal conductivity. The consequences are operating disturbances due to jerky motions, noise, vibration and material damage.

It is obvious that air bubbles must be removed from the fluid as quickly as possible.

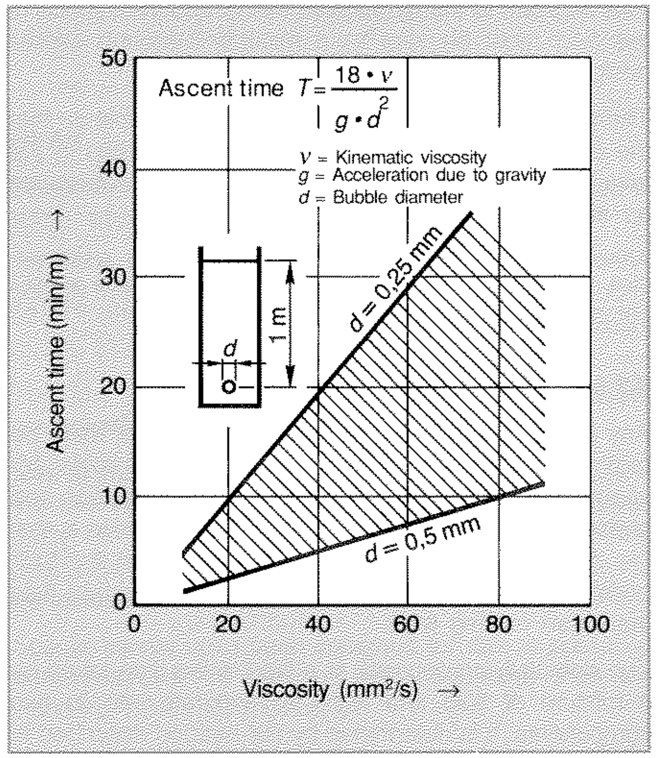


Diagram 6: *Ascent time of air bubbles in mineral oil (according to Hayward) [2]*

The air separation capacity of fluids is determined according to DIN 51 381. The method measures the amount of time taken for the dissolved air in the fluid to be separated down to 0.2% by volume.

The tables in DIN 51 524, Parts 1 and 2 list the maximum times for 50°C referred to the fluid viscosities.

ISO VG 10 ISO VG 22 ISO VG 32	maximum 5 min
ISO VG 46 ISO VG 68	maximum 10 min
ISO VG 100	maximum 14 min

Tabelle 10: *Air separation capacity in minutes at 50°C according to DIN 51 524, Parts 1 and 2*

3.2.5 Foaming

Foaming due to the air bubbles that rise to the surface of the fluid must be kept to a minimum through careful design of the tank.

The surface of fluid must be as large as possible so that the air bubbles can escape quickly. Baffle plates in the tank are just as effective in improving the air separation as air-separating strainers.

Although hydraulic fluids contain anti-foaming additives, impurities such as water, dirt and the products of ageing increase the tendency of the fluid to foam.

3.2.6 Demulsifying capacity

Any water that penetrates into hydraulic fluid must be removed as quickly as possible because it has an adverse effect on the viscosity and corrosion protection and causes deposits. The important factor in this case is that the dwell time must be as long as possible in the tank because water separates out faster from stagnant fluid than from moving fluid.

The demulsifying capacity of a fluid is the time taken by a mixture of fluid and water to separate into its two components. It is determined by the method described in DIN 51 599.

3.2.7 Resistance to oxidation

The ageing of hydraulic fluid depends on its composition and can vary between fluids although they are of the same standard. The process is accelerated by the air dissolved in the fluid at high pressures, the temperature and the metals with which the fluid comes into contact as well as by impurities such as dirt, rust and water.

The products of ageing can make valves sticky and block filters and heat exchangers. The demulsification capacity is reduced, as is the protection against corrosion and wear.

A longer dwell time for the fluid in the tank, good filtering and cooling together with regular checking of the fluid quality can counteract these effects.

3.2.8 Corrosion protection

Hydraulic fluid not only has to prevent the rusting of steel components it also has to be compatible with non-ferrous metals and alloys.

The corrosion protection properties for steel can be determined according to DIN 51 585 and for copper according to DIN 51 587.

Hydraulic fluids which attack lead or bearing materials containing lead should not be used even though they might satisfy the minimum requirements of DIN 51 524.

4 Fire-resistant fluids and their selection

Fire-resistant fluids were originally developed in order to reduce the fire risk from hydraulic systems installed near naked flames, molten metal or other high temperature equipment or in otherwise hazardous areas with a risk of fire or explosion.

The fire resistance of such fluids is achieved either through its water content or through its chemical composition. Pure water is no longer used in modern hydraulic systems due to its low viscosity and poor protection against wear and corrosion.

According to VDMA 24 317 fire-resistant fluids can be subdivided as follows:

HFA group fluids: Oil-in-water emulsion

HFB group fluids: Water-in-oil emulsion

HFC group fluids: Solutions of polymers in water

HFD group fluids: Anhydrous synthetic fluids.

Different fire-resistant fluids must not be mixed with each other, not even those of the same group produced by different manufacturers.

In all other respects the specification of the fire-resistant fluids is the same as that of mineral oils, the fire resistance is simply an additional property.

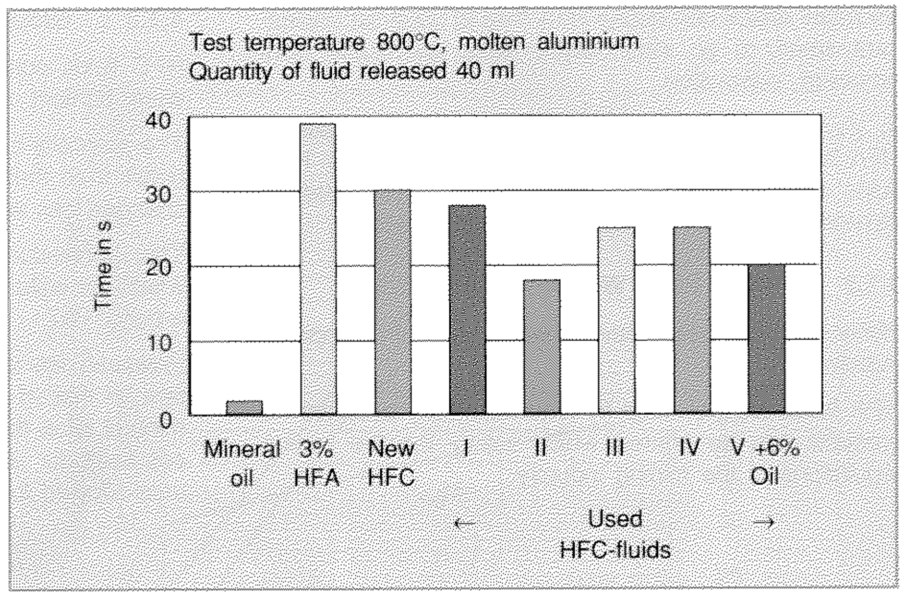


Fig. 21:
Time from release of fluid to ignition [3]

"Fire-resistant" does not mean inflammable. It simply means that there is a delay before fluid released on to a hot object ignites. The purpose of the delay is to allow personnel time to escape from the danger zone *(see Fig. 21)*.

4.1 Fluid groups

4.1.1 Group HFA: Oil-in-water emulsions

With a maximum flammable component of only 20% oil-in-water emulsions would be the ideal hydraulic fluids if their other properties such as viscosity, corrosion protection, wear protection and pour point were the same as those of mineral oil.

Unfortunately, such a perfect fluid is still not yet available today.

The two major fluid groups are:

HFAE

a Oil-in-water emulsion comprising an emulsifying oil and water. Used mainly in mining. The specification is given in DIN 24 320.

b Micro-emulsion with organic chemicals. Mainly used in industrial equipment due to the better wear protection than **a)**. Fluids with a water content of 95% by weight have proved the most popular (HFA 95/5).

HFAS

Solutions of salts in water or organic esters dissolved in water. Most are not compatible with mineral oil.

Due to the high water content the fire risk is less than with other fluids if, as in most cases, the oil content is restricted to 3 to 5%.

The low viscosity is a disadvantage since it causes greater internal and external leakage.

Due to the inferior protection against wear and corrosion, the low viscosity and the high vapour pressure, valves designed for mineral oil can only be used for operating pressures up to 70 bar and a reduced service life has to be taken into account when planning a system.

Special valves have been designed for use at operating pressures over 70 bar but they cost more than the standard valves for mineral oil.

Designers must also ensure that flow velocities at throttling points are low and the materials are rustproof. And they must remember that leakage with spool valves can be high and that clean fluids are essential because of the close tolerances.

The prescribed mixing ratio of the emulsion concentrate must be checked; any deviations can affect the stability of the emulsion, its effect on the seal materials and the anti-corrosion capacity. Increasing the concentration can lead to greater attack of the seals and to more corrosion of zinc. Reducing the concentration spoils the corrosion protection for steel.

Microbial attack can cause HFA fluid to have an unpleasant smell, to go slimy, block filters and suffer from emulsion separation.

In view of these substantial disadvantages the use of HFA fluids is restricted to a very few applications. HFA fluids are currently being developed with additives for increasing the viscosity up to 40 mm²/s at 40°C .

4.1.2 Group HFB: Water-in-oil emulsions

Water-in-oil emulsions have a water content of approximately 40%.

HFB fluids are not used in West Germany because no products can be offered which satisfy the fire testing regulations of the mining industry.

4.1.3 Group HFC: Aqueous polymer solutions

HFC fluids achieve their fire resistance by means of a water content of about 35 to 50%.

The viscosity-temperature characteristic of the fluids is better than that of normal mineral oils, i.e. the viscosity changes less with increasing temperature *(see Diagr. 7)*.

This behaviour is expressed with the viscosity index which, for water glycol, is over 150 and, for mineral oil, is around 100.

However, the viscosity-pressure behaviour is very different to that of mineral oil. HFC fluids are poorer than mineral oil in this respect *(see Diagram 8)*. This expresses itself, for example, in the reduced operating pressures allowed for pumps.

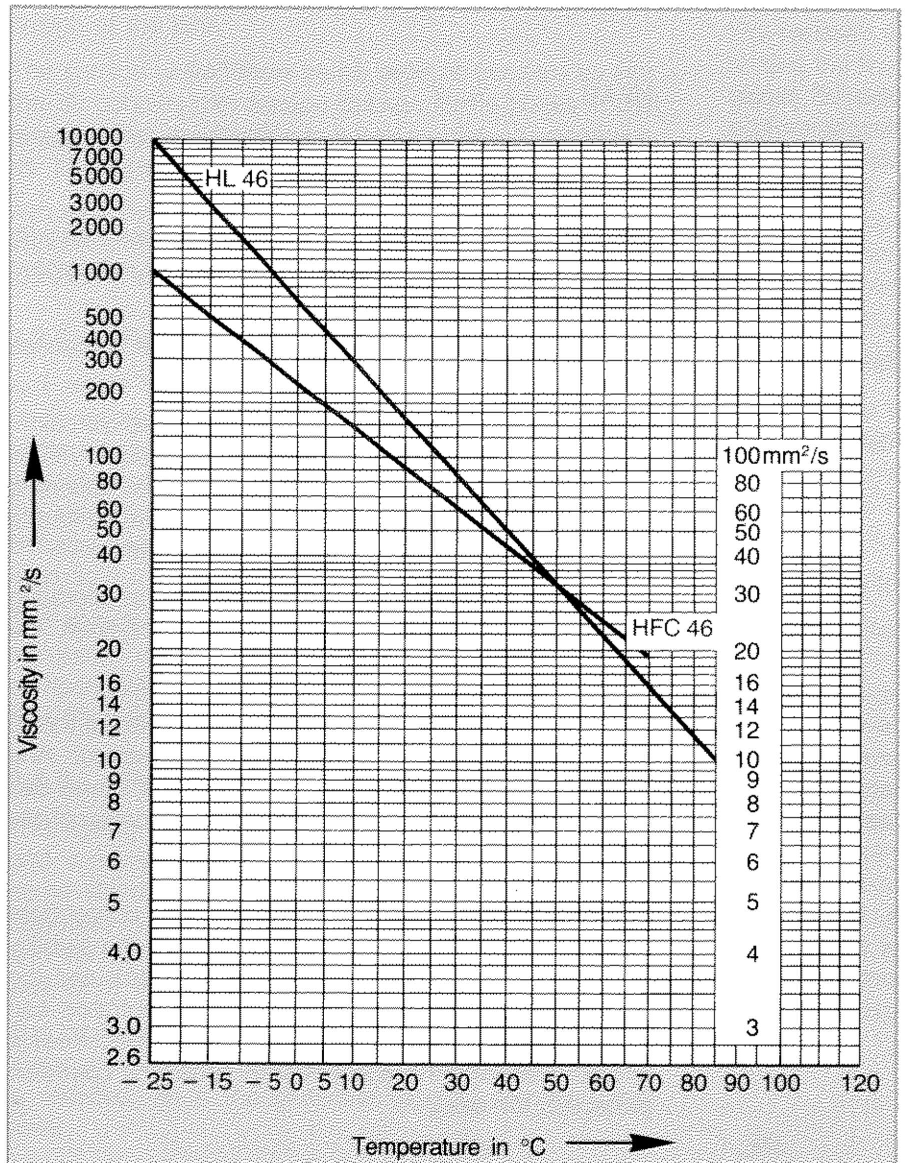


Diagram 7: *Viscosity-temperature characteristic of HFC 46 water-glycol fluid with 45% water content compared with HL 46 mineral oil [4]*

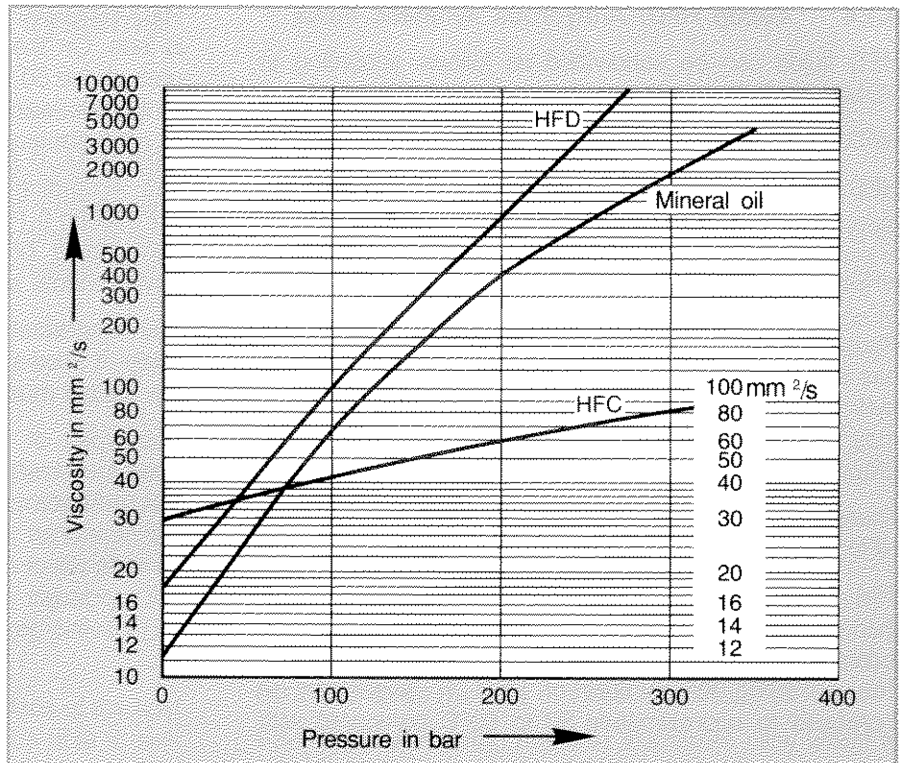


Diagram 8:
Viscosity-pressure characteristics of HFC water-glycol fluid compared with mineral oil and HFD phosphate ester [4]

The ability of HFC fluids to dissolve air is considerably less than that of mineral oil. This means that hydraulic systems using water-glycol fluid have a greater tendency to cavitation and erosion than those using mineral oil. An essential prerequisite for the satisfactory operation of hydraulic systems using water-glycol fluid is to adhere to the fluid supplier's instructions for the monitoring and care of the fluid and also checking of the free acid content. The monitoring covers checking of the viscosity of the water content, the reserve alkalinity of the pH value and the amounts of solid and liquid contaminants. The additional check on the free acid content still has to be the subject of separate agreement at the moment.

Free acids, e.g. formic acid and acetic acid, are produced by the HFC fluids as products of ageing. They render the corrosion and wear protection ineffective.Their concentration must not exceed 0.15 %.

The reserve alkalinity is gradually reduced as the fluid ages. There is no point in readjusting it since the ageing of the water-glycol continues just the same.

HFC fluids should be checked at six-monthly intervals provided the operating temperature does not exceed 40 °C . Shorter intervals between checks will be necessary if the temperature is higher.

Since contaminant separation is poorer compared with mineral oil, the check for solid and liquid contaminants in HFC fluids is very important. Whereas solid contaminants can settle out provided the tank is designed properly, liquid contaminants such as mineral oil can only be detected by regular testing. Residual amounts of mineral oil reduce the fire resistance and the air separation capacity and so should be restricted to 0.1 %.

The life of HFC fluids is limited since the compatibility of old fluid with metals, paintwork and seals is poorer than new fluid.

HFC fluids must not be mixed - either between themselves or with other fluids - because the properties deteriorate and routine test figures can sometimes be rendered inaccurate.

4.1.4 Group HFD: Anhydrous synthetic fluids

HFD fluids can be subdivided as follows according to VDMA 24 317:

HFD...R = Phosphate ester

HFD...S = Chlorinated hydrocarbons

HFD...T = Blends of HFDR and HFDS

HFD...U = Other compositions.

Phosphate ester containing no chlorinated hydrocarbons has become one of the most popular fluids. Chlorinated hydrocarbons (PCB's), which are difficult to degrade, may no longer be used in hydraulic systems above ground.

Therefore, the following information is restricted to Group HFD...R fluids.

The viscosity-temperature characteristics are poorer than those of mineral oil. This is expressed by a viscosity index of less than 80 measured according to ISO 2909. The operating temperature of HFD...R fluids can be increased to approximately 50 to 60°C compared with HFC fluids (40°C) because the tendency to evaporation is less. On the other hand, continuously high temperatures above 50 to 60°C necessitate more frequent testing of the fluid and will result in a shorter life.

Most HFD...R fluids are sensitive to the ingress of moisture. According to VDMA 24 317 the water content should not exceed 0.1% by volume. This necessitates the use of dehydrating breathers on hydraulic systems in moist environments such as by the sea or near rivers.

The neutralization number must be monitored continuously. It shows how much decomposed ester there is in the fluid; it must not exceed 0.3 mg KOH/g.

The types of seals, hoses and accumulator bladders used for mineral oil are not resistant to phosphate ester. Fluoro-elastomers such as Viton must be used.

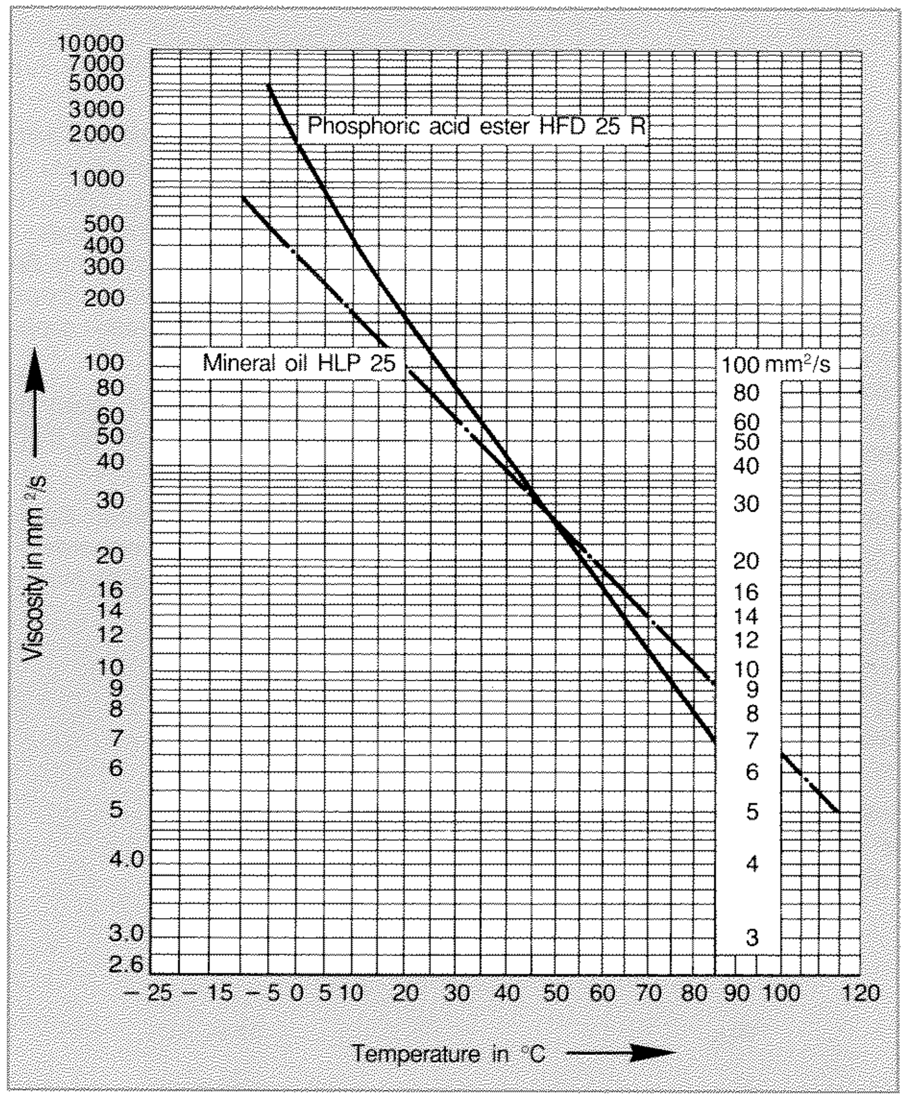


Diagram 9:
Viscosity-temperature characteristics of HFD 25 phosphate ester fluid compared with HLP 25 mineral oil [5]

5 Designing hydraulic systems

The fundamental aspects of designing hydraulic systems are dealt with in DIN 24 346. It contains various rules to ensure a uniform design approach to such systems and covers both equipment and safety aspects. Consequently, it is only necessary here for us to examine the differences between systems designed for fire-resistant fluids and mineral oil-based fluids.

5.1 Tanks

Tanks must be of adequate size so that there is a long dwell time for the fluid. In the case of mineral oil a guide figure is 3 to 5 times the pump delivery per minute. In the case of HFC and HFD fluids, however, due to the poor air separation and contaminant separation capacity, the tank should be at least 5 to 8 times the pump delivery per minute.

Air bubbles must be able to rise to the surface of the fluid (see Diagram 6) and solid particles must be able to sink to the bottom of the tank . Baffle plates or dividers should be provided as necessary.

The tank bottom must have a fall to one side and a fluid drain that can also be used to drain off water must be provided at the lowest point.

Aluminium alloys such as those commonly used for access covers are not resistant to HFC fluids.

The zinc based paints used for internal corrosion protection in mineral oil systems are not resistant to HFC and HFD fluids.

5.2 Pumps

Consultation with the manufacturer is advisable when selecting a pump. Most pumps have been designed to use mineral oil and probably incorporate parts of non-ferrous metal or other alloys which must not be allowed to come into contact with fire-resistant fluids.

When using HFA, HFB and HFC fluids, bearing life is reduced to about 20% compared with mineral oil and phosphate ester.

Permitted pump operating pressures are also reduced compared with mineral oil when using fire-resistant fluids. Suction lines must be larger for the HFC and HFD fire-resistant fluids because of the higher specific gravity. The suction velocity should not exceed 0.5 m/s and it is advisable to mount the pump lower than the tank so that the fluid can run to the suction.

5.3 Valves

Valves designed for mineral oil systems are usually also suitable for HFC and HFD fluids although, in the case of HFC fluids, the maximum permitted operating pressure is restricted compared with mineral oil. There is no point in giving figures because of the different design features of the valves. The appropriate information can be taken from manufacturers' documentation.

Valves suitable for HFA fluids are specially designed and are not comparable with valves designed for mineral oil.

5.4 Filters

The contaminant separation capacity of fire-resistant fluids is not as good as that of mineral oil so filters must be 2 to 3 times the size of those for mineral oil.

Filter bodies are often made of aluminium alloy which is an unsuitable material for HFC fluids. Filter elements containing galvanized parts are also unsuitable.

Filters used in HFC fluid systems must have grey cast iron or spheroidal graphite cast iron bodies. The steel parts of the elements can be electro-less nickel plated or phosphated and the mesh made of stainless steel.

5.5 Seals

The same seals used for mineral oil can be used for fire-resistant fluids with the exception of HFD fluids. Viton seals must be used for HFD fluids.

DIN 53 521 and DIN 53 505 give details of the tests necessary to establish the characteristics of different seal materials.

The following list provides a summary of different types of seal material for different fluids.

Fluid group	Suitable elastomer
HL+HLP	NBR,Viton
HFA	NBR,Viton
HFB	NBR,Viton
HFC	NBR,SBR,EPDM,IIR,NR
HRD-R	Viton,EPDM,IIR

Abbreviations

NBR	– nitrile rubber (acrylic nitrile butadiene)
Viton	– fluoro-carbon rubber, e.g. Viton
SBR	– styrene-butadiene rubber
EPDM	– ethylene propylene diene rubber
IIR	– butyl rubber
NR	– natural rubber

6 Changing fluids

VDMA 24 314 gives instructions for the changing of hydraulic fluids to DIN 51 524, i.e. HL and HLP fluids, and VDMA 24 317 deals with fire-resistant fluids.

If it is intended to change over to HFA fluids complying with DIN 24 320, major structural modifications to the system and components will be necessary, e.g. specially designed valves if the operating pressure exceeds 70 bar.

Note to Table 11

According to our own experience we regard some of the values for the maximum amount of residual old fluid as too high. We recommend that the residual fluid be limited to less than 0.1 % by volume.

Table 11 has been taken from VDMA 24 314.

Change of Fluid		Residual old fluid % by volume	Cleaning aids	Flushing fluid, maximum	Materials no longer suitable after the changeover				Reduced limits for		Modifications needed		
					Hoses bladders, seals	Paint, coatings	Metals, bearing materials	Filter materials2)	Tank temperature	Negative suction head	Viscosity temperature characteristics	Air separation capacity	Solids separation capacity
HL HLP [3]	HFB	6	Non-linting textile and paper cloths, compressed air	HFB	Polyurethane (Vulkollan Asbestos Leather Cork	All normal products except epoxy-resin and Desmodur/ Desmophen based paint	Zinc [1] Cadmium [1] Magnesium	Paper Cotton Cellulose	60 °C	Follow the specifications of the manufacturers of the pumpe or fluid, or avoid negative suction pressures.	—	—	Filter in delivery or return lines also in by-pass, no suction filter. Filterelements designed for system requirements.
HL HLP [3]	HFC	0,1		1.2 Propylene-glycol then HFC				Paper Cellulose	55 °C, or makers' figure for closed-circuit temperature, if necessary < 55 °C		—	Longer delay for fluid from return line to tank suction	
HL HLP [3]	HFD	3	Non-linting textile and paper cloths, **NO** compressed air	HFD	All normal elastomeres except [1] FPM (Viton ®) PTFE (Teflon ®) SI (silicones) EPDM, IIR (Butyl)		Lead/steel/Zinc, Cadmium Aluminium alloys				Tank or other external heating for low ambient temperatures according to makers' instructions		
HFB	HL HLP	1	Non-linting textile and paper cloths, compressed air	HL HLP	—	—	—	—	—		—	—	—
HFB	HFC	0,5		HFC	—	—	Zinc [1] Cadmium Magnesium	—	—		—	Longer delay for fluid from return line to tank suction	
HFB	HFD	0,1	Non-linting textile and paper cloths, **NO** compressed air	HFD	All normal elastomers except [1] FPM, PTFE,	All normal products except epoxy-resin and Desmodur/ Desmophen based paints	Lead/Cast iron, Steel/Zinc, Cadmium Aluminium alloys				Tank or other external heating for low ambient temperatures according to makers' instruction		
HFC	HL HLP	0,1	Non-linting textile and paper cloths, hot water, steam	HL HLP									
HFC	HFB	0,1		HFB	IIR (butyl rubber)		—	Cotton	—		—	—	Filter in delivery or return lines, also in by-pass, no suction filter. Filter element design according to system requirements
HFC	HFD	0,1	Non-linting textile and paper cloths, hot water steam warm air	HFD	All normal elastomers except [1] FPM (Viton ®) PTFE (Teflon ®) SI (silicones) EPDM, IIR		Lead/Steel/Zinc, Cadmium Aluminium alloys				Tank or other external heating for low ambient temperatures according to makers' instruction		
HFD	HL HLP	1	Non-linting textile and paper cloths, normal solvents	HL HLP									
HFD	HFB	0,1		HL then HFB	IIR (butyl rubber) EPDM			Paper Cotton Cellulose	60 °C				
HFD	HFC	0,1		HL then HFC	FPM if necessary		Zinc [1] Cadmium [1] Magnesium	Paper Cellulose	55 °C, closed-circuit temperature as makers' instructions, if necessary < 55 °C				

1) According to composition of fluid 2) Always fit new filter after the changeover and follow the makers' instructions. A blank rectangle marked (—) means that there is usually no difference in behaviour after the changeover.

3) HLP= International designation HM

Tabelle 11

7 The major relevant standards

DIN 24 320	Fire resistant fluids; group HFAE, characteristics, requirements
DIN 24 346	Hydraulic fluid power; hydraulic systems; general rules for application
DIN 51 354 Part 2	Testing of lubricants; mechanical testing of lubricants in the FZG gearrig test machine, gravimetric method for lubricating oils A/8,3/90
DIN 51 381	Testing of lubricating oils, governor oils and hydraulic fluids; determination of air release properties
DIN 51 389 Part 2	Determination of lubricants; mechanical testing of hydraulic fluids in the vane-cell-pump; method A foranhydrous hydraulic fluids
DIN 51 389 Part 3	Determination of lubricants; mechanical testing of hydraulic fluids in the vane-cell-pump; method B for aqueous not easily inflammable hydraulic fluids
DIN 51 502	Lubricants and Related Materials; Designation of the Lubricants and Marking of Lubricant Containers, Lubrication Equipment and Lubrication Points
DIN 51 517 Part 1	Lubricants; Lubricating Oils; Lubricating Oils C; Minimum Requirements
DIN 51 517 Part 2	Lubricants; Lubricating Oils; Lubricating Oils CL; Minimum Requirements
DIN 51 517 Part 3	Lubricants; Lubricating Oils; Lubricating Oils CLP; Minimum Requirements
DIN 51 519	Lubricants; ISO viscosity classification for industrial liquid lubricants
DIN 51 524 Part 1	Pressure fluids; hydraulic oils; HL hydraulic oils; minimum requirements
DIN 51 524 Part 2	Pressure fluids; hydraulic oils; HLP hydraulic oils; minimum requirements
DIN 51 550	Viscometry; Determination of Viscosity; General Principles
DIN 51 558 Part 1	Testing of Mineral Oils; Determination of the Neutralization Number, Colour-indicator titration
DIN 51 561	Testing of Mineral Oils, Liquid Fuels and Related Liquids; Measurement of Viscosity Using the Vogel-Ossag Viscometer; Temperature Range: approximately 10 to 150 °C
DIN 51 562 Part 1	Viscosity; determination of kinematic viscosity using the Ubbelohde viscometr; viscosity relative increment at short flow times
DIN 51 566	Testing of lubricants; determination of foaming characteristics
DIN 51 585	Testing of Lubricants; Testing of Corrosion-protection Properties of Steam Turbine Oils and Hydraulic Oils Containing Additives
DIN 51 587	Testing of Lubricants; Determination of the Ageing Behaviour of Steam Turbine Oils and Hydraulic Oils Containing Additives
DIN 51 599	Testing of Lubricating Oils; Determination of Demulsification Capacity according to the Stirring Method
DIN 51 759	Testing of liquid mineral oil products; method of test for copper corrosion; copper strip test

DIN 51 848 Part 1	Testing of mineral oils; precision of test methods, general introduction, definitions and their application to mineral oil standards which contain requirements
DIN 53 505	Testing of rubber, elastomers and plastics; Shore hardness testing A and D
DIN 53 521	Determination of the behaviour of rubber and elastomers when exposed to fluids and vapours
DIN 53 538 Part 1	Standard reference elastomers; Acrylonitrile-butadiene rubber (NBR), cross-linked by peroxide, for characterizing liquids with regard to their effect on NBR
DIN EN 7	Determination of Ash from Petroleum Products
DIN ISO 2592	Petroleum products; determination of flash and fire points, Cleveland open cup method
DIN ISO 2909	Petroleum Products; Calculation of Viscosity Index from Kinematic Viscosity
DIN ISO 3016	Petroleum oils; determination of pour point
DIN ISO 3733	Petroleum products and bituminous materials; determination of water, distillation method
ISO/DIS 6071	Hydraulic fluid power - Fire resistant fluids - Classification and designation
ISO 6743/4-1982	Lubricants, industrial oils and related products (class L) - Classification - Family H (Hydraulic systems)
VDMA 24 314	Hydraulics in fluid technology; conversion of gases; guidelines
VDMA 24 317	Fluid technology; hydraulics; flame-resistant gases; guidelines

8 References

[1] F. Eckhardt — Ölhydraulische Anlagen, Betrieb und Wartung
Fa. Mobil Oil AG in Deutschland, Hamburg

[2] M. Frauenstein — Hinweise über Druckflüssigkeiten für Konstrukteure
Fa. Mobil Oil AG in Deutschland, Hamburg

[3] Prof. Dr. H.W. Thoenes, Essen — Derzeitiger Markt und technische Entwicklungen von Hydraulikflüssigkeiten
o + p Nr. 3/1985

[4] Company literature — Fa. Wenzel und Weidmann GmbH, Eschweiler
"Schwerentflammbare Hydraulik-Flüssigkeit UK-Ecubsol-Hydrotherm"

[5] Company literature — Fa. Houghton-Hildesheim, Hildesheim
"Houghto Safe, die feuerresistenten Hydraulik-Fluids"

Notes

The Thermal Design of Hydraulic Systems

Hans H. Faatz

1 Introduction

Hydraulic systems utilize a process of energy conversion and transmission. They have major advantages over other systems of energy conversion in that they achieve a very high energy density. The electrical energy is converted into hydraulic energy by the motor-driven pump in a simple manner incurring very few losses. The resulting energy flows are easy to control and regulate and they allow great repeat accuracy.

The processes of energy conversion and transmission involve losses in which mechanical and hydraulic energy is converted into heat.

The operating temperature is one of the factors governing the efficiency of a hydraulic system. If the temperature is too low the flow resistance is increased and difficulties are experienced with the suction to the pump. If the temperature is too high there are more fluid leaks so losses and wear are greater.

When hydraulic systems are installed in extremely cold places or outdoors it can sometimes be necessary to preheat the fluid to operating temperature before starting the system. So that the fluid does not overheat due to the losses in the system that are converted into heat it is sometimes necessary to cool the fluid by means of a heat exchanger.

The size and nature of the heating and/or cooling system for a hydraulic installation depends on the demands made on the system, its precision and life and the type of fluid being used. In many cases it is necessary to calculate a heat balance for the hydraulic system.

2 Basic principles

Energy, work and heat are physical quantities of a similar nature with the same SI units.

	J	kJ	kW h	kcal	kgf m
1 J = (= 1 N m = 1 W s)	1	0.001	$2.78 \cdot 10^{-7}$	$2.39 \cdot 10^{-4}$	0.102
1 kJ =	1000	1	$2.78 \cdot 10^{-4}$	0.239	102
1 kW h =	3600000	3600	1	860	367000
1 kcal =	4200	4.2	0.00116	1	427
1 kgf m =	9.81	0.00981	$2.72 \cdot 10^{-6}$	0.00234	1

Table 12: *Conversion table for units of energy, work and heat*

$$\text{where } 1\ \text{Nm} = \frac{1}{9.81} = 0.102\ \text{kpm}$$

Energy flow, power and heat flow are defined as the amount of energy, work and heat flowing in unit time.

$$\text{Power} = \frac{\text{Work}}{\text{Time}} = \frac{\text{Nm}}{\text{s}} = \frac{1}{9.81} \cdot \frac{\text{kpm}}{\text{s}} = 1\ \text{W}$$

$$\text{Heat flow} = \frac{\text{Heat}}{\text{Time}} = \frac{\text{J}}{\text{s}} = 0.86\,\frac{\text{kcal}}{\text{h}}$$

	W	kW	kcal/s	kcal/h	kgf m/s
1 W = (= 1 N m/s = 1 J/s)	1	0.001	$2.39 \cdot 10^{-4}$	0.860	0.102
1 kW =	1000	1	0.239	860	102
1 kcal/s =	4190	4.19	1	3600	427
1 kcal/h =	1.16	0.00116	1/3600	1	0.119
1 kgf m/s =	9.81	0.00981	0.00234	8.43	1

Table 13: *Conversion table for units of power, energy flow and heat flow as the quotient of energy and time (energy per unit time)*

The old unit of power that has been superseded and gone out of current use is the horse power (HP). It was defined as 1 HP = 75 kgfm/s.

Thus $1\ \text{kW} = \frac{1}{1.36}\ \text{HP} = 0.736\ \text{HP (metric)}$

The conservation of energy law is relevant when considering the heat balance of hydraulic systems:

$$\Sigma \dot{Q} = \text{constant}\ \frac{\text{kcal}}{\text{h}} \qquad (1)$$

Where $\Sigma \dot{Q}$ is the sum of all heat flows into and out of the hydraulic system.

All thermal calculations are performed with the heat flow $\dot{Q}$ in kcal/h.

Calculations on hydraulic systems use the power P in kW. As energy per unit time, heat flow and power are similar physical quantities and their mathematical relationship is shown in *Table 13.*

For the heat balance of hydraulic systems according to Equation 1: P_W = constant in kW.

Where P_W is the power relevant to heat, e.g. that gained by a hydraulic system due to losses and which must be removed by cooling.

2.1 Heat flow

The volumetric flows entering a heat exchanger contain a certain amount of energy, so-called "inner energy". The inner energy is equal to the volumetric flow multiplied by the density of the medium, its specific thermal capacity and its absolute temperature.

Thus $\dot{U} = \dot{V} \cdot \rho \cdot c \cdot T$

$\dot{U}$ = Inner energy of mass flow in kW
$\dot{V}$ = Volumetric flow in m³/s
c = Specific thermal capacity in kJ/kg K
T = Absolute temperature in K
ρ = Density of medium in kg/m³

During its time in the heat exchanger the hot volumetric flow gives up some of its inner energy. The difference corresponds to the heat flow.

$$\dot{Q}_W = \dot{U}_{WE} - \dot{U}_{WA} = \dot{V}_W \cdot \rho_W \cdot c_W \cdot (T_{WE} - T_{WA}) \qquad (3)$$

If we assume that the heat exchanger does not give up any heat to its surroundings, the cold volumetric flow will absorb the same amount of heat as that given up by the hot volumetric flow. Thus, as the hot volumetric flow, the following applies:

$$\dot{Q}_K = \dot{U}_{KA} - \dot{U}_{KE} = \dot{V}_K \cdot \rho_K \cdot c_K \cdot (T_{KA} - T_{KE}) \qquad (4)$$

The heat flow that can be transmitted through a heat exchanger can be determined directly by measuring the volumetric flow and the inlet and outlet temperatures of one of the media of known density and known specific thermal capacity.

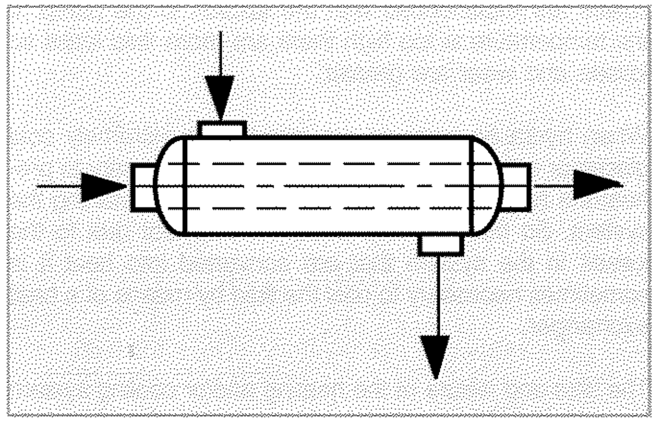


Fig. 22: *Heat exchanger*

2.2 Heat transfer

2.2.1 Principle of the heat exchanger

In a heat exchanger, heat is transferred from one medium (a liquid or gas) at temperature T_1 to another medium at temperature T_2. The heat transfers from one to the other by convection at a surface of area A (casing wall or pipe). The heat is transferred through the surface by conduction and, at the surface to the other medium, the heat is again transferred by convection. According to the conservation of energy law *(1)* the flow of energy during this process is constant.

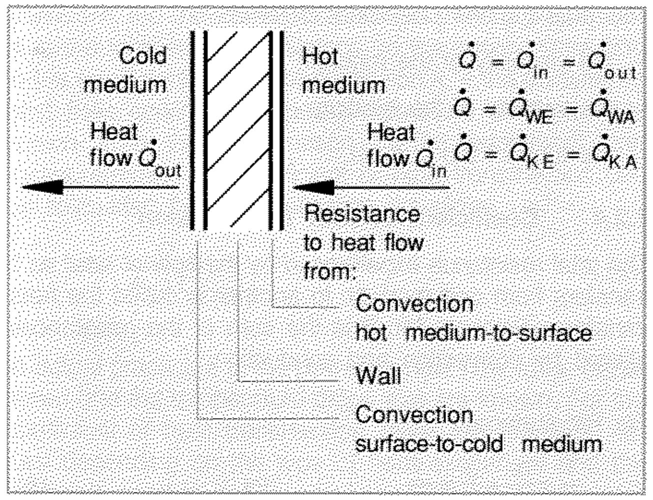


Fig. 23: *Heat transfer*

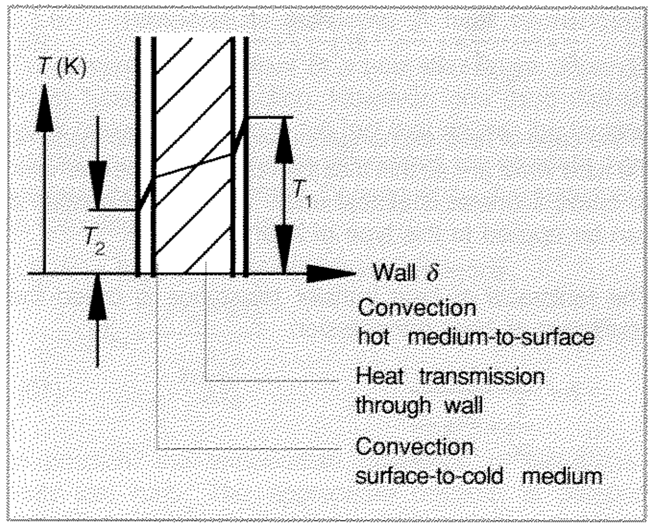


Fig. 24: *Temperature gradient*

2.3 Thermal conductance

2.3.1 Conduction of heat through a surface

When heat flows perpendicularly through a flat wall there is a linear temperature drop from temperature T_1 on one side to temperature T_2 on the other side. The actual drop depends on the thickness of the wall and the coefficient of thermal conductivity which is related to the material λ = kW/mK.

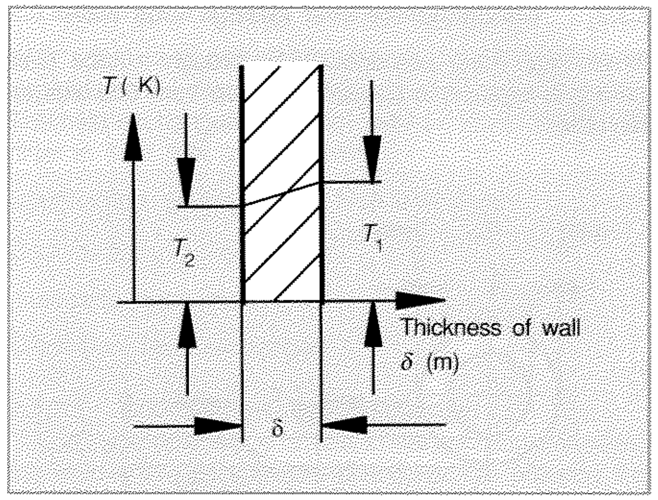


Fig. 25: *Conduction of heat through a flat wall*

The heat flowing perpendicularly through a flat wall is directly proportional to the area. Thus, the heat flow through a flat wall is as follows:

$$\dot{Q}_W = \dot{Q}_{in} = \dot{Q}_{out} = \frac{\lambda}{\delta} \cdot A \cdot (T_1 - T_2) \qquad (5)$$

$\dot{Q}_W$ = Heat flow in kW
λ = Coefficient of thermal conductivity in kW/m K
d = Thickness of flat surface in m
A = Flow area in m²
T = Temperature in K

The coefficient of thermal conductivity λ depends on the particular material and the temperature. For example, in the case of 99% pure aluminium at a temperature of 20 °C; λ = 180 (W/m K).
At a temperature of 100 °C ; λ = 187 W/m K

For steel containing approximately 0.1 % carbon and at a temperature of 100 °C ; λ = 45 W/m K.
At 300 °C ; λ = 40 W/m K

The following are typical coefficients quoted in W/m K at 20 °C

Red casting brass	~52
Cast iron	~43
Copper	~320
Brass	~68 to 96
Steel	~48
Stainless steel	~13

2.3.2 Conduction of heat through tube walls

When heat flows perpendicularly through a tube wall there is a logarithmic temperature drop from one side to the other.

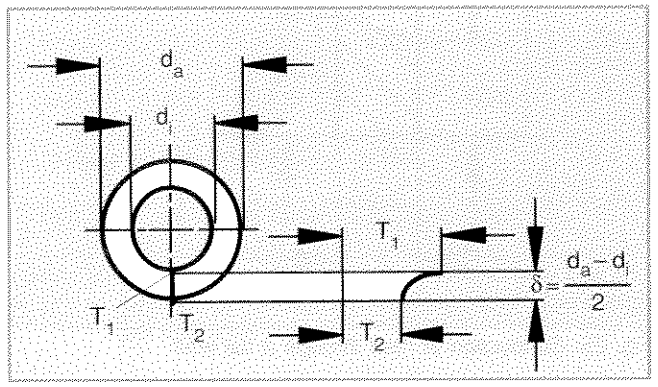


Fig. 26: *Conduction of heat through a tube wall*

Similarly to *Equation 2*, the heat flow through the tube is

$$\dot{Q} = \frac{2\pi \cdot L \cdot \lambda}{\ln \frac{d_a}{d_i}} \cdot (T_1 - T_2) \qquad (6)$$

2.3.3 Convection

The flow of heat from a liquid or gas at temperature T_1 to a surface at temperature T_{w1} is:

$$\dot{Q}_W = \alpha_W \cdot A\,(T_1 - T_{W1}) \qquad (7)$$

For the transfer of heat from the surface to a cold medium:

$$\dot{Q}_K = \alpha_K \cdot A\,(T_{W2} - T_2)$$

Where is the heat transfer coefficient (kW/m²K)

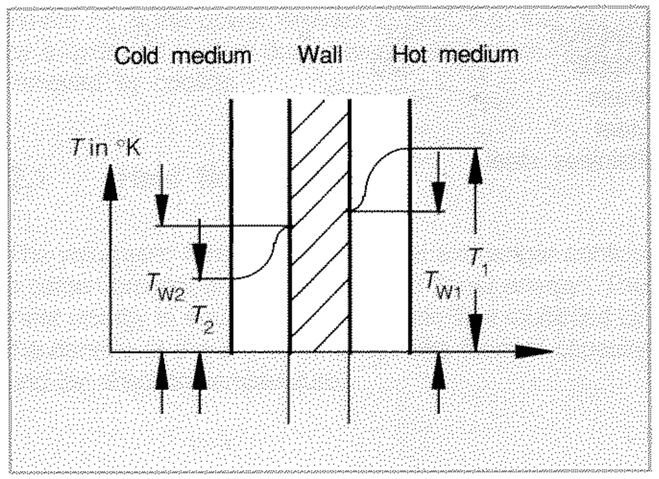


Fig. 27: *Transfer of heat by convection*

The heat transfer coefficient depends on:

- the viscosity of the medium
- the velocity of the medium
- the form of the surface.

2.3.4 Coefficient of heat transmission

The total resistance experienced by the flow of heat as it transfers from one medium to another is called the heat transfer resistance 1/k. It is the sum of the resistances offered by convection and conduction.

For flat surfaces

$$\frac{1}{k} = \frac{1}{\alpha_1} + \frac{\delta}{\lambda} + \frac{1}{\alpha_2} \qquad (8)$$

For pipes, referring to the outside diameter d_a:

$$\frac{1}{k} = \frac{1}{\alpha_i} \cdot \frac{d_a}{d_i} + \frac{d_a}{2 \cdot \lambda} \cdot \ln\left(\frac{d_a}{d_i}\right) + \frac{1}{\alpha_a} \qquad (9)$$

In the case of heat exchangers used in hydraulic systems the tube walls are so thin that the ratio of outside diameter to inside diameter is almost unity. Therefore, *Equation 8* can be applied to this type of heat exchanger.

The reciprocal of heat transfer resistance is called the coefficient of heat transmission

$$k = \frac{1}{\frac{1}{\alpha_1} + \frac{\delta}{\lambda} + \frac{1}{\alpha_2}} \qquad (10)$$

2.3.5 Ascertaining the coefficient of heat transmission

The total heat flow transferred by convection and conduction is:

$$\dot{Q} = k \cdot A \cdot \Delta T_m \qquad (11)$$

$\dot{Q}$ = Heat flow in kW
k = Coefficient of heat transmission in kW/m² K
A = Cooling area in m²
T = Mean temperature difference between the two media in K *(see Equation 13)*

Therefore, the transferred heat flow is directly proportional to the temperature difference, to the cooling area and to the coefficient of heat transmission k. If the transferred heat flow, cooling area and temperature difference are known from the construction of the equipment and measurements, the coefficient of heat transmission can be calculated from the following equation.

$$k = \frac{Q}{A \cdot \Delta T_m} \qquad (12)$$

In actual practice it is the power or heat flow to be transferred and the temperature drops that are known. For the coefficient of heat transmission there are empirical values depending on the type of heat exchanger and the materials used in it. The only variable factors remaining for the manufacturer of a heat exchanger are the surface area and the volumetric flow of the media.

2.3.6 Temperature gradients in heat exchangers

The temperature gradient of a heat exchanger depends on its type of construction. However, one thing that all types have in common is the fact that neither the temperatures nor the temperature differences are constant over the heat exchange surface. Consequently, it is necessary to calculate a mean value.

ΔT_m is a logarithmic temperature difference

$$\Delta T_m = \frac{\Delta T_A - \Delta T_E}{\ln \frac{\Delta T_A}{\Delta T_E}} \qquad (13)$$

In practice, the temperature difference is determined from test runs. The appropriate medium is pumped through the test subject and the volumetric flow and input and output temperatures measured. *Equations 2, 3 and 4* are then used with the measurements to obtain the heat transfer flow:

$$\dot{Q} = \dot{V} \cdot \rho \cdot c \cdot \Delta T \qquad (14)$$

where

$$\Delta T = T_{out} - T_{in}$$

2.3.7 Types of heat exchanger

With active heat exchangers the temperature difference depends on the direction of flow of the hot and cold media. The heat exchangers normally used in hydraulic systems differ in their flow arrangements:

Counter-flow or single-pass heat exchanger

Counter-flow/parallel-flow or

multi-pass heat exchanger

Cross-flow heat exchanger.

2.3.7.1 Counter-flow heat exchanger

This is the simplest type of active heat exchanger in which the media flow in opposite directions. On entering the heat exchanger the cold medium first encounters the hot medium which has already been cooled. The temperature difference between T_{KE} and T_{WA} is comparatively small.

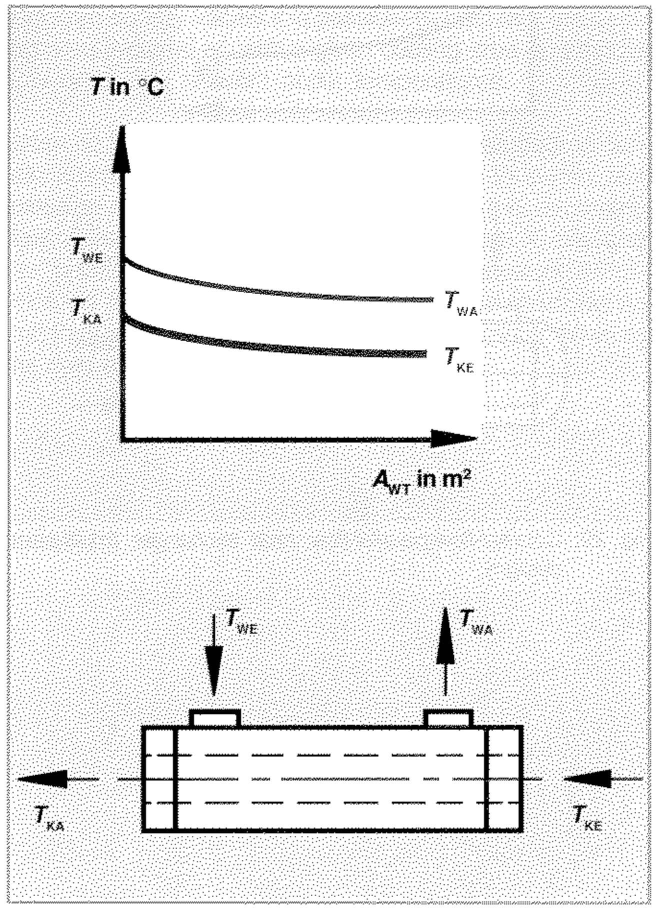


Fig. 28: *Counter-flow (single-pass heat exchanger)*

2.3.7.2 Counter-flow/parallel-flow heat exchanger

In this case, one of the media, generally the cold medium, is reversed in direction. Therefore, it flows twice through the other medium which is not reversed in direction. The end result is a counter-flow and also a parallel flow. The design necessitates a split cover. The temperature difference between T_{KE} and T_{WA} is higher than with the parallel-flow heat exchanger.

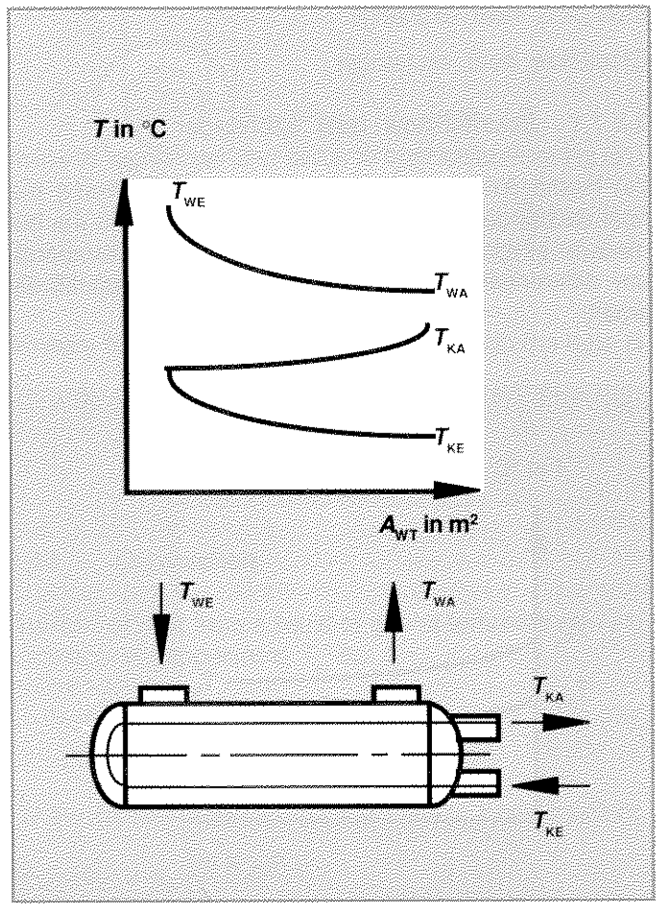


Fig. 29: *Counter-flow/parallel-flow heat exchanger (multi-pass design)*

2.3.7.3 Cross-flow heat exchanger

Cross-flow heat exchangers are mostly of the oil/air type.

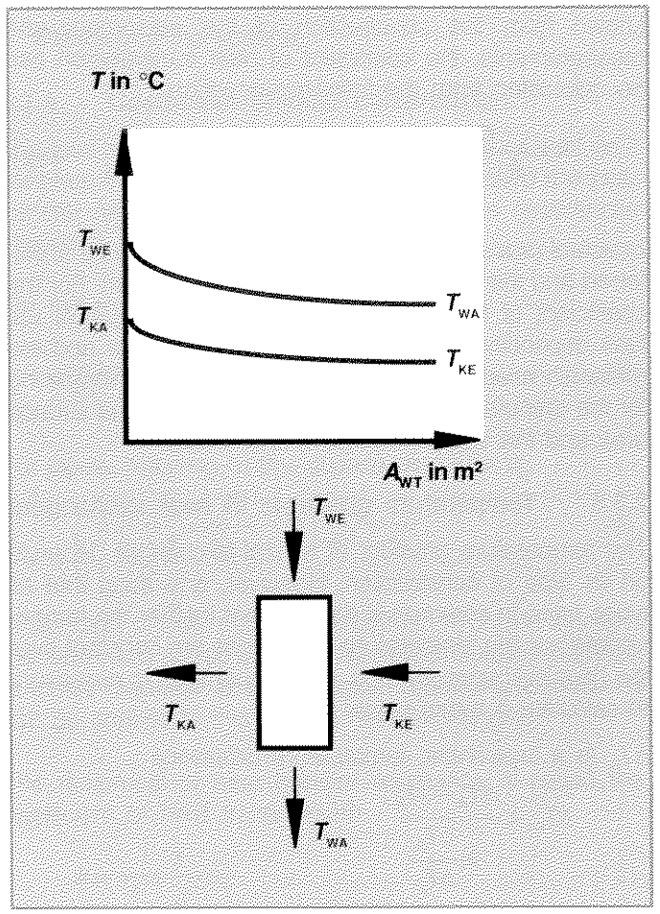


Fig. 30: *Cross-flow heat exchanger (air-cooled system)*

2.3.8 Heat exchanger calculation by the NTU method

Since the mean temperature difference is not always simple to calculate, other methods of design must be sought.

In their book "Compact Heat Exchangers" Kays and London describe a method which allows heat exchangers to be calculated with the aid of graphic charts.

The starting point is the "energy value" W of a medium.

$$W = \dot{V} \cdot \rho \cdot c \tag{15}$$

$\dot{V}$ = Volumetric flow in m³/s
ρ = Density of medium in kg/m³
c = Specific heat capacity of the medium in kW/kg K

The energy value is the internal energy of the mass flow per unit temperature *(see Equation 2)*.

$$W = \frac{\dot{Q}}{\Delta T} \tag{16}$$

For the cold medium

$$W_K = \dot{V}_K \cdot \rho_K \cdot c_K$$

For the hot medium

$$W_W = \dot{V}_W \cdot \rho_W \cdot c_W$$

The smaller of the two values W_{min} is the governing factor for the effectiveness of a heat exchanger since the law of energy conservation *(1)* is applicable regardless of other factors of influence. In making a calculation, ascertain both values and then use the smaller one (W_{min}). The amount of heat flow that a heat exchanger can handle depends on the temperature difference between the input and output media as well as the energy value W_{min}. The maximum possible heat flow $\dot{Q}_{max}$ occurs when the maximum entry temperature difference (ETD)arises at the lower energy value. Referred to figures 28 to 30: $ETD = T_{WE} - T_{KE}$.

$$\dot{Q}_{max} = W_{min} \cdot ETD \tag{17}$$

The effectiveness of a heat exchanger is defined as the ratio of the actual dissipated heat flow to the maximum heat flow that it is possible to dissipate.

$$\varepsilon = \frac{\dot{Q}}{\dot{Q}_{max}} = \frac{W_K \cdot (T_{KA} - T_{KE})}{W_{min}\,(T_{WE} - T_{KE})} = \frac{W_W \cdot (T_{WE} - T_{WA})}{W_{min}\,(T_{WE} - T_{KE})} \tag{18}$$

In the case of oil/water heat exchangers W_{min} is generally equal to W_K and, on this assumption,

$$\varepsilon = \frac{t_{KA} - t_{KE}}{t_{WE} - t_{KE}}$$

In addition to what has been said so far it is necessary to introduce a value for "Number of Heat Transfer Units" or NTU.

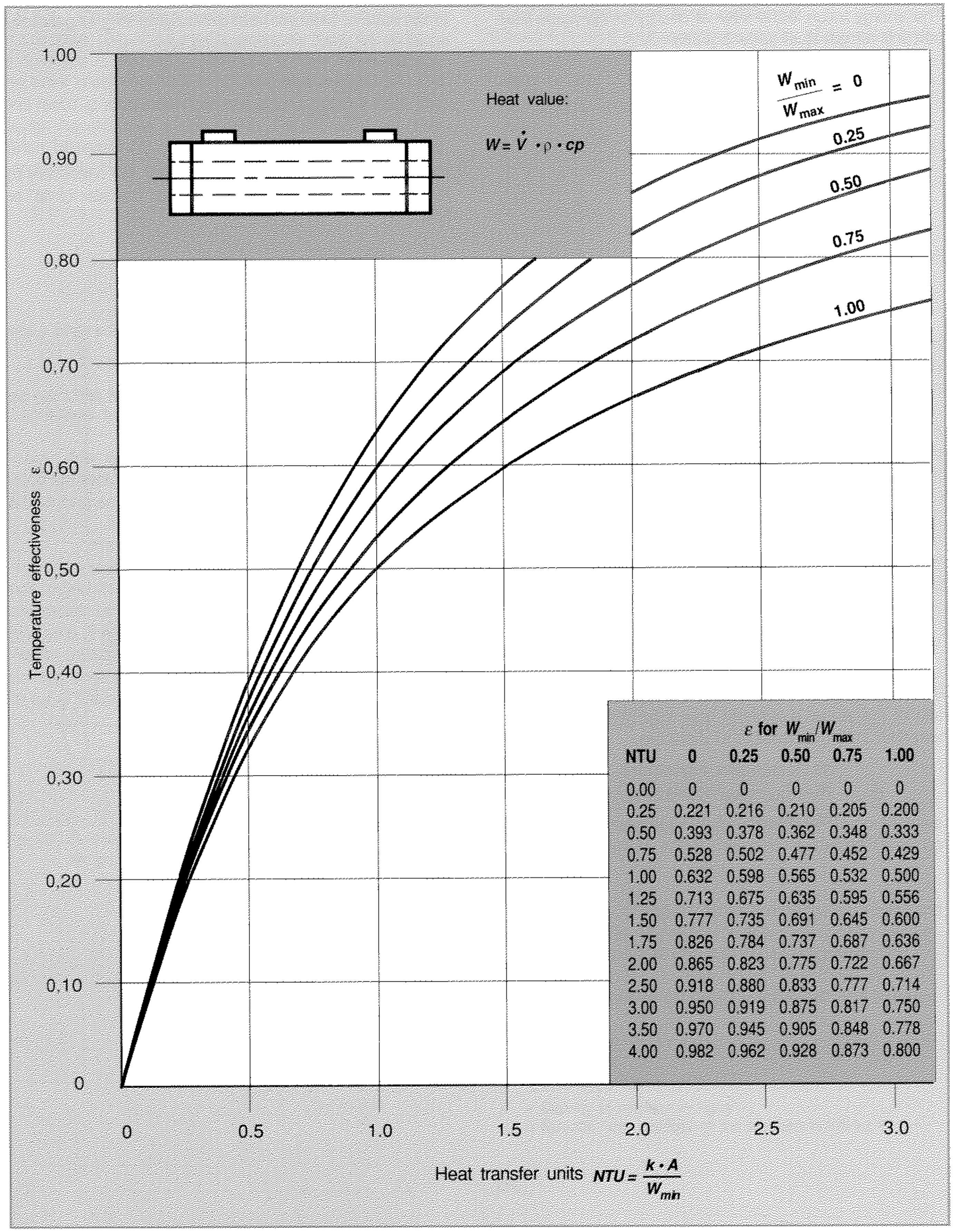
$$NTU = \frac{k \cdot A}{W_{min}}$$

A dimensional analysis shows that this value and ε are both dimensionless.

The relation between the effectiveness ε and *NTU* can be plotted graphically for specific types of heat exchanger.

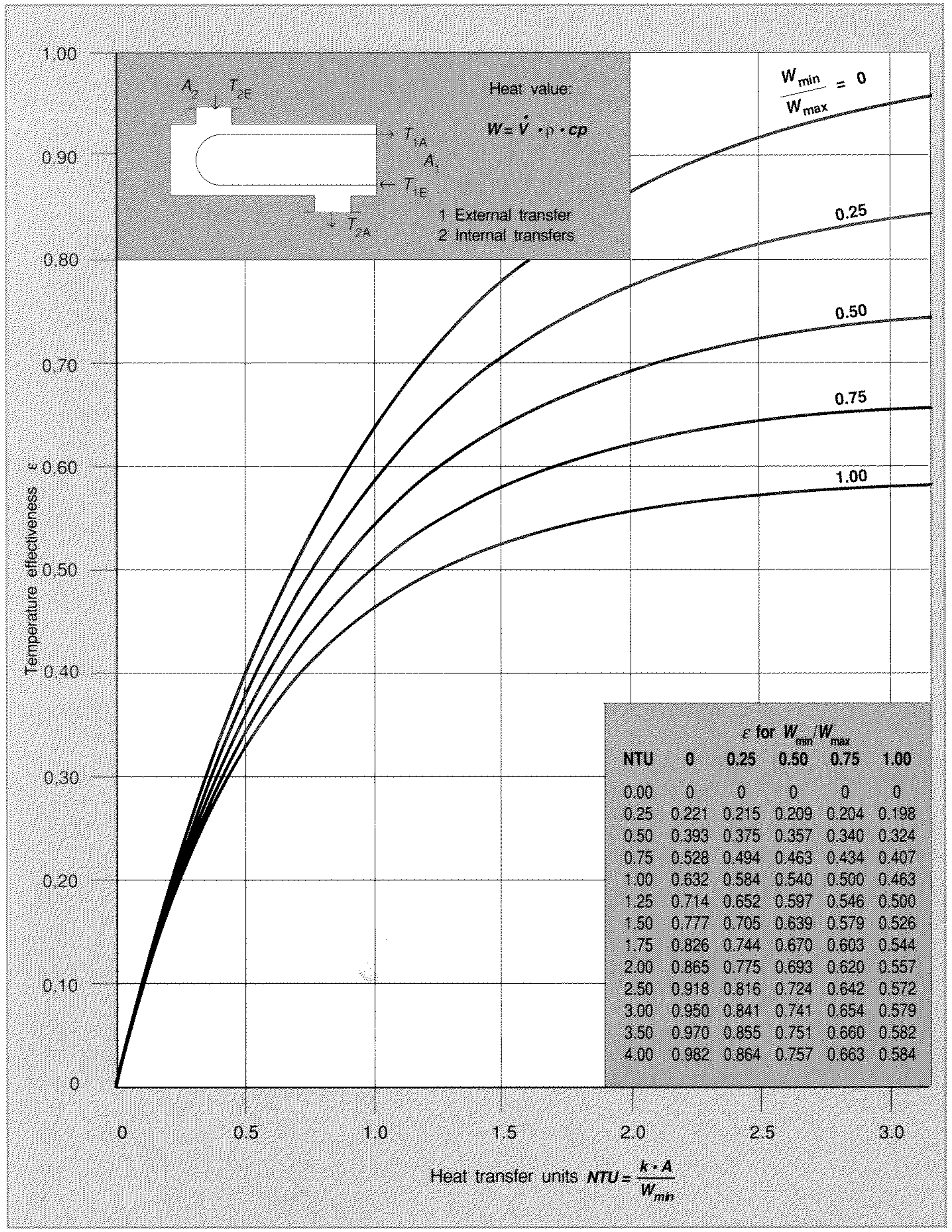
Using the graphs and the equations together with predetermined energy values it is possible to calculate the cooling area required or, with a given cooling area and media flow rates, the relevant outlet temperatures.

Since, in practice, there are certain basic values and certain sizes of heat exchanger from which to begin, the selection diagrams are constructed so that the volumetric flow of the hot medium (fluid) is given on the X axis and the heat-exchange rating per inlet temperature difference on the Y axis. The diagrams are also arranged so that they refer to a specific ratio of the volumetric flows of the cold and hot media to each other.



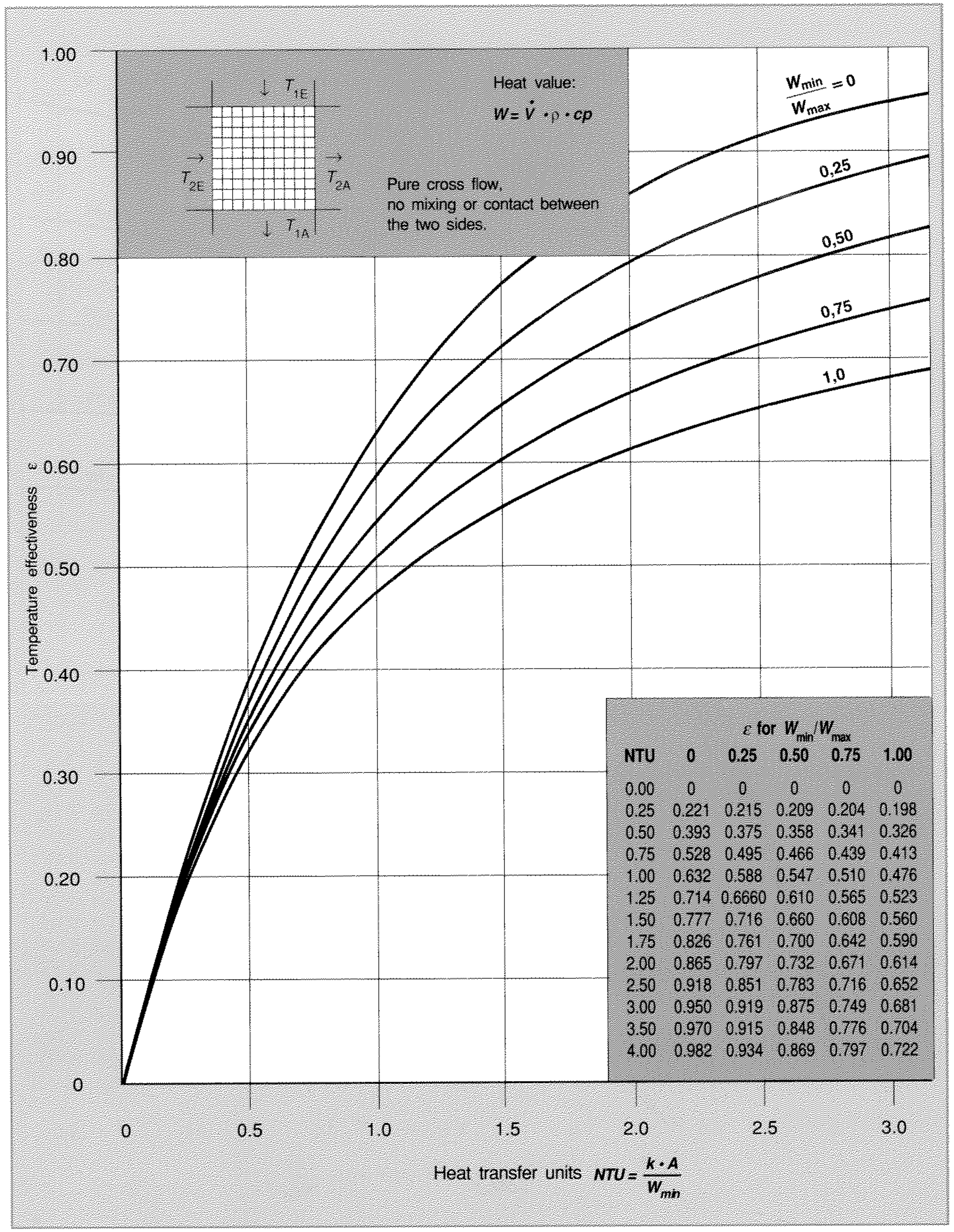
	ε for W_{min}/W_{max}				
NTU	**0**	**0.25**	**0.50**	**0.75**	**1.00**
0.00	0	0	0	0	0
0.25	0.221	0.216	0.210	0.205	0.200
0.50	0.393	0.378	0.362	0.348	0.333
0.75	0.528	0.502	0.477	0.452	0.429
1.00	0.632	0.598	0.565	0.532	0.500
1.25	0.713	0.675	0.635	0.595	0.556
1.50	0.777	0.735	0.691	0.645	0.600
1.75	0.826	0.784	0.737	0.687	0.636
2.00	0.865	0.823	0.775	0.722	0.667
2.50	0.918	0.880	0.833	0.777	0.714
3.00	0.950	0.919	0.875	0.817	0.750
3.50	0.970	0.945	0.905	0.848	0.778
4.00	0.982	0.962	0.928	0.873	0.800

Diagram 10: *Rating of a counter-flow heat exchanger*
Effectiveness according to NTU value
Parameter: ratio of water values according to Kays and London



	ε for W_{min}/W_{max}				
NTU	**0**	**0.25**	**0.50**	**0.75**	**1.00**
0.00	0	0	0	0	0
0.25	0.221	0.215	0.209	0.204	0.198
0.50	0.393	0.375	0.357	0.340	0.324
0.75	0.528	0.494	0.463	0.434	0.407
1.00	0.632	0.584	0.540	0.500	0.463
1.25	0.714	0.652	0.597	0.546	0.500
1.50	0.777	0.705	0.639	0.579	0.526
1.75	0.826	0.744	0.670	0.603	0.544
2.00	0.865	0.775	0.693	0.620	0.557
2.50	0.918	0.816	0.724	0.642	0.572
3.00	0.950	0.841	0.741	0.654	0.579
3.50	0.970	0.855	0.751	0.660	0.582
4.00	0.982	0.864	0.757	0.663	0.584

Diagram 11: *Rating of a counter-flow/parallel-flow heat exchanger; with even exchange of heat*
Effectiveness according to NTU value
Parameter: ratio of water values according to Kays and London



NTU	ε for W_{min}/W_{max} 0	0.25	0.50	0.75	1.00
0.00	0	0	0	0	0
0.25	0.221	0.215	0.209	0.204	0.198
0.50	0.393	0.375	0.358	0.341	0.326
0.75	0.528	0.495	0.466	0.439	0.413
1.00	0.632	0.588	0.547	0.510	0.476
1.25	0.714	0.6660	0.610	0.565	0.523
1.50	0.777	0.716	0.660	0.608	0.560
1.75	0.826	0.761	0.700	0.642	0.590
2.00	0.865	0.797	0.732	0.671	0.614
2.50	0.918	0.851	0.783	0.716	0.652
3.00	0.950	0.919	0.875	0.749	0.681
3.50	0.970	0.915	0.848	0.776	0.704
4.00	0.982	0.934	0.869	0.797	0.722

Diagram 12: *Rating of a cross-flow heat exchanger*
Effectiveness according to NTU value
Parameter: ratio of water values according to Kays and London

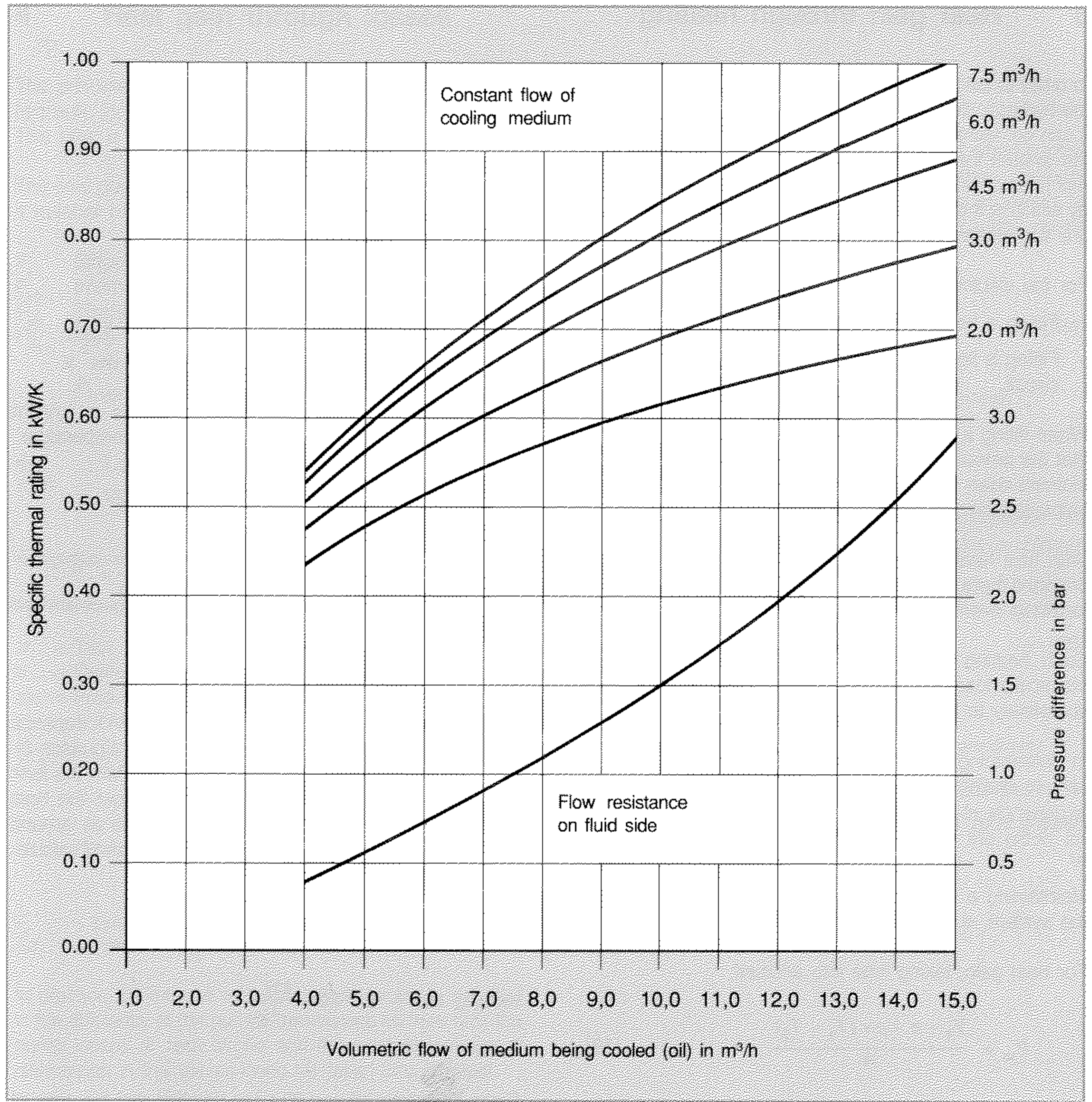


Diagram 13: *Plot of specific thermal rating against volumetric flow for an oil/water heat exchanger with specific values of cooling media flow*

3 Heat gain through heating

In order to satisfy the requirement that the temperature of the hydraulic fluid should be kept as constant as possible, it can sometimes be necessary to inject heat energy into the system through a suitable heat exchanger. The normal practice is to convert electrical energy into heat energy which is then imparted to the fluid. Hot water or steam are sometimes used as alternatives to electricity.

For the electrical method there are two options:

- Fitting a heating element in the tank
- Fitting a flow heater in a separate heater-filter-cooler circuit.

When fitting a heating element in the tank it is important that the amount of heat given off from the surface of the element does not exceed 0.7 W/cm², otherwise localized carbonization of the fluid might occur.

With flow heaters the surface loading can be 2 W/cm² provided there is a minimum flow of fluid to ensure that no localized overheating of the fluid occurs.

Such heating arrangements in hydraulic systems are practically loss-free and the energy input is transferred completely to the fluid.

In determining the heater capacity required it is necessary to distinguish between:

Heating from T_1 to T_2 in time Z

or

Maintaining a temperature when heat is being lost from the system, e.g. in cold surroundings.

For heat gain, taking time into account,

$$P_W = \frac{V_T \cdot c \cdot \rho \cdot (T_1 - T_2)}{H} \qquad (19)$$

P_W = Heat gain required in kW
V_T = Quantity of tank fluid to be heated in dm³
c = Specific heat in kW h/kg K
π = Density in kg/dm³
T_1 = Required fluid temperature in K
T_2 = Outlet temperature, usually the ambient temperature in K
H = Heating time in h

For mineral oil:

$$\rho = 0{,}89 \ \frac{\text{kg}}{\text{dm}^3}$$

$$c = 0{,}00052 \ \frac{\text{kW h}}{\text{kg °C}}$$

and

$$\rho \cdot c = 0{,}89 \cdot 52 \cdot 10^{-5} = 0{,}0004628 = 4{,}63 \cdot 10^{-4} \ \frac{\text{kW h}}{\text{dm}^3 \text{ K}}$$

For maintaining a constant temperature in a system which is losing heat due to a low ambient temperature:

$$\dot{Q}_W = k \cdot A(T_1 - T_2) \qquad (20)$$

$\dot{Q}_W$ = Required heat gain in kW
k = Coefficient of heat transmission of heat-dissipating surface in kW/m² K
A = Heat-dissipating surface area of tanks, equipment and pipes in m²
T_1 = Required fluid temperature in K
T_2 = Ambient temperature in K

In this case the coefficient of heat transmission k depends largely on the heat transfer coefficient *(see Equation 8)*. Assuming that most of the heat is radiated from the tank and the velocity of the fluid in the tank is low, the heat transfer depends solely on the motion of the air around the tank. The following can be used for practical calculation purposes:

stagnant air: $k = 0{,}01 \ \frac{\text{kW}}{\text{m}^2\text{K}}$

moving air: $k = 0{,}02 \ \frac{\text{kW}}{\text{m}^2\text{K}}$

From *Equations 19 and 20* it can be seen that the amount of heat to be supplied depends on the temperature differences. In this case the temperature can be used in °C

4 Heat gain through losses

The process of energy conversion and transmission in hydraulic systems involves losses taking the form of heat. This heat is absorbed and carried by the hydraulic fluid.

The total power loss P_{vtot} of a hydraulic system comprises a number of separate losses as follows:

P_{v1} Efficiencies of components
P_{v2} Internal leakage
P_{v3} Throttles
P_{v4} Flow resistances

$$P_{v\,tot} = \Sigma P_v = P_{v1} + P_{v2} + P_{v3} + P_{v4} \qquad (21)$$

4.1 Losses due to component efficiencies

$$P_{v1} = \frac{\dot{V} \cdot p}{600 \cdot \Pi\eta} \ \text{[kW]} \qquad (22)$$

$\dot{V}$ = Total volumetric flow in dm³/min
p = Operating pressure in bar
$\Pi\eta$ = Product of all efficiencies, e.g. pumps and motors

4.2 Losses due to internal leakage

$$P_{v2} = \frac{\dot{V}_L \cdot \Delta p}{600} \qquad (23)$$

$\dot{V}_L$ = Internal leakage in dm³/min
Δp = Pressure difference in bar

In practice the internal leakage of the pumps is included in the efficiency of the pumps themselves so here only the pump losses at zero stroke and the internal leakage at clearances in the valves need be taken into account.

4.3 Losses at throttles

Substantial losses are sometimes incurred when controlling the flow of fluid by means of throttles, metering grooves and orifice plates. Such losses must always be allowed for, particularly with directional control valves, proportional valves and servo valves.

$$P_{v3} = \frac{\dot{V}_1 \cdot p_1}{600} + \frac{\dot{V}_2 \cdot p_2}{600} + \frac{\dot{V}_n \cdot p_n}{600} \qquad (24)$$

$\dot{V}$ = The volumetric flow at the throttle in dm³/min
p = The pressure drop at the throttle in bar

4.4 Losses from flow resistances

The flow of hydraulic fluid through components and pipework involves friction losses which cause a pressure drop in the hydraulic system.

$$P_{v4} = \frac{\dot{V} \cdot \Sigma\Delta p}{600} \qquad (25)$$

$\dot{V}$ = Volumetric flow in dm³/min
$\Sigma\Delta p$ = Total pressure drop in bar (sum of all pressures)

If necessary, several calculations have to be performed for the different flow paths for the fluid in a hydraulic system. In practice the losses arising in the pipes are radiated again from the pipes.

5 Heat loss through components

Heat is lost through the components of a hydraulic system, the tank and pipework at a rate depending on the surface area, wall thickness and velocity of the media. In practice only the heat loss from the tank is generally taken into account. The heat radiated from other components can be neglected or included as an extra safety factor.

5.1 Heat loss through heat exchangers

In practice the losses in a hydraulic system which take the form of heat must be dissipated through a heat exchanger. The heat lost from the tank can be taken into account.

5.2 The tank as heat exchanger

The simplest heat exchanger is the fluid storage tank itself.
For the tank:

$$\dot{Q} = k \cdot A\,(T_{tank} - T_{ambient}) \qquad (26)$$

Due to the normally low velocities of both the fluid and the air and the comparatively large wall thickness, the coefficient of heat transmission k is low. With stagnant air and a low fluid velocity, $k = 0.012$ kW/m² K.

Tests have revealed that only the area actually in contact with the fluid should be used for A in the equation.

The areas and power dissipation of the standard tanks of DIN 24336 are listed in *Table 14.*

Nominal tank size	A in m²	ΔT= 20 K	ΔT= 30 K	ΔT= 40 K
63	0,89	0,21	0,32	0,42
100	1,16	0,28	0,42	0,56
160	1,58	0.38	0.57	0.76
250	2,12	0,51	0,76	1,02
400	2,98	0,72	1,07	1,44
630	3,91	0,94	1,41	1,88
800	4,75	1,14	1,71	2,28
1000	5,4	1,30	1,94	2,60

Table 14: *Power dissipation P_B in kW of tanks when $k = 0.012$ kW/m² K*

With a temperature difference of 30 K the heat flow from a freestanding tank is approximately 0.36 kW/m².

Pipes and components are also effective heat exchangers in large, extensive installations. The amount of heat dissipated can be ascertained in the same way as the heat dissipation from tanks. It is not normally taken into account in the heat balance.

5.3 Active heat exchangers

"Active heat exchangers" in hydraulic systems are the types that were described on *pages 73 and 74.*

The cooling capacity rating required of the heat exchanger depends on the heat balance for the total system *(see page 83).* The amount of heat fed into the system by the losses must be equal to the amount of heat dissipated through the heat exchanger and tank.

Thus, $\dot{Q} = \dot{Q}_{A\,tank} + \dot{Q}_{A\,heat\,exchanger}$

In practice, calculation of the power losses in hydraulic systems usually only takes account of the losses due to internal leakage in pumps, motors and valves and the throttling losses with proportional valves and servo valves. In many cases, particularly with small tanks, the amount of heat dissipated by the system tank is neglected. Even with large tanks installed in unventilated places and therefore not dissipating their heat to the atmosphere the amount of dissipated heat is not taken into account in the calculation. Sometimes it is even possible for heat to be gained by the hydraulic system through the tank and pipework.

6 Heat balance for hydraulic systems

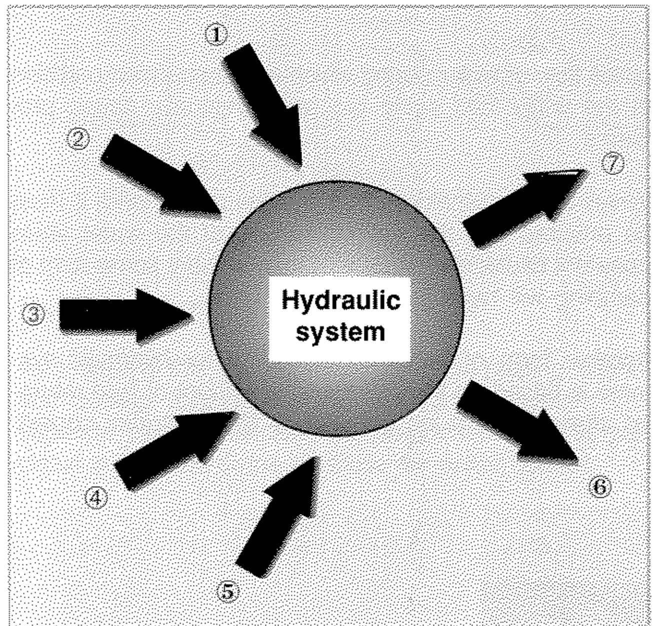
The temperature of the hydraulic fluid depends on

- the power losses
- the place of installation
- the surface area of heat-radiating components (such as the tank).

The maximum permitted fluid temperature depends on

- the type of fluid
- the requirements of the system.

A heat balance according to *Equation 1* must be drawn up according to the various factors of influence and the maximum permitted fluid temperature.



Heat gain from

1 Heater and/or environment, e.g. warm room
2 Power losses from pumps and motors
3 Power losses from internal leakage
4 Power losses from throttles (proportional valves and servo valves)
5 Power losses from flow resistances

Heat loss from

6 Components, especially tanks
7 Active heat exchangers

Fig. 31: *Heat balance for hydraulic systems*

7 Controlling the thermal status

In simple hydraulic systems with small energy flows the fluid tank is sufficient as a heat exchanger to dissipate the power losses from the system.

In the case of hydraulic units with pressure-compensated pumps the leakage fluid from the pump is very often passed through a air/oil cooler incorporated in the bell housing.

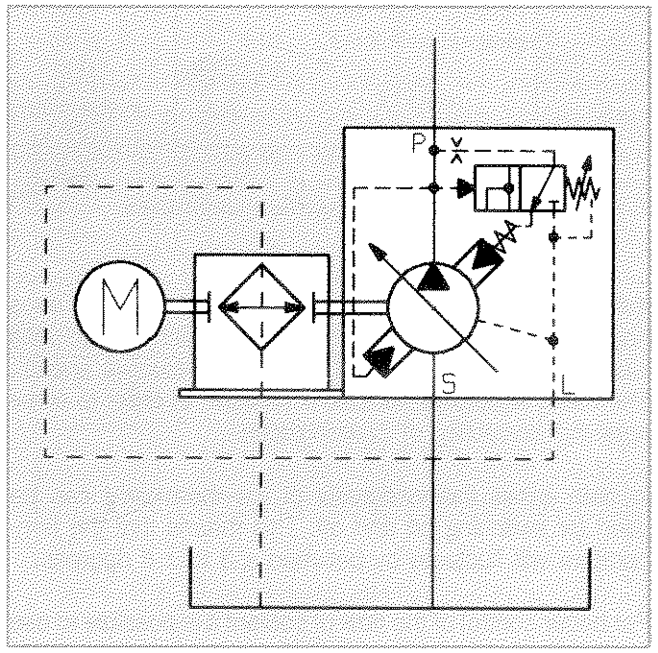


Fig 32: *Circuit diagram of a pump unit with oil-air cooler*

As the performance demands on hydraulic systems rise so too do the demands for a constant fluid temperature. This has to be taken into account when designing hydraulic systems.

The simplest type of cooling employs switching valves. In order to cool the operating fluid when a certain temperature is reached - the limits can be adjusted - a thermostatic valve allows cooling water to pass to a heat exchanger. It is the fluid returning from the system to the tank that is cooled.

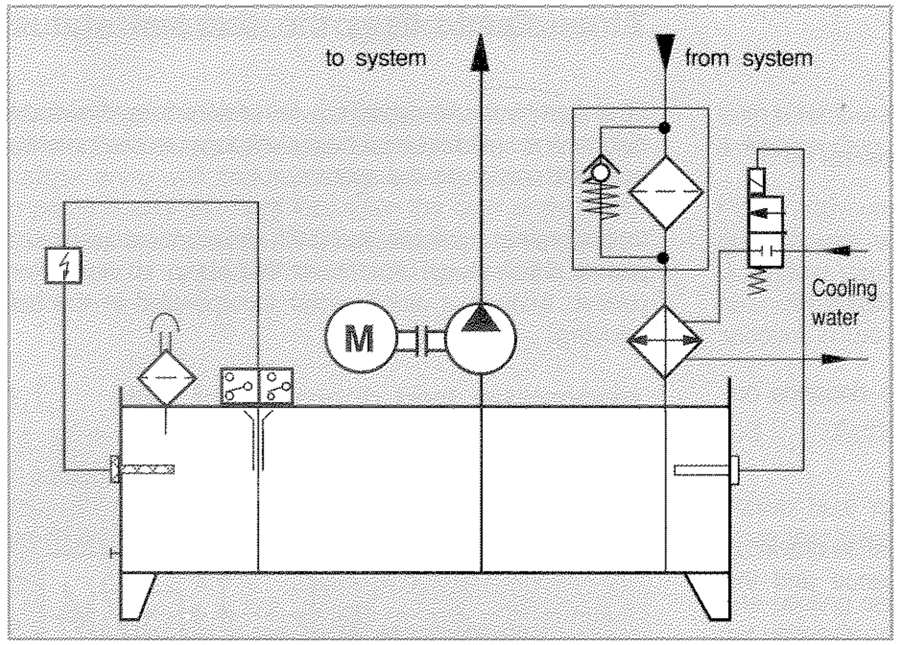


Fig. 33: *Circuit diagram of on/off control*

If the fluid temperature falls, a thermostat switches on a heater coil in order to heat the fluid. The element provides localized heating of the fluid through a protective sheath. The rating should not exceed 0.7 W/cm² in order to eliminate the possibility of carbonization.

In both cases the hysteresis of the thermostat can be used for switching on and off. This means that the temperature will fluctuate by between ± 3 to ± 8 degrees.

For better control of the flow of coolant it is possible to use a proportional valve instead of the thermostatic water valve with a simple open/closed function. Valves with a proportional function have better control characteristics. However, at low flows they sometimes provide an inadequate flow of water during which any suspended particles in the water can settle out and eventually cause problems.

When using switching valves, on the other hand, it must be ensured that the sudden closing of the water valve does not cause any difficulties; if necessary some form of damping will have to be incorporated.

Another form of temperature control for hydraulic systems is based on the fact that the amount of fluid returned after control is too great to be filtered and/or cooled.

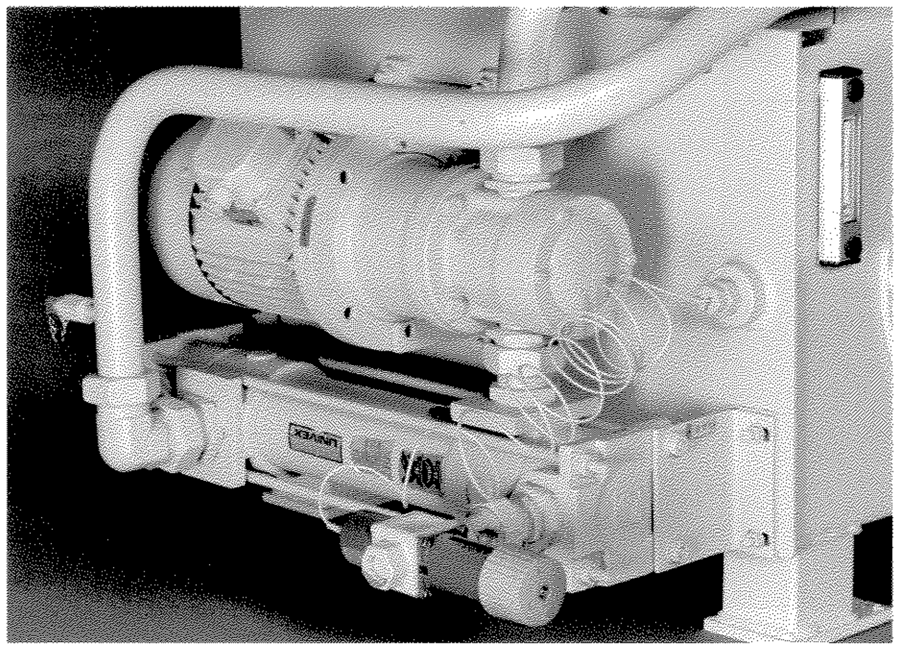
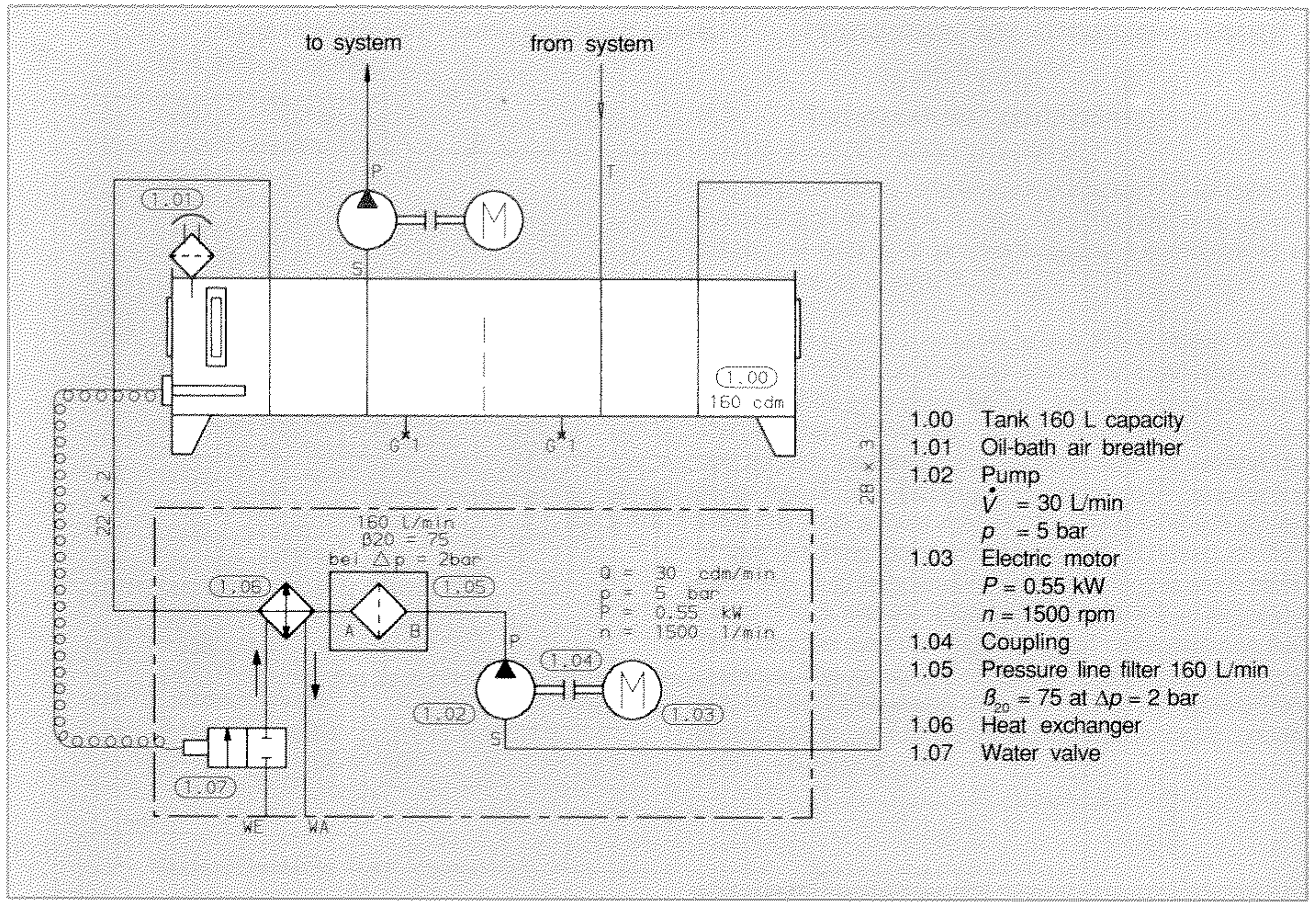


Fig. 34



1.00 Tank 160 L capacity
1.01 Oil-bath air breather
1.02 Pump
$\dot{V}$ = 30 L/min
p = 5 bar
1.03 Electric motor
P = 0.55 kW
n = 1500 rpm
1.04 Coupling
1.05 Pressure line filter 160 L/min
β_{20} = 75 at Δp = 2 bar
1.06 Heat exchanger
1.07 Water valve

Fig. 35: *Circuit diagram of a filter-cooler unit*

As a result of this, it is common to find the heating, filtering and cooling arrangements of a hydraulic system in a separate circuit. Basically, the components of this subcircuit will be the same as those for on/off control.

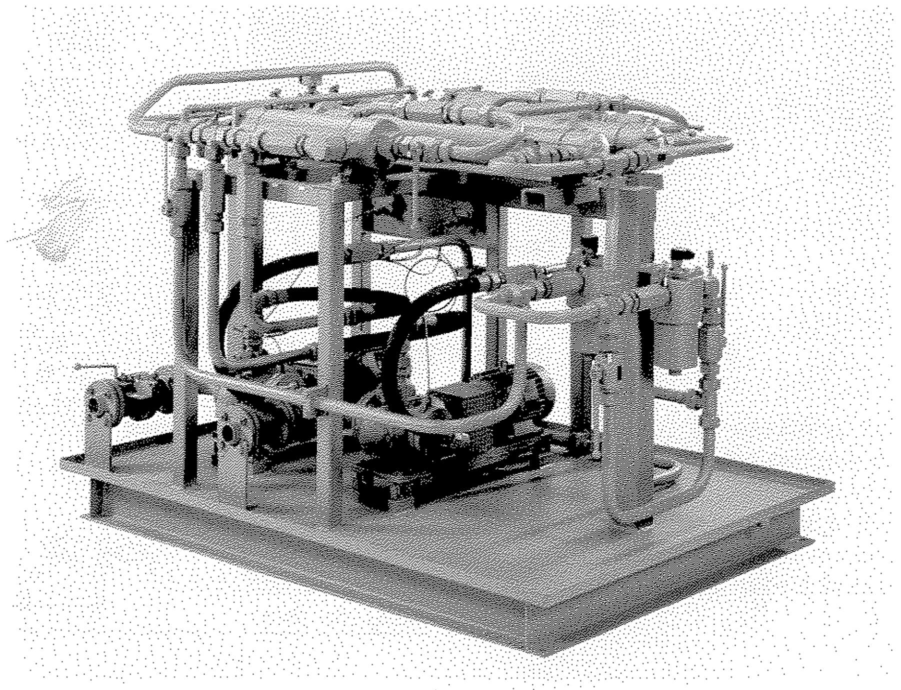


Fig. 36

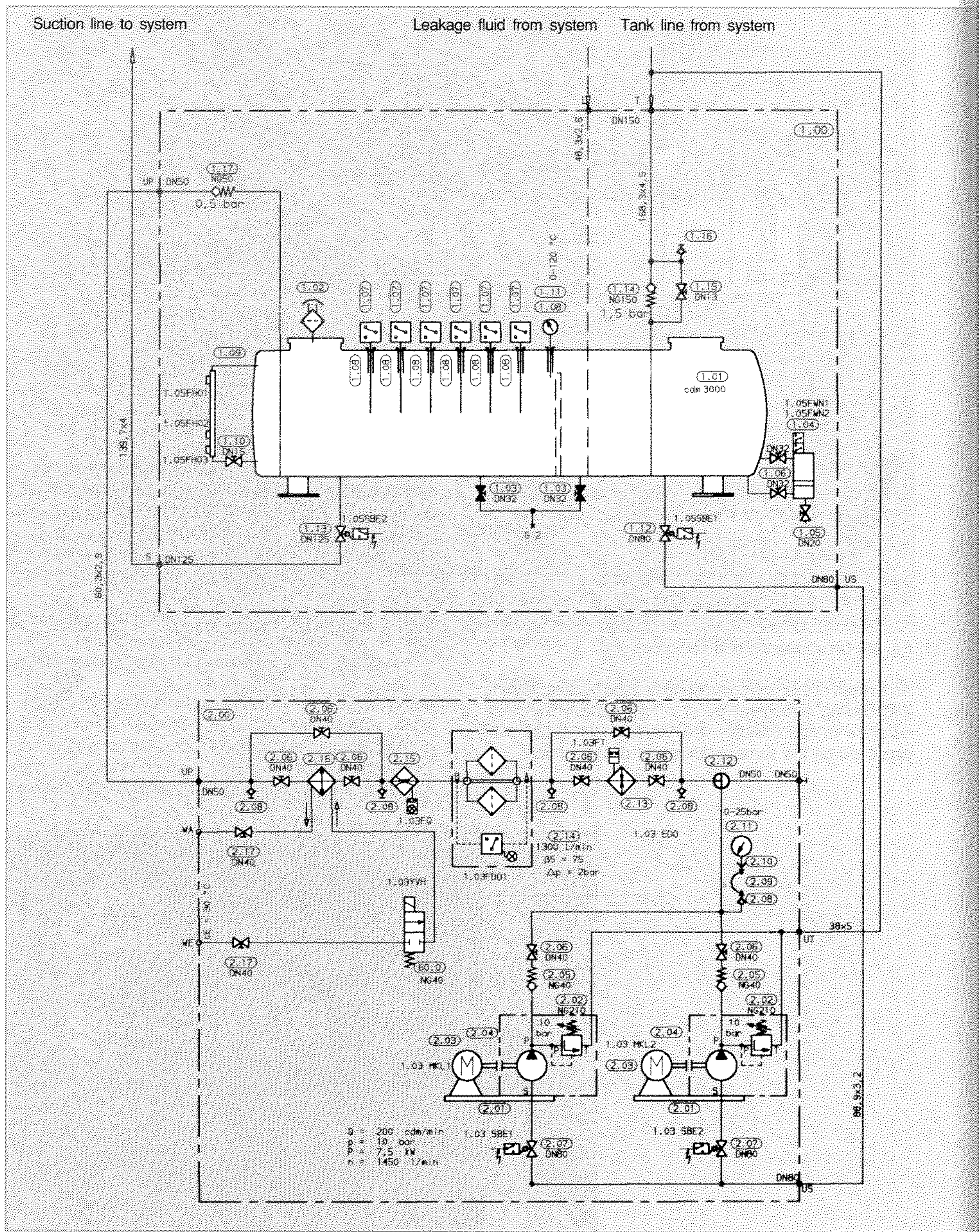


Fig. 37: *Circuit diagram of a heater-filter-cooler unit*

1.00 **Tank unit comprising:**

1.01 Tank, $\dot{V}$ = 3000 L capacity
1.02 Oil-bath breather
1.03 Ball valve
1.04 Water alarm
1.05 Ball valve
1.06 Ball valve
1.07 Thermostats (from left to right)
40°C cooling "off" 1.05 FT04
50°C cooling "on" 1.05 FT03
45°C heater "off" 1.05 FT06
35°C heater "on" 1.05 FT05
60°C fluid "too hot" 1.05 FT02
25°C fluid "too cold" 1.05 FT01
1.08 Sleeve
1.09 Float switch
1.10 Flange valve
1.11 Thermometer, 0 to 120°C
1.12 Gate valve with limit switch
1.13 Gate valve with limit switch
1.14 Check valve,
nominal size 150, $p_{cracking}$ = 1.5 bar
1.15 Ball valve
1.16 Threaded coupling
1.17 Check valve,
nominal size 50, $p_{cracking}$ = 0.5 bar

2.00 **Heater-filter-cooler unit comprising:**

2.01 Base frame
2.02 Helical screw pump,
$\dot{V}$ = 200 L/min, p = 10 bar
2.03 Electric motor P = 7,5 kW, n = 1450 rpm
2.04 Coupling
2.05 Check valve, size 40
2.06 Ball valve
2.07 Gate valve with limit switch
2.08 Threaded coupling
2.09 Hose
2.10 Connector
2.11 Pressure gauge
2.12 Three-way valve
2.13 Flow heater, 9 kW rating
2.14 Duplex return line filter
1300 L/min, β_5 = 75 at Δp = 2 bar
2.15 Flow switch
2.16 Heat exchanger
2.17 Shut-off valve
2.18 Water stop valve

8 Hardware for thermal control

8.1 Temperature controller

The functions of a temperature controller - also called a thermostat in its simplest form - are the control of temperature, the display of temperature and the monitoring of temperature in a system.

Most temperature controllers utilize the principle of fluid expansion. The temperature sensor is connected by a capillary tube to a diaphragm in the switching controller. The volume of the fluid in the sensing system varies directly with the temperature. The change in volume deflects the diaphragm which is linked to a snap-action switch through a lever mechanism. By simple adjustment of the switching point the device can be used for limit switching, fluid temperature control or temperature monitoring. Shock and vibration must not affect their accuracy. The controller and thermometer must have separate fluid systems. The sensor is contained in a protective sleeve and it must be ensured that the part of the sleeve containing the sensor is immersed in the hydraulic system fluid.

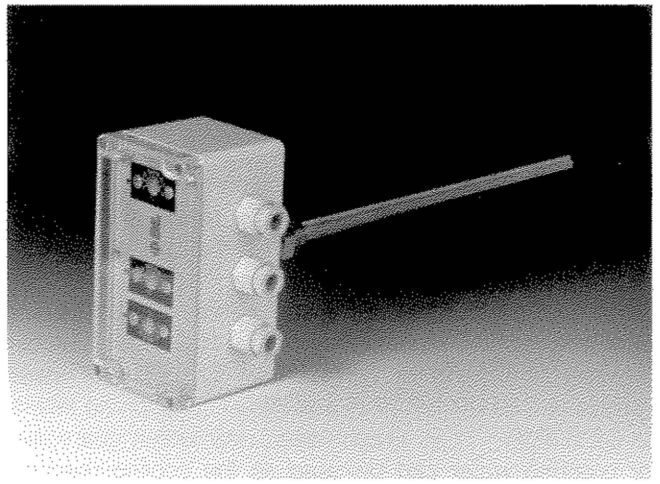


Fig. 38: *Thermostat*

8.2 Thermostatic water valve

Thermostatic water valves are used for regulating the flow of cooling water. They also employ the principle of fluid expansion. The temperature sensor is enclosed in a protective sleeve and connected by capillary tube to the actual water valve. The opening temperature of the valve can be preset. Both direct-operated and pilot-operated types are used.

Some of the valves incorporate a damping device in order to produce smoother closing.

There are other valves the opening of which, and therefore water throughput, is proportional to the temperature. However, when using such valves it must be anticipated that the efficiency of the oil-water heat exchanger will be lower.

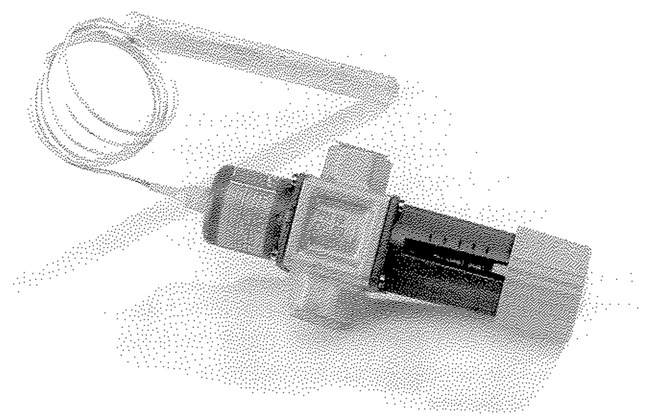


Fig. 39: *Thermostatic water valve*

8.3 Fluid-water heat exchanger

Typical oil-water heat exchangers are the counter-flow and counter-flow/parallel-flow types described on pages 5 and 6. Both types are available in several different versions - simplex tube-type heat exchanger, duplex tube-type heat exchanger, spiral-tube heat exchanger. The one factor common to them all is the attempt made at intensive contact between the two media at the heat-exchange tubes by a variety of mechanical methods.

Such heat exchangers are used for most of the different operating media - HLP mineral oil to DIN 51 524, HFA oil-in-water emulsions to CETOP RP 77 H, HFC water glycols to CETOP HP 77 H and HFD-R phosphate esters to CETOP RP 77 H. Fresh water, river water, sea water or brackish water can be used as the cooling medium depending on the type of materials chosen. Correct selection of materials is very important when ordering equipment.

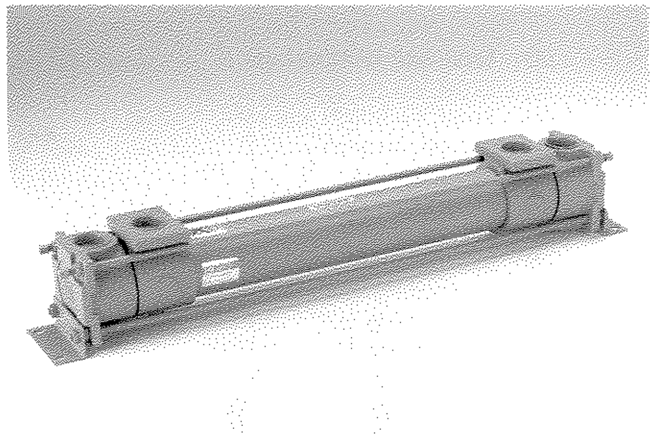


Fig. 40: *Water-oil heat exchanger*

8.4 Air-oil heat exchanger

In an air-oil heat exchanger, or radiator, the cooling air is drawn through the cooler by a fan. The fan can be driven by an electric motor or a hydraulic motor. All the usual hydraulic fluids can be accommodated. When ordering such heat exchangers, however, it is necessary to state whether they will be installed in a normal environment or in a salt-laden atmosphere.

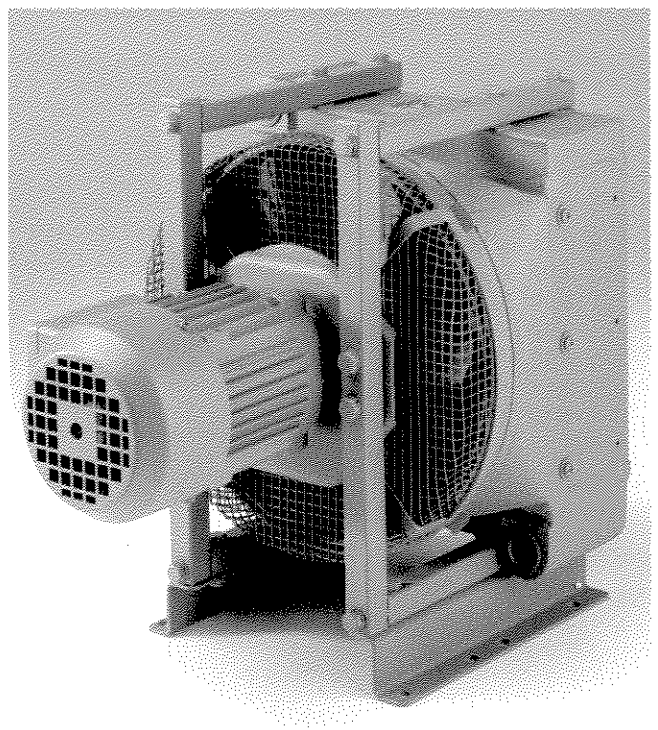


Fig. 41: *Air-oil heat exchanger*

8.5 Immersion heater

Electric immersion heaters are pure resistance heating devices. They are used for heating the fluid in hydraulic systems. The surface power density must be such that there can be no localized overheating of the fluid even when it is stagnant. The normal practice is to fit the heater horizontally below the surface of the fluid. By incorporating a suitable protective sleeve it will be possible to change the heater in the event of a malfunction without having to drain the tank.

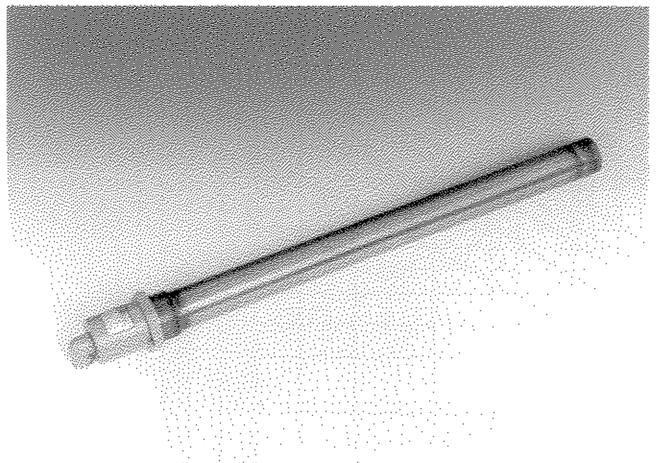


Fig. 42: *Immersion heater*

8.6 Flow heater

Flow heaters are used in circulating units, heater units and cooler-filter units. They are heat exchangers containing a fluid-filled resistance heater. The heat from the element passes first to the primary fluid and then through tubes to the operating fluid of the system which is circulated through the heater. It is essential for the heater only to be switched on when fluid is flowing through it and there must also be over temperature protection. Such heaters can be mounted in any position.

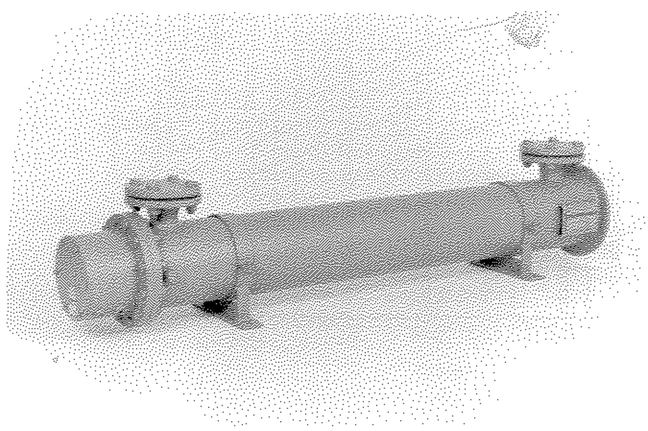


Fig. 43: *Flow heater*

9 Practical applications and typical calculations

Design calculations for heat exchangers make use of the graphs of effectiveness in relation to *NTU* values.

For project design purposes the practice is to make use of the pre-designed sizes of heat exchanger that are already available on the market.

Practical experience has shown that the simplest method is for designers to use the graph of power(kW)/temperature(Kelvin) in relation to the volumetric flow of the hotter medium with a given ratio, volumetric flow of hot medium to volumetric flow of cold medium.

Both kW/Kelvin and volumetric flow curves have been plotted logarithmically in the diagrams in order to obtain almost straight curves.

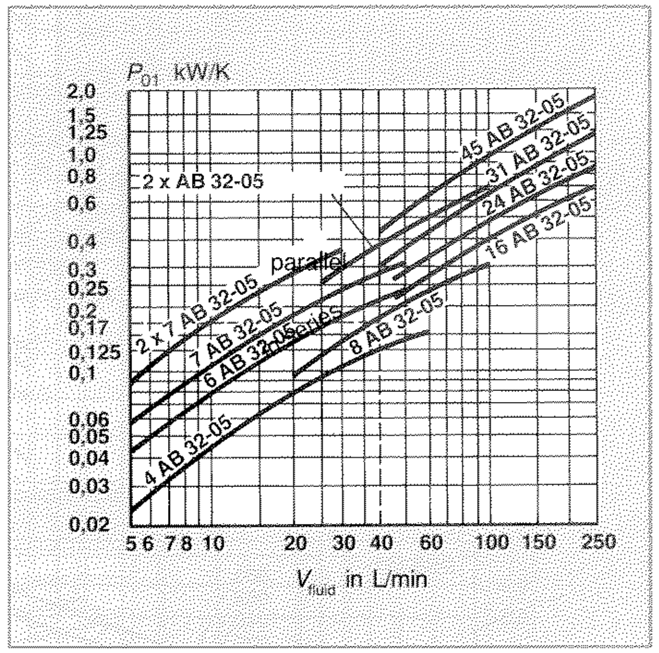


Diagram 14: *Power diagram for heat exchangers of spiral finned-tube and double-tube types*

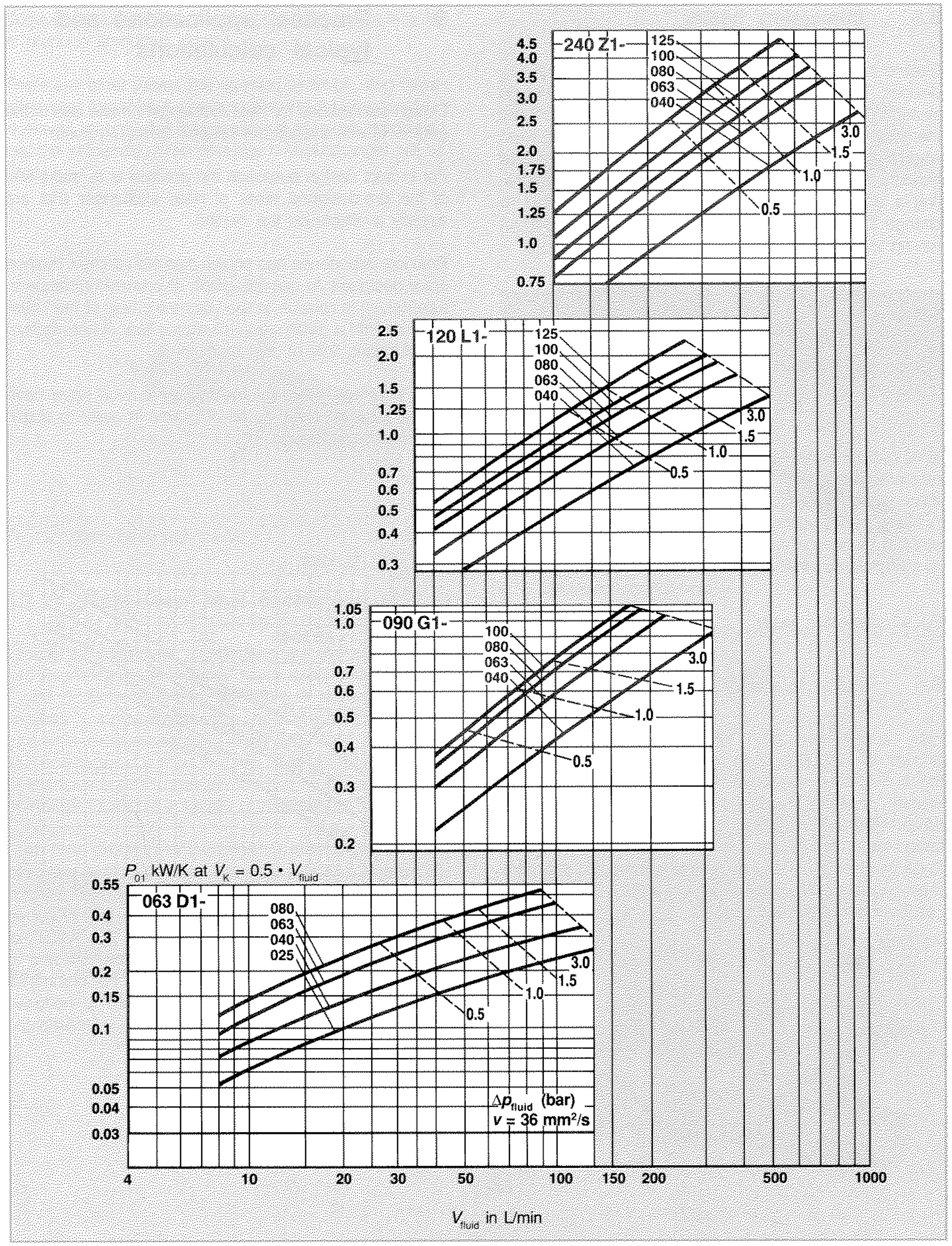


Diagram 15: *Performance diagram for heat exchangers with increased water consumption ($\dot{V}_K$ = 0,5 x $\dot{V}_{fluid}$)*
e.g. for drinking water and well water

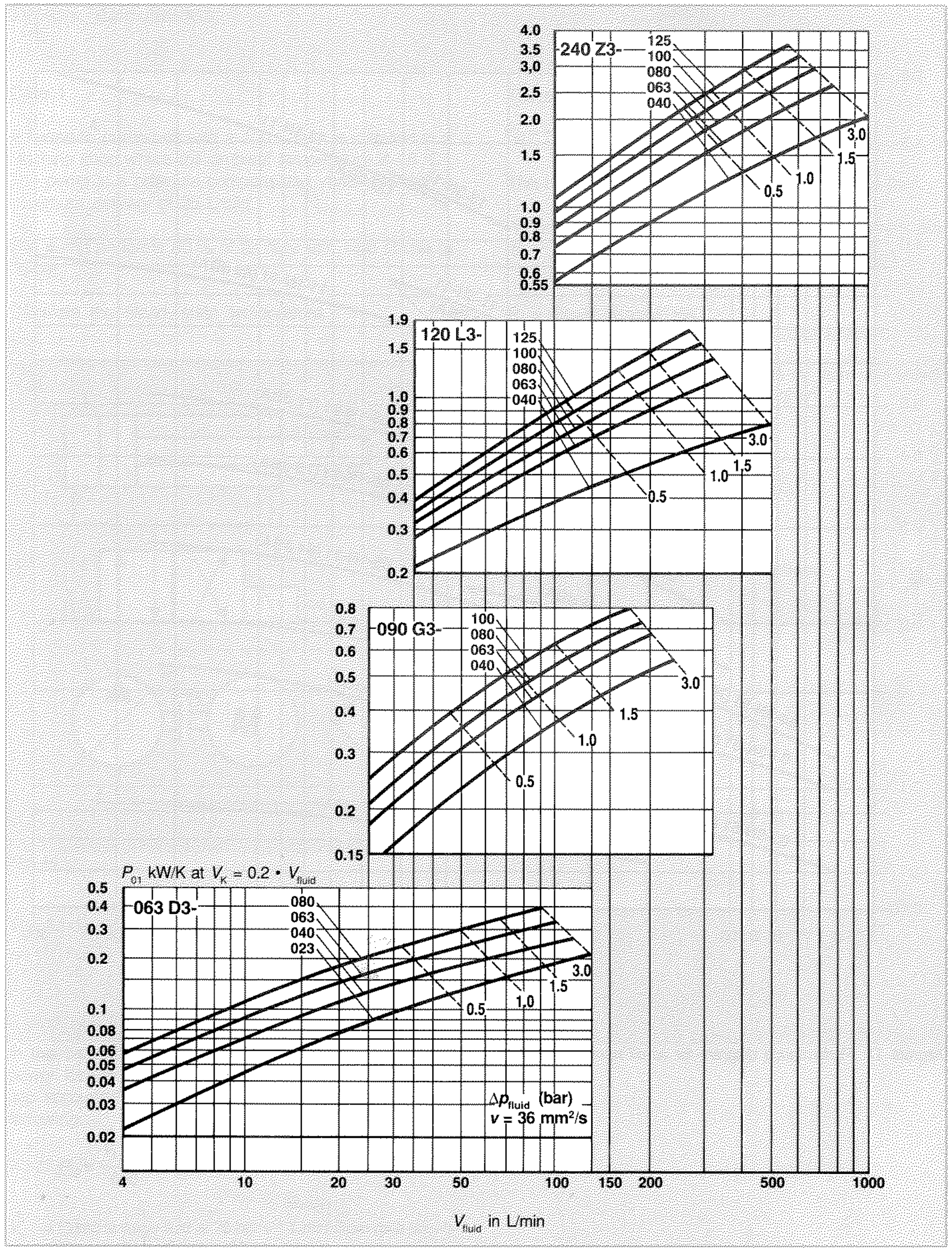


Diagram 16: *Performance diagram for heat exchangers of low water consumption ($\dot{V}_K$ = 0,2 x $\dot{V}_{fluid}$)*
e.g. for industrial water, stream and river water, seawater and brackish water

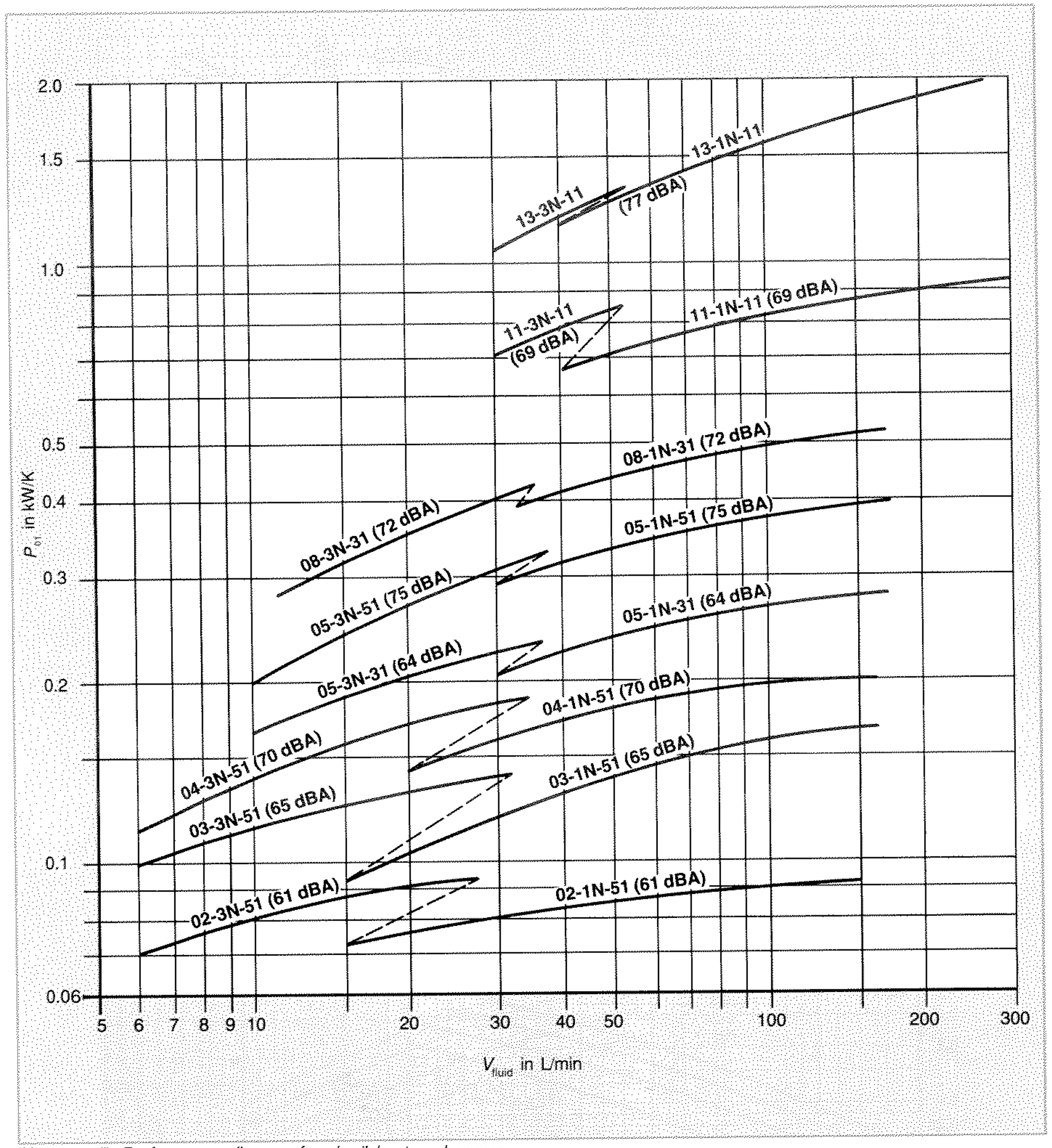


Diagram 17: *Performance diagram for air-oil heat exchangers*

Typical calculations

Case 1

A hydraulic pump unit with a 100 L tank is installed in a machine shop where the ambient temperature is 18 °C. The pump is a pressure-compensated vane type with a maximum delivery of 25 L/min.

The pump performs 80 % of its duty with zero stroke at 70 bar.

Calculate the steady-state temperature.

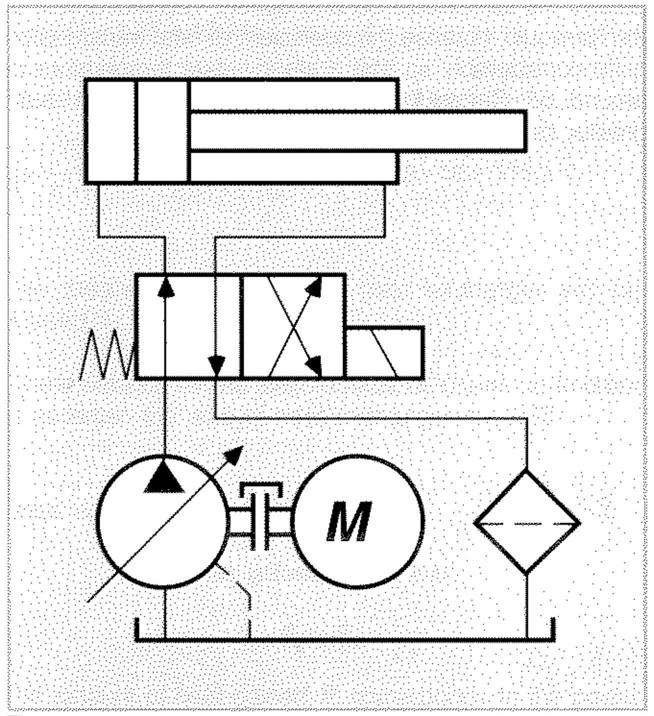


Fig. 44

The fundamental law applicable is:

Heat gained = Heat lost

$$P_v = P_w$$

In this case the heat gained by the system is from the power losses incurred by zero-stroke operation. Taking into account the duty factor *(DF)*, *Equation 23* gives the following:

$$P_{V2} = \frac{\dot{V}_L \cdot p}{600} \cdot DF$$

$\dot{V}_L$ = Pump leakage fluid at 70 bar = 1 L/min (from data sheet)
p = Zero-stroke pressure = 70 bar (given)
DF = Duty factor = 0.8 (given)

Therefore:

$$P_{V2} = \frac{1 \cdot 70}{600} \cdot 0{,}8 = 0{,}093 \text{ kW}$$

The heat is dissipated through the sides of the tank.

According to *Equation 26* the steady-state temperature is:

$$T_2 = \frac{P_V}{k \cdot A} + T_1$$

P_v = Power losses in kW
k = Coefficient of heat transmission 0.01 kW/m²°C (from references)
A = Radiating surface of tank = 1.16 m² (from data sheet)
T_1 = Ambient temperature = 18 °C (given)

In this particular example:

$$T_2 = \frac{0{,}093}{0{,}01 \cdot 1{,}16} + 18 = 26 \text{ °C}$$

No separate heat exchanger is needed.

Case 2

In the installation described in Case 1 a proportional valve is to be used to control the cylinder which will be working for 20 % of the duty with a pressure difference of 30 bar.

Re-calculate the steady-state temperature.

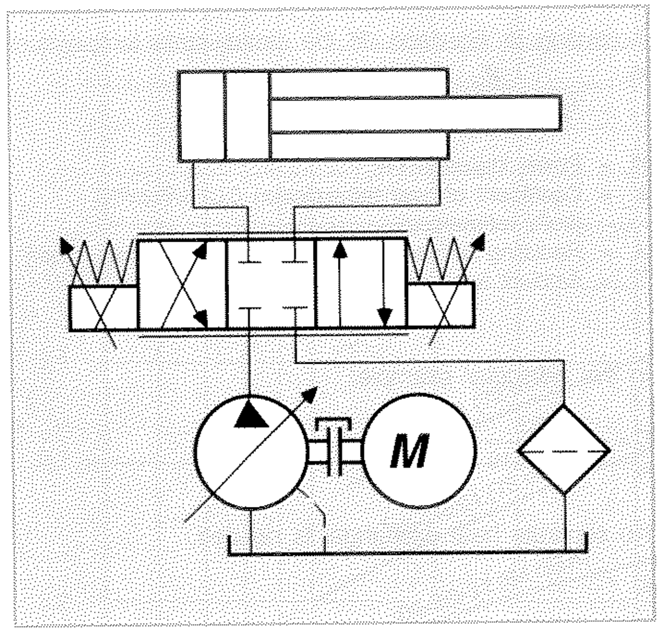


Fig. 45

The power losses P_{v3} arising from the throttling at the proportional valve must be added to the power losses P_{v2} calculated in Case 1. According to *Equation 24* the losses are:

$$P_{V3} = \frac{\dot{V}_L \cdot p}{600} \cdot DF$$

$\dot{V}$ = Volumetric flow of pump = 25 L/min (given)
Δp = Pressure difference at throttle = 30 bar (given)

Therefore:

$$P_{V3} = \frac{25 \cdot 30}{600} \cdot 0{,}2 = 0{,}25 \text{ kW}$$

The total power losses are:

$$P_V = P_{V2} + P_{V3}$$

$$P_V = 0.093 + 0.25 = 0.343 \text{ kW}$$

Therefore, the steady-state temperature is:

$$T_2 = \frac{P_V}{k \cdot A} + T_1$$

$$T_2 = \frac{0{,}343}{0{,}01 \cdot 1{,}16} + 18 = 47{,}6 \text{ °C}$$

No separate heat exchanger is required in this case either since the steady-state temperature of 50 °C normally permitted for hydraulic systems has not been exceeded. In fact, the ambient temperature could even rise to 22 °C before any extra provision for heat dissipation would be necessary.

In this particular case some additional air circulation around the tank would be one option. The movement of the air would increase the coefficient of heat transmission quite considerably so that a steady-state temperature of around 50 °C could still be maintained with an even greater rise in the ambient temperature.

Case 3

The system described in Cases 1 and 2 is to have an additional function. It means that the system pressure will have to be increased to 100 bar. Also, the duty factor of the proportional valve will be increased from 20 % to 70 % and the pressure drop at the throttling point from 30 bar to 60 bar. Moreover, the pump will only be running for 30 % of the time at zero stroke. Once again a steady-state temperature of 50 °C must not be exceeded.

The power losses must be re-calculated according to the known equations:

$$P_V = P_{V2} + P_{V3} = \frac{1 \cdot 100}{600} \cdot 0{,}03 + \frac{25 \cdot 60}{600} \cdot 0{,}7 =$$

$$= 0{,}05 + 1{,}75 = 1{,}8 \text{ kW}$$

Therefore, the steady-state temperature is:

$$T_2 = \frac{P_V}{K \cdot A} + T_1 = \frac{1{,}8}{0{,}01 \cdot 1{,}16} + 18 = 155{,}2 + 18 = 173{,}2 \text{ °C}$$

This means that a separate heat exchanger will be needed.

A heat balance for the system must be drawn up in order to ascertain the size of heat exchanger required.

Once again:

Heat gained = Heat lost

$$P_V = P_W$$

The amount of heat gained P_v corresponds to the power losses P_{v1} and P_{v3}

$$P_V = P_{v1} + P_{v3}$$

The amount of heat lost P_w comprises the amount of heat P_T that can be dissipated from the fluid tank and the amount of heat P_k that will have to be dissipated by a separate heat exchanger.

$$P_W = P_T + P_k$$

With an additional steady-state temperature T_2 of 50°C and an ambient temperature T_1 of 18 °C , the amount of heat dissipated from the tank according to *Equation 26* becomes:

$$P_T = (T_2 - T_1) \cdot k \cdot A$$

For this particular example:

$$P_T = (50 - 18) \cdot 0{,}01 \cdot 1{,}16 = 0{,}37 \text{ kW}$$

Therefore, the amount of heat to be dissipated by the heat exchanger is:

$$P_k = P_{v1} + P_{v3} - P_T = 1{,}8 - 0{,}37 = 1{,}43 \text{ kW}$$

First investigate the possibility of fitting a air-oil heat exchanger between pump and motor.

Such heat exchangers cool the leakage fluid from the pump and it can be assumed that the temperature of the leakage fluid will be about 20 °C higher than the steady-state temperature of the system. Also the volumetric flow of the leakage fluid will be a uniform 1 L/min all the time the pump is running.

Thus, according to the catalogue data, a air-oil heat exchanger incorporated in the pump base will be able to dissipate 0.2 kW of heat.

Obviously, therefore, there is no sense in using this type of heat exchanger. It must be either a separate air-oil heat exchanger or a fluid-water unit.

In this case the designer decides on a water-oil unit of low water consumption.

The size of the heat exchanger is ascertained from *Diagram 16.* The performance diagrams for heat exchangers show the amount of power that can be dissipated in relation to the volumetric flow of the fluid per degree temperature difference.

$$P_0 = \frac{P_k}{T} \quad \frac{\text{kW}}{\text{°C}}$$

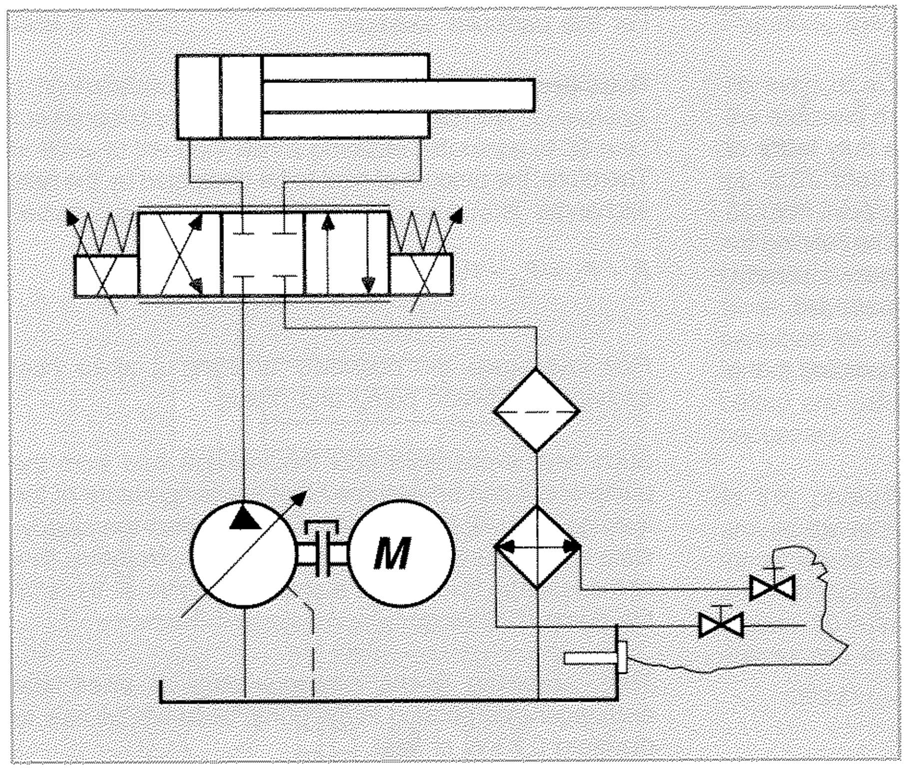


Fig. 46

In selecting a heat exchanger it is necessary to calculate the average volumetric flow through the heat exchanger of the fluid that needs to be cooled. In this case the total return flow is to pass through the heat exchanger. Thus:

$$\dot{V}_k = \dot{V}_{max} \cdot DF = 25 \cdot 0.7 = 17.5 \text{ L/min}$$

The smallest heat exchanger shown in *Diagram 17* has a specific thermal capacity of 0.07 kW/°C at a volumetric flow of 17.5 L/min. Therefore, in dissipating 1.43 kW the temperature would be:

$$T_{2theor.} = \frac{P_k}{P_0} = \frac{1.43}{0.07} = 24\ °C$$

This means that the heat exchanger is about twice as big as strictly necessary so a thermostatic water valve must be fitted to the water inlet in order to regulate the thermal status of the system. The end result for a steady-state temperature of 50°C is an approximate water consumption of:

$$\dot{V}_{water} = 0.2 \cdot \dot{V}_{fluid} \cdot DF = 0.2 \cdot 17.5 \cdot 0.5 = 1.75 \text{ L/min}$$

As a further alternative to the water-oil heat exchanger the air-oil type can now be investigated on the same basis.

A water-oil heat exchanger will be chosen if there is cooling water available at the place of installation because there is always the danger of fouling and a subsequent reduction in efficiency with air-oil heat exchangers.

10 Symbols and subscripts

Symbols

Symbol	Units	Quantity
$\dot{Q}$	kcal/h kJ/h	Heat flow
$\dot{U}$	kW	Internal energy
$\dot{V}$	m^3/s L/min	Volumetric flow
c	kJ/kg K	Specific thermal capacity
T	K	Absolute temperature
ρ	kg/m^3	Density of medium
λ	kW/m K	Coefficient of thermal conductivity
δ	m	Wall thickness
a	m^2	Flow area
d	m	Diameter of tube
α	kW/m^2 K	Heat transfer coefficient
k	kW/m^2 K	Coeefficient of heat transmission
ΔTm	K	Mean temperature difference between two media
p	bar	Pressure
P	kW	Power
H	h	Time
NTU		Number of heat transfer units
ε		Effectiveness of heat exchanger

Subscripts

Symbol	Quantity
WE	Hot inlet
WA	Hot outlet
KE	Cold inlet
KA	Cold outlet
in	Heat gained
out	Heat lost
1	Temperature of hot medium
2	Temperature of cold medium
a	external
i	internal
w	hot
k	cold
WT	Heat exchanger

11 References

AB-Projektnormen, AB 44-16
Mannesmann Rexroth GmbH, Lohr

Compact Heat Exchangers, Kays and London

Technische Information
Kühlerfabrik Längerer&Reich GmbH&Co KG, Filderstadt

Notes

Hydraulic Accumulators

Dr.-Ing. Norbert Achten

1 Introduction

Accumulators have a wide variety of applications in hydraulic systems because of their very useful properties.

Their main applications are for:

- Energy storage
- Emergency operation
- Leakage fluid make-up
- Volume compensation
- Shock absorption
- Pulsation damping

As its name suggests, a hydraulic accumulator is a pressure vessel that is able to store a specific volume of fluid sufficient to perform its intended purpose. When necessary, the fluid that the accumulator has taken from the system is returned to it without the need for any external supply of energy. The storage of pressure energy in the volume of fluid can be effected by either weight, spring or gas (i.e. hydro-pneumatically). Since hydro-pneumatic accumulators are the most popular the following chapter will concentrate on them.

2 Types of hydro-pneumatic accumulator

As *Fig. 47* shows, hydraulic accumulators can be classified according to two features:

- the energy carrier and
- the separating element.

The purpose of all hydraulic accumulators is to store pressure energy. In the mechanical types (i.e. weight-loaded and spring-loaded) it is performed by a change in potential energy. In contrast, with the gas-loaded accumulator it is the internal energy of a gas that is changed. For this type of accumulator, classification according to the separating element is ideal because they can be classified as either with or without a separating element.

Hydraulic accumulators with a separating element can be divided into:

- bladder-type accumulators
- diaphragm-type accumulators
- piston-type accumulators

The mode of operation of these accumulators utilizes the compressibility of a gas for storing a fluid. Nitrogen is often the energy carrier. Basically, a hydro-pneumatic accumulator comprises a fluid part and a gas part and a gas-tight separating element. The fluid part of the accumulator is connected to the hydraulic circuit so that, as the pressure rises, the gas in the gas part of the accumulator is compressed. Then, as the pressure in the system falls, the compressed gas expands and forces the stored fluid back into the system.

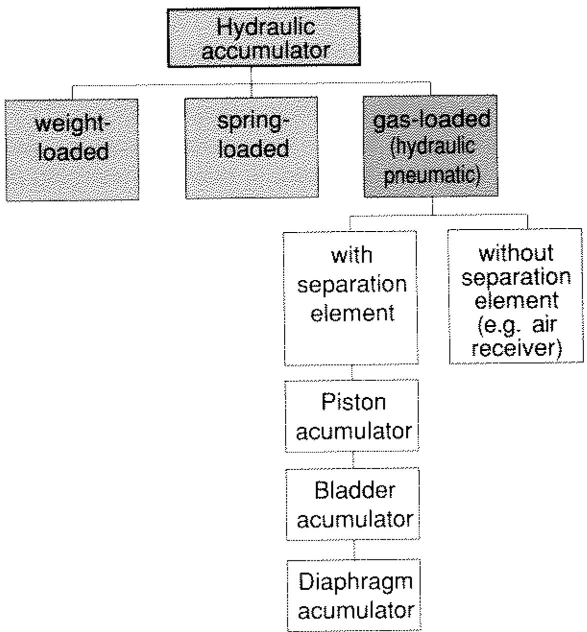


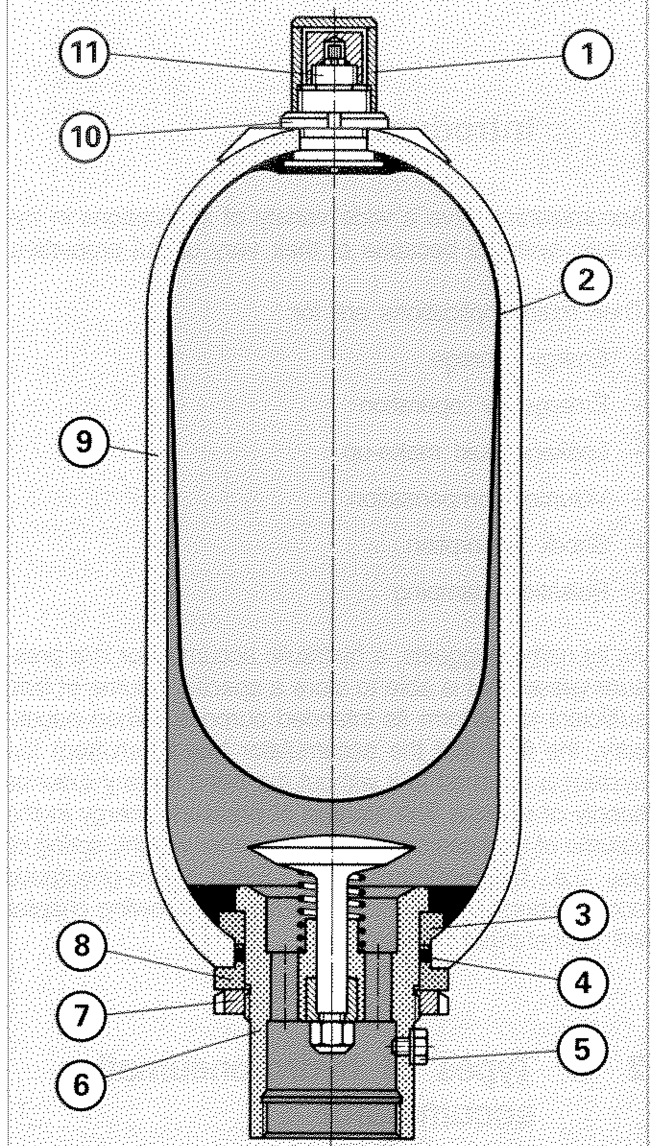
Fig. 47: *Classification of hydraulic accumulators*

2.1 Bladder-type accumulators

The bladder-type accumulator shown in *Fig. 48* comprises a strong pressure vessel, able to withstand the full system pressure, and an internal bladder containing the gas and made of a rubber-like material (elastomer). The bladder is filled through the gas valve at the top. The main purpose of the fluid valve at the bottom of the pressure vessel is to prevent the bladder being ejected with the out-flowing fluid.

For this purpose theopening area of the valve is sized so that a maximum volumetric flow depending on the size of the accumulator (approximately 120 L/s) cannot be exceeded. Volumetric flows of up to 140 L/s are possible with certain special designs called high-flow accumulators *(Fig. 49)*. The special feature of this design is a perforated disc in the connection fitting with the greater necessary opening area for the higher volumetric flow. An alternative version of the high-flow design is shown in *Fig. 50*. In this case the accumulator can be used at operating pressures up to 290 bar and the connection fitting contains a pre-loaded check valve which again prevents the bladder being ejected if there is a sudden drop in system pressure or the accumulator is emptied completely. The stem of the valve also incorporates a damping device so that the valve itself is not harmed by the high velocity flow during opening and closing.

Generally speaking, it is necessary for the hole in the pressure vessel for fitting the fluid valve to be of larger diameter than that for the gas valve. Consequently, it is normal to insert and remove the bladder from the fluid end. In a few exceptional cases when removing the accumulator in order to change the bladder would involve a large amount of dismantling, or fast bladder changing is essential, it is also possible to insert and remove the bladder from the gas end (the "top repairable" type, *see Fig. 51*). The mode of operation of the bladder-type accumulator is as follows, referring to *Fig. 52*: The bladder is filled with nitrogen to a certain pressure specified by the manufacturer according to the operating regime. At this point the fluid valve is closed. If, now, the charging pressure of the accumulator is exceeded in the system, the valve opens and the hydraulic fluid flows into the accumulator. As the pressure increases further the gas is compressed up to the maximum operating pressure p_2. The change in gas volume in the bladder between minimum and maximum operating pressure represents the useful fluid capacity.

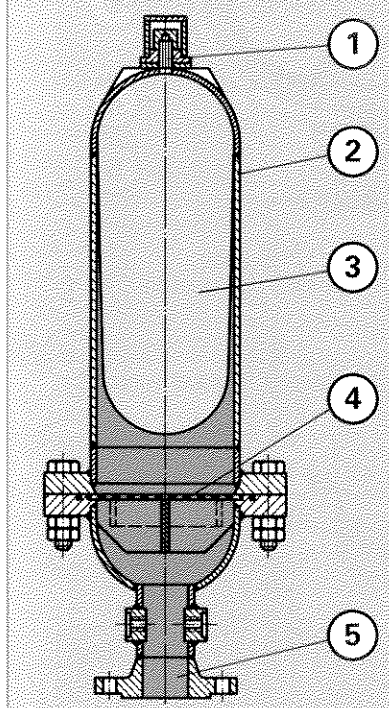


1 Sealing cap
2 Bladder
3 Split ring
4 Sealing ring
5 Vent screw
6 Fluid valve
7 Clamp nut
8 Spacer ring
9 Pressure vessel
10 Retaining nut
11 Gas valve

Fig. 48: *Hydro-pneumatic bladder-type accumulator*

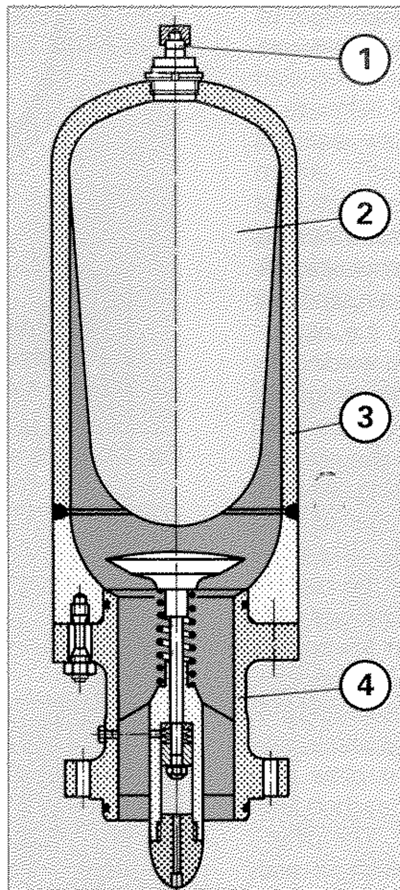
Hydraulic Accumulators

Although bladder-type accumulators can technically be mounted in any position, the vertical is preferred. In vertically mounted or inclined accumulators, the fluid valve must always be at the bottom.



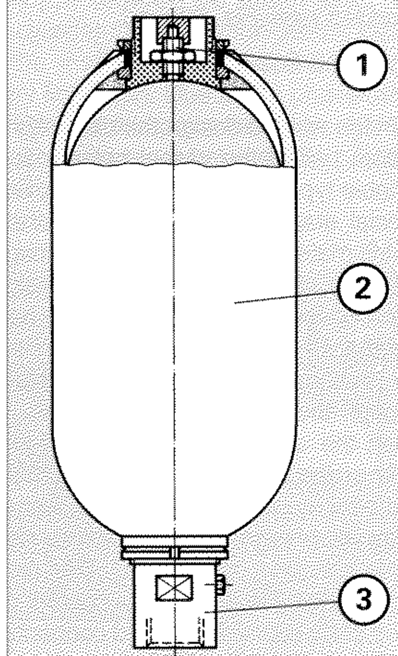
1 Gas valve
2 Pressure vessel
3 Bladder
4 Perforated disc
5 Fluid connection

Fig. 49: *High-flow low-pressure bladder-type accumulator*



1 Gas valve
2 Pressure vessel
3 Bladder
4 Fluid valve

Fig. 50: *High-flow high-pressure bladder-type accumulator*



1 Gas valve
2 Pressure vessel
3 Fluid valve

Fig. 51: *Hydro-pneumatic bladder-type accumulator, "top repairable" design*

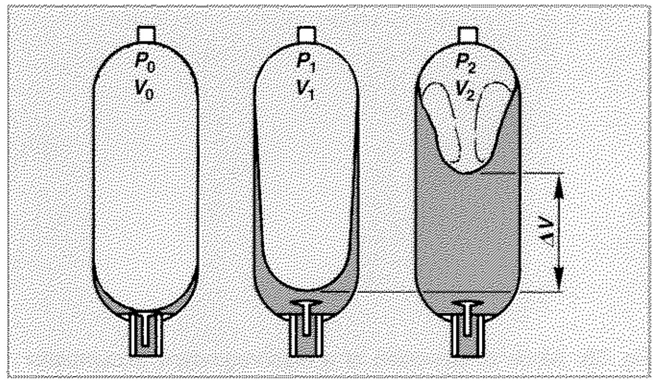


Fig. 52: *Schematic illustration of the operation of a bladder-type accumulator*

2.2 Diaphragm-type accumulator

The diaphragm-type accumulator shown in *Fig. 53* comprises a strong steel pressure vessel which is usually spherical or cylindrical. Inside the vessel is a separating element in the form of a diaphragm made of an elastic, malleable material such as an elastomer. For certain applications where high demands are made on the life of the elastometric material, as when corrosive fluids are present, it is an advantage to change the diaphragm at regular intervals. Consequently, there are two different designs:

- welded *(see Fig. 53)* and
- threaded body *(see Fig. 54).*

With the welded design the diaphragm is pressed into the bottom half of the vessel before the seam is welded. A suitable type of welding, such as electron beam, and the special diaphragm arrangement ensure that the elastometric material does not suffer damage during welding. In the case of the threaded body design the diaphragm is held between the top and bottom halves which are held together by a nut. In both designs there is a valve plate at the bottom to prevent the diaphragm being ejected through the fluid connection. There is a danger of this, when the accumulator is emptied completely. The principle of the diaphragm-type accumulator can be described best by referring to *Fig. 55.* At the start, the gas side of the diaphragm is connected to nitrogen at the appropriate charging pressure p_0. This causes the diaphragm to mould itself to the internal contour of the vessel and the valve plate seals off the fluid connection. In the same way as the bladder-type accumulator, the valve plate lifts when the minimum operating pressure p_1 is reached and hydraulic fluid can flow into the accumulator. The difference between the two gas volumes at minimum and maximum operating pressure represents the useful capacity for fluid. Although diaphragm-type accumulators can technically be mounted in any position the vertical is preferred.

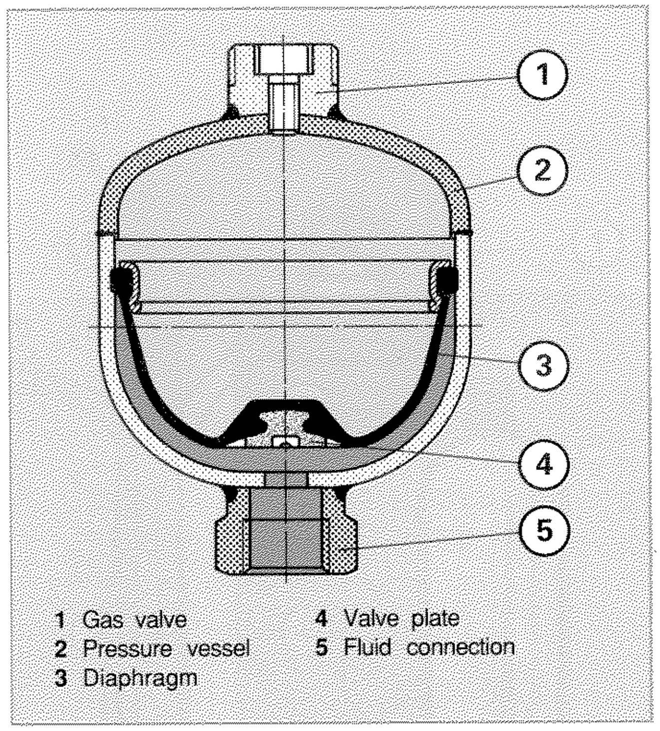


Fig. 53: *Diaphragm-type accumulator, welded version*

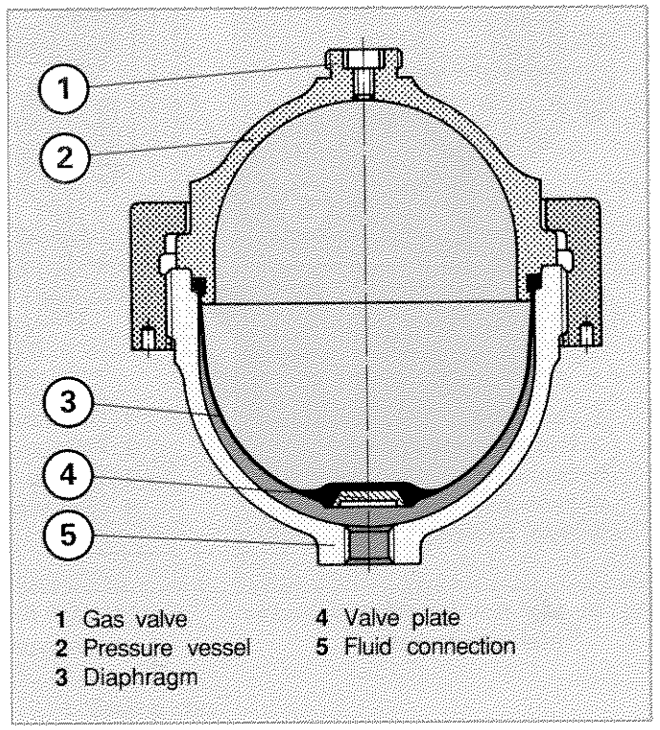


Fig. 54: *Diaphragm-type accumulator, threaded body version*

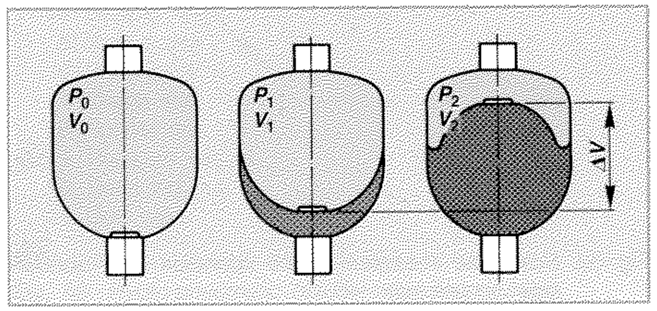


Fig. 55: *Mode of operation of a diaphragm-type accumulator*

2.3 Piston-type accumulator

The construction of a typical piston-type accumulator is illustrated in *Fig. 56.* Its principal components are the outer cylinder tube, the piston with its sealing system and the end covers which also contain the fluid and gas connections. The cylinder performs two functions - it stores the fluid pressure and also guides the piston which forms the separating element between the gas and fluid parts. The mode of operation of this type of accumulator is as follows, referring to *Fig. 57.*

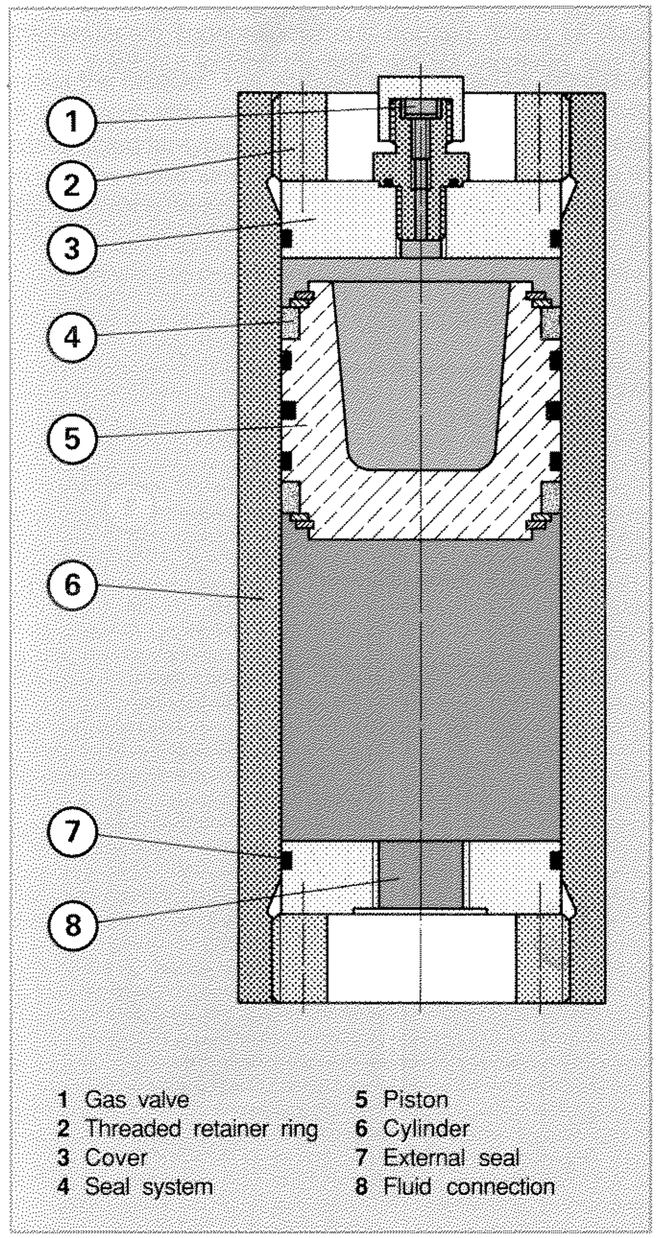


Fig. 56: *Piston-type accumulator*

Filling the gas space with nitrogen to the appropriate charging pressure forces the piston on to the end cover at the fluid end, so covering the fluid inlet connection. As the fluid pressure in the system rises and passes the minimum operating value, the piston is forced towards the gas end and compresses the gas in the cylinder. The compressed gas volume between V_1 and V_2 represents the useful volume available V. In order for there to be balanced pressures between the two pressure spaces it is essential for the friction between the piston seals and the inner wall of the cylinder to be very low as the piston moves. Therefore, the internal surface of the cylinder tube must have a very fine finish. Due to the unavoidable presence of some friction at this point, however, it is impossible for a difference in pressure between the gas and fluid spaces to be eliminated completely.

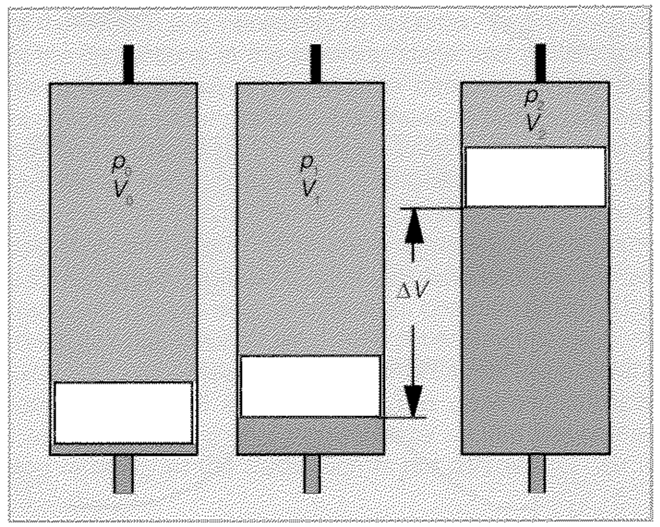


Fig. 57: *Mode of operation of a piston-type accumulator*

Diagram 18 clarifies this frictional effect. It shows graphs of the fluid and gas pressures against time for a single accumulator cycle with two different types of sealing system. It shows clearly that a low-friction sealing system causes less differential between the two pressures and so provides better operating characteristics. However, the frictional resistance is not constant; it increases with the operating pressure. At lower operating pressures the frictional resistance predominates over the piston motion so use of the accumulator at low pressure levels is not normally sensible.

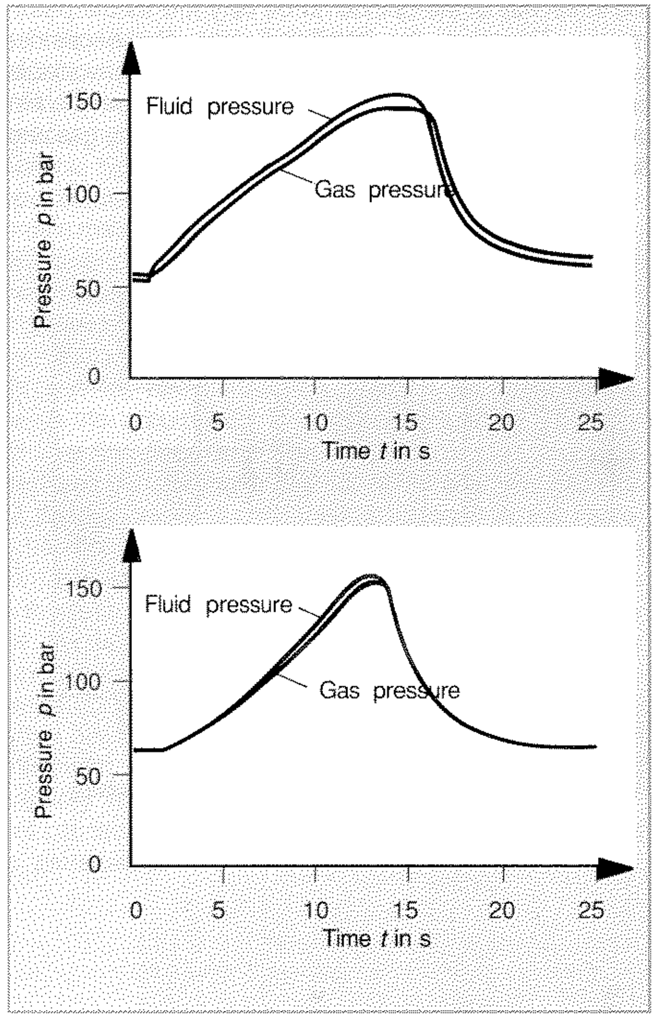
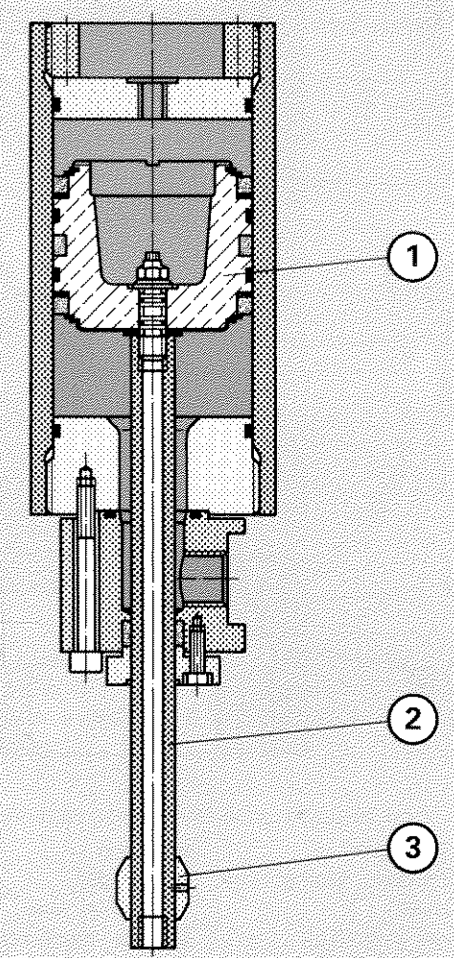


Diagram 18: *Graphs of pressure versus time for a piston-type accumulator with a normal sealing system (top) and a low-friction sealing system (bottom)*

Certain functions of the hydraulic system, such as stopping of the supply pump or monitoring of the charging of the accumulator, can be achieved directly by modifying the piston. As shown in Fig. 58 a piston rod is attached to the piston and extended through one of the end covers. This allows a variety of control functions to be achieved in a number of different ways, either:

- mechanically with switching cams or
- electrically with permanent magnets or inductive proximity switches.

Yet another method of determining the position of the piston is to use an ultrasonic measuring system which employs a microprocessor making direct use of data such as piston position and simultaneous measurement of gas pressure for the different control functions.

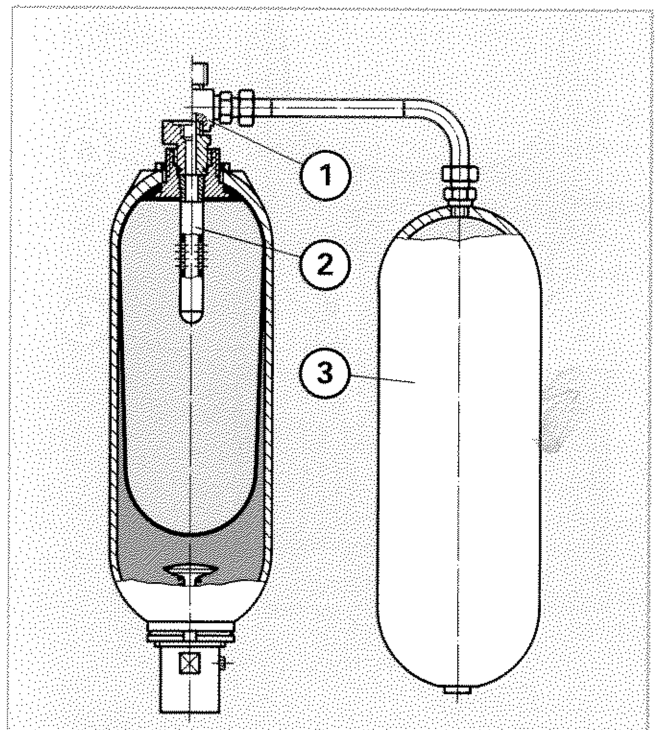


1 Piston
2 Piston rod
3 Switching cam

Fig. 58: *Piston-type accumulator with external piston rod*

2.4 Adding nitrogen bottles

For certain applications it can be an advantage to increase the gas volume by adding extra nitrogen bottles. One typical such application is when the difference between the minimum and maximum operating pressures is small. The volume of nitrogen in the accumulator is then only compressed slightly and the useful portion of the storage volume is insufficient for the purpose. Depending on the operating conditions the gas volume can be doubled by adding extra nitrogen bottles. *Fig. 59* shows such an arrangement with a bladder-type accumulator. The gas end of the accumulator has a modified connection for the nitrogen bottle and, so that the bladder does not suffer any damage when the accumulator is being charged, there is also an additional Crepin tube inside the bladder. The nitrogen volume can be increased in the same way with piston-type accumulators. In both cases, bladder-type and piston-type, the designer must match the maximum connected gas volume accurately to the operating temperature and ambient temperature.



1 Adapter
2 Crepin tube
3 Nitrogen bottle

Fig. 59: *Bladder-type accumulator with added nitrogen bottle*

2.5 Hydraulic dampers

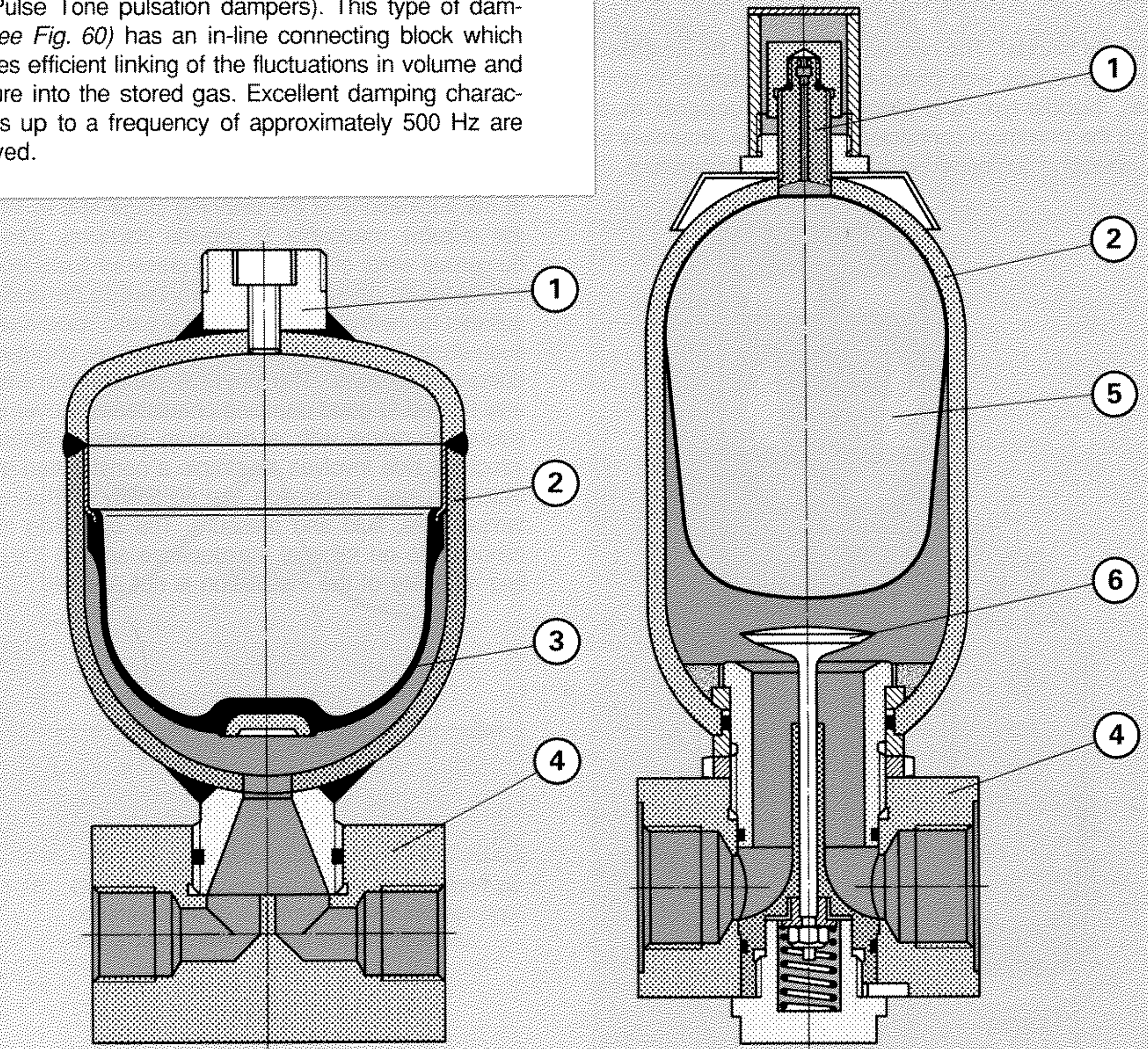
Fluctuations in pressure can occur in hydraulic systems as a result of changes in the flow of fluid due to a variety of circumstances in the system such as:

- irregularities in the displacement pump
- spring-mass systems (pressure compensators in valves)
- sudden linking of spaces at different pressures
- operation of fast opening and closing
- starting and stopping of displacement pumps.

Associated with these circumstances are fluctuations in volumetric flow and pressure which have a negative effect on the service life of all components in the system. Depending on the origin, fluctuations in pressure vary between pressure shocks and more regular pulsations. In order to ensure that they do not affect the functioning of the system it is necessary to ascertain the magnitude of the fluctuations at the planning stage and to incorporate suitable measures to provide damping. There are numerous ways of damping out pressure fluctuations but the hydraulic damper has shown itself to be particularly suitable for hydraulic systems. The basic specification for such dampers can be divided into physical, structural and operating sections. The physical parameters are primarily related to good damping characteristics over a large frequency range with a minimum pressure drop. The structural side involves as simple a construction as possible with easy installation and suitability for the relevant temperature, fluid and pressure. The operating aspects concentrate on minimal maintenance so that the operating reliability of the installation is not affected.

2.5.1 Construction and mode of operation

Depending on the mode of operation, hydraulic dampers employ the principle of hydro-pneumatic bladder-type and diaphragm-type accumulators or a fluid silencer. In the case of the hydro-pneumatic dampers the compressibility of a gas (usually nitrogen) is utilized for the damping. With a bladder-type accumulator, for example, the bladder is compressed or expanded according to the magnitude of the pressure fluctuations. The diaphragm-type accumulator behaves similarly. Since using normal bladder accumulators or diaphragm accumulators does not always produce good damping due to the imperfect link between the hydraulic fluid and the gas volume, special hydro-pneumatic dampers have been developed (e.g. Pulse Tone pulsation dampers). This type of damper *(see Fig. 60)* has an in-line connecting block which provides efficient linking of the fluctuations in volume and pressure into the stored gas. Excellent damping characteristics up to a frequency of approximately 500 Hz are achieved.



1 Gas valve
2 Pressure vessel
3 Diaphragm
4 "Pulse Tone" connector block
5 Bladder
6 Check valve

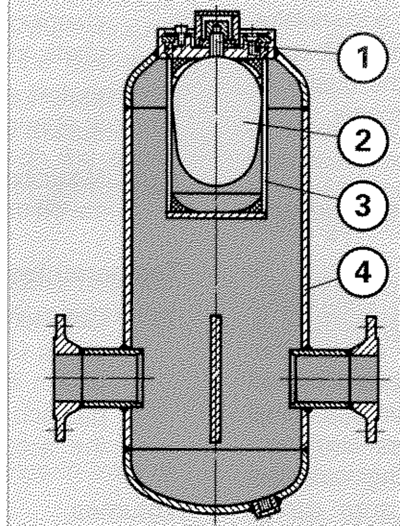
Fig. 60: *Hydro-pneumatic dampers, Pulse Tone diaphragm accumulator (left) Pulse Tone bladder accumulator (right)*

Another form of pulsation damping, specifically for reducing fluctuations in pressure on the suction side, can be effected with the suction stabilizer shown in *Fig. 61*. Basically, it comprises a small volume of gas surrounded by a volume of fluid many times larger. It performs the function of a storage vessel and considerably reduces the acceleration effects of the flow.

Special hydraulic dampers, called shock absorbers *(see Fig. 62)*, have been developed for damping out the pressure shocks that can be associated with the fast opening and closing of valves and the starting and stopping of pumps.

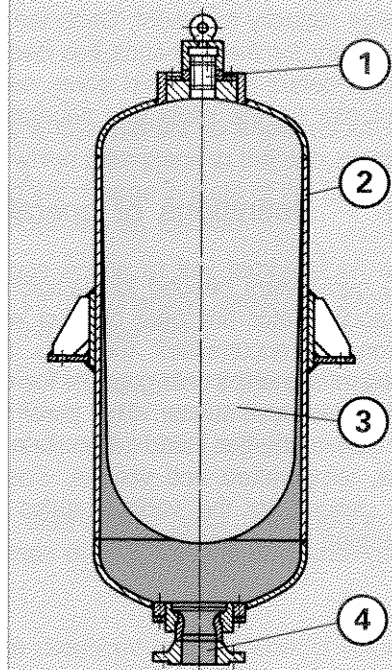
In construction, the shock absorber is like a bladder-type accumulator and stops further propagation of the pressure shock through the conversion of potential energy into kinetic energy and vice versa.

Damping can also be achieved without an additional volume of gas by means of a fluid silencer. The design of such a silencer is shown in *Fig. 63* and is totally different from the hydro-pneumatic dampers. In this case the fluctuations in volume and pressure are reduced by careful flow design employing such features as a resonator or an expansion chamber.



1 Gas valve
2 Bladder
3 Strainer basket
4 Pressure vessel

Fig. 61: *Suction stabilizer*



1 Gas valve
2 Vessel
3 Bladder
4 Fluid connection

Fig. 62: *Shock absorber*

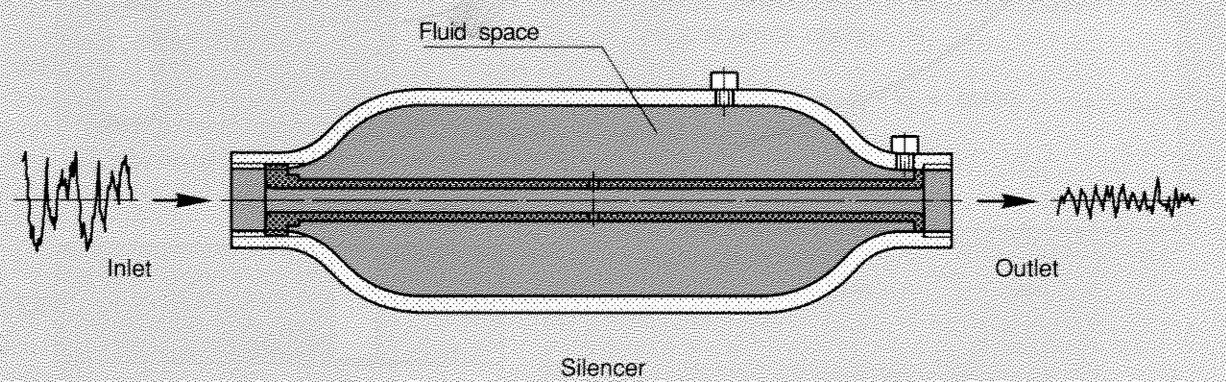


Fig. 63: *Fluid noise transmission damper(silencer)*

3 Design of hydro-pneumatic accumulators

The performance required of a hydro-pneumatic accumulator varies according to the particular application. In the design the initial factors of interest are only the requirements regarding useful volume and pressure energy. In addition there are certain secondary factors to be taken into account, mostly specific to the type of installation, e.g. mobile equipment needs a maximum ratio of energy capacity to weight. Once the size of accumulator necessary for the demands has been established, component details can be laid down, such as the quality of elastomer for the seals and separating elements.

3.1 Definitions of operating parameters

The parameters needed for designing a hydro-pneumatic accumulator are best clarified with a diagram of a piston-type accumulator *(see Fig. 64)*. Of course, the same relationships hold good for the other types of hydro-pneumatic accumulator. The parameters for describing the state of the gas - also called state variables - are pressure, temperature and volume. The following variables are defined for the various states which arise during operation of an accumulator

Pressures

p_0 Charging pressure of the gas space with the fluid space depressurized.

p_1 Minimum pressure required to open the valve. With bladder and diaphragm accumulators this pressure is normally about 10% higher than the charging pressure. The charging pressure can be made lower with piston accumulators.

p_2 Maximum operating pressure of the hydraulic system with bladder and diaphragm accumulators

p_0/p_2 Maximum permitted pressure ratio for operating conditions

Temperatures

T_i Gas temperature corresponding to the various states (i = 0, 1, 2). The temperature of the fluid affects the heat exchange with the compressed gas and so is only needed indirectly for designing the accumulator.

Volumes

V_0 Effective gas volume at charging pressure

V_1 Gas volume at minimum pressure

V_2 Gas volume at maximum operating pressure

ΔV Useful volume

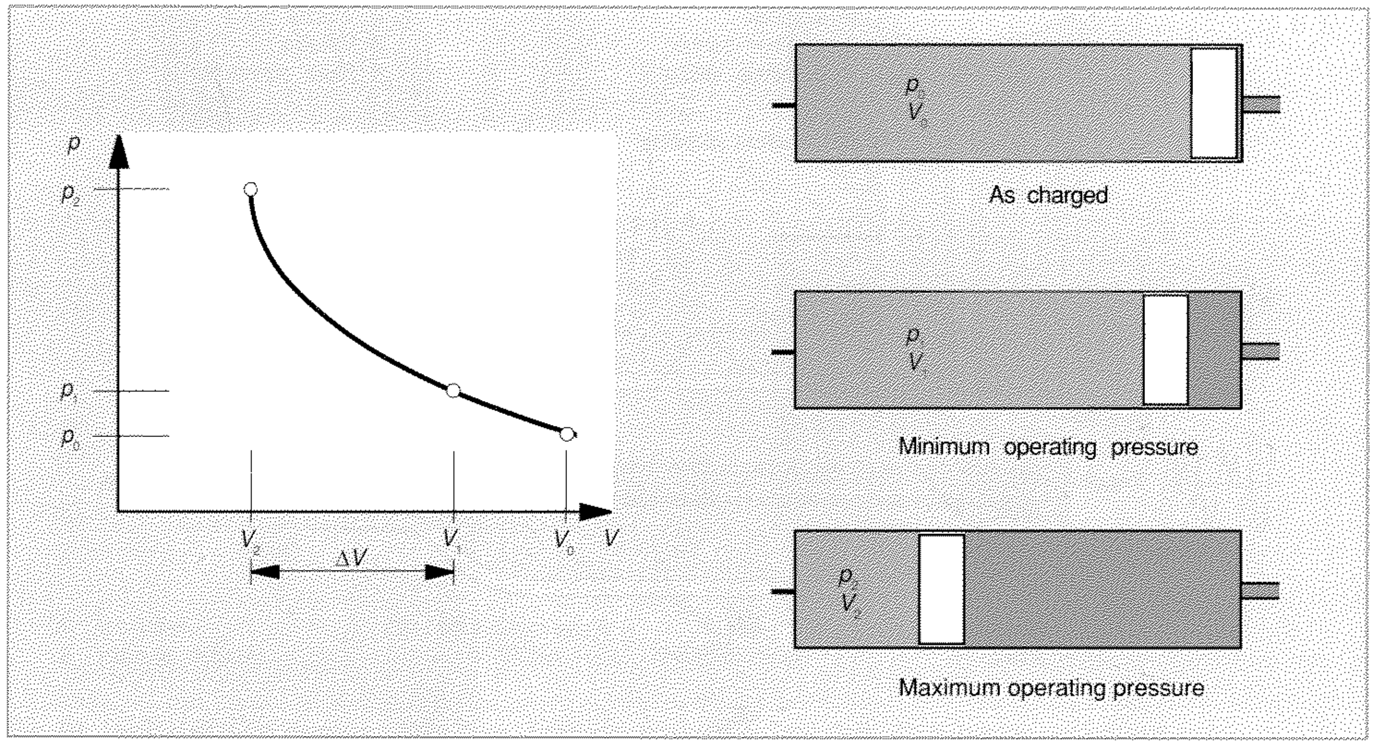


Fig. 64: *Diagram of operating state showing variables applicable to a piston-type accumulator*

3.2 The fundamental physics

In the thermodynamic sense the volume of gas in a hydraulic accumulator can be regarded as a homogeneous closed system with appropriate state variables. Although the basic physical equations for accumulator design will be explained by referring to a piston-type accumulator it does not restrict the universal nature of the equations in any way (the friction between piston and cylinder wall is neglected in this case). The flow of hydraulic fluid into or out of the accumulator has a direct relation to the change of state of the volume of gas inside. Firstly, the hydraulic fluid causes an exchange of work with the gas and, secondly, an exchange of heat between the surroundings and the gas if the gas temperature is not equal to the ambient temperature. The "surroundings" mean the separating element, the accumulator vessel and the hydraulic fluid.

In *Fig. 65*, moving the piston an infinitely small distance ds to change the volume by an amount dV requires the following amount of work

$$dW_V = -p \cdot A \cdot ds = -p \cdot dV \qquad (1)$$

The change in volume also involves a change in state of the gas.

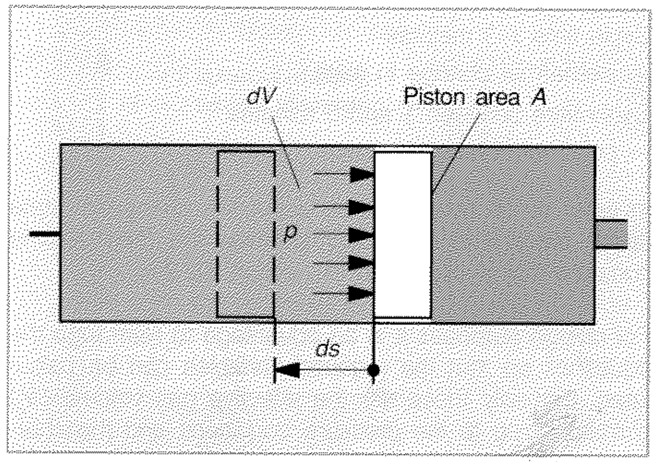


Fig. 65: *Volumetric relationship*

Note on Equation 1:
There is a recognized mathematical sign rule to establish whether the system is absorbing work (+) or expending work (-). According to this rule, the work is considered to be positive with compression (dV<0).

Assuming that it is an ideal gas, the relation between pressure, temperature and volume can be described with the equation of state

$$p \cdot V = m \cdot R \cdot T \qquad (2)$$

Where R is a constant dependent only on the type of gas. For nitrogen (N_2) it is:

$$R = 297 \frac{\mathrm{J}}{\mathrm{kg\,K}}$$

In order to continue, it is important to know about the individual processes which take place in the accumulator in order to understand the change in state of the gas. The states and their changes are as follows:

a) Charging of the gas space at low temperature with subsequent change in the charging pressure through heat exchange with the surroundings.

b) The charging or discharging cycle of the accumulator by the hydraulic fluid takes place over a time span sufficient for a complete exchange of heat with the surroundings to occur.

c) The charging or discharging cycle takes place so rapidly that no exchange of heat with the surroundings is possible.

In the change of state described in **a)** no work of volume change is expended, i.e. no change in volume takes place. This change of state is called isochore and can be described by the following simplified equation

$$\frac{p}{T} = \frac{p_1}{T_1} = \text{const.} \qquad (3)$$

The change of state described in **b)** is called isothermal and takes place without change in temperature if complete exchange of heat with the surroundings is assumed. The mathematical relation between the state variables can be derived from the thermal equation of state and, for an isothermal change, is

$$p \cdot V = p_1 \cdot V_1 = \text{const.} \qquad (4)$$

The change of state described in **c)** is called adiabatic. In this case there is only an exchange of work between the hydraulic fluid and the gas and the relevant equation is

$$p \cdot V^{\kappa} = p_1 \cdot V_1^{\kappa} = \text{const.} \qquad (5)$$

The relation between temperature, volume and pressure can also be obtained from the thermal equation of state

$$T \cdot V^{\kappa-1} = T_1 \cdot V_1^{\kappa-1} \quad \text{and} \qquad (6)$$

$$T \cdot p^{(1-\kappa)/\kappa} = T_1 \cdot p_1^{(1-\kappa)/\kappa} \qquad (7)$$

In these equations κ represents the adiabatic index which can be taken as 1.4 for a diatomic gas such as nitrogen under normal conditions *(see Diagram 19).*

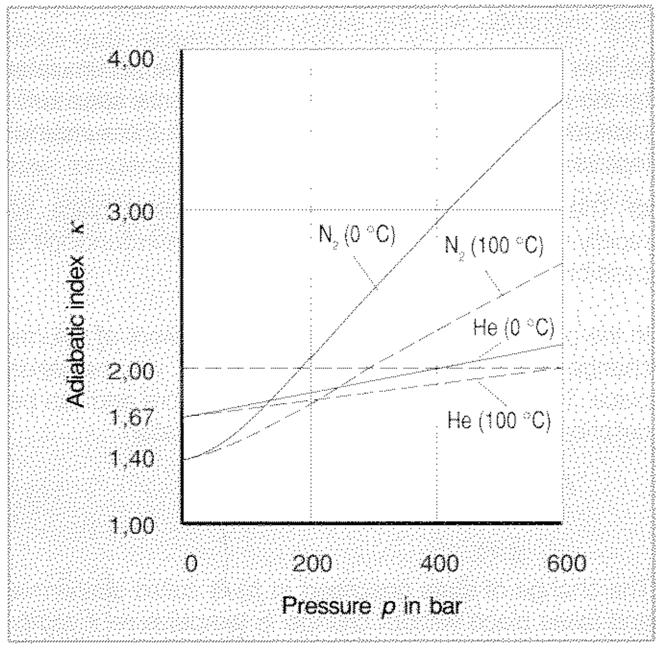


Diagram 19: *Adiabatic index of nitrogen and helium in relation to pressure at 0 and 100°C*

Since an accumulator never functions precisely according to theory with no exchange of heat, the change of state will lie somewhere between the isothermal and the adiabatic. This type of state change is called polytropic. The mathematical relationships are similar to those for the adiabatic change of state but with the adiabatic index replaced by the polytropic index *n.* The *p-V* diagram *(Diagram 20)* shows the different changes in state and it is obvious that the isothermal and adiabatic are extremes of the polytropic.

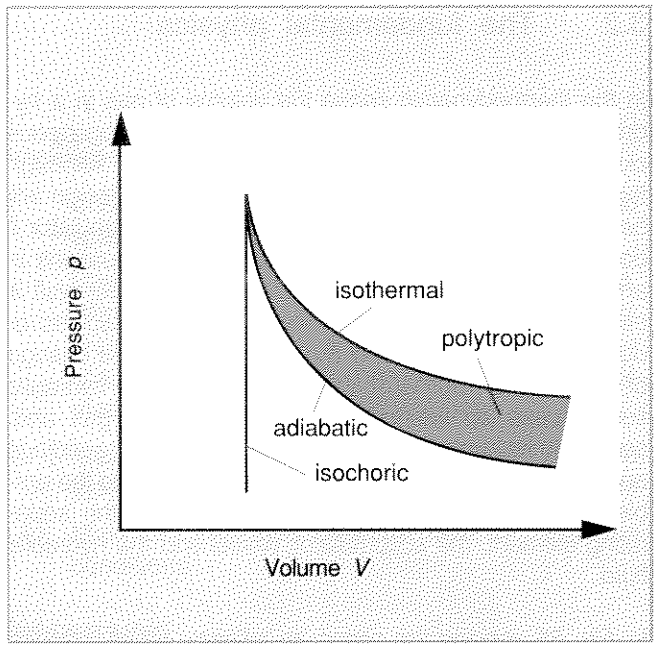


Diagram 20: *p-V diagram of change in state*

Accordingly, the equations to be used for designing a hydraulic accumulator depend on the time effect of the charging or discharging process. The following limits can serve as a useful rule-of-thumb for using the appropriate equation:

Cycle time < 1 minute
→ adiabatic change in state
Cycle time > 3 minutes
→ isothermal change in state
Cycle time between 1 and 3 minutes
→ polytropic change in state.

In order to be more precise about the change of state taking place it is necessary to know the thermal time constant that is dealt with in *Section 3.2.2.*

For design purposes it is an advantage to rearrange the equations so that the required variables can be calculated. Primarily these are the effective gas volume (V_0) corresponding to the pressure conditions and the charging pressure p_0. *Table 15* lists the basic equations for accumulator design.

Also when designing an accumulator, there are certain empirical values to be adhered to which ensure, firstly, optimum utilization of the accumulator volume and, secondly, long service life. *Table 16* lists the relevant empirical values for different types of accumulator.

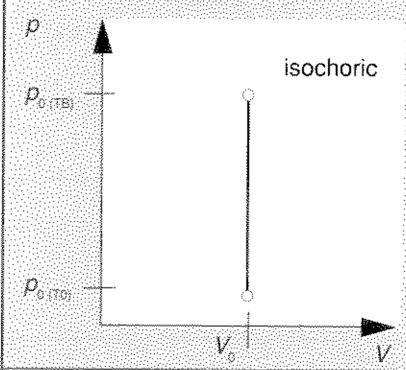
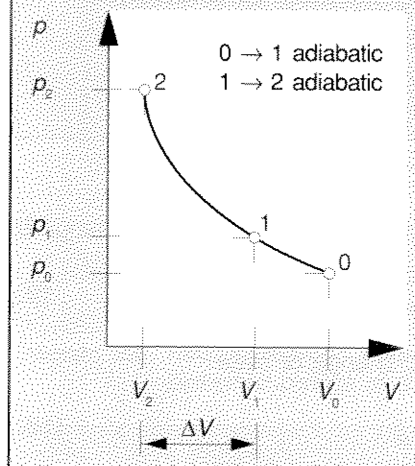
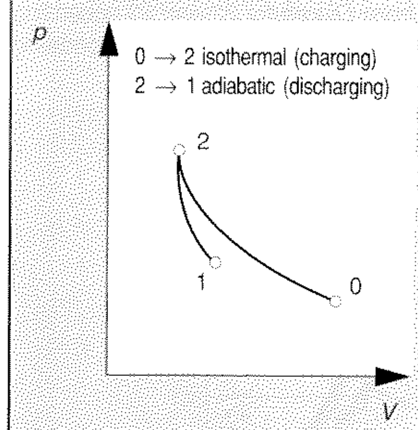
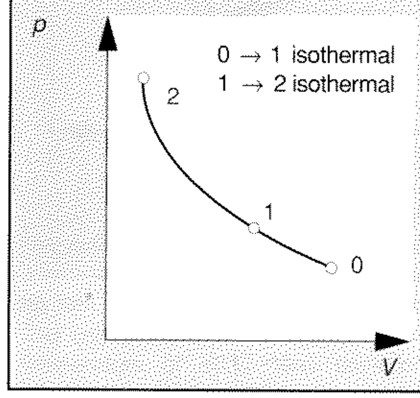
Cycle (change of state)	Equation	Remarks
	$p_{0(T0)} = p_{0(TB)} \cdot \frac{T_0}{T_B}$	$p_{0\,(T0)}$ = Charging pressure at charging temperature T_0 $p_{0\,(TB)}$ = Charging pressure at operating temperature T_B **Application:** Calculation of the charging pressure when the operating temperature is different from the charging temperature
	$\Delta V = V_0\left[\left(\frac{p_0}{p_1}\right)^{\frac{1}{n}} - \left(\frac{p_0}{p_2}\right)^{\frac{1}{n}}\right]$ $V_0 = \frac{\Delta V}{\left(\frac{p_0}{p_1}\right)^{\frac{1}{n}} - \left(\frac{p_0}{p_2}\right)^{\frac{1}{n}}}$	$n = \kappa$ = 1,4 for nitrogen **Application:** Energy storage
	$\Delta V = V_0 \frac{p_0}{p_2}\left[\left(\frac{p_2}{p_1}\right)^{n} - 1\right]$ $V_0 = \frac{\Delta V \cdot \frac{p_2}{p_0}}{\left(\frac{p_2}{p_1}\right)^{\frac{1}{n}} - 1}$	**Application:** Emergency functions, safety functions
	$\Delta V = V_0\left(\frac{p_0}{p_1} - \frac{p_0}{p_2}\right)$ $V_0 = \frac{\Delta V}{\frac{p_0}{p_1} - \frac{p_0}{p_2}}$	**Application:** Leakage fluid make-up

Table 15: *Basic equations for accumulator design*

Condition	Bladder accumulators	Diaphragm accumulators	Piston accumulators
Gas charging pressure p_0	$\leq 0{,}9 \cdot p_1$ $= 0{,}6$ to $0{,}9 \cdot p_m$ (shock absorption) $= 0{,}6 \cdot p_m$ (pulsation damping)	$\leq 0{,}9 \cdot p_1$	$\leq p_1 - 5$ bar ≥ 2 bar (low-frictionpiston) ≥ 10 bar (Normal piston)
Max. permitted pressure ratio p_2/p_0	$\leq 4:1$	$\leq 6{:}1$ to $8{:}1$ (welded version) $\leq 10{:}1$ (screwed version)	No restrictions
Max. fluid flow	to 15 L/s according to size to 140 L/s at High-Flow version	to 6 L/s	Max. piston velocity = 3,5 m/s (low-friction version) = 2 m/s (normal version)

Table 16: *Conditions of application for hydro-pneumatic accumulators*

In the case of the arrangement with added nitrogen bottles the designer must also examine the useful volume of the accumulator. He should begin with an isothermal change from charging pressure to maximum operating pressure. The increased useful volume $\Delta V'$ can be calculated from:

$$\Delta V' = V_{0G}\left(1 - \frac{p_0}{p_2}\right). \qquad (8)$$

For bladder accumulators with added nitrogen bottles $\Delta V'' = 0.75 \cdot V_{OG}$ should not be exceeded because of excessive distortion of the bladder. V_{OG} is the total effective gas volume (accumulator plus nitrogen bottles). The increased useful fluid volume $\Delta V'$ must always be less than the effective gas volume of the accumulator. The value of gas volume must be chosen so that these conditions are fulfilled.

3.2.1 Deviations from the ideal gas

The equations of state described in the previous section are only applicable to an ideal gas. Practical gases such as nitrogen *(see Diagram 21)*, however, do not follow the ideal gas laws, particularly at higher pressures. This behaviour is called "real" or "imperfect". The mathematical relation between the state variables (p, T and V) for a real gas can only be given by an approximate equation. The use of such an equation with sufficient accuracy is very tiresome in practice and requires a large amount of computing time which can only be provided by a mainframe computer. For this reason it is advisable to introduce correction factors that allow for the behaviour of the real gas.

Consequently, the volume with an isothermal change of state becomes

$$V_{real} = C_i \cdot V_{ideal} \qquad (9)$$

and with an adiabatic change of state

$$V_{real} = C_a \cdot V_{ideal}\,. \qquad (10)$$

The correction factors C_i and C_a in *Equations (9) and (10)* can be taken directly from *Diagrams 22 and 23* according to the pressure ratio p_2/p_1 and the maximum operating pressure.

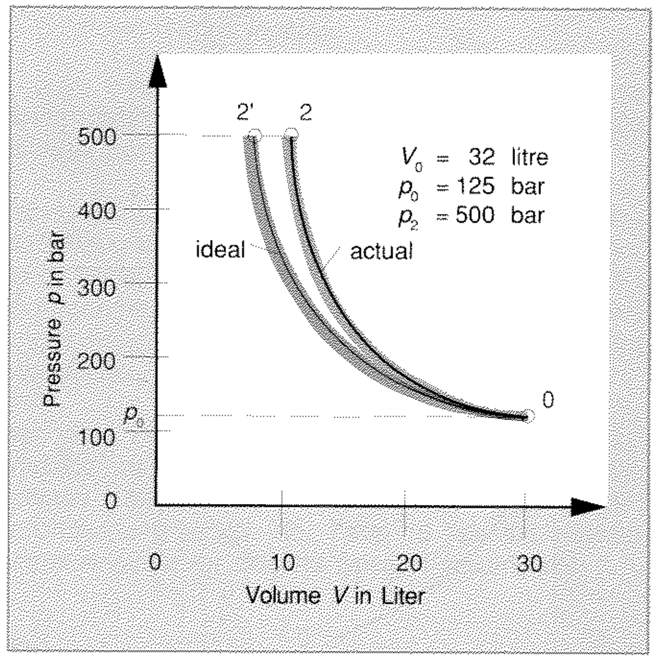


Diagram 21: *p-V diagram comparing the ideal and real gas behaviour of nitrogen in compression*

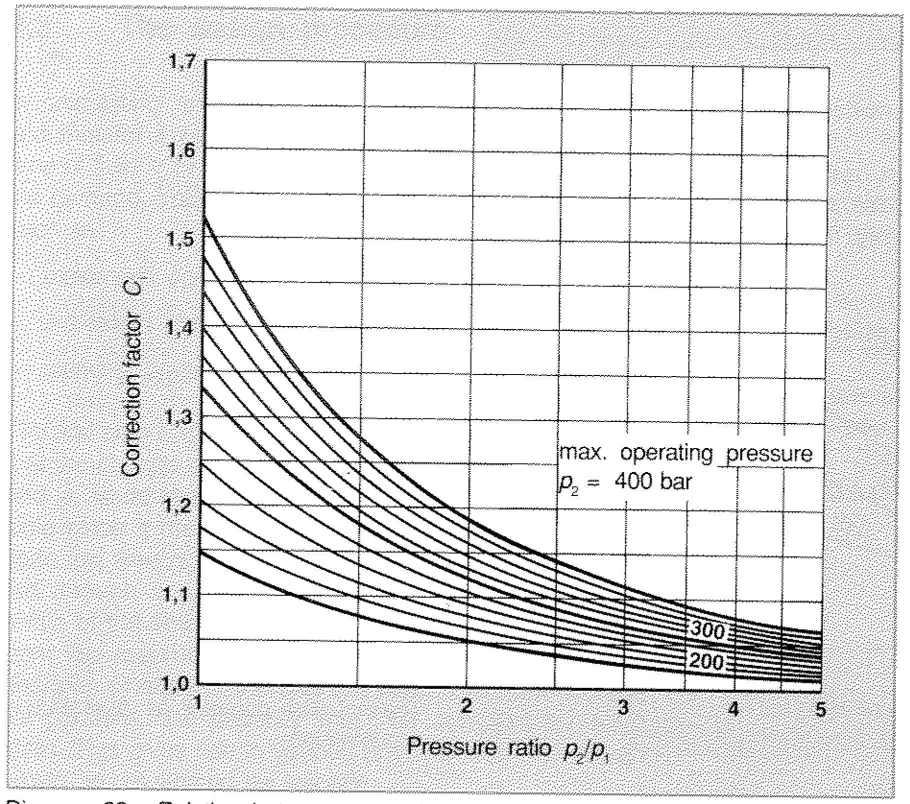


Diagram 22: *Relation between correction factor C_i and pressure ratio p_2/p_1 for an isothermal change of state*

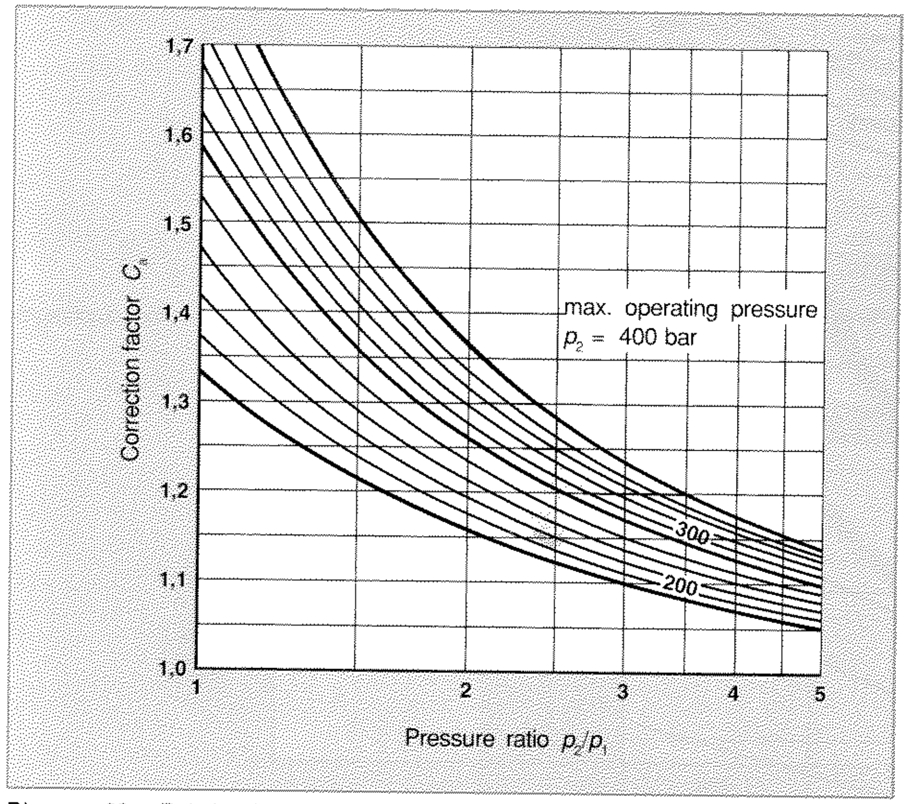


Diagram 23: *Relation between correction factor C_a and pressure ratio p_2/p_1 for an adiabatic change of state*

The correction factors are referred to a temperature of 50°C. Any deviations resulting from changes in temperature can be neglected in the permitted temperature range (-10 to +80°C).

3.2.2 Thermal time relationships

Approximate time limits were quoted in *Section 3.2* in order to establish the type of change of state. In order to arrive at a more accurate design for an accumulator it will be necessary to analyze the processes of thermodynamic exchange. In the case of intermittent operation, in particular when there is a rapid sequence of changes, the processes are determined from the intensity of the heat transfer. Describing and analyzing the thermal time response of hydraulic accumulators makes use of the thermal time constant

$$\tau = \frac{c_V \cdot m}{\alpha \cdot A} \qquad (11)$$

Where c_v is the specific thermal capacity at constant volume, m the mass of the gas, α the heat transfer coefficient and A the total area of heat transfer.

The time constant can be determined very easily by experiment. Since it depends on the charging pressure and the type and size of accumulator, it must be determined by test for each different type of accumulator. *Diagrams 24, 25 and 26* show the results of tests according to |1| for the different types. The thermal time constants are plotted in relation to the charging pressure for various nominal volumes of the different sizes of accumulator. Using these time constants and a suitable simulation program an accumulator can be designed for a given cycle of operation.

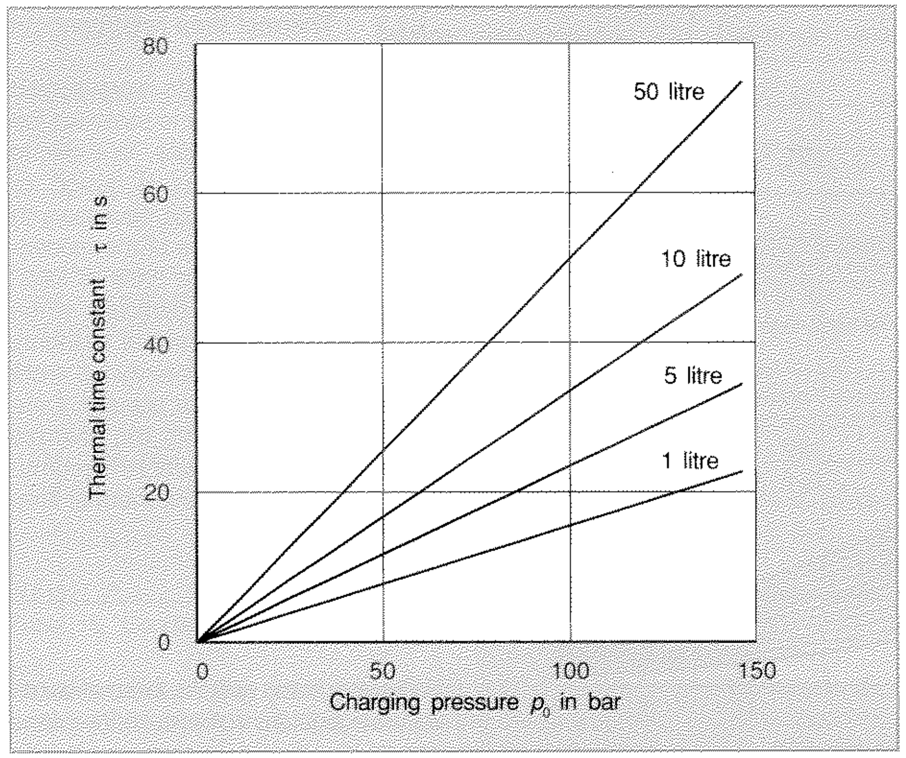


Diagram 24:
Thermal time constant for bladder-type accumulators

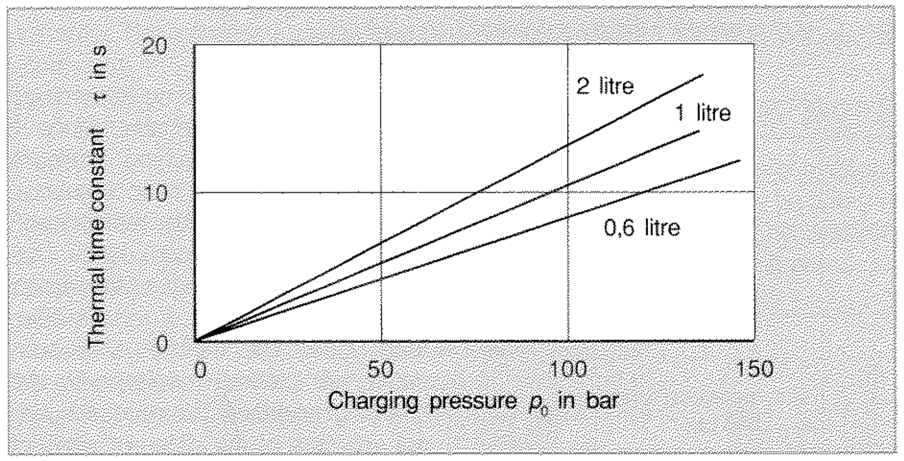


Diagram 25:
Thermal time constant for diaphragm-type accumulators

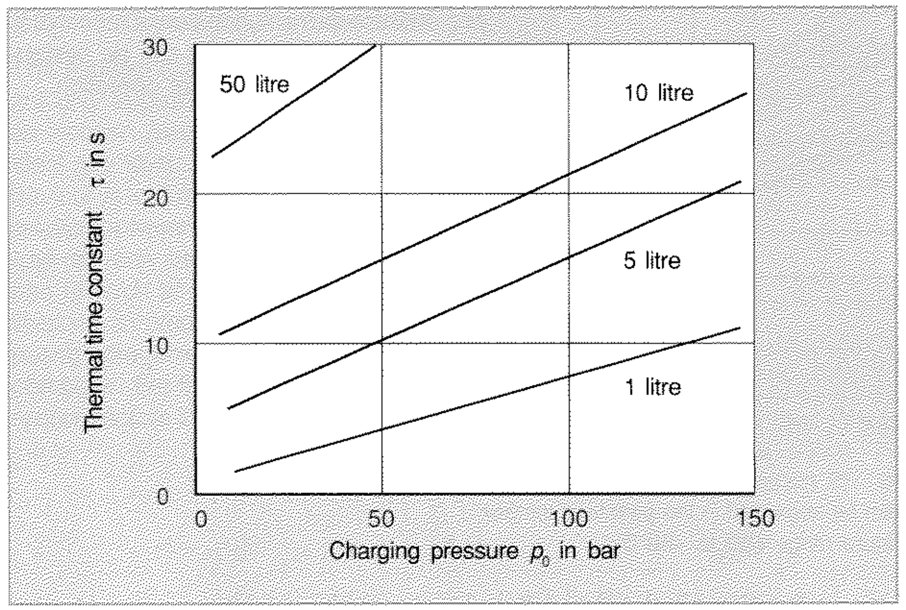


Diagram 26:
Thermal time constant for piston-type accumulators

3.3 The design procedure

In calculating and establishing the appropriate size of accumulator it can be assumed that the necessary volume of fluid *V* or the necessary energy *W* is available for the demand. Taking account of certain peripheral factors, such as

- maximum operating pressure
- maximum and minimum operating temperature
- working pressure differential

The first step in the design procedure is to assume initially that the change of state between the working pressure p_1 and p_2 is adiabatic. This limiting assumption is permissible since it means that the other possible changes of state are always fulfilled. The design can then be corrected through a subsequent check of the calculations in terms of the time response and the associated deviation from the assumed adiabatic change of state. Since the changes of state of the gas are related to the operating temperature, the capacity must be sufficient for each state. This gives rise to various restrictions on the design which are examined as follows:

– Restriction 1a and b

The required volume of fluid V_{req} or the required energy W_{req} must still be available from the accumulator at a maximum attainable operating temperature.

– Restriction 2

At the minimum operating temperature the permitted value of working pressure difference $p_{permiss}$ must not be exceeded. The various restrictions can be explained by referring to a diagram of a polytropic change of state *(see Diagram 27)*.

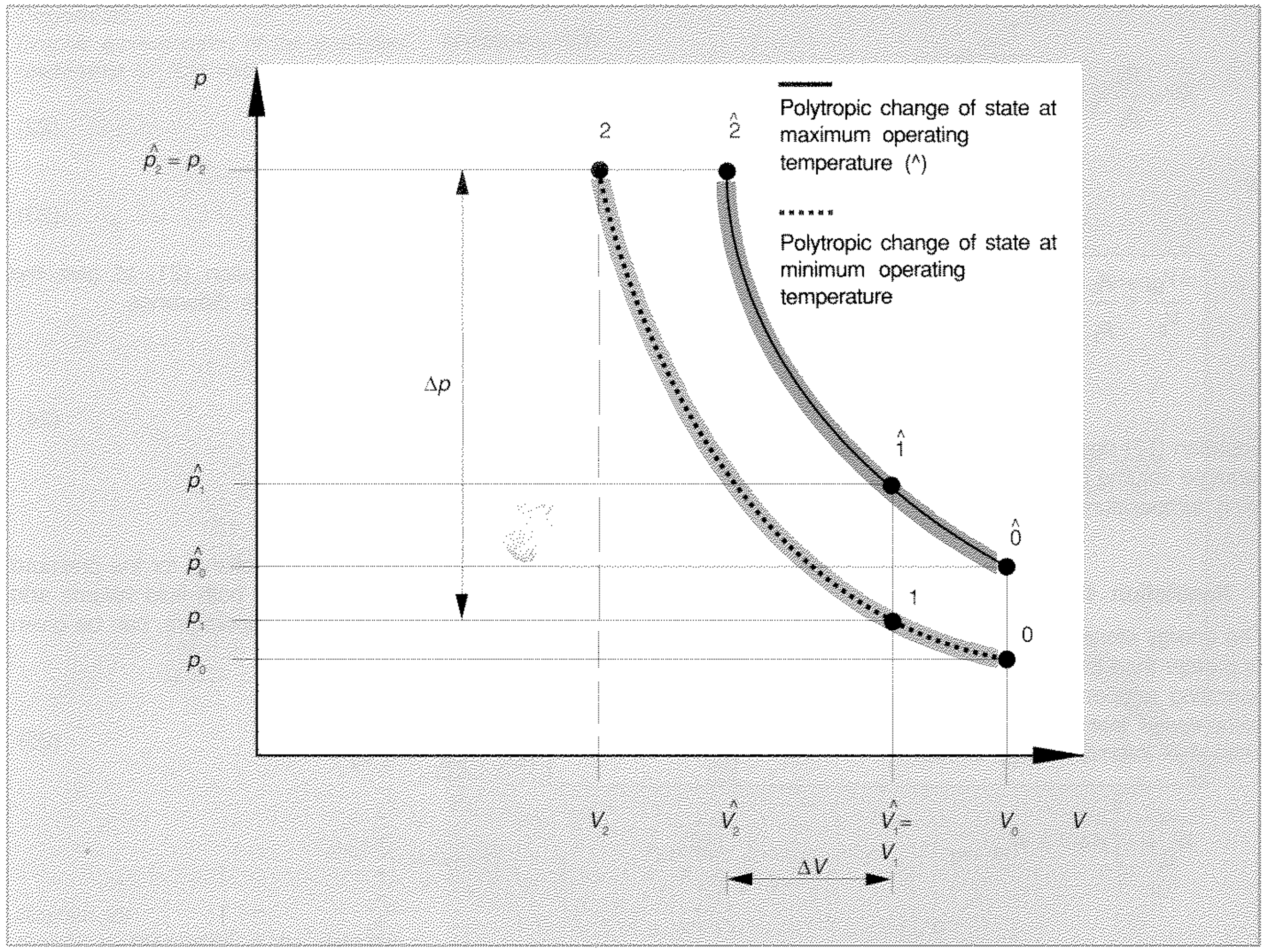


Diagram 27: *p-V diagram of a polytropic change of state at maximum and minimum operating temperature*

– **Restriction 1a**

The volume of fluid is calculated from:

$$\Delta V = \hat{V}_1 - \hat{V}_2 \geq \Delta V_{erf}\,. \qquad (12)$$

The changes of state are:

Filling:

isochoric $$\frac{p_0}{T_0} = \frac{\hat{p}_0}{T_{max}}\,, \qquad (13)$$

Charging to minimum operating pressure:

isothermal $$\hat{p}_1 \cdot \hat{V}_1 = \hat{p}_0 \cdot V_0\,, \qquad (14)$$

Compression to maximum operating pressure

adiabatic $$\hat{p}_2 \cdot \hat{V}_2^{\kappa} = \hat{p}_1 \cdot \hat{V}_1^{\kappa}\,. \qquad (15)$$

Substituting the three Equations (13), (14) and (15) in Equation (12) gives

$$\Delta V = \frac{\hat{p}_0}{\hat{p}_1} V_0 \left(1 - \left(\frac{\frac{T_{max}}{T_0} p_0}{\frac{\hat{p}_0}{\hat{p}_1}\hat{p}_2}\right)^{\frac{1}{\kappa}}\right) \geq \Delta V_{req} \qquad (16)$$

or

$$\frac{\Delta V_{erf}}{V_0} \leq \frac{\hat{p}_0}{\hat{p}_1} \left(1 - \left(\frac{\frac{T_{max}}{T_0}}{\frac{\hat{p}_0}{\hat{p}_1}} \cdot \frac{p_0}{\hat{p}_2}\right)^{\frac{1}{\kappa}}\right). \qquad (17)$$

Equation (17) describes the restricted relationship for the required volume of fluid at the maximum attainable operating temperature T_{max}.

– **Restriction 1b**

The energy stored by the accumulator must be equal to or greater than the required energy W_{req} at the maximum operating temperature. Work is done when the gas is compressed from point $\hat{1}$ to point $\hat{2}$. The associated change in internal energy is then

$$W_{\hat{1}\hat{2}} = -\int_{\hat{1}}^{\hat{2}} p\,dV \geq W_{12\ req} \qquad (18)$$

With the equation for an adiabatic change of state, Equation (18) becomes

$$W_{\hat{1}\hat{2}} = \frac{\hat{p}_1 \cdot \hat{V}_1}{\kappa - 1}\left(\left(\frac{\hat{p}_1}{\hat{p}_2}\right)^{\frac{1-\kappa}{\kappa}} - 1\right). \qquad (19)$$

The restricted relationship for the energy at maximum operating temperature is obtained by substituting Equations (13) and (14) in Equations (18) and (19)

$$\frac{W_{\hat{1}\hat{2}erf}}{\hat{p}_2 \cdot V_0} \leq \frac{p_0 \cdot T_{max}}{\hat{p}_2 \cdot T_0 (\kappa - 1)}\left(\left(\frac{\frac{T_{max}}{T_0}}{\frac{\hat{p}_0}{\hat{p}_1}} \cdot \frac{p_0}{\hat{p}_2}\right)^{\frac{1-\kappa}{\kappa}} - 1\right). \qquad (20)$$

– **Restriction 2**

The difference in pressure at minimum operating temperature between operating states 1 and 2 is:

$$\Delta p = p_2 - p_1 \leq \Delta p_{permit}\,. \qquad (21)$$

With equations

$$\frac{p_0}{T_0} = \frac{p_0}{T_{min}} \qquad (22)$$

and $$p_1 \cdot V_1 = p_0 \cdot V_0 \qquad (23)$$

it is possible to rearrange Equation (21) to give the following relationship for pressure difference

$$\Delta p = p_2 - \frac{\frac{T_{min}}{T_0}}{\frac{p_0}{p_1}} \cdot p_0 \qquad (24)$$

Thus, the second restriction relationship for the pressure ratio p_0/p_2, which is applicable to both fluid volume and energy at minimum operating temperature, can be given as:

$$\frac{p_0}{p_2} \geq \frac{\frac{p_1}{p_0}}{\frac{T_{min}}{T_0}}\left(1 - \frac{\Delta p_{zul}}{p_2}\right) \qquad (25)$$

Graphical representations of the restriction relationships of Equations (17), (20) and (25) are given in *Diagrams 28 and 29*. They show clearly the range of validity within which a design is permitted under the given conditions. The point of intersection of the restriction curves characterizes the optimum design. However, this is not always attainable in practice because of the steps in vessel sizes, and therefore gas volumes. Nevertheless, economic efficiency demands that every attempt be made to get the design as close as possible to this point.

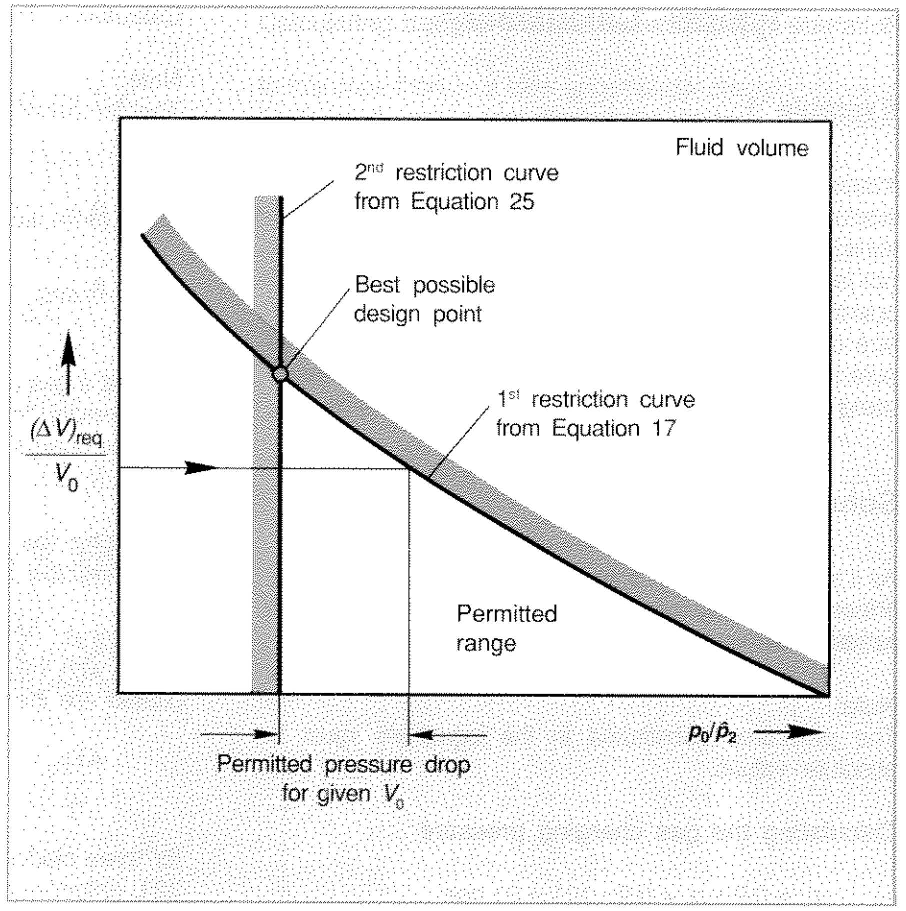


Diagram 28:
Graphic representation of the restriction relationship for fluid volume

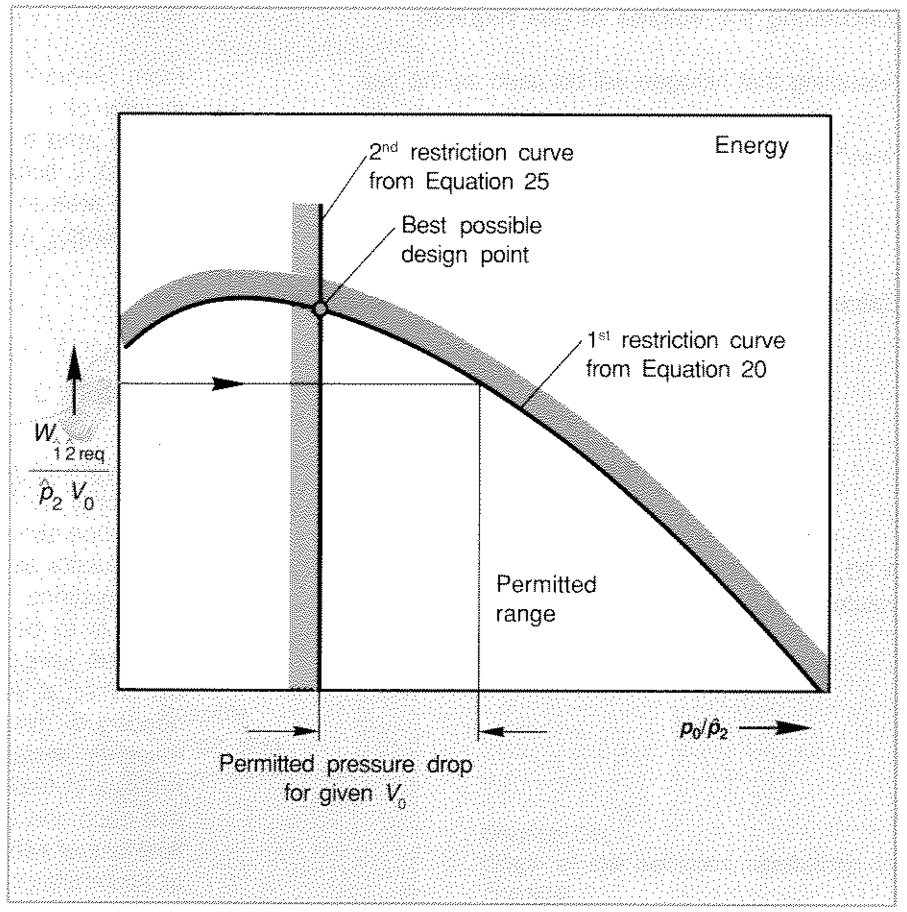


Diagram 29:
Graphic representation of the restriction relationship for energy

4 Typical Calculations

Example 1

In an injection moulding machine 5 litres of fluid are to be made available in 2.5 s. The maximum operating pressure is 200 bar and the minimum working pressure must not fall below 100 bar. The charging time is 8 s and the operating temperature is given as 45 °C. Calculate the size of accumulator required and the gas charging pressure at 22 °C using a real gas. Finally, check the results using the restriction relationships.

Solution

As it is a high-speed process (the outflow time is less than 1 minute) the change in state of the gas can be regarded as adiabatic.

Note:
The pressures substituted in the equations must be absolute values.

Calculating the gas charging pressure

$$p_0 = 0{,}9 \cdot p_1 = 0{,}9 \cdot 101 \approx 91 \text{ bar}$$

Calculating the volume of gas required

Assuming that nitrogen will be used as the gas:

$$V_{0\,ideal} = \frac{\Delta V}{\left(\frac{p_0}{p_1}\right)^{0{,}714} - \left(\frac{p_0}{p_2}\right)^{0{,}714}} = \frac{5}{0{,}9^{0{,}714} - \left(\frac{91}{201}\right)^{0{,}714}} = 13{,}87 \text{ L}$$

Calculating the correction factor from Diagram 23

$$C_a = 1.16 \text{ where } \frac{p_2}{p_1} \approx 2{,}0$$

$$V_{0\,real} = C_a \cdot V_{0\,ideal}$$
$$= 1.16 \cdot 13.87 = 16.09 \text{ L}$$

A 20 Litre bladder-type accumulator with an effective gas volume of 17.4 L is selected.

Calculating the gas charging pressure at 20°C

Note:
The temperatures used in the equations must be in Kelvin.

$$p_{0(T0)} = p_{0(TB)} \cdot \frac{T_0}{T_B}$$

$$p_{0(20°C)} = 91 \cdot \frac{20+273}{45+273} = 83{,}8 \text{ bar}$$

So that a gas charging pressure of 91 bar is available at an operating temperature of 45°C, the accumulator may only be charged to 83.8 bar at 20°C .

Checking the results with the restriction relationships for fluid volume

In *Diagram 30* the restriction curve to Equation (17) is drawn with the governing variables for the example. Also plotted are the volume ratios

$$\frac{\Delta V_{req}}{V_0}$$

for three bladder accumulators of sizes 10 L , 20 L and 32 L (effective gas volumes of 9 L , 17.4 L and 32.5 L).

The 2nd restriction curve to Equation (25) provides a value of 0.452 for the given variables.

From *Diagram 30* it can be seen that the smallest accumulator of 10 L nominal volume does not intersect the design area and so does not satisfy the requirements. The 20 L accumulator is at the optimum point of the design area so it is the correct choice. Although the 32 L accumulator also satisfies the requirements it is oversized.

Example 2

In a hydraulic system ,various cylinders are operated by directional control valves. The installation is for emergency operation and is to be driven by stored energy. The accumulator also has to make up the leakage losses at the directional control valves. For this, a low-capacity pump is to be started every 5 minutes. The pressure switches keep the pressure between 180 bar and 200 bar. When emergency operation is initiated, 8 L of fluid is needed to maintain certain functions and the pressure must not fall below 110 bar. The system incorporates 5 directional control valves with a leakage fluid loss of 30 cm³/min each and 2 directional control valves with a leakage fluid loss of 140 cm³/min each. Calculate the required size of accumulator and the required charging pressure.

Solution

Calculating the total volumetric flow of leakage fluid

$$\dot{Q}_L = 5 \cdot 30 \text{ cm}^3\text{/min} + 2 \cdot 140 \text{cm}^3\text{/min} = 430 \text{ cm}^3\text{/min}.$$

The useful volume of the accumulator required to make up the leakage fluid is therefore

$$\Delta V = \dot{Q}_L \cdot t = 430 \text{ cm}^3\text{/min} \cdot 5 \text{ min} = 2{,}15 \text{ L}.$$

The charging pressure can be calculated from

$$p_0 = 0{,}9 \cdot p_1 = 0{,}9 \cdot 111 \text{ bar} = 100 \text{ bar}.$$

Calculating the gas volume in order to make-up leakage fluid

As it is a slow process (the delivery time is more than 3 minutes) it can be assumed to be an isothermal change of state

$$V_0 = \frac{\Delta V}{\frac{p_0}{p_1} - \frac{p_0}{p_2}} = \frac{2{,}15}{\frac{100}{181} - \frac{100}{201}} = 39{,}1 \text{ L}$$

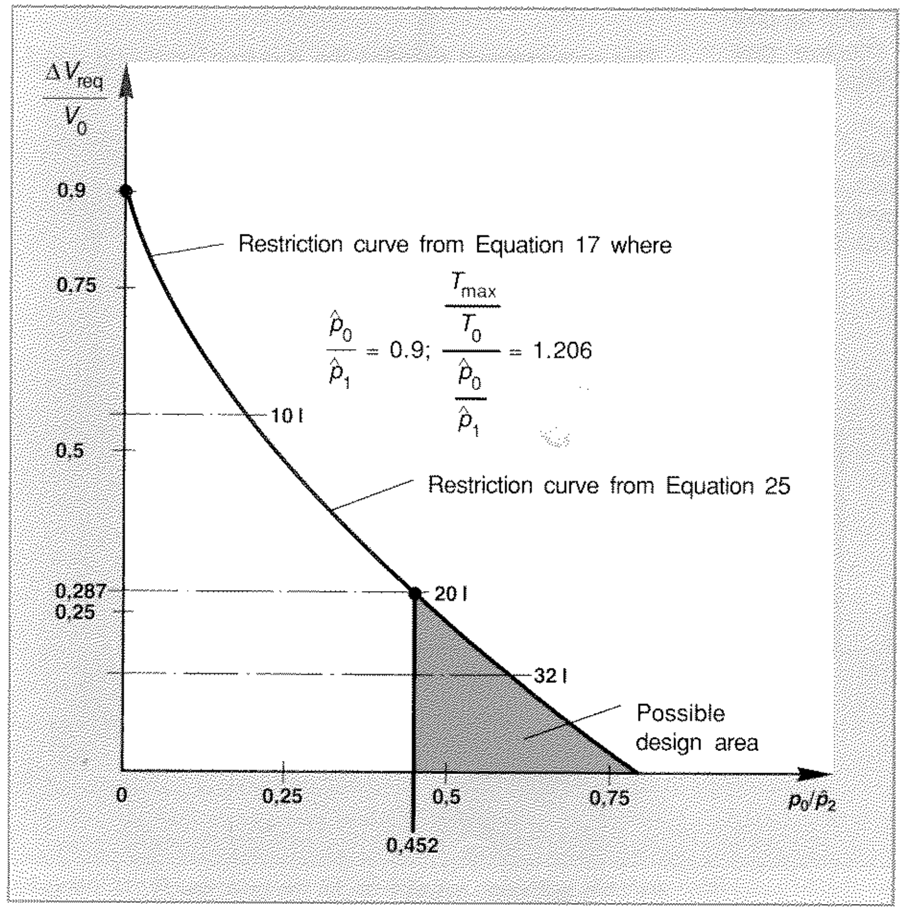


Diagram 30:
Graph of the restriction relationship for Example 1

Calculating the gas volume for emergency operation

In this case there is a slow accumulator charging process (isothermal) and a fast discharging process (adiabatic).

Since there is the possibility during emergency operation of the accumulator being charged to a pressure of only 181 bar, this must also be taken as the maximum pressure for the design.

$$V_0 = \frac{\Delta V \cdot \frac{p_2}{p_0}}{\left(\frac{p_2}{p_1}\right)^{\frac{1}{n}} - 1} = \frac{8 \cdot \frac{181}{100}}{\left(\frac{181}{111}\right)^{\frac{1}{1{,}4}} - 1} = 34{,}6 \text{ L}$$

In selecting the accumulator the larger gas volume is the governing factor and a 50 L accumulator with an effective gas volume of 47.5 L is chosen.

Example 3

Between a maximum operating pressure of 180 bar and a minimum working pressure of 120 bar, a piston-type accumulator is to supply 35 L of fluid in 2 seconds. The accumulator must be refilled in 4 minutes.

Solution

Calculating the gas charging pressure

$$p_0 = p_1 - 5\text{ bar} = 121\text{ bar} - 5\text{ bar} = 116\text{ bar}$$

Calculating the gas volume required

The calculation is performed for an adiabatic change of state (the delivery time is less than 1 minute).

$$V_0 = \frac{\Delta V}{\left(\frac{p_0}{p_1}\right)^{0{,}714} - \left(\frac{p_0}{p_2}\right)^{0{,}714}} = \frac{35}{\left(\frac{116}{121}\right)^{0{,}714} - \left(\frac{116}{181}\right)^{0{,}714}} = 144{,}3\text{ L}$$

This volume of gas can be provided by a piston-type accumulator with an effective gas volume of 150 L . For economic reasons, however, it is advisable to employ an accumulator with a smaller gas volume and to add separate nitrogen bottles in an arrangement such as one 50 L piston-type accumulator (with an effective gas volume of 52.5 l) and two 50 L nitrogen bottles.

Checking the useful storage volume

According to Equation (8) it is necessary for the enlarged useful volume V' to be less than the effective gas volume V_0 of the accumulator. The total effective volume for the arrangement chosen is

$$V_{0G} = 1 \cdot 52{,}5\text{ L} + 2 \cdot 50\text{ L} = 152{,}5\text{ L}.$$

Therefore:

$$\Delta V' = V_{0G}\left(1 - \frac{p_0}{p_2}\right) = 152{,}5\left(1 - \frac{116}{181}\right) = 54{,}8\text{ L}.$$

Thus $\Delta V' > V_0$.

This shows that the volume of the accumulator selected is too small. When another calculation is performed for an arrangement with one 60 L accumulator (with an effective gas volume of 62.5 l) and two 50 L nitrogen bottles the total effective volume becomes

$$V_{0G} = 1 \cdot 62{,}5\text{ L} + 2 \cdot 50\text{ L} = 162{,}5\text{ L}$$

and the increased useful volume

$$\Delta V' = 162{,}5\left(1 - \frac{116}{181}\right) = 58{,}4\text{ L}.$$

This arrangement satisfies the required operating conditions.

5 Typical applications

5.1 Energy storage

From the pattern of power demand for a plastics injection moulding machine, shown in *Diagram 31*, it can be seen that maximum power is only needed for a short time during the high injection velocity into the mould. For economic reasons it would not be sensible to cover this peak demand entirely with pump power. However, it is sensible to rate the pump for a medium power demand and to cover the shortfall with an accumulator. The arrangement of the accumulator for this application is shown in *Fig. 66*.

Benefits

Smaller hydraulic pump, lower power, less heat, simpler maintenance and installation, lower operating costs. In addition there is the extra shock and pulsation damping which allows a longer service life to be anticipated.

Typical applications

Bladder-type and piston-type accumulators for energy storage in injection-moulding and blow-moulding machines, transfer lines, steelworks machinery, rolling mills, construction machinery, machine tools, hydraulic presses and shears, transport systems, marine engineering and power stations, trip-out systems on steam turbines and nuclear power plants.
Diaphragm-type accumulators are used for energy storage in pilot control circuits, braking systems, machine tools and jig and tool making.

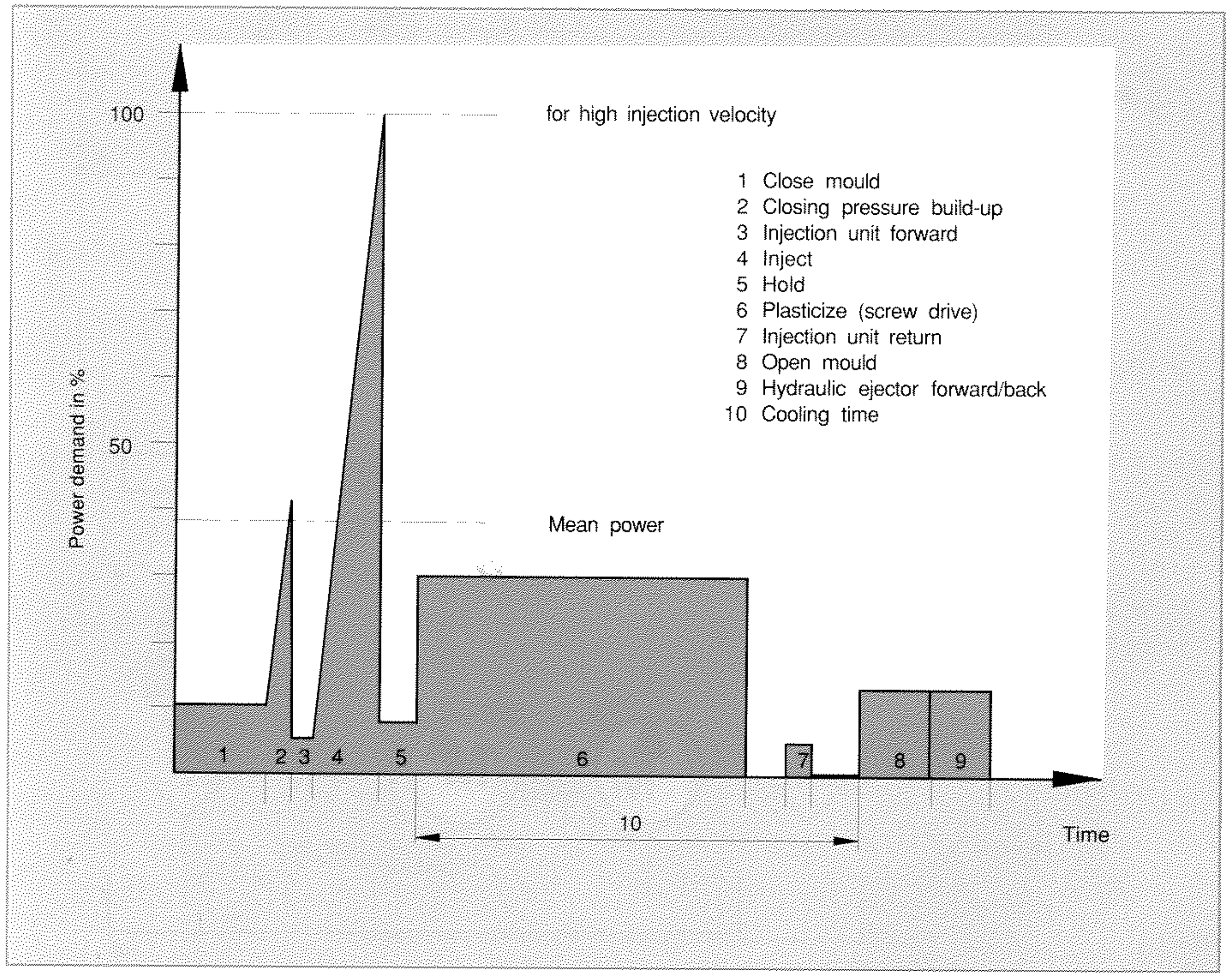


Diagram 31: *Power diagram for an injection moulding machine*

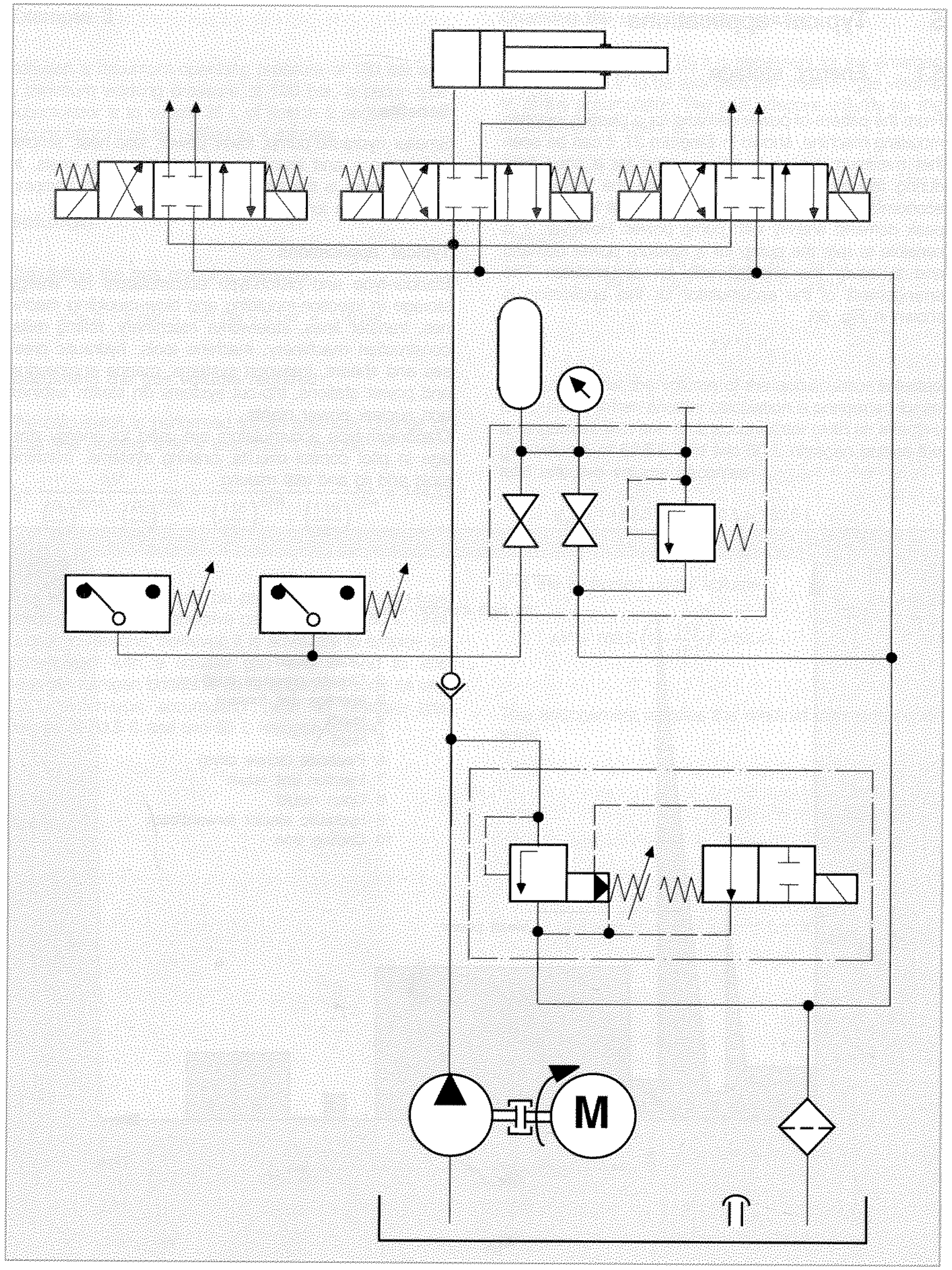


Fig. 66 *Circuit diagram for energy storage on an injection moulding machine*

5.2 Emergency operation

In an emergency such as a power failure, it is possible to use stored energy to operate a device such as a cylinder for a working stroke or closing stroke. *Fig. 67* shows a circuit diagram for emergency operation with a solenoid valve which, when triggered by the emergency, opens and allows the fluid stored under pressure in the accumulator to flow to the piston rod end of the cylinder causing it to retract.

Benefits

The energy stored is available immediately, it can be stored indefinitely, there is no fatigue or inertia, safety is at a maximum and maintenance at a minimum.

Typical applications

Bladder-type and diaphragm-type accumulators for closing bulkhead doors, dampers, sluices, bunker valves, silos and transport devices in the event of a power failure.

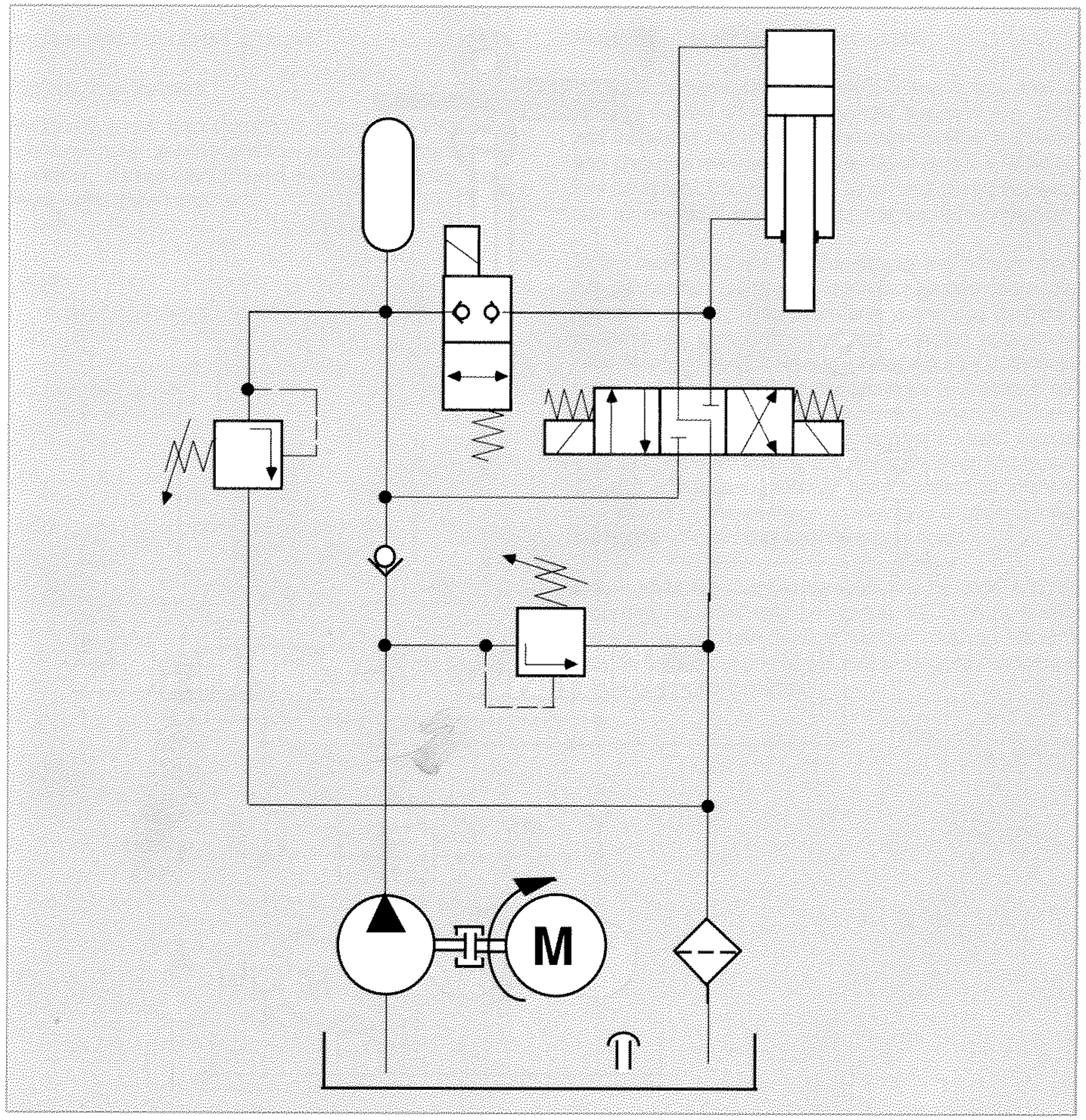


Fig. 67: *Circuit diagram for emergency operation*

5.3 Force balancing

Hydraulic accumulators can be used for force or stroke balancing. This can be necessary, for example, in a continuous production process such as steel rolling when uneven loads from the material being rolled could cause the rolls to adopt an off-centre position. Balancing the rolls produces a uniform wall thickness. *Fig. 68* shows a circuit diagram for a roll balancing system incorporating an accumulator and a direct-mounted safety and shut-off block.

Benefits

Ensures uniform forming, smooth force balancing and therefore less load on foundations and roll stand, elimination of counterweights and therefore a reduction in total weight.

Typical applications

Bladder-type, diaphragm-type and piston-type accumulators for machine tools, roll stands and crane jibs.

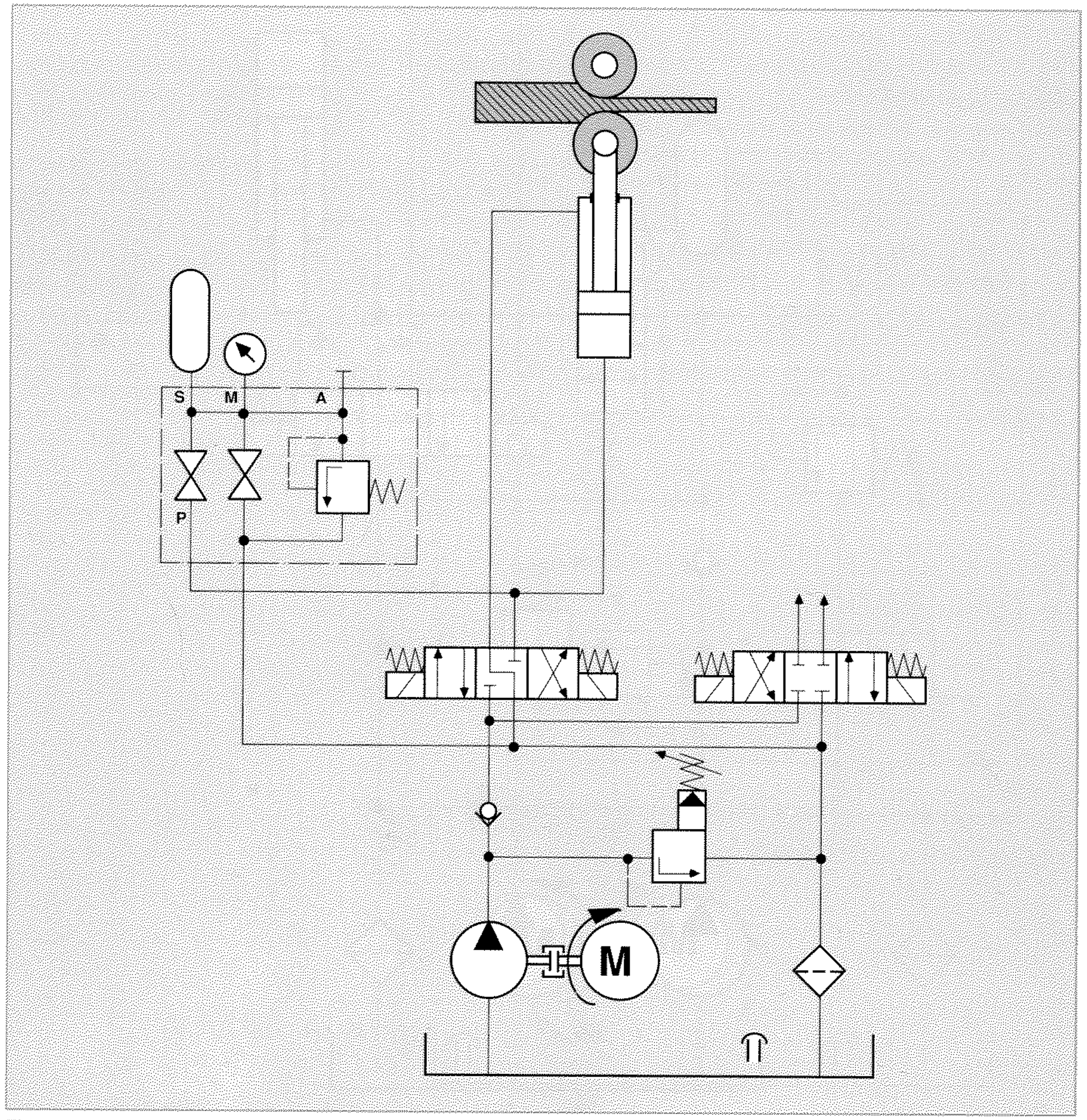


Fig. 68: *Circuit diagram for a roll balancing system*

5.4 Leakage fluid make-up

The pre-tensioning force exerted by a hydraulic cylinder can only be maintained if the leakage fluid losses in the system are made up. An accumulator is ideal for this task and Fig. 69 shows how it is used. It can be seen that the leakage fluid make-up from the accumulator is discharged into the cylinder below the piston and the pump is only started again when the pressure falls below a preset value.

Benefits

The pump does not run continuously, there is less heat wastage and service life is longer.

Typical applications

Bladder-type and diaphragm-type accumulators for leakage fluid make-up in jig and tool making, presses, lifting platforms, clamps and fixtures for machine tools, conveyor belts, roll stands, etc.

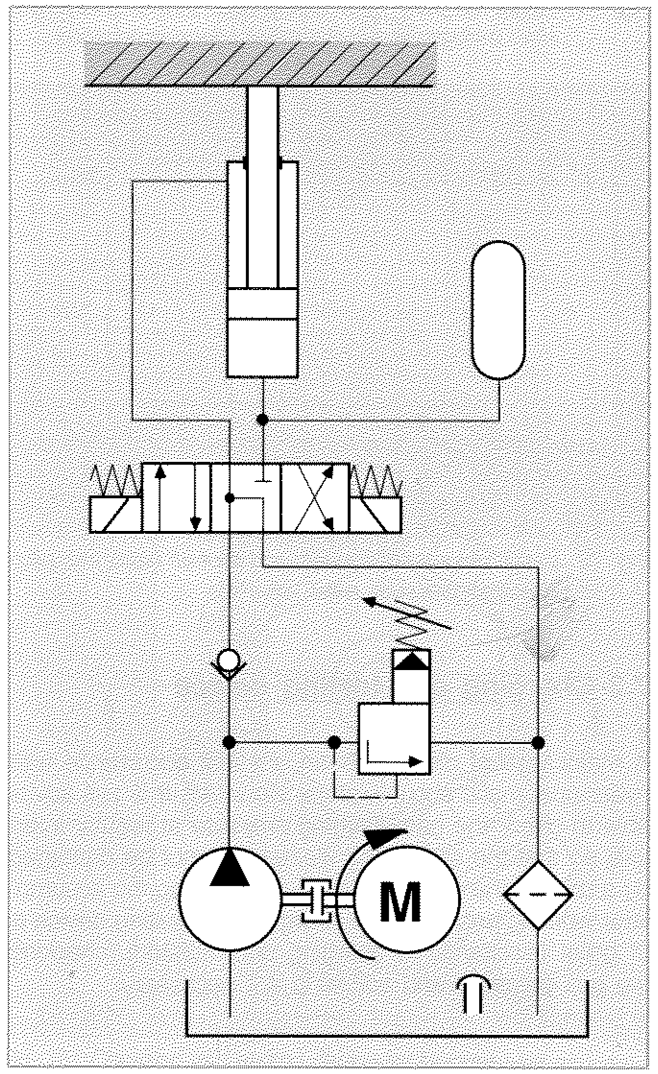


Fig. 69: *Circuit diagram for leakage fluid make-up*

6 Safety regulations

As pressure vessels, hydraulic accumulators are subject to the regulations governing pressure vessels which came into force in West Germany on July 1, 1980. Their design, installation and operation are covered by the TRB regulations. The pressure vessels of the accumulators are classified as follows according to the maximum permitted operating pressure p in bar, the capacity l in litres and the product of pressure and capacity ($p \cdot l$):

Class II: $p > 1$ bar and $p \cdot l < 200$

Class III: $p > 1$ bar and $200 < p \cdot l < 1000$

Class IV: $p > 1$ bar and $p \cdot l > 1000$

Hydraulic accumulators of Classes III and IV may only be put into service after an authorized inspector (i.e. TÜV) has inspected them (initial inspection, design inspection and pressure test) and certified that they are in satisfactory condition. This inspection by an authorized inspector can be waived if type-test approval has been obtained.

In the case of Class II accumulators the manufacturer himself certifies satisfactory manufacture and successful pressure testing. The operator's authorized inspector performs an acceptance test and provides appropriate certification. In-service tests are carried out by authorized inspectors on Class IV accumulators. The interval between internal inspections is 10 years when non-corrosive fluids are used, otherwise it is every 5 years. A pressure test is performed by the inspector every 10 years. Class II and III accumulators are inspected at intervals determined by the operator according to experience with the mode of operation and operating fluid.

Only inert gases such as nitrogen may be used in accumulators. No work may be carried out on accumulator pressure vessels until the fluid has been drained. No work may be carried out on the gas side of an accumulator until it has been depressurized.

7 Accessories for hydro-pneumatic accumulators

7.1 Safety and shut-off block

The safety and shut-off block is a device for protecting, isolating and depressurizing hydro-pneumatic accumulators. It conforms to the relevant safety regulations, especially those for pressure vessels (TRB 403 and 404 in West Germany) which require suitable devices for:

- measurement of pressure
- protection against over-pressure
- isolation.

With a simple arrangement of the various components in a compact unit, each one of these points is covered and additional benefits incorporated such as:

- minimum space requirement
- fast installation
- connections for all types of accumulator with imperial and metric fluid ports and flanged and welding connections
- extra valves such as pilot operated check valves, relief valves, flow valves, combined flow and check valves in add-on or integral form.

The construction of a safety and shut-off block is illustrated by the circuit diagram in *Fig. 70.* Basically, the unit comprises the valve block, the main shut-off cock and the depressurizing valve. It also incorporates the necessary connections to the tank, pressure gauge, accumulator and supply.

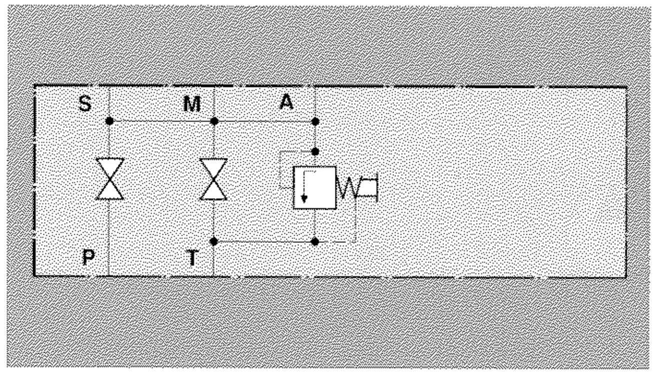


Fig. 70: *Circuit diagram of a safety and shut-off block with manual depressurization*

By fitting an extra solenoid operated two-way valve *(see Fig. 71)* it is also possible to depressurize the accumulator automatically. Another device that can be fitted is a pilot-operated pressure relief valve *(see Fig. 72)* which allows the accumulator to be depressurized quickly or in a controlled manner.

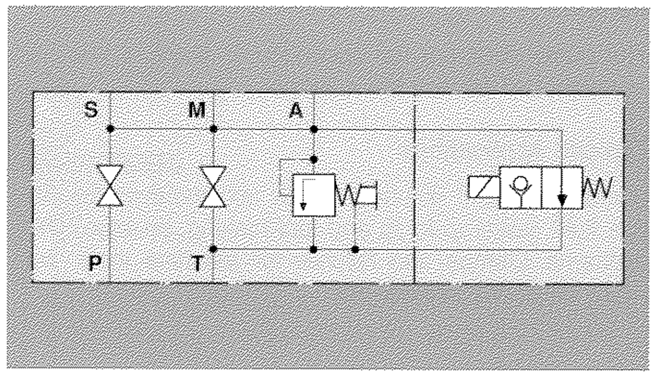


Fig. 71: *Circuit diagram of a safety and shut-off block with electro-magnetically-controlled depressurization*

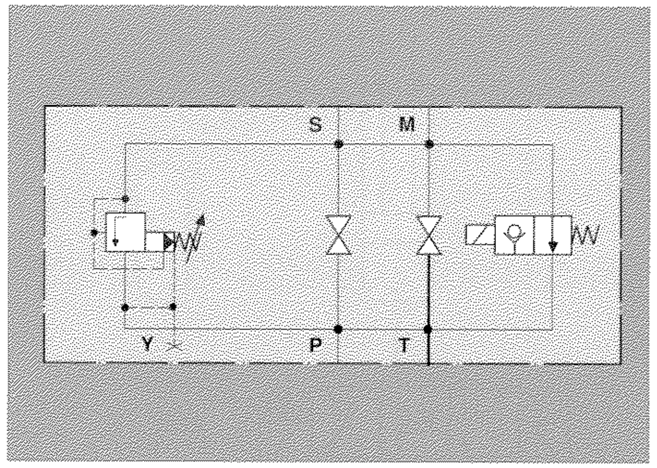


Fig. 72: *Circuit diagram of a safety and shut-off block with pilot-operated pressure relief valve*

7.2 Charging and testing device

A major loss of nitrogen is an unusual occurrence with hydro-pneumatic accumulators. However, it is still advisable to check the gas charging pressure at regular intervals so that the separating element cannot be damaged by a drop in the charging pressure and stop the accumulator working properly. The charging and testing device is of great assistance in charging accumulators, checking the gas pressure and adjusting it if necessary.

In order to charge an accumulator with the appropriate gas, the charging and testing device must be connected up as shown in *Figs. 73 and 74* with a screw connection to the gas valve of the accumulator and a flexible hose to a standard nitrogen bottle. If the gas pressure is only being checked or reduced, the charging hose will not be required. The recommended intervals between tests of the value of gas charging pressure stated on the rating plate or accumulator body are - at least once during the first week after installation, once more after a further 4 months and then annually.

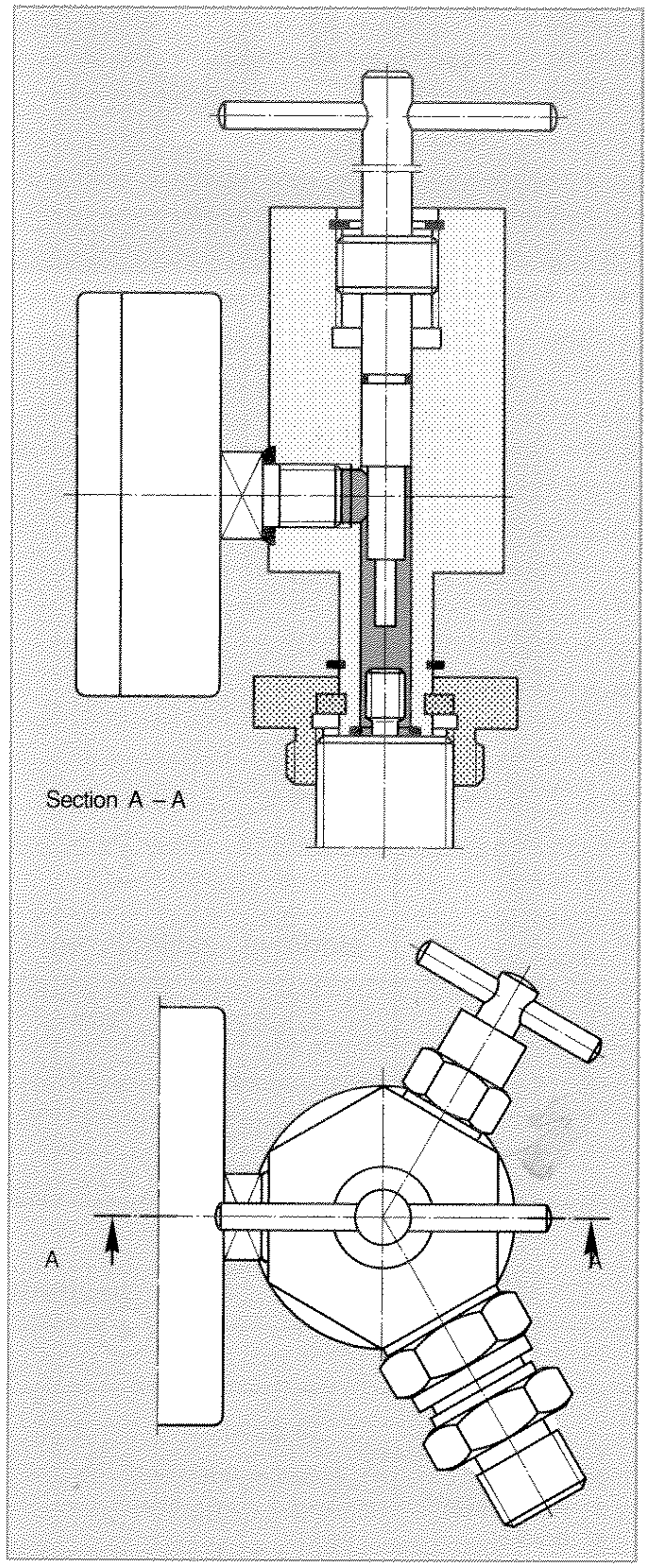


Fig. 73: *Charging and testing device for bladder-type accumulators*

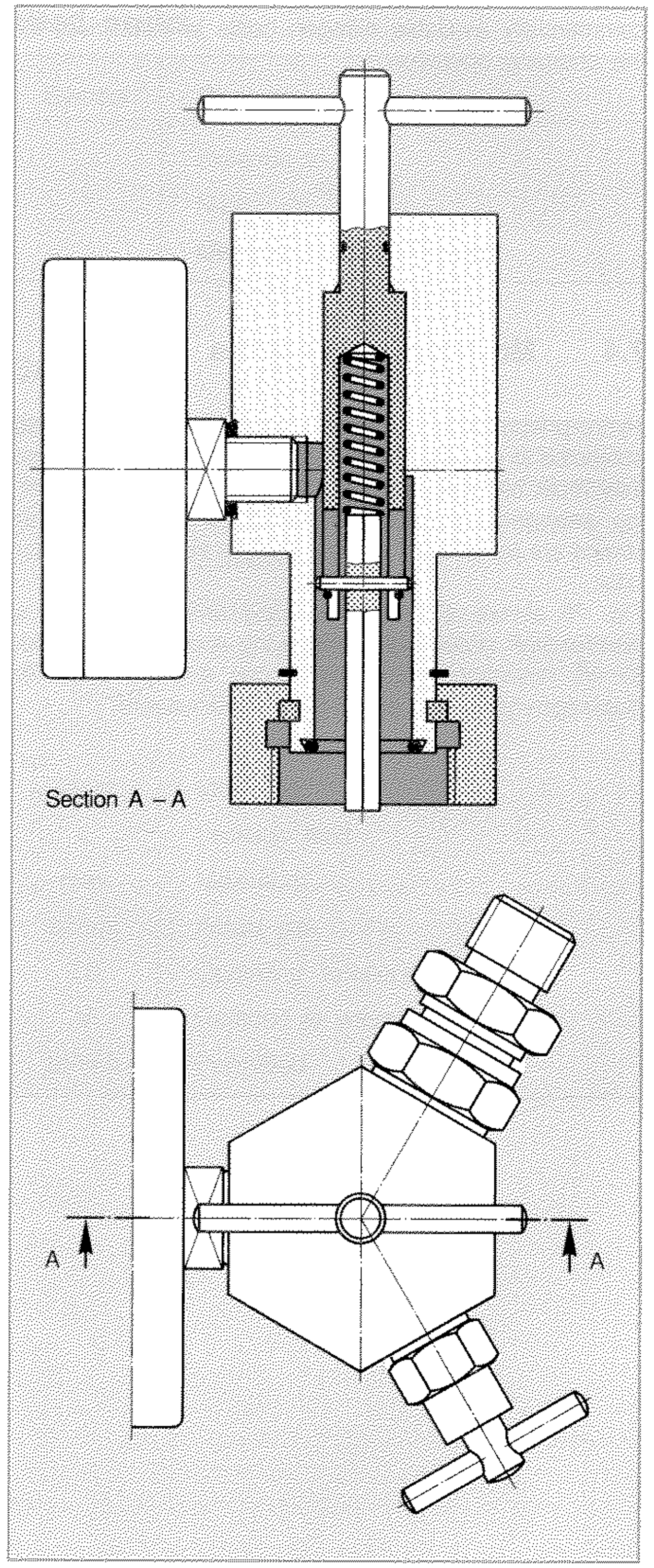


Fig. 74: *Charging and testing device for piston-type and diaphragm-type accumulators*

7.3 Nitrogen charging unit

The nitrogen charging unit shown in *Fig. 75* is suitable for charging small accumulators or for topping up the charging pressure of large multiple accumulator systems. The accumulator charging pressure can be increased using the unit; this is mostly necessary when the pressure of a commercially available nitrogen bottle is insufficient for the charging operation.

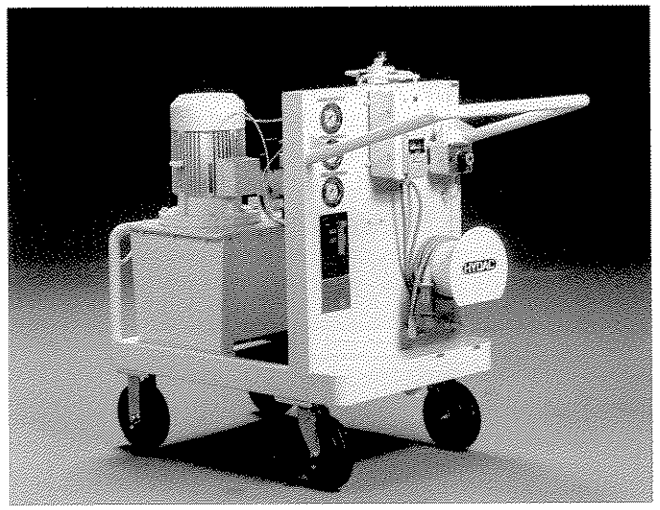
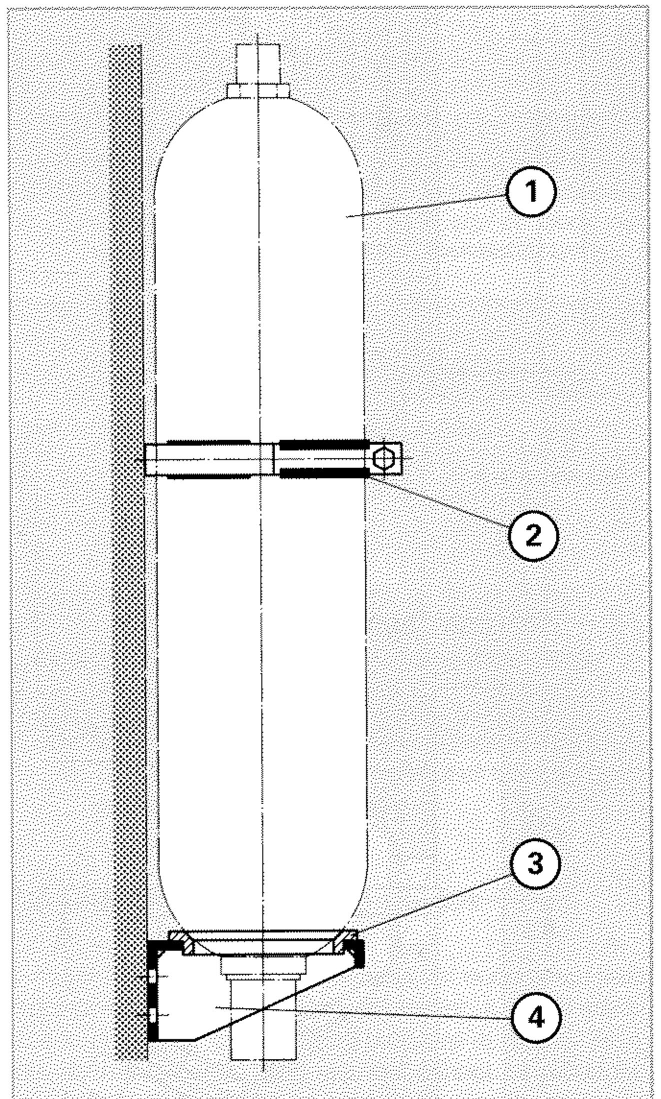


Fig. 75: *Mobile nitrogen charging unit*

7.4 Mounting accessories

Hydro-pneumatic accumulators are heavy and also subjected to acceleration forces due to the flow of fluid so they must be firmly supported and secured. The mounting arrangements must be such that no stresses or strains are transmitted to the pipework from the accumulator.

Fig. 76 shows a typical mounting arrangement for a bladder-type accumulator employing a bracket and clip. Similar mounting arrangements can also be used for diaphragm-type and piston-type accumulators.



1 Bladder accumulator
2 Clip
3 Rubber seat
4 Bracket

Fig. 76: *Mounting arrangement for a bladder-type accumulator*

8 Symbols and subscripts

Symbols

Symbol	Units	Quantity
A	m^2, cm^2, mm^2	Area
c_v	J/(kg K)	Specific thermal capacity at constant volume
m	kg	Mass
p	bar	Pressure
$\dot{Q}$	cm^3/min, L/s	Flow
R	J/(kg K)	Gas constant
s	m, cm, mm	Travel, stroke
T	°C, K	Temperature
t	s, min	Time
V	L	Volume
W	J	Work
α	W/(m^2 K)	Heat transfer coefficient
τ	s	Thermal time constant

Dimensionless symbols

Symbol	Quantity
C	Correction factor
n	Polytropic index
κ	Adiabatic index

Prefixes

Symbol	Quantity
Δ	Difference
d	Differential
$\int$	Integral

Head marks

Symbol	Quantity
$\dot{}$	Referred to time
$\hat{}$	Maximum
$'$	Deviation from initial value

Subscripts

Symbol	Quantity
0, 1, 2	Changed state
a	Adiabatic
B	Operating status
req	Required
tot	Total
i	Isothermal
ideal	Ideal gas
L	Leakage fluid
max	Maximum
actual	Actual gas
TB	Operating temperature
V	Referred to volume
permit	Permitted

9 References

[1] Rupprecht, K.-R.
Austauschvorgänge bei Hydrospeichern.
Ölhydraulik und Pneumatik,
30 (1986) 1, S. 42 – 47.

Notes

Safety regulations for accumulators in hydraulic systems

Hans H. Faatz

1. Introduction

1.1 General

The new pressure vessel specifications (Druckbeh.V) (valid for the Federal Republic of Germany), together with the associated technical regulations for pressure vessels (TRB) have been in force since 27.02.1980 and have been applicable to hydraulic accumulators in hydraulic systems and power units.

The pressure vessel specifications were produced by the specialist chemical industry panel at the central office for accident prevention and industrial health (ZefU) within the industrial associations. To all intent, they are laws of the land. This also applies to the technical regulations. Although the pressure vessel specifications were originally conceived for the chemical industry, e. g. for autoclaves etc., they must also be applied to hydraulic systems.

The authors of the pressure vessel specifications and the technical regulations have naturally used the terms normally found in the chemical industry. The requirements of fluid power technology have only partially been covered.

This area is covered by the term "Hydraulic accumulators in hydraulic systems", in the proposed standard for machine building (NAM) for the fluid power industry in the DIN standard.This should cover the application of pressure vessel specifications and technical regulations within the fluid power industry. The requirements of the standard confine themselves to hydraulic power units and hydraulic systems in machines and installations in which hydraulic accumulators are used to store energy.

1.2 The division into groups

The pressure vessel regulations always apply to hydraulic accumulators in hydraulic systems if, during the operation of the accumulator, a pressure of more than 0.1 bar or less than -0.2 bar can occur.

The product of pressure x volume *(p • l, p in bar and l in Litres)* is the factor which is applied in setting the test groups into which accumulators are divided. There are, in fact, seven groups. In general, only groups II, III, and IV are applicable to hydraulic accumulators in hydraulic systems.

Group II:

This group covers all hydraulic accumulators with a permissible operating pressure of more than 1 bar, and in which the product of pressure x volume *(p • l)* is not more than 200.

Group III:

This group covers all hydraulic accumulators with a permissible operating pressure of more than 1 bar, and in which the product of pressure x volume *(p • l)* is more than 200 and not more than 1000.

Group IV:

This group covers all hydraulic accumulators with a permissible operating pressure of more than 1 bar, and in which the product of pressure x volume *(p • l)* exceeds 1000.

Group I applies to all pressure vessels subject to negative pressures.

Groups V, VI, and VII apply to all pressure vessels subject to pressures above 500 bar.

The acceptance details and the associated tests are laid down in the individual group specifications. They apply to both the manufacturer and to the operator of hydraulic accumulators.

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2 Terms

Within the pressure vessel specifications and the technical regulations, terms are used which are practically never found or are totally unusable in the fluid power industry. It is therefore necessary to define the terms laid down in the pressure vessel specifications and the technical regulations to make them acceptable to the fluid power industry.

An explanation of these terms related to fluid power technology, and as far as is required, is laid out in the following sections. This does not strictly comply with the standards, but is of more practical use.

2.1 Hydraulic systems/ hydraulic power units

Hydraulic systems consist of an electric motor driven pump delivering fluid, a fluid reservoir, control valves and hydraulic motors together with the necessary pipes and hoses for the operation of the system.

Hydraulic power units consist basically of a pump delivering fluid, driven by an electric motor, the control valves and the fluid reservoir.

(Similar to the official document of the European Community, 19.04.85).

2.2 Hydraulic Accumulators

2.2.1 Hydraulic accumulators as energy storage devices in hydraulic systems

Gas loaded hydraulic accumulators used in hydraulic systems, with of without a separating wall between the liquid and the gas, are pressure vessels in the terms of the pressure vessel specifications and the technical regulations. In general, accumulators employed in hydraulic systems are of piston, bladder, or membrane design. Only occasionally are so-called air loaded accumulators to be found. The first three types all have a separating wall or membrane between the liquid and the gas, while the latter does not.

Pipes, pumps, valves, cylinders, filters, and isolating valves do not , at the moment, come within the scope of the pressure vessel specifications and the technical regulations (of 27. 02. 1980)

2.2.2 The labelling of hydraulic accumulators

Hydraulic accumulators must at all times have durable and easily readable labels showing:

- Manufacturer or supplier
- Manufacturer's number
- Year of manufacture
- Permissible operating pressure
- Internal volume
- Permissible operating temperature
  if more than 50 °C or less than 10 °C
- Official test approval number
  (for officially tested accumulators)

2.3 Operational pressure (above atmospheric zero)

For the operation of pressure vessels in fluid technology and under the pressure vessel specifications and the technical regulations, certain important pressure ratings must be observed.

- Working pressure
- Permissible operational pressure of the hydraulic accumulator
- Permissible operational pressure of the hydraulic system.

2.3.1 Working pressure

This is designated as the pressure at any moment in time during the operational process at a predetermined point in the hydraulic system.

2.3.2 Permissible operational pressure of the hydraulic accumulator

Permissible operational pressure of the hydraulic accumulator is that pressure at which the accumulator is permitted to operate. It is also known as the nominal pressure. This is the pressure at which the hydraulic accumulator must be labelled for continuous operation. Under certain circumstances, the permissible operating pressure of the accumulator may be different dependent upon acceptance society *(see section 5).*

2.3.3 Permissible pressure of the hydraulic system

The permissible pressure of the hydraulic system is the pressure to which the hydraulic system is limited. This pressure is dependent upon the duty of the system and is determined by the project engineer or the operating company.

2.4 Pressure measuring devices

In hydraulic systems, pressure measuring devices are normally pressure gauges. It is important that this pressure gauge is installed on the fluid side of the accumulator. A pressure gauge may be fitted on the gas side, but is not obligatory.

The range of indication of the pressure gauge must be at least 1.5 times the operating over pressure of the hydraulic system.

It must be possible to test the indication of the pressure gauge e.g. by means of a test connection in the vicinity of the accumulator or an isolating cock to DIN 16 262 or DIN 16271, or by removing it and testing on a separate test stand.

The permissible over-pressure of the system is to be displayed in the vicinity of the pressure gauge. The indication must be durable.

The pressure gauge employed must be compatible with the system fluid and may not be made unworkable by the system fluid.

Damage to the pressure gauge must not cause any kind of danger.

2.5 Safety devices to prevent excessive pressures occurring

Corresponding to the difference between the definitions of operating pressure of the hydraulic accumulator and the hydraulic system, the safety devices must differentiate between limiting the system pressure and preventing the operating pressure of the accumulator from being exceeded.

2.5.1 Safety devices to prevent excessive pressure in the hydraulic system

This can either take the form of a pressure regulated pump or a pressure relief valve.

2.5.2 Safety devices to prevent excessive pressure at the hydraulic accumulator

Safety devices to prevent excessive pressure in an accumulator are defined as safety valves to AD guidance sheet A2. These are officially design tested valves. The valves are subject to an acceptance test in the manufacturer's factory by an official inspector. At this time, they are correctly tested and the setting and their compatibility with the fluid to be used is checked. The setting is then sealed (normally by a lead seal) so that they cannot be set to higher pressures.

These safety valves are described as TÜV valves, as the setting is normally carried out in the manufacturer's factory by an inspector from the TÜV.

The pressure safety valves fitted to hydraulic accumulators must be self operating and must limit the pressure in the accumulator to a maximum excess pressure of 10%.

The accumulator safety valves should not be called upon to operate during the normal operation of the machine. The pressure should therefore be set sufficiently far above the permissible excess pressure of the hydraulic system.

Regardless of the setting of the permissible excess pressure within the hydraulic system, it is recommended that the accumulator safety valve is set so that the operational pressure of the accumulator cannot be exceeded by more than 10%. It must be noted that at this time the whole of the pump flow may be passing through the accumulator safety valve.

And now an example

In a hydraulic system with a system pressure of 100 bar, a hydraulic accumulator with a permissible pressure of 210 bar is to be installed.

With a pressure drop of 31 bar across the relief valve due to the pump volume, the accumulator safety valve may only be set at:

$p_{max.}$ = 210 bar + 10% – 31 bar = 200 bar.

If one pressure relief valve is not sufficient to pass all the pump flow, a number of pressure relief valves may be installed in parallel.

The accumulator safety valve must be connected to the pressure source, on the hydraulic accumulator or the pressure line by its own connector. The valve must not be able to be made unworkable by the system fluid.

It must be ensured that the pressure in the accumulator cannot flow back to the pressure source. Generally a non return valve is fitted between the pump and the accumulator safety valve. This non return valve can only be omitted if the pump design contains its own integral non return valves.

The tank line from the accumulator safety valve must pass the fluid which may pass along it safely to the tank. As this pipe may be subject to sudden shock loading, it must be held firmly in place. In addition, it must be noted that the pipework must not lead to a further rise in pressure at the accumulator safety valve.

Accumulator safety valves must not be designed as fire safety valves.

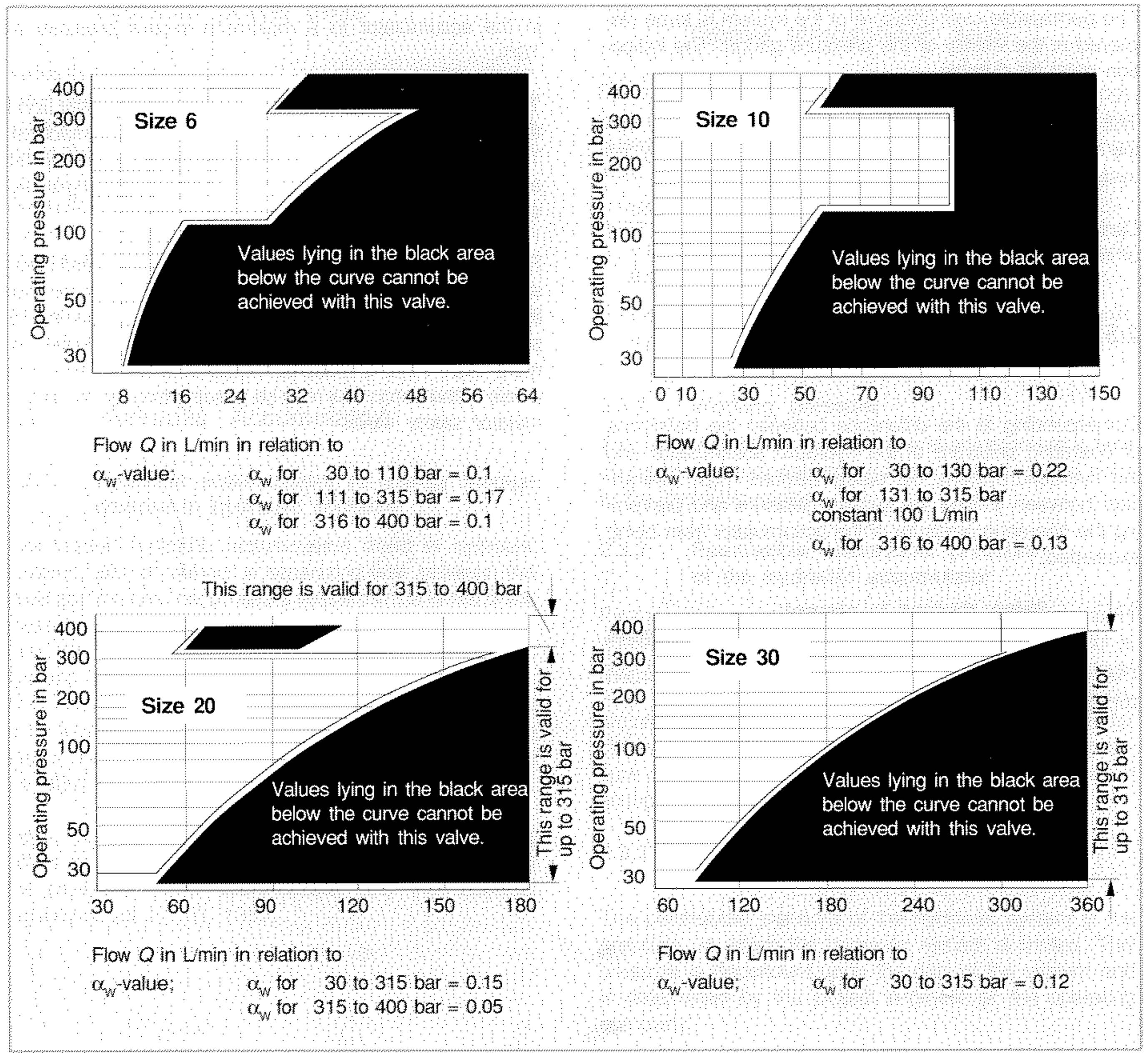


Figure 77: *Pressure/flow dependence of design tested pressure relief valves*

2.6 Isolating devices

Isolating devices are poppet valves, spool valves and cocks.

It should be possible to isolate the accumulator from the pressure line so that maintenance and possible replacement is possible during operation of the system.

All isolating devices must be easily accessible. It must be possible to open and close them under operating conditions and an indication of "open - closed" must be made. It must be impossible to accidently remove spindles by screwing them out.

2.7 Pressure unloading valves

Pressure unloading valves are hand operated devices with which it is possible to unload the fluid side of accumulators in which the gas is physically separated from the fluid. In this process the fluid must be throttled so that it passes safely to the tank. The lever position of these valves must again carry an indication "open-closed".

Pressure unloading valves are "pressure warning devices" in the sense of the pressure vessel regulations.

3 Typical circuits

There follows a few illustrations of hydraulic accumulators as energy storage devices in hydraulic systems. No claim is made that this selection of circuits is complete. Other circuits are certainly possible. They are valid for all accumulators with a separating wall between the gas and the fluid.

3.1 Typical circuit of a hydraulic accumulator without a self operating unloading device.

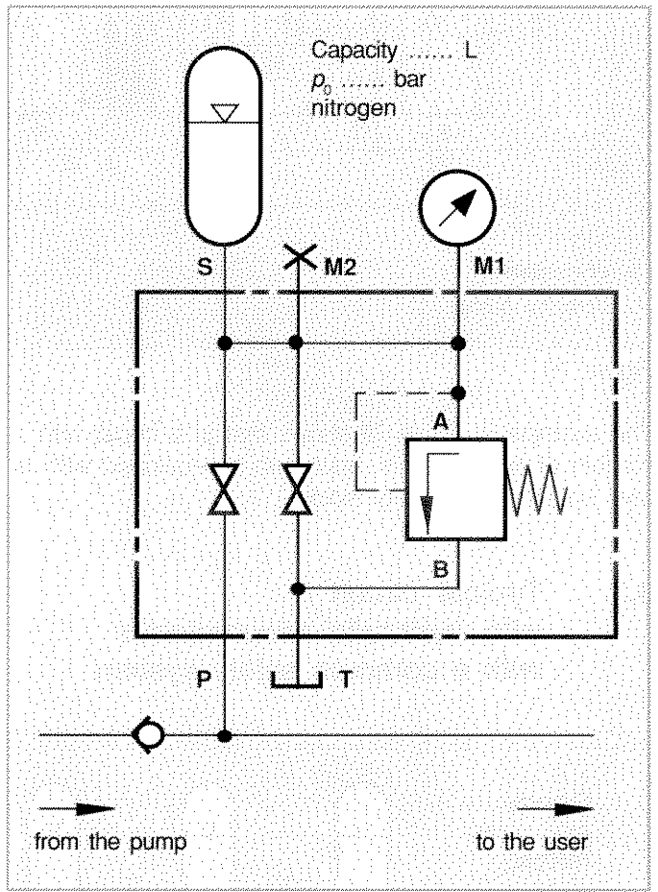


Figure 78

The pressure relief valve shown here can be unloaded by means of a directional poppet valve so that the accumulator is not under pressure when electrical power is not present.

3.2 Typical circuit with a number of accumulators each with its own safety and isolating block

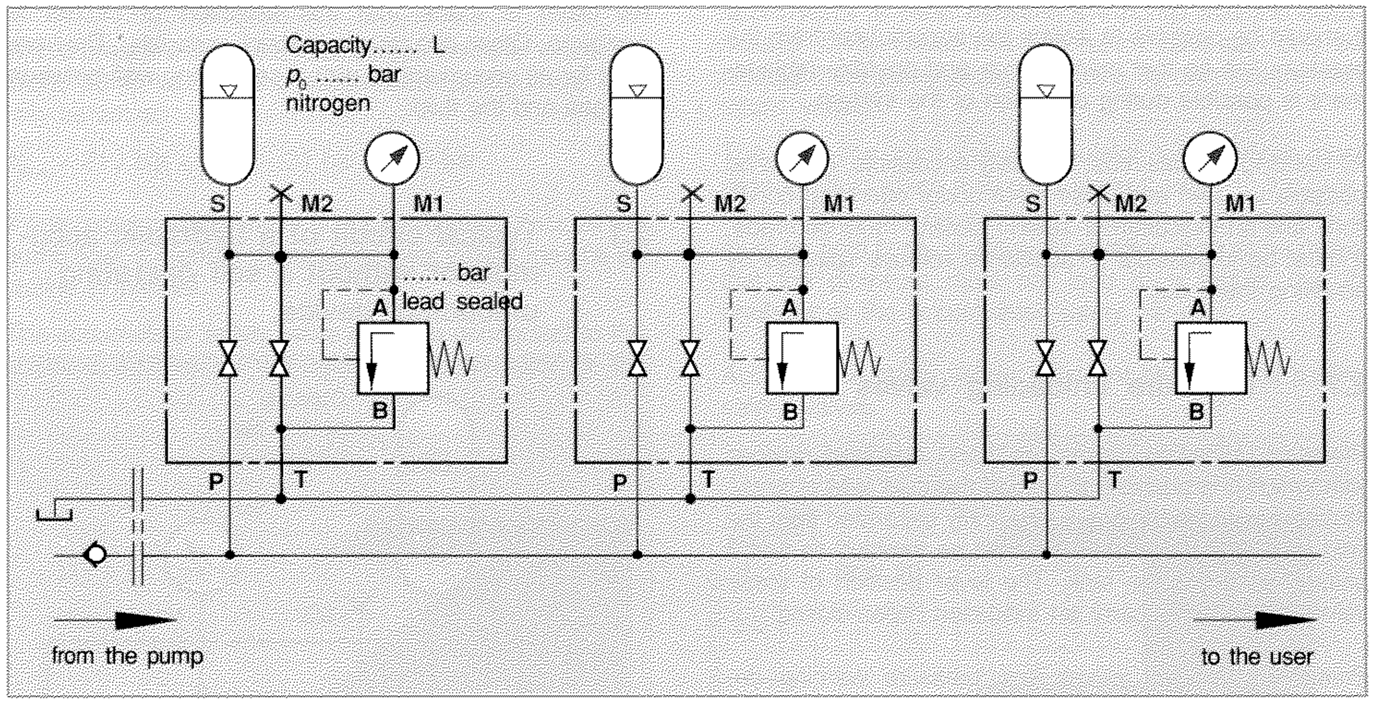


Figure 79

The example in *figure 79* shows that in a circuit with a number of accumulators, each can be equipped with its own isolating and safety block.

3.3 Typical circuit with hydraulic accumulators with a common safety system

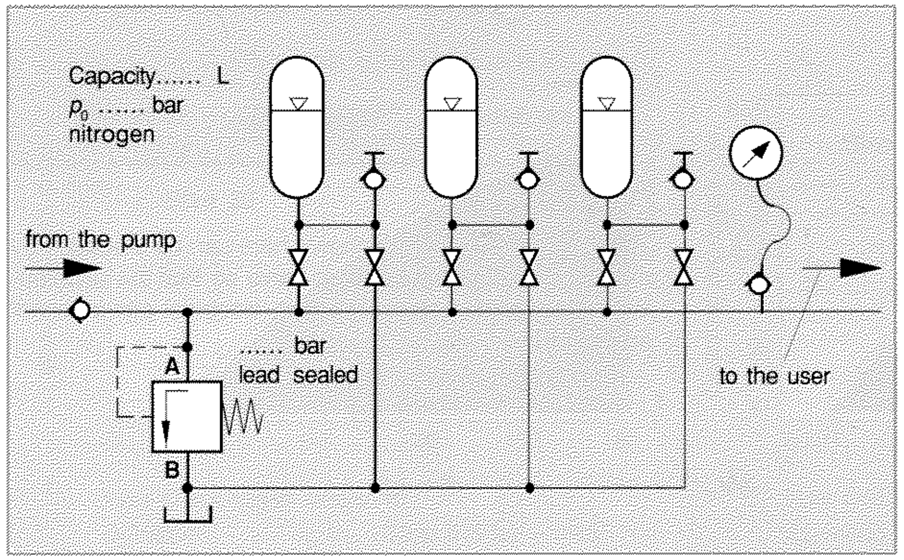


Figure 80

The example in *figure 80* shows that a number of hydraulic accumulators can be safeguarded with a single safety valve. It is recommended, but not the law, that it should be possible to isolate each individual accumulator from the system. If this isolation is included, it must be possible to unload each accumulator individually. It is again to be recommended, but again not the law, that it should be possible to check the pressure in each accumulator separately.

3.4 Typical circuit of one or more accumulators backed up by gas bottles.

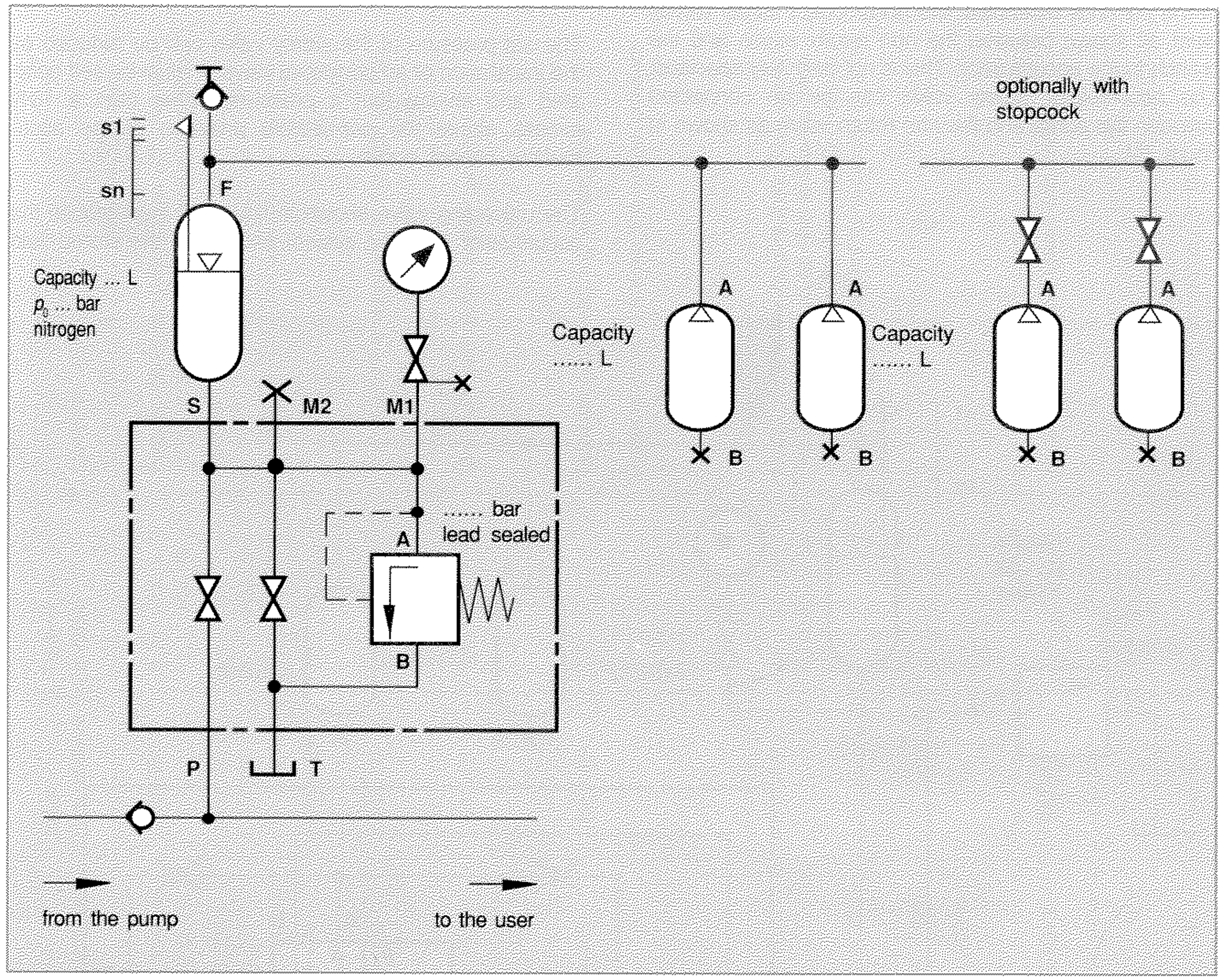


Figure 81

It is a sensible arrangement to fit a connection "B" on the gas bottles to allow condensation to be drained off.

3.5 Typical circuit of a hydraulic accumulator with automatic isolation should electrical power fail

In the previous examples, the accumulator safety valves were all shown between the non return valves and the branch to the accumulator. This is not definitely specified. The accumulator safety valve can be be connected between the pump and the non return valve as shown in *figure 82*.

The set pressure of the safety valve can be reduced by the installation of an additional valve. It must be self operated and ensure that a pressure higher than a 10% excess above the permissible pressure of the accumulator cannot occur.

The unloading valve shown in figure 82 must isolate the accumulator from the system and return the stored fluid

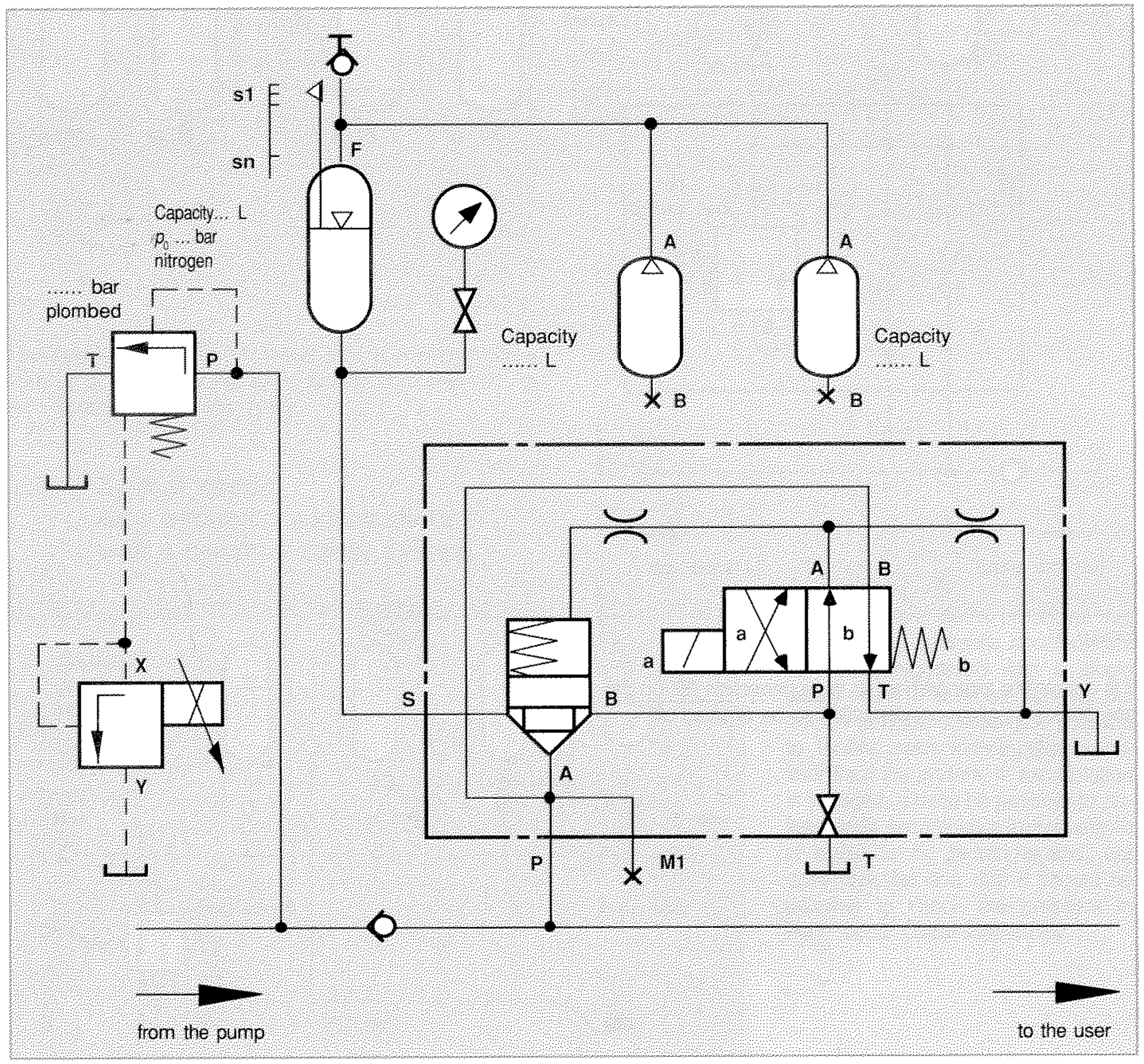


Figure 82

4 The relationship between the sections of this chapter and the relevant official regulations

Section	Title	Guidelines	Issue/ Proclamation	Reference section	Remarks
2.2.1	Hydraulic accumulators	Druckbeh.V. Anhang II § 12 TRB 002 TRB 801	27.02.80 2.84/5.86 2.84/5.86	§ 3, Abs. 1, 2 Abs. 4 1.1 4 Abs. 5	Bladder, piston and membrane accumulators come under pressure vessels (Druckbehälter V).
2.2.2	Labelling hydraulic accumulators	TRB 401	11.83/9.85	2.1	
2.3.2	Permissible operational overpressure of the hydraulic accumulator.	TRB 002	2.84/5.86	1.4.1	The pressure is according to the label on the accumulator.
2.3.3	Maximal operational over-pressure of the hydraulic system in which the hydraulic accumulator is employed.	TRB 002	2.84/5.86	1.4.2	Maximum permissible system pressure.
2.4	Pressure measuring devices	TRB 403	1.84/9.85	2.1 2.1.2 2.1.3 2.1.4 2.1.5 2.2 a, b	Pressure gauge with red warning line. Pressure range 1.5 times the system pressure.
2.5	Safety devices to prevent excessive pressures occuring.	TRB 403	1.84/9.85	3.1 3.1.1 a 3.1.2 3.1.4 3.1.5 3.1.7 3.1.8 3.2 3.5	Safety valve to the AD sheet A2 (TUV valves). The pump flow must be able to flow via these valves. In doing this the permissible pressure of the accumulator may not be exceeded by more than 10%.
2.6	Isolating devices	TRB 404	1.84/9.85	3.1 3.2 3.3 3.5	Poppet valves, spool valves and cocks.
2.7	Pressure unloading valves	TRB 404	1.84/9.85	4.1	Unloading the accumulator to tank.

Table 17

5 Acceptance specifications for hydraulic accumulators in various countries

Country	Tested by/to	Approved by	Remarks
Africa (other)	TÜV	TÜV	See German Federal Republic, but with documentation in English
Africa (South)	LRIS/ASME code	LRIS or TÜV	In general, the final customer specifies the approval required. If this is not given, to LRIS.
Algeria	Service des Mines Algeria	Service des Mines Algeria	The Service des Mines France is not generally recognised by the Service des Mines Algeria. In exceptional cases the Service des Mines France is accepted. In case of doubt, the customer must decide. Approval by the Service des Mines Algeria is very expensive to obtain. The permissible operating pressures are lower than the Service des Mines France.
America (South)	LRIS/ASME code	LRIS	
Australia	Department of Labour and Industry to Australian standards	LRIS or TÜV	The permissible operating pressures are lower than for TÜV
Austria	TÜV Vienna, to their own standards	TÜV	
Belgium	Apragaz Brussels	Apragaz	
Bulgaria	TÜV	TÜV	Some technical details must be cleared by the final customer.
Canada	National Board (with U stamp) Hydac-ASME code	Lloyd's Register Insurance Inc. (Authorised Inspector)	U stamp (as for the USA) is generally accepted. However, in some provinces the additional testing conditions are expensive both in time and money.
Czechoslovakia	TÜV	TÜV	Some technical details must be cleared by the final customer. An accumulator passport must be obtained for each pressure vessel.
Denmark	Derekoratet for Arbajdsog Fabriktil-synet (available for standard production series accumulators)	TÜV	Name plate in Danish
Finland	TTIC (pretesting to be paid for)	TÜV	

Country	Tested by/to	Approved by	Remarks
France	Service des Mines	Service des Mines	The approval of welded accumulators is very troublesome. Standard are easier.
German Democratic Republic	Staatl. Amt für Technische Überwachung (TÜ), Technical (TÜ) inspection of the type presented	TÜ approved inspector	Some technical details must be cleared by the final customer. Safety valves must have the approval of the TÜ.
German Federal Republic	TÜV	TÜV	
Great Britain	LRIS, British standards	LRIS	
Holland	Stoomwezen Büro of the relevant district	Stoomwezen Büro of the relevant district	
India	LRIS, Indian standard	LRIS	Pre-testing and approval by LRIS. Operational overpressures less than for TÜV.
Italy	ISPESL - Rome	TÜV to Italian standards, partly through ISPESL themselves	TÜV approval accepted up to a nominal volume of 25 L. Above this approval by ISPESL is required. The operational overpressure is reduced by around 20%. Special safety valves must be fitted on the gas side.
Luxembourg	TÜV	TÜV	If required, approval can also be obtained from the Inspection du Travail et Mines (a private organisation). Up to now this approval has not been required.
New Zealand	LRIS in Croydon (GB) to New Zealand standards	LRIS	For approval by LRIS, an "as built drawing" is required for approval by LRIS in Croydon. Only then will the unit be approved.
Poland	UDT	TÜV, Hydac has the right to use the UDT official stamp of approval	Documentation in Polish
Portugal	LRIS	LRIS	Pre-testing and approval lies between Portugal and LRIS. In part, a lower operational over pressure is permitted than for the TÜV.
Rumania	TÜV	TÜV	Some technical details must be cleared by the final customer.
Spain	TÜV	TÜV	TÜV documentation will be validated by the Spanish Consulate.

Country	Tested by/to	Approved by	Remarks
Sweden	AB Statens Anläggningsprovning (SA) und Arbetars-kyddsstyrelesen	TÜV (under contract from SA)	The oil valve and the split ring must be made of a different material to standard
Switzerland	Schweizerischer Verein für Druckbehälter (SVDB)	TÜV	Entry test by SVDB which must be paid for.
USA	National Board (U stamp) Hydac-ASME code	Lloyd's Register Insurance Inc. (Authorised Inspector)	The U stamp is not strictly required in some states. It is most strongly recommended for importation purposes.
USSR	Gost-Norm (covered by TÜV approval)	TÜV	Accumulator pass port
Yugoslavia	TÜV	TÜV	Some technical details must be cleared by the final customer.

LRIS = Lloyd's Register Industrial Services (Hamburg)
TÜV = Technischer Überwachungsverein (Federal Republic of Germany) (Technical monitoring association)
ASME = The American Society of Mechanical Engineers (USA)
AD = Arbeitsgemeinschaft Druckbehälter (Federal Republic of Germany)
TÜ = Staatl. Amt für Technische Überwachung (State office for monitoring technical standards) German Democratic Republic
UDT = Urzqd Dozoru Technicznego (Poland)
TTIC = Teknillinen Tarkastuslaitos (Finland)
SdM = Service des Mines (France)

Notes

Filtration in Hydraulic Systems

Martin Reik

1 Introduction

Efficient and effective filtration in hydraulic systems is absolutely essential in order to prevent malfunctions and, at the same time, to increase the service life of important and expensive components.

Any analysis of the causes of hydraulic system failure will show that a majority of them are due to solid particles contaminating the fluid.

Such contamination is a result of inadequate filtration.

The repair costs of components can only be kept under control by preventive maintenance of the whole system. Constant checking of the fluid (*see Section 4.8*) provides a background for the condition of the fluid at any time. The necessary counter-measures can then be adopted as soon as deterioration sets in and any resulting damage can be minimized.

The constant clamour for better performance from hydraulic components means that fits and clearances are becoming ever tighter. Whereas in past years an absolute filtration rating of between 80 and 100 µm was usual for hydraulic systems, nowadays the minimum value is around 20 µm. When servo valves are used in a system the figure can be as low as 3 µm.

A correct choice of filter is essential as early as the project design stage of a system. However, the initial good intentions of project engineers are often overridden by price considerations once the contract is awarded. Changing the size of filter and the filtration rating is a simple method of reducing a quoted price without seeming to have any adverse effect on the functioning of the system. Retro-fitting of a more suitable filter, however, is complex and expensive. Also, the overall impression of the system suffers from the fact that the filters are less than the best. This often spoils the carefully nurtured image of the supplier.

It cannot, therefore, be emphasized strongly enough that there should be no "cutting of corners" as far as filters are concerned. Any extra costs incurred through the use of larger but optimum filters will definitely and quickly be recouped through less maintenance and downtime.

Using filters with more filtration surface area reduces the surface loading for the same throughput. This produces a disproportionate increase in filter life (*see Diagrams 32 and 40*).

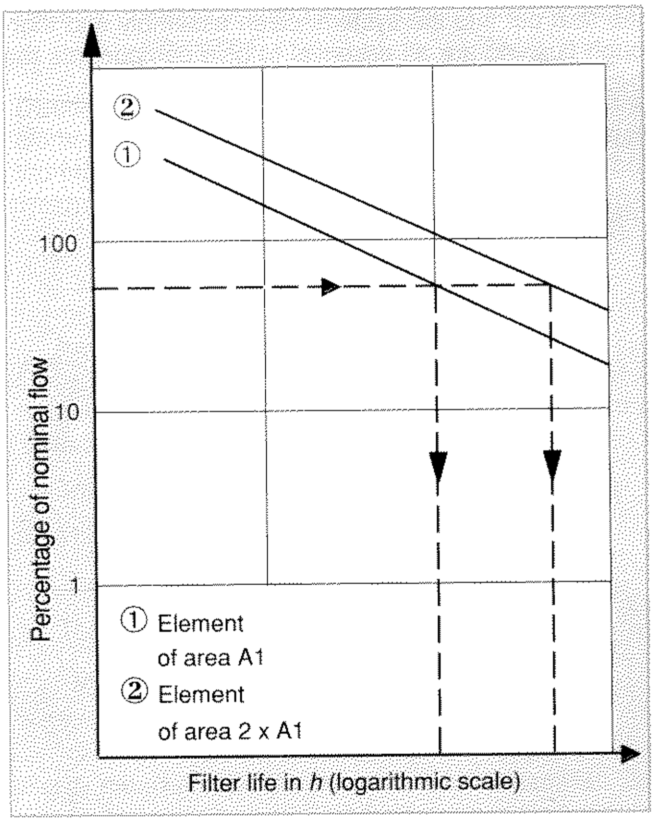


Diagram 32: *Extended filter life with more filter area*

When selecting the filtration rating

- remember that the component with the tightest clearances governs the rating figure for the whole system
- and filtration ratings must be selected for those components. Power units must not simply be equipped with the finest filter available from the manufacturer.

2 Functions of filters in hydraulic systems

2.1 The effect of solid particle contamination

General

Tests on hydraulic systems have shown that reducing the amount of solid particles in the hydraulic fluid makes an important contribution to longer component life and functional reliability.

Solid particles are often produced by high mechanical or hydraulic stresses and, when they are allowed to circulate unhindered in the system, cause severe wear. Quite naturally, this in turn produces more solid particles. Contamination entering the system from outside can sometimes initiate or accelerate the condition. The chain reaction of solid particle production and accumulation can be minimized by the use of a good filter. Effective filtering of the fluid put into the system, clean assembly and thorough flushing are all essential to give components the best start in life.

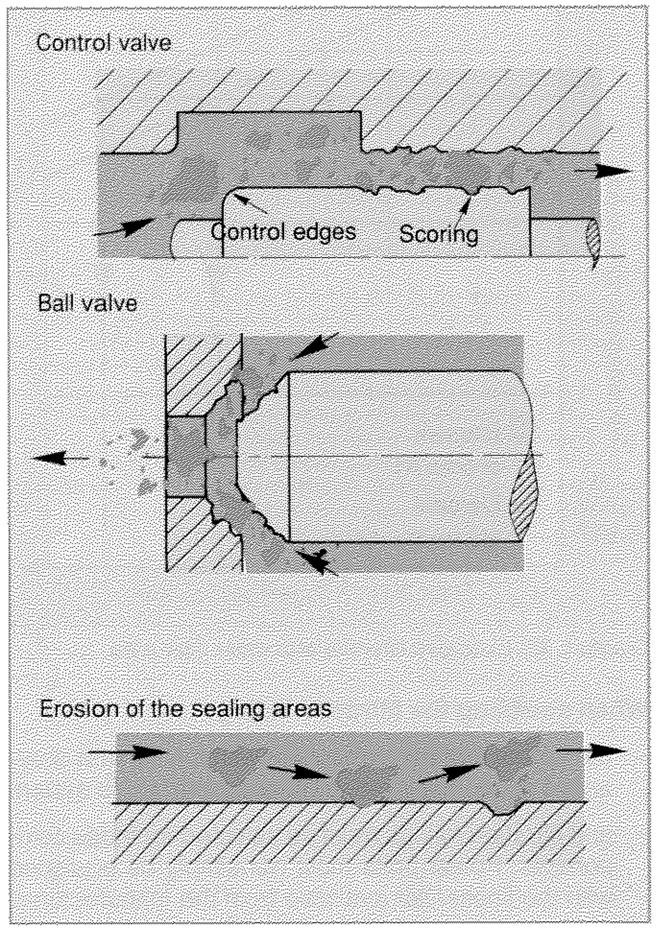


Fig. 83: *Surface damage due to solid particles*

Control valves and pumps

It will be clear from *Fig. 83* that solid particles substantially smaller than the clearance between two surfaces, e.g. an oscillating spool, cause no damage; they simply pass through the gap. If the relative motion is only occurs occasionally, however, there is a danger of the gap becoming silted up which can cause the valve to malfunction. Any solid particles larger than the gap build up in front of it and, to begin with, have no adverse effect on the valve. However, at high operating pressures or flow velocities there is a danger of the motion of the spool crushing the particles and forcing them into the gap.

Particles of about the same size as the gap itself are the most dangerous of all for the components because they cause scratches and therefore heavy wear. The cutting action of the original particles produces new particles and a chain reaction is initiated. Component failure is very often due to these solid particles.

Control lands frequently suffer from erosion due to the high flow velocities prevailing there. The erosion is made worse by any solid particles in the fluid. The end result is a change in the control characteristics of the component.

Seated valves (poppet valves)

Particles can become trapped between the valve and its seat, so causing it to leak.

Throttles and orifices

Particles of dirt in the hydraulic fluid become stuck in the orifices and the restriction interferes with the accuracy of flow control.

Bearings

With sintered bearings the particles of dirt get into the pores or, if they are hard particles, they can be pressed into the relatively soft sintered material. Severe scoring of the shaft is the result. The blocking of lubricating grooves is also a possibility, causing the bearing and shaft to run hot.

Erosion by solid particles

Due to the pressure gradient across a gap, solid particles are forced through the gap at approximately the same velocity as the flow within the gap. Due to its mass, each particle possesses certain energy which is given up when the particle strikes the surface. This causes other particles to be detached from the surface, so increasing the amount of solid particles in the fluid.

The effects of solid particles in a system

- Increased leakage
- Jamming of pistons and spools
- Component failure
- Changes in control characteristics

2.2 Types of contamination

The following are the different kinds of contamination encountered in hydraulic fluids (see Table 18).

Hard and sharp particles

These are mainly responsible for the wear of components. Their precise effect on the component depends on their shape and what they are made of.

Hard and sharp particles cause deep scratches and so are more dangerous than soft, spherical particles. They must be filtered out by means of filters in the systems which must be of a size and rating appropriate to the type of contamination anticipated.

Soft and gelatinous particles

These can cause blockages in working clearances and so lead to component failure. They also interfere with lubrication by gumming up lubricating channels.

Good system filters will take out these particles but they eventually block the filter element so a reduced filter life must be anticipated.

Dissolved substances in the fluid

These do not cause any wear of the components but they can lead to changes in the lubricating characteristics, faster ageing, carbonization and the deterioration of the filtration capacity of the fluid.

Material	Effect
Carborundum Scale, rust particles	Very severe damage
Steel Iron Brass Bronze Aluminium	Severe damage
Laminated fabric Fibres Seal residue Rubber particles from hoses Paint particles Oxydation products from fluid	Slight damage

Table 18: *Effects of solid particle contamination on working clearances*

Dissolved substances cannot be filtered with normal filter elements so all the fluid must be changed and the system thoroughly flushed out.

2.3 Effect of contamination on component wear

Generally speaking, all solid particles cause wear in hydraulic components. However, the actual amount of wear depends on the following parameters:

- The material of the solid particles
- The size of the solid particles
- The ratio of particle size to working clearance
- The shape of the particles
- The working pressure
- The flow velocity

Hard, mineral particles in even small quantities can cause serious damage. The frequency of damage depends on the operating pressure. The higher the pressure in the system, the more the particles are forced into the working clearances and the greater is the damage.

2.4 The origins of solid particle contamination

Before or during commissioning

In spite of thorough flushing of the equipment after assembly it is impossible to remove entirely every last particle of dirt from the components and pipework. Tests have shown the following substances to be present on many occasions:

core sand, weld spatter, swarf, scale, fluff,
rust, packaging residue, paint.

The hydraulic fluid itself with which the system is filled can be a major source of contamination so the following procedures should be adopted before the system is commissioned.

a New fluid should always be put into the system through the filter or a separate filter unit similar to that shown in Fig. 84. The filtration rating of the flushing filter or filling unit must be at least the same as the filtration rating for normal operation of the system.

b The existing hydraulic fluid in the system should be cleaned with a separate filter unit.

c Start the hydraulic pump. Keep the separate filter unit operating and this will ensure that the large amount of solid particle contamination to be expected in the fluid returning from the hydraulic power unit will also be filtered out.

d After a predetermined amount of flushing, take a fluid sample and determine the amount of contamination. Dependent upon the results, more flushing may be needed.

As *Diagram 33* shows that, the amount of flushing depends on the size of the tank, the cleanliness of the installation, the components being used, the required cleanliness of fluid and the cleanliness of the new fluid added.

The amount of flushing required can be gauged only very roughly before the actual flushing begins and it must be anticipated that high-precision components might be damaged during the flushing. Therefore, this type of component, e.g. servo valves and proportional valves, are best not installed until the flushing has been completed.

Of course, an installation should always be flushed again by the operator if any subsequent modifications are made to the piping, after repair work or if the equipment is moved to a new site.

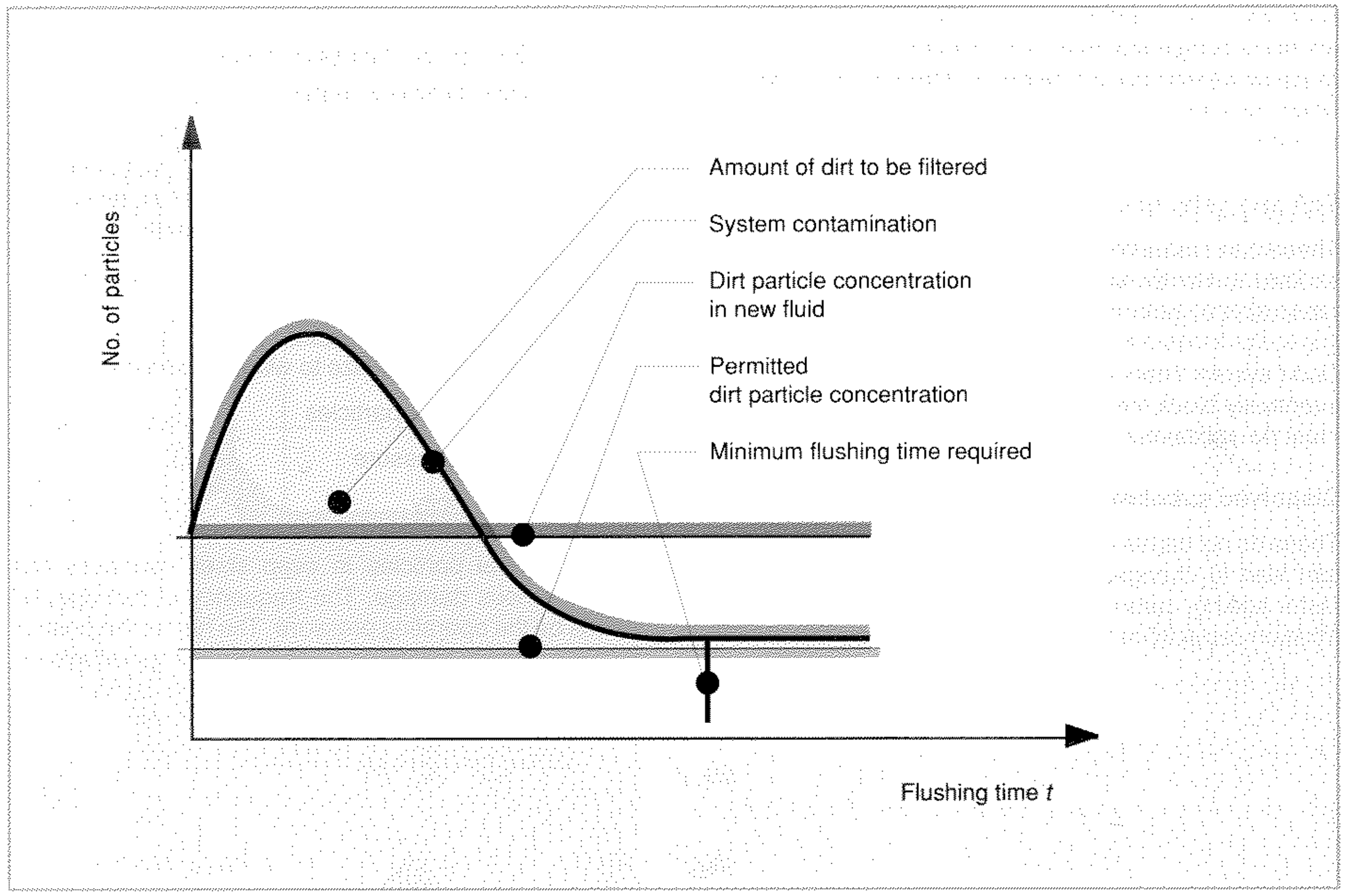


Diagram 33: *Variation in particle concentration during flushing of a hydraulic system*

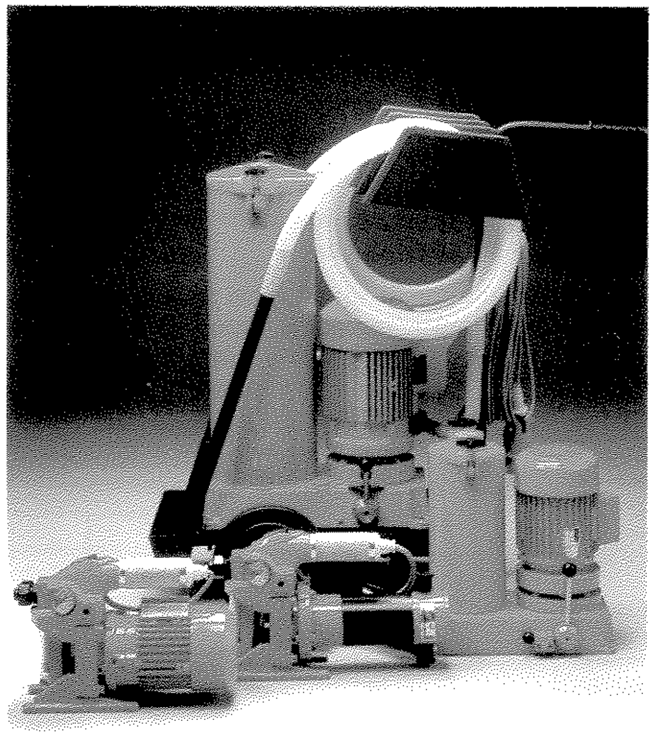


Fig. 84: *Separate filter unit for cleaning hydraulic systems and filling them with fluid*

During operation of the system

This type of contamination can be divided into:

- internal contamination
- external contamination

Internal contamination means all the contamination originating from inside the system itself, e.g. due to wear at control lands, cylinders and pistons, particles of rubber from hoses and seals, particles of paint and the products of oxidation of the fluid.

External contamination is a result of dirt penetrating badly sealed tanks, unsuitable air filters and damaged seals on pistons and piston rods.

The task of the filters installed in the system is to filter out the internal and external contamination and so interrupt the chain reaction that produces contamination in the first place.

Tests have shown that, when using very fine filters in well-kept units with good external sealing, it is possible to achieve a much higher total number of service hours with less downtime.

2.5 The task of the hydraulic filter

The filters installed in hydraulic systems are of the same critical importance to its overall function as every other component with which it is associated. Correctly sized and installed filters ensure that the costs of maintenance, repair and downtime are kept to a minimum. Their use improves the efficiency of the hydraulic system and hence of the total installation of which it forms part. The filters also have a great effect on the operator's opinion of the availability and reliability of the system.

A correctly sized filter must perform the following tasks:

- Remove solid particle contamination from the hydraulic fluid
- Prevent functional disturbances due to solid particle contamination
- Prevent variations in switching times due to damaged control lands
- Reduce downtime between maintenance shutdowns
- Increase component life
- Permit preventive maintenance
- Prevent aging of the fluid due to chemical processes (resulting from solid particle contamination)
- Maintain the lubricity of the fluid
- Extend the life of the fluid
- Maintain high reliability between maintenance shutdowns
- Ensure long maintenance intervals for the filter
- Ensure continuous filtering of solid particle during service
- Have a high dirt holding capacity
- Ensure reliability and availability of the hydraulic system
- Ensure proper functioning of the filter under changing pressure and flow conditions in the system

3 Requirements for hydraulic filters

3.1 Testing standards

The filters used in hydraulic systems are subjected to a variety of tests on the elements and housings.

A filter element is assessed from criteria laid down in testing standards. These standards can either be applied individually or in combination depending on the requirements.

Testing standards are listed in the Appendix.

3.2 Filter elements

3.2.1 Materials for filter elements

The effectiveness of a filter element is governed by the type of mat employed. Filter mat is sometimes also called a matrix. The types of material used for filter mat allow filter elements to be divided into two broad groups:

- Surface filters
- Depth filters

Advantages and disadvantages of various materials

General

Surface filters and depth filters vary in terms of dirt holding capacity and filtration capacity according to their different construction (*see Diagrams 34 and 35*).

Surface filters

Fabrics in a variety of forms are used as the material in this case *(see Table 19)*.

Due to their construction the filters possess a defined filtration rating referred to cubic particles which are about the same size as or larger than the mesh size of the filter. Under certain circumstances it is possible for long, thin particles such as fibres to pass through these filters.

The free filter area available for filtering is small depending on the filtration rating. ("Free filter area" means the area through which the fluid flows.) With surface filters the free filter area is approximately 30 to 40% of the total filter mat area. With a filtration rating below 25 µm the free filter area is even less.

Elements with a filtration rating of over 40 µm can be well cleaned quite simply. When the filtration rating is less than 40 µm it is advisable to supplement the cleaning process with an ultrasonic bath.

Due to the simple cleaning, low initial pressure drop and high differential pressure stability, especially with braided mesh, this type of filter element is chiefly used as safety filters in hydraulic systems, in lubrication systems and back flushing systems.

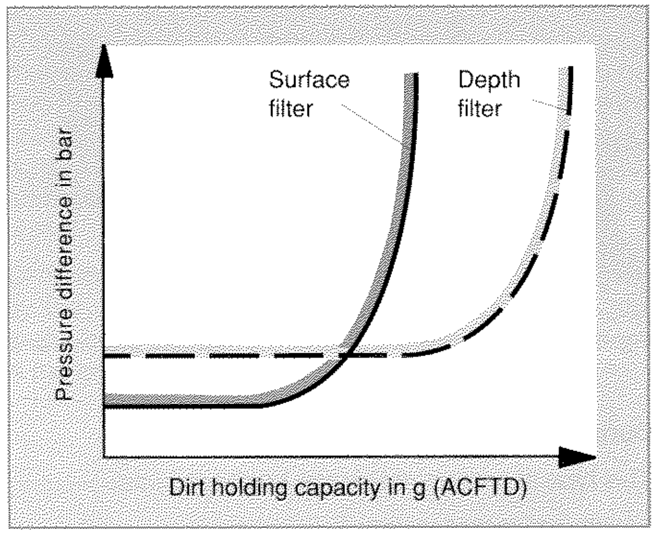


Diagram 34: *Dirt holding capacity of surface filters and depth filters*

Filtration in Hydraulic Systems

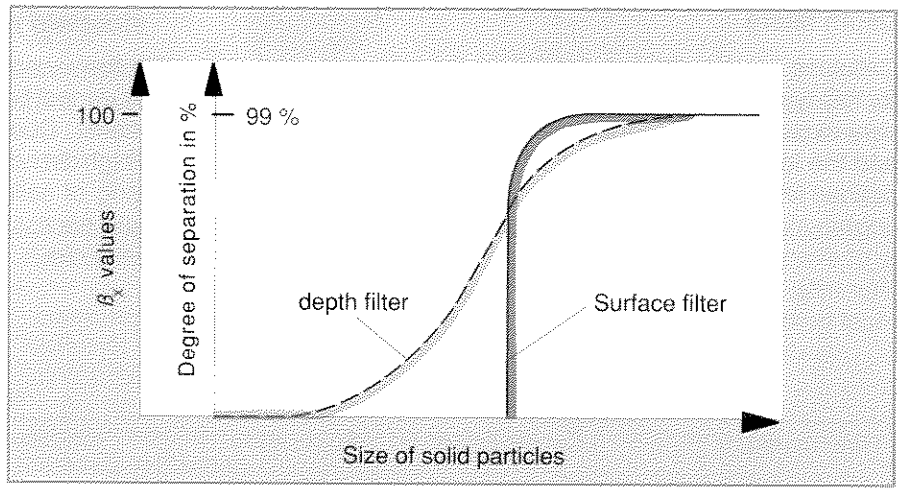


Diagram 35:
Filtration capacity of surface filters and depth filters

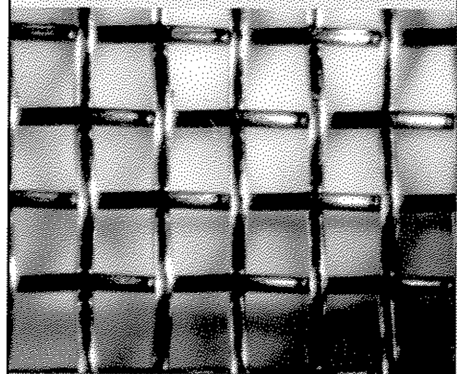
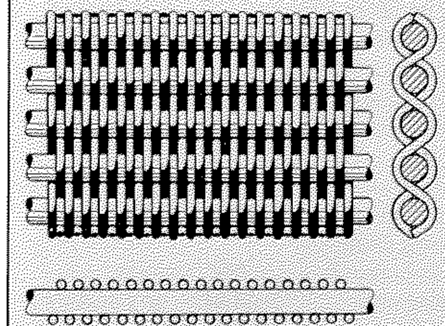
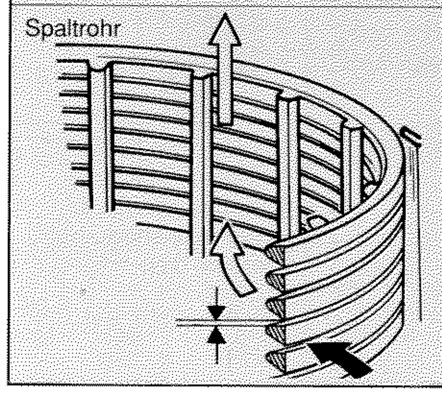
Construction		Application	Advantages	Disadvantages
Wire mesh 	Square mesh, stainless steel, galvanized iron or phosphor bronze	Lubricating oil filters, coarse filters, protection filters, suction filters For filtering water, fire-resistant fluids, at high operating temperatures, special fluids	Elements can be cleaned Low pressure drop	Filtration ratings below 10 µm impossible Multi-pass impossible *(see section 3.2.3)* Small free filter area (aprox. 30 to 40%)
Braided mesh 	Different wire gauges for warp and weft Stainless steel mesh	Coarse filters and protection filters	Elements can be cleaned Low pressure drop Very high pressure differences possible (up to Δp= 420 bar)	Multi-pass impossible *(see section 3.2.3)* Small free filter area
Spaltrohr 	Triangular-section wire wound on to a former at different pitch angles Stainless steel wire	Backflushing filters or coarse filters	Elements can be cleaned during operation Suitable for corrosive media, water, fire-resistant fluids	Filtration ratings below 50 µm impossible Small free filter area Multi-pass impossible *(see section 3.2.3)*

Table 19: *Materials for surface filters*

Depth filters

Cellulose, plastic, glass and metal are the materials used for this type of filter *(see Table 20)*. The pore structure is very closely related to the type of fibrous material used and the length and thickness of the fibres. There is no defined filtration rating from the construction. The resulting labyrinth effect causes particles of dirt of a wide variety of shapes and sizes to be trapped in the depth of the filter mat. It is possible to draw a "filtration profile" which must be determined by experiment.

With the exception of metal non-woven elements, depth filters cannot be cleaned and are regarded as disposable elements. Due to their excellent filtration performance and high dirt holding capacity they are chiefly used for the filtering of solid particles under 20 μm in size. This is usually necessary with systems that are particularly sensitive to dirt.

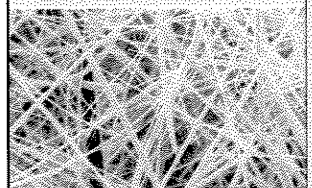
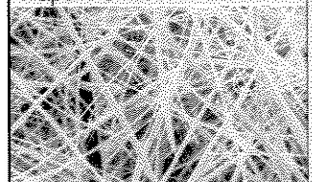
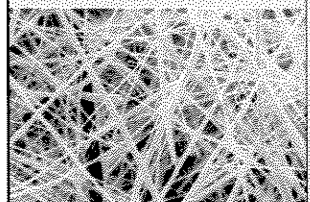
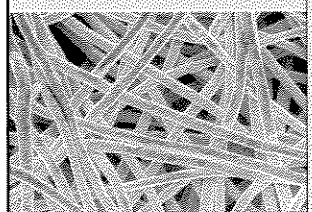
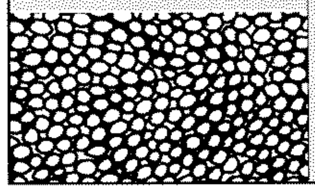
Filter material	Construction	Application	Advantages	Disadvantages
Paper non-woven 	Organic fibres, random layering with binding agent	Suction filters Return line filters Fine filters Disposable elements	Low cost Low pressure drop	Multi-pass restricted Medium dirt holding capacity Low pressure drop strength
Phenolic resin-impregnated Paper non-woven 	Organic fibres, random layering, impregnated with phenolic resin	Fuel filters Engine oil filters Fine filters Disposable elements	Low cost Simple element construction Large filter area	Multi-pass restricted, poor dirt holding capacity, not suitable for all fluids, low pressure drop strength
Glass fibre non-woven 	Glass fibre, random layering with binding agent	Very fine filters for precision components Disposable elements	Fine filtering possible with glass fibre, good dirt holding capacity, absorption of particles over wide pressure drop range, good chemical resistance, suitable for all hydraulic systems	High pressure drop Cannot be cleaned Low flow resistance
Metal non-woven 	Stainless steel wire random layering, sintered and calendered	Fine and very fine filtering, for high operating temperatures, high pressure drops, all fluids, limited cleaning of elements	Low pressure drop, good dirt holding capacity, Multi-pass possible with high-quality non-wovens, good fatigue properties, high temperature resistance, good compatibility with fluids	Very expensive Limited cleaning, depending on pressure drop and filtration rating
Sintered metal 	Metal granules sintered together. The diameter of the granules determines the filtration rating.	Protection filters	Low manufacturing costs	Only suitable for low flow rates, small free filter area, sensitive to pressure shock, high pressure drop

Table 20: *Materials for depth filters*

3.2.2 Constructive features of filter elements

The constructional features of filter elements are determined by the different conditions under which they are expected to function.

		Application	Advantages	Disadvantages
Pressure range	Low pressure	Low working pressure, filters with by-pass, working filters	Cheap elements	Damaged by severe, rapid pressure shocks
	High pressure	High working pressure, filters without by-pass, protection filters	Universal application	Expensive
Filtermat construction	Single layer	Automotive	Cheap	Poor pressure drop strength, poor filtration capacity
	Multiple layer	Hydraulic systems and lubricating systems	Good filtration capacity, high pressure drop strength	Expensive
	Star pleated	Hydraulics, Lubrication, Fuel	Large filter area in small space	Limited cleaning possible
	Shell strainer	Lubricating oil systems	Easily cleaned	Small filter area
	Basket strainer	Lubricating oil systems	All dirt removed when element is changed.	More complex construction
Flow direction	From inside to outside	For low pressure drops	All dirt removed when element is changed.	More complex construction
	From inside to outside	For high pressure drops	Can be cleaned, depending on material	Filtered dirt not trapped in element.
Bonding of filter mat to end caps.	Adhesive	For mineral oil up to 100 °C	Simple and cheap	Not suitable for high temperatures, not suitable for all fluids
	Soldered	For mineral oil 100 °C, over 100 °C, for corrosive media	For high operating temperatures and corrosive media	Expensive, complex
	Crimped	For mineral oil 100 °C, over 100 °C, for corrosive media	For high operating temperatures and corrosive media	Expensive, complex

Table 21: *Constructional features of filter elements*

3.2.3 Verifying the filtration performance to DIN ISO 4572 (Multi-pass Test)

This test enables the filtration capacity and dirt holding capacity of filter elements to be determined.

It is an internationally standardized method and so allows a direct comparison to be made between elements with the same filtration rating produced by different manufacturers.

In order to make a comparison the conditions of the test must be recorded in the test log. Any modifications to the method of testing, as is common practice nowadays in all countries, must be stated.

Arrangement of the Multi-pass Test rig and the test procedure *(Fig. 85)*

The test rig incorporates two hydraulic circuits.

The test system with tank, test fluid, pump, cooler/heater, flowmeters, filter with test element and electronic particle counter.

The dirt injection system with tank, pump, cooler/heater, injection nozzle and injection fluid. In this tank the injection fluid is contaminated with the test dust (ACFTD).

Before the commencement of the test, both systems are cleaned with ultra-fine filters and the actual test is not begun until the prescribed figure of contamination particle count in the systems has been achieved.

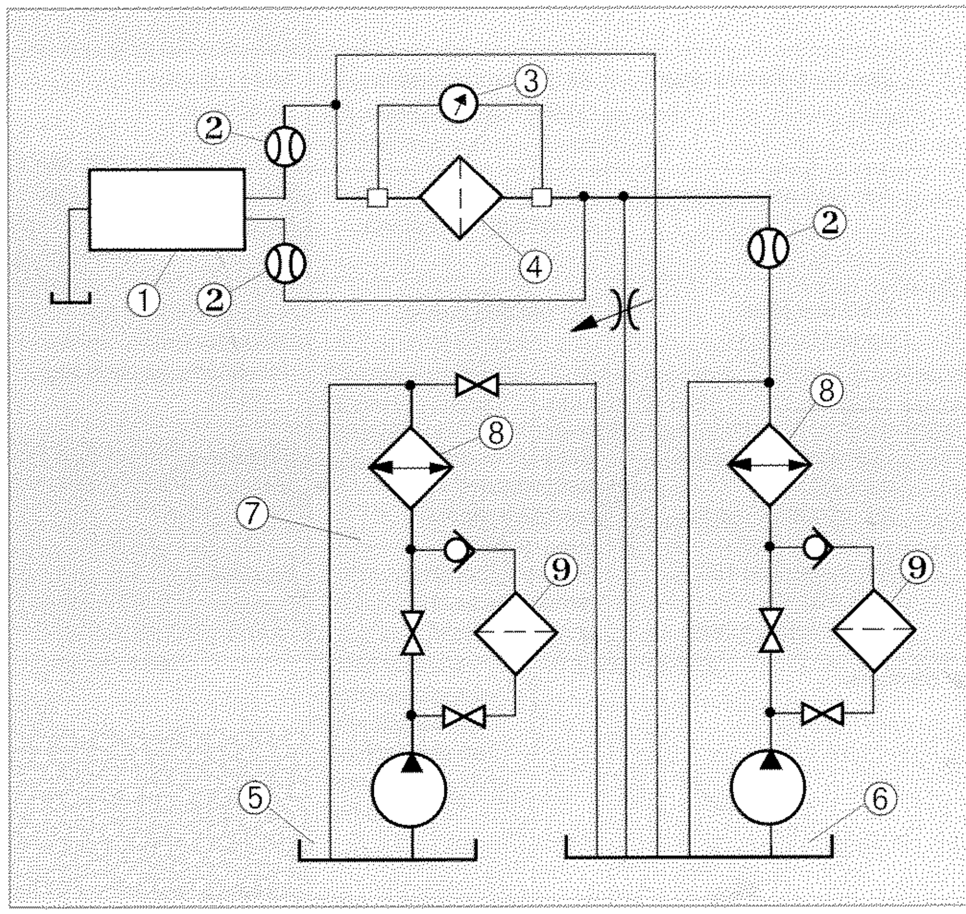
Test sequence

The filter element is subjected to a constant circulating flow of hydraulic fluid into which a small quantity of fluid with a specific contamination is injected.

The now contaminated test fluid is fed to the element and fluid samples are taken upstream and downstream of the test filter and analyzed in the electronic particle counter. The pressure drop across the element caused by the contamination is also measured. The retention rate for filtration rating is defined by the degree of separation β_x, in which X denotes the particle size.

Any contamination not retained by the test filter element remains in the system and so simulates real operating conditions.

The β_x value always refers to particles either equal to or larger than the particle size X under consideration. A change in the pressure drop across the filter element also changes the β_x value.

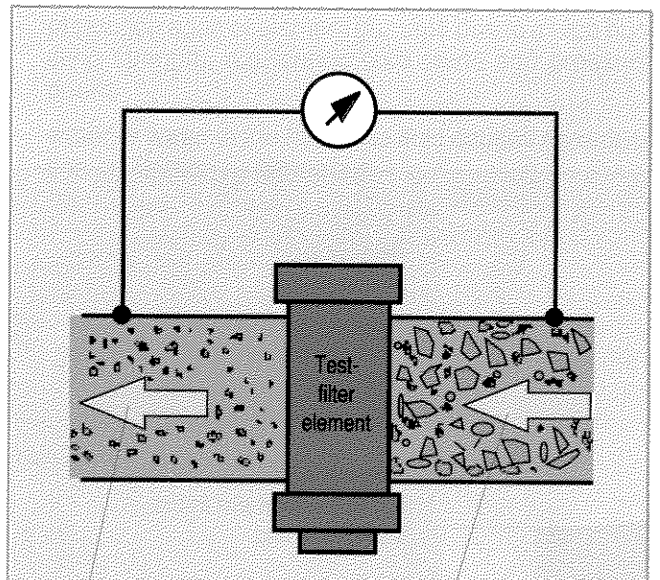


1 Electronic particle counter
2 Flow meter
3 Differential pressure
4 Test filter
5 Tank of injection fluid
6 Tank of test fluid
7 Dirt injection system
8 Heat exchanger
9 Cleaning filter

Fig. 85: *Simplified circuit diagram of the Multi-pass Test rig*

Determining the degree of separation ($\beta_{x\ value}$) *(Fig. 86)*

The number of dirt particles larger than a specific particle size X counted upstream of the filter element is divided by the number of dirt particles counted downstream of the filter element (same particle size X, same pressure drop, at the same point in time). The resulting dimensionless number represents the degree of separation β_x



$n_{Z\ outlet}$ 1000 particles $\geq 10\ \mu m/100ml$ = 0,08 mg/l (ACFTD)

$n_{Z\ inlet}$ 100 000 particles $\geq 10\ \mu m/100ml$ = 10 mg/l (ACFTD)

$$\beta_X = \frac{n_{inlet} \geq X\ \mu m}{n_{outlet} \geq X\ \mu m}$$

Numerical example: $\beta_{10} = \frac{100\,000}{1000} = 100$

Particle size in µm

β_X =	2	50 % filter efficiency average pore size = minimum particle capture size
β_X =	20	95 % filter efficiency (nominal retention rate)
β_X =	75	98,6 % filter efficiency
β_X =	100	99 % filter efficiency (absolute retention rate)

Fig. 86: *Determining the degree of separation ($\beta_{x\ value}$)*

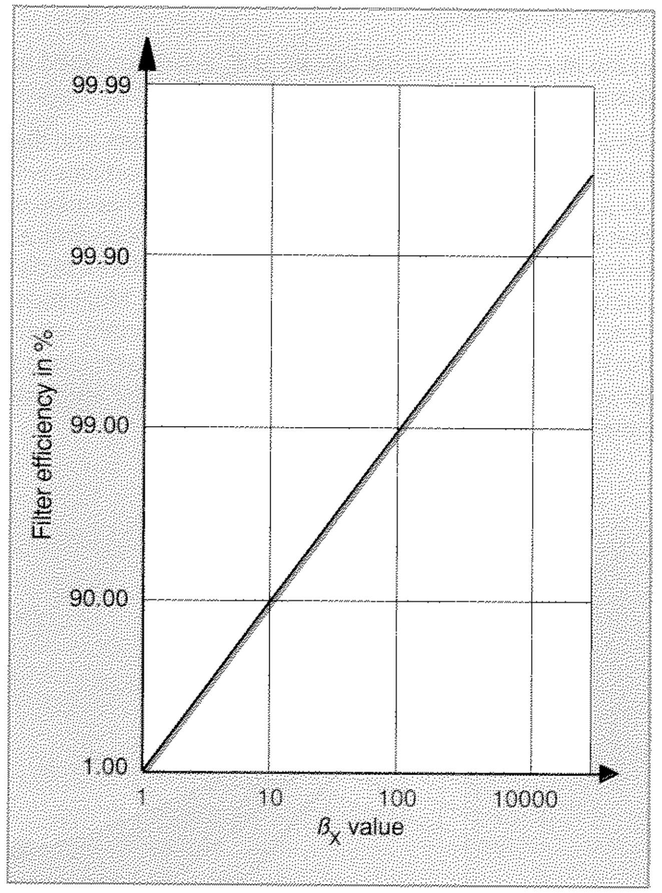


Diagram 36: *Degree of separation(βx value) versus filter efficiency in %*

Definition of filtration rating

Earlier data on filtration rating was based on a variety of in-house tests performed by different filter manufacturers. Only with the introduction of the degree of separation β_x, taking into account the resulting pressure drop, has it become possible to compare filtration rating data from different manufacturers.

Nominal filtration rating

There are no usable values of degree of separation laid down for this. For the user it means that only part of the dirt is actually filtered out which could be filtered out with an optimum filter.

Definition: $\beta_X \leq 20$
This corresponds to a filter efficiency of 95%.

Absolute filtration rating

Above a β_X value of ≤100 or a filter efficiency of 99%, the filtration rating is called the absolute retention rate *(see Diagram 36)*.

Notes on β_x values

In the Multi-pass Test the values of β_x are determined at a constant dirt concentration.

On account of the labyrinth effect of depth filters and the resulting porous structure a certain range of particles will be able to pass through the filter element. This means that the *ßx values* change with different dirt concentrations, different kinds of dirt and different structures of dirt compared with the "ideal dirt" used for the Multi-pass Test. This circumstance is particularly important when the hydraulic fluids used in practice are to be employed for verifying the filtration performance of a filter element *(see Diagram 37)*.

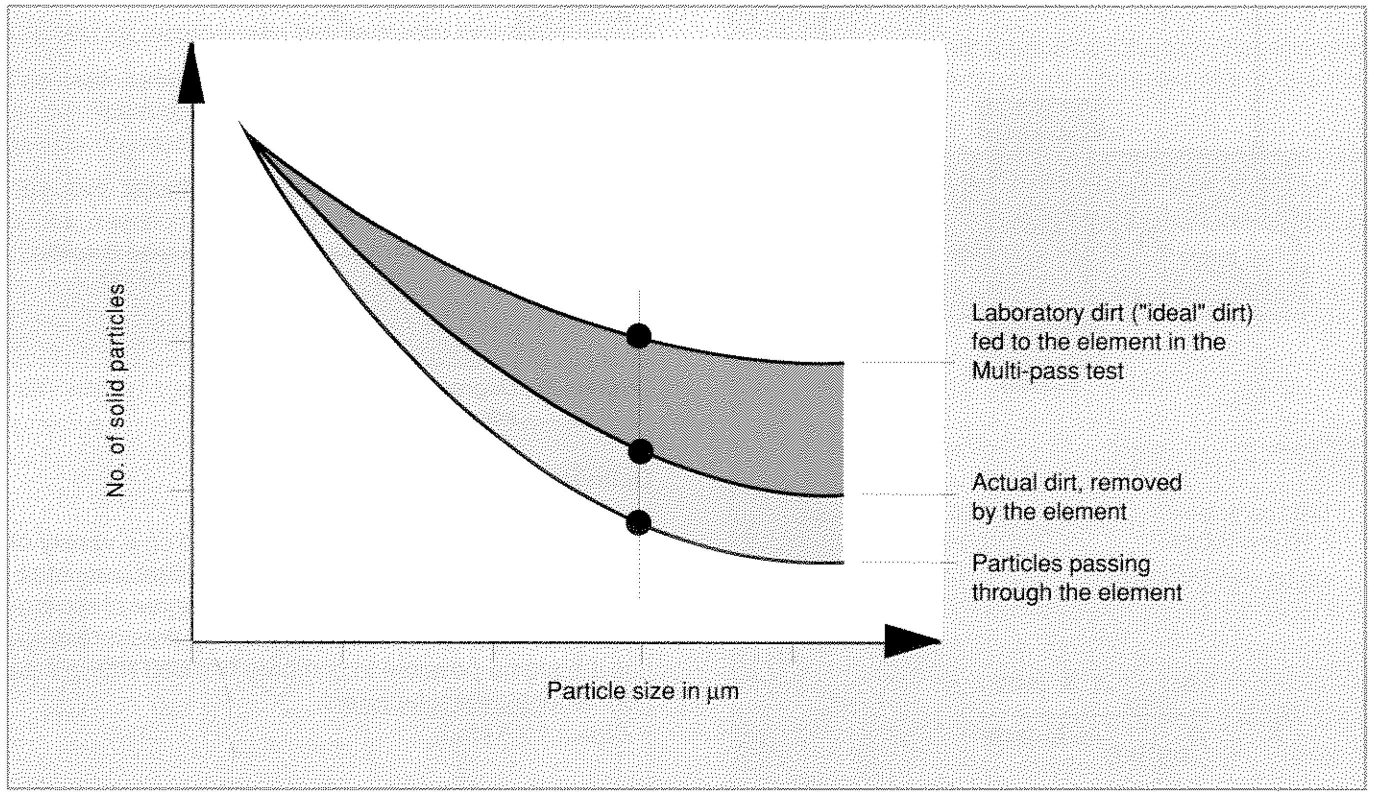


Diagram 37: *Variation in β_x values with laboratory dirt and real dirt*

3.2.4 Properties of filter elements of multi-layer mat construction

The experience that has been accumulated from actual practice and test rig experiments has led to the development of filter elements of multi-layer mat construction called Betamicron® *(Fig. 87)*.

Investigations have also shown that only with this mat construction is it possible to maintain the required levels of cleanliness.

The flow through the filter elements must always be from the outside to the inside.

So that as much filter area can be packed into a small volume, the filter mat should be pleated or corrugated to a star form. The actual construction of the filter mat depends on the permitted value of element pressure drop.

High-quality adhesives are used to attach the filter mat to the end caps of the element and to join the mat ends. The strength of these adhesives is temperature-sensitive and decreases sharply at high temperatures.

Betamicron® multi-layer elements possess a number of key features:

- a precise pore size
- excellent separation of very fine particles over a wide range of pressure drops, i.e. adherence to defined β_x values *(see Diagram 38)*
- high dirt holding capacity through a large specific capture area
- good chemical resistance
- protection against element damage due to a high bursting strength, e.g. during cold starts and pressure peaks
- water or water in the hydraulic fluid causes no reduction in filtration performance.

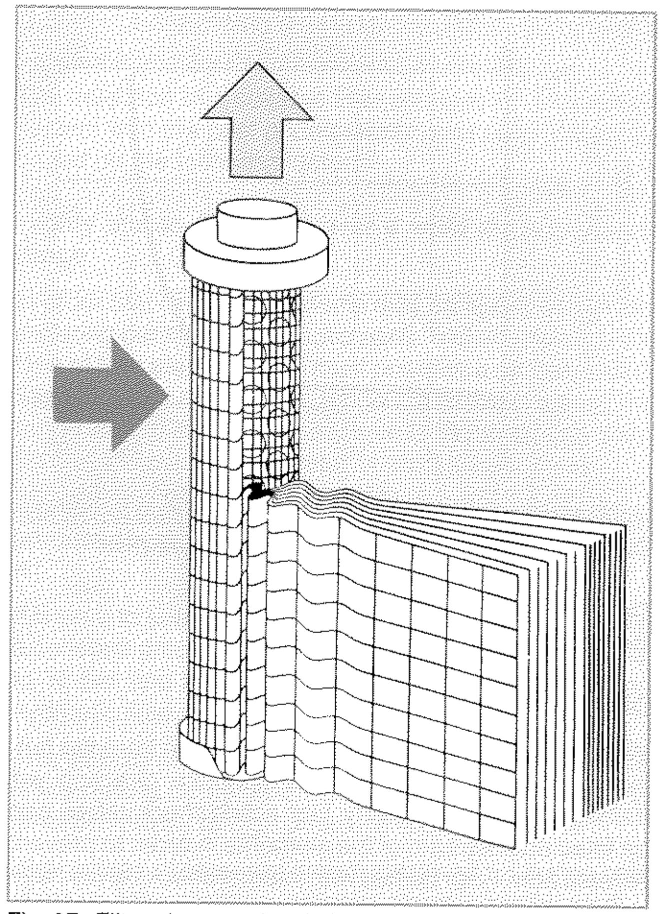


Fig. 87: *Filter element of multi-layer mat construction*

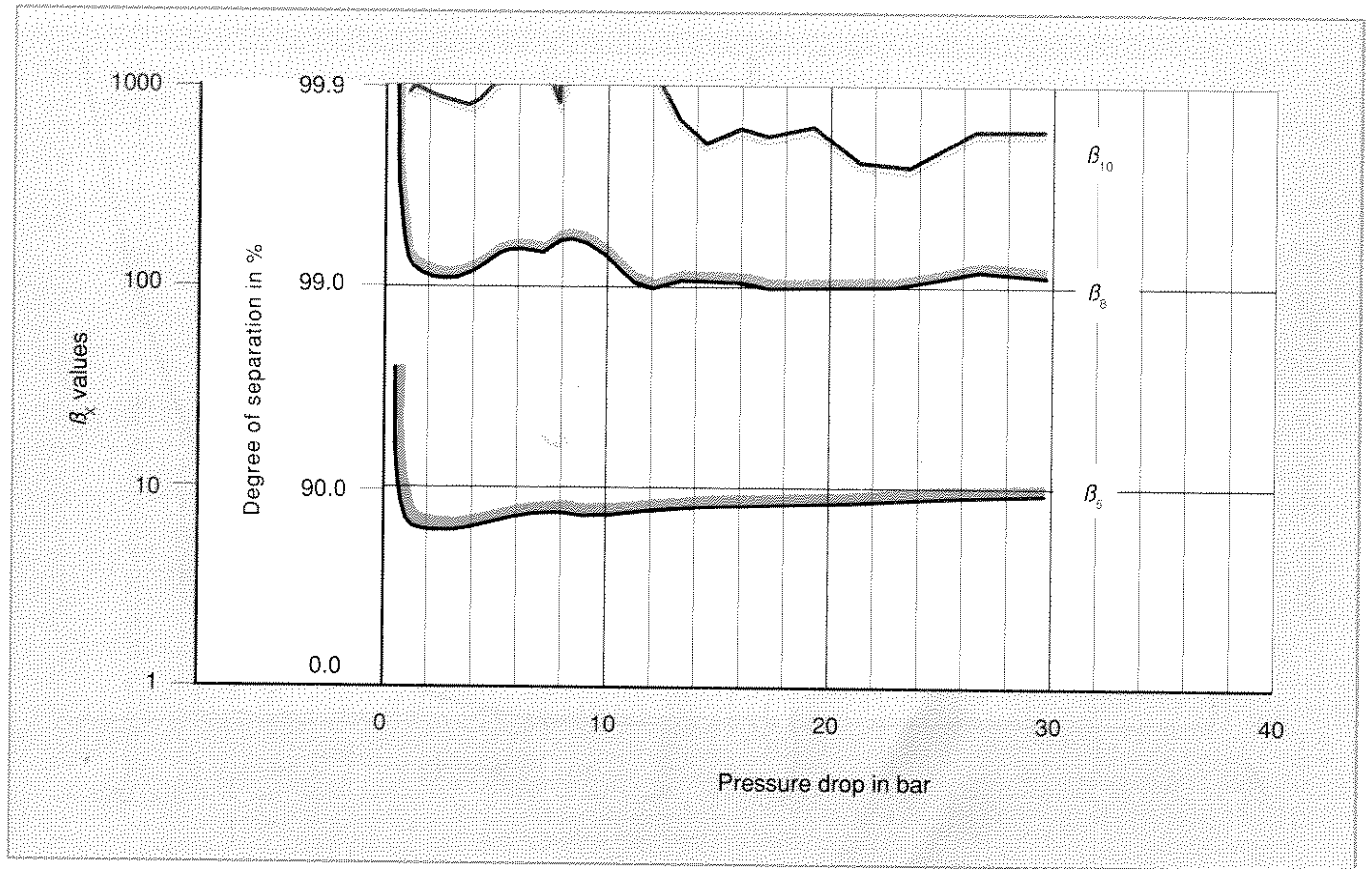


Diagram 38: *β_x values for different pressure drops across the filter element*

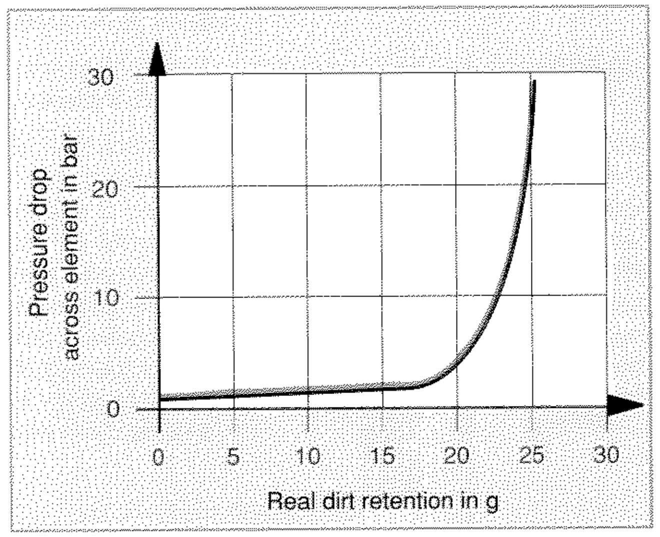


Diagram 39: Curve of dirt retention in a filter element

Constructional features of Betamicron® multi-layer elements

Direction of flow

With these filter elements the flow must be from the outside to the inside; flow in the opposite direction will damage them. If necessary, fast-acting check valves can be fitted downstream of the elements in order to prevent reverse flow. Filter bodies with integral check valves have proved ideal for such applications.

Star-shaped pleating

The filter mat of the elements is pleated into a star shape in order to compress as much filter area as possible into the element in order to achieve a long service life.

Filter element life

This means the number of hours for which a filter element can be used while delivering fluid of the required cleanliness.

The filter must be changed before the maximum permitted pressure drop across the element is reached and the clogging indicator is triggered with the fluid at operating viscosity.

Under certain unfavourable circumstances, such as when fluid temperatures are high or there are frequent and severe variations in the flow, it can be necessary to restrict the maximum service life of the element regardless of the clogging indicator signal. If the service life were unlimited in these circumstances there would be the possibility of fatigue failure in the filter material which would cause a deterioration in the filtering efficiency. In the worst case the clogging indicator would not operate at all.

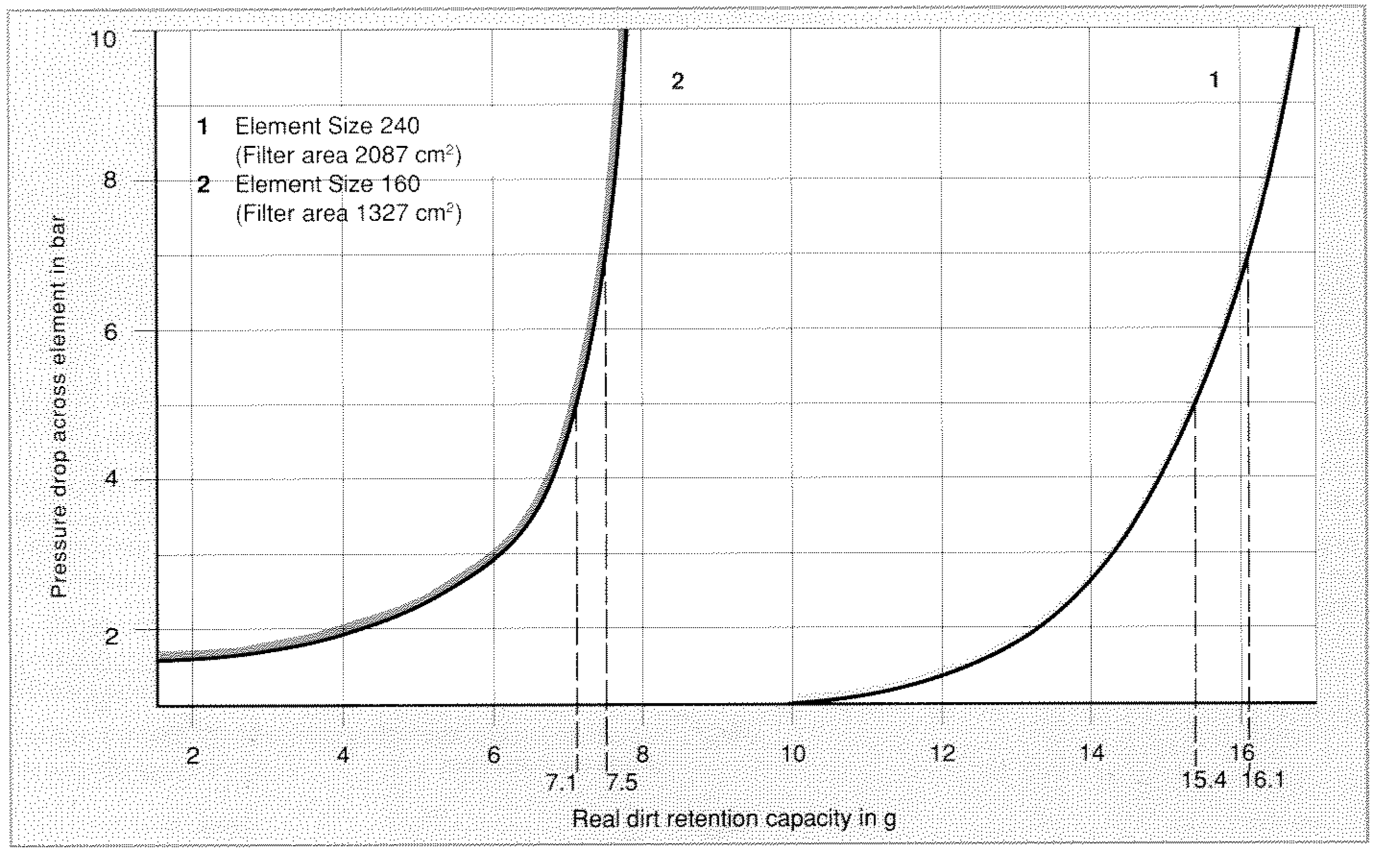


Diagram 40: Dirt retention in different sizes of element with the same volumetric flow 120 L/min

Filtration in Hydraulic Systems

If there is no clogging indicator in the system, filter changing will have to be organized by time schedule, including adequate reserves of service life to ensure that the filtering is always satisfactory.

It is impossible to calculate filter element life theoretically in advance during the project design stage of a system.

In order to provide as large a useful range of pressure drop as possible for the dirt retention of a filter element, and therefore its life, it is advisable when determining its size to begin with the smallest possible pressure drop when the element is clean *(see Diagrams 39 and 40)*. The graphs show the pressure drop across the element with increasing clogging and service life. It is obvious that the low initial pressure drop of a larger filter element provides more real dirt retention capacity than a smaller filter element with a higher initial pressure drop. In both cases the by-pass valve, clogging indicator or element pressure-drop strength set the upper limit for element loading.

3.3 Selection criteria for filter elements

The selection of a suitable filter element for a hydraulic system with the best price/performance ratio should be based on the following factors:

Highly stable $ß_x$ values over a wide pressure drop range

In order that hydraulic systems may be operated without suffering damage due to solid particle contamination the type of filter element used must possess a constant filtration efficiency over a wide range of pressure drops. The range should extend to a multiple of the response pressure of the clogging indicator or the by-pass valve.

A graph of $ß_{10}$ values for filter elements of the same rating produced by different manufacturers is shown in *Diagram 41*.

It is clear to see how only filter elements 1 and 3 maintain a constant efficiency over the range of pressure drop up to 10 bar and therefore are suitable for the filtration of hydraulic fluid.

This stability of $ß_x$ value is of most importance to hydraulic filters having no by-pass valve and therefore having to function reliably at high pressure drops.

High values of pressure drop typically occur during cold starts or if the clogging indicator alarm is not heeded.

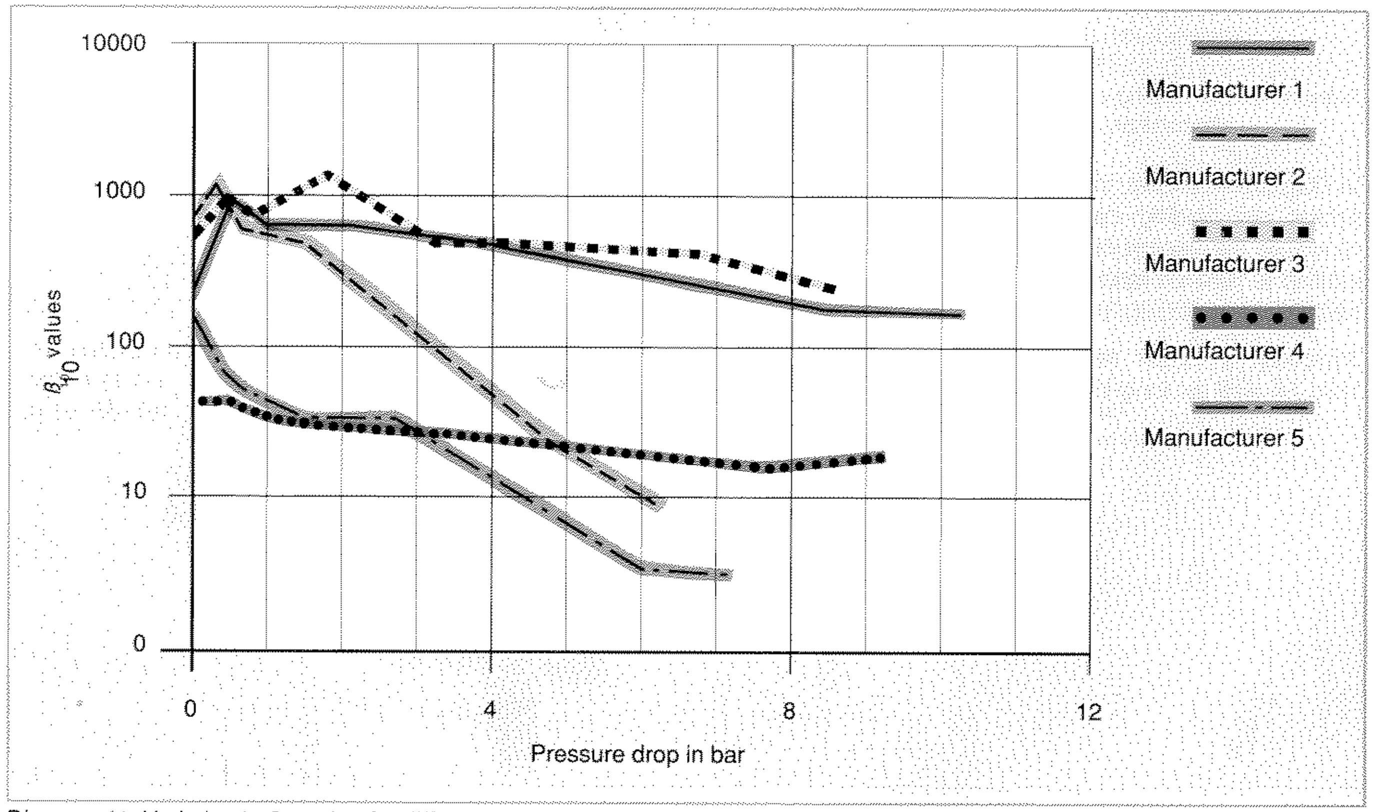


Diagram 41: Variation in $ß_{10}$ value for different makes of filter of comparable sizes and identical performance data

Dirt retention of filter elements

Another important criterion in assessing the price/performance ratio of filter elements is the dirt retention or dirt holding capacity.

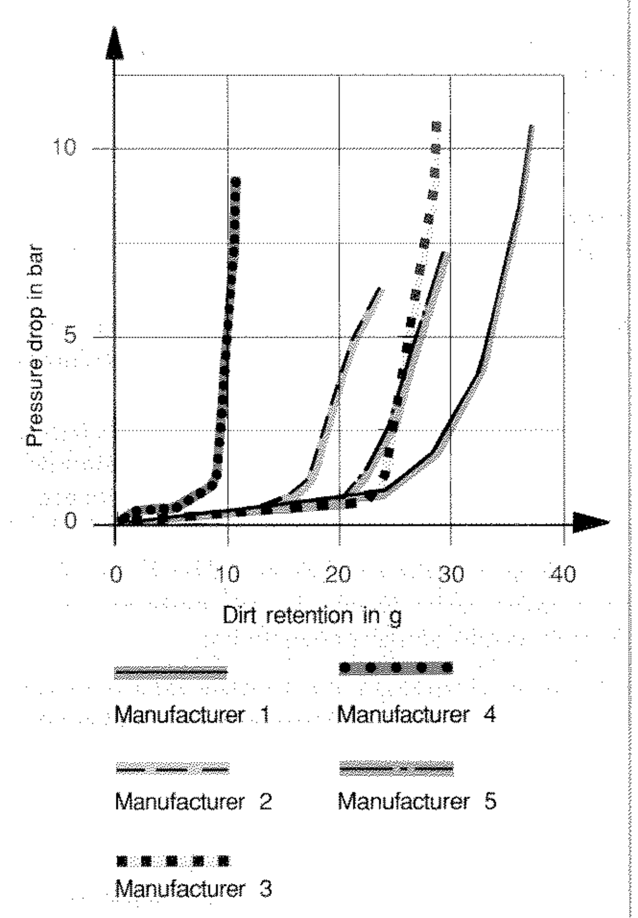


Diagram 42: Dirt holding capacity of different makes of filter of comparable sizes

As *Diagram 42* shows, the filter element of Make 1 has the highest dirt holding capacity. This factor, which also influences the service life, is a further important factor in addition to filtration rating and price in the overall assessment of the suitability of filter elements.

Obviously, a longer service life means lower service costs as well as longer maintenance intervals.

Specific dirt retention of filter elements

Investigation of the specific dirt retention provides an even clearer assessment of the price/performance ratio of filter elements. The figure is obtained by dividing the total dirt retention of a filter element at a certain pressure drop by the effective filtering area of the element, which gives a figure of dirt retention per cm² filter area *(see Diagram 43)*.

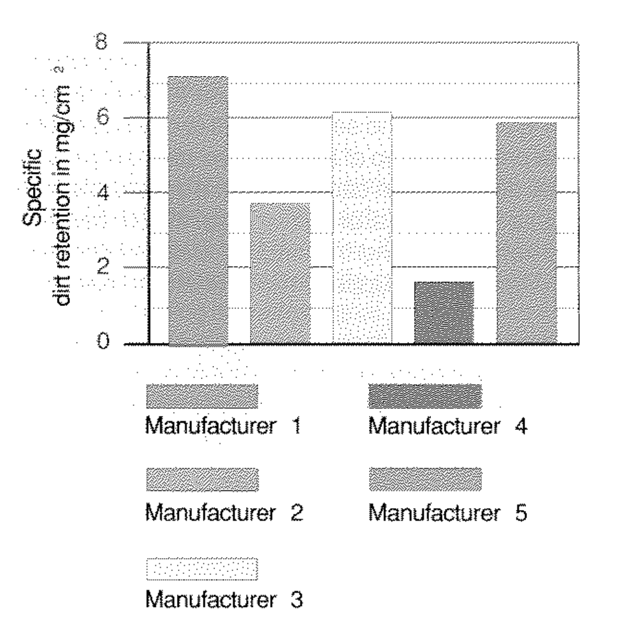


Diagram 43: Specific dirt retention of different makes of filter at a pressure drop of 3.5 bar

3.4 Filter housings

3.4.1 Requirements

Filter housings must satisfy the following requirements:

Low pressure drop across the housing

A low pressure drop across the housing must be achieved by means of a good flow shape inside the housing and primarily around the inlet and outlet ports.

Durable housing construction

Filter housings must be designed so that they have a long operating life at the given value of operating pressure. This means that they must successfully withstand a pulsation test.

Bursting pressure of the housing

In order to verify the maximum operating pressure for filter housings several licensing authorities specify that the burst pressure of the housing must be tested. The burst pressure is the pressure at which the housing ruptures.

Housing material

The materials used for the housings and seals must be suitable for the hydraulic fluid to be filtered.

3.4.2 Types of filter housing

The different types and designs of housings for pressure-line and return line filters are listed in *Table 22*.

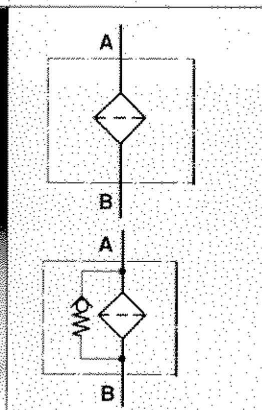
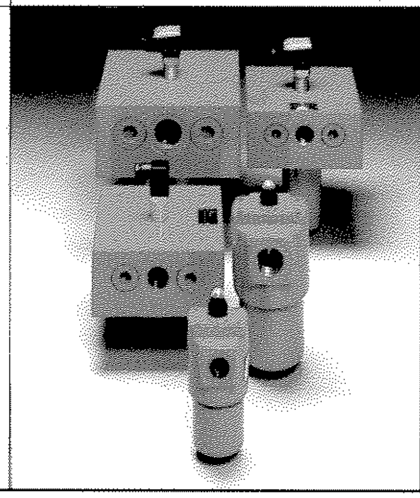
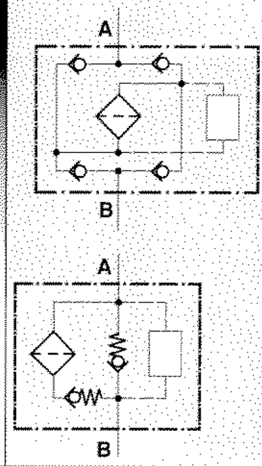
Type Pressure rating		Symbol	Application	Remarks
Low pressure, up to 100 bar Medium pressure, up to 210 bar High pressure, up to 420 bar			Pressure lines Control lines Safety filters	
For reversible fluid flow high pressure up to 420 bar			Safety filters for cylinders, proportional- or servo valves	

Table 22 *(Part 1)*

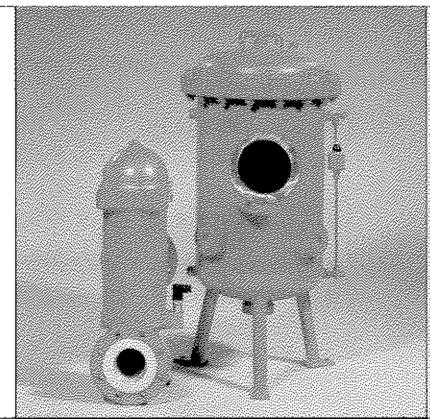
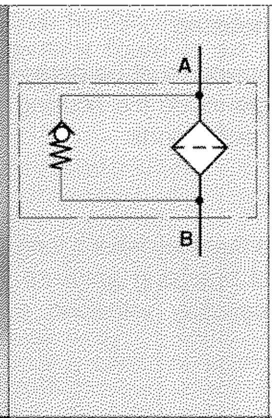
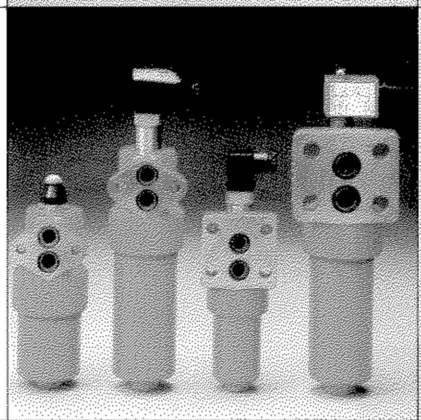
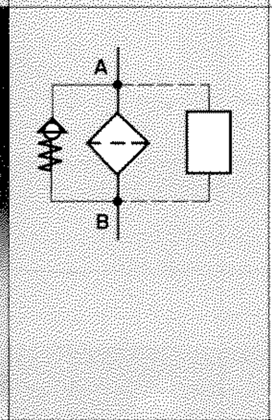
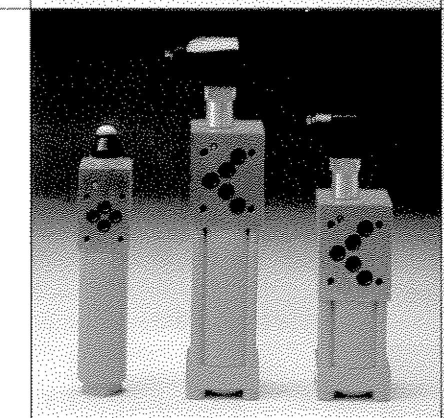
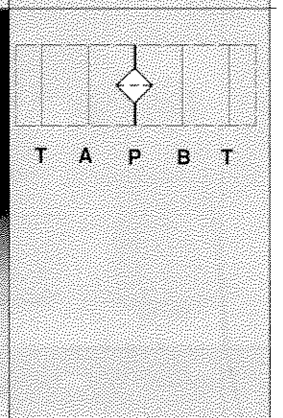
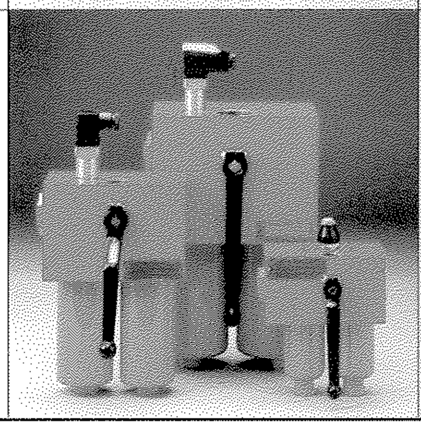
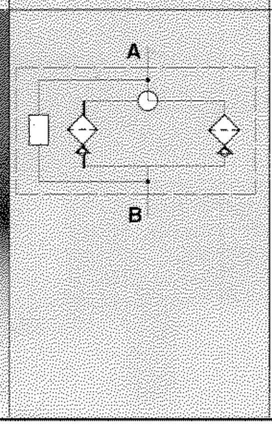
Type Pressure rating		Symbol	Application	Remarks
Low pressure, up to 25 bar			For large quantities of fluid	
Flange mounting, up to 315 bar			Manifold mounting	No filter piping required.
Sandwich plate model, up to 315 bar			Safety filters for precision valves, for vertical and horizontal stacking systems.	Can be fitted directly under the valve.
Pressure-line filter, duplex, up to 315 bar			For systems which cannot be shut down for element changing. Turbine control lines.	

Table 22 *(Part 2)*

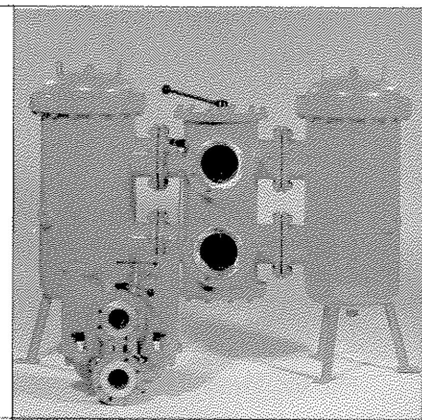
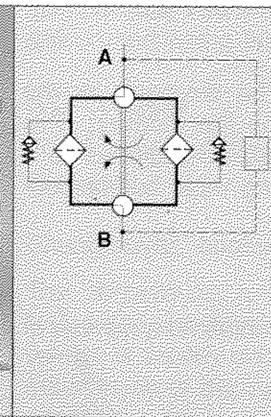
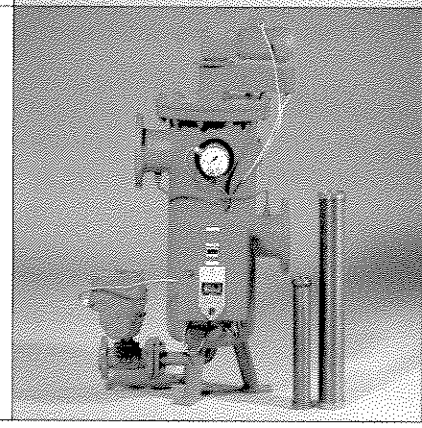
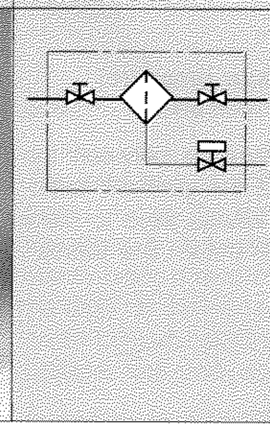
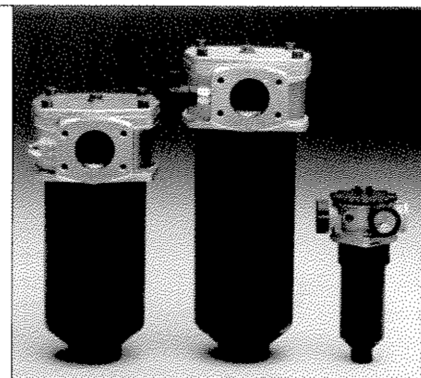
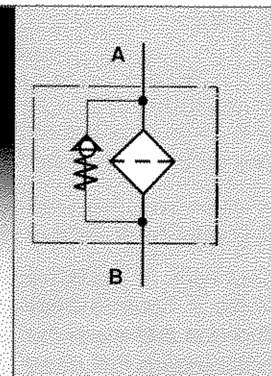
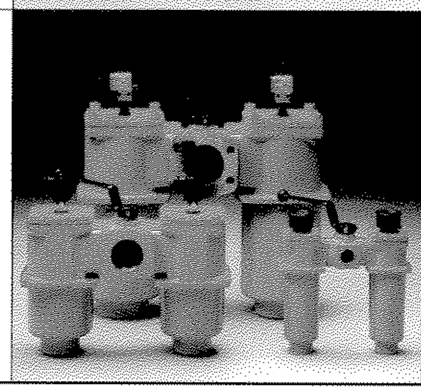
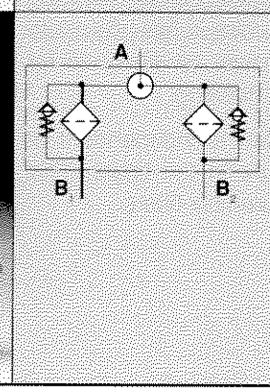
Type Pressure rating		Symbol	Application	Remarks
Pressure line filters				
Low pressure, duplex, up to 25 bar			Systems to API, oil supply systems. For systems which cannot be shut down for element changing.	
Automatic, up to 16 bar			Filtering of machining oils. For severe contamination.	No filtering below 20 µm
Return line filter				
Single, up to 25 bar			Mounted on the tank.	
Duplex, up to 25 bar			Mounted on the tank. For systems which cannot be shut down for element changing.	

Table 22 *(Part 3)*

3.5 Clogging indicators

Basically, hydraulic filters should always be fitted with a clogging indicator to monitor the state of clogging of the filter element.

3.5.1 Requirements

The body of the indicator must be designed for the maximum operating pressure of the filter housing. This means that the indicators must also be subjected to a pulsation test. The response setting must be reproducible.

3.5.2 Key features

Clogging indicators differ from each other in a number of key features:

Type of indication

Back pressure indicators (absolute pressure) *(Fig. 88)*

These indicators measure the difference between the pressure in the filter housing and the ambient atmospheric pressure. They are nearly always fitted to filters which discharge directly to the tank (return line filters).

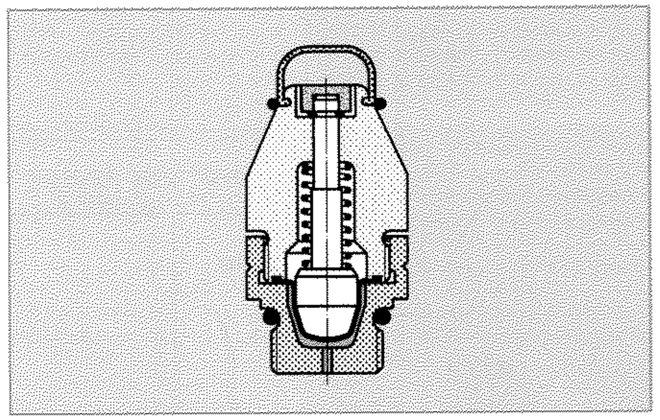


Fig. 88: *Visual pressure head indicator for return line filters*

Differential pressure indicators *(Fig. 89)*

These indicators measure the difference between the pressure on the dirty side and the pressure on the clean side. The ambient atmospheric pressure is not taken into account. The body of the indicator must be designed for the operating pressure of the filter housing.

The value of pressure drop indicated is independent of the instantaneous operating pressure in the filters. This type of indicator is used for pressure-line filters.

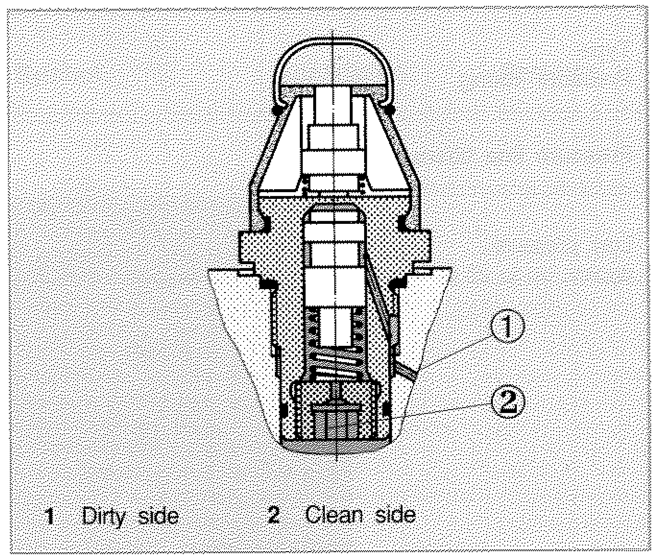


Fig. 89: *Visual Differential pressure indicator*

Processing the indicated signal

Visual

In this case the set pressure is indicated by means of a pressure gauge or a red pin which emerges from the indicator.

Electrical

Electric indicators are used when the signal is to be processed by machine control systems or transmitted to a control room. Such indicators can also be fitted in inaccessible places with the signal indicating the need to change the filter brought electrically to a convenient point *(Fig. 90)*.

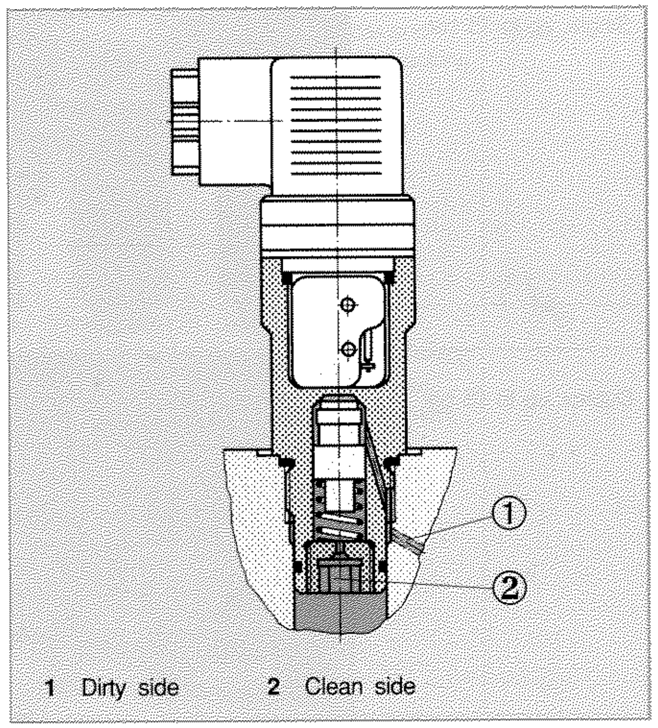


Fig. 90: *Electric pressure drop indicator*

Visual/electric

Electric indicators also have an electric light source to give a local signal for the operator or maintenance personnel in addition to the main electrical signal.

Electronic

There are electronic clogging indicators available for special applications. They are mainly employed for dynamic operating conditions. The electronic units suppress the indicator function up to an operating temperature of, say, 32 °C. Pressure peaks of up to 9 seconds are also suppressed so that they cannot trigger the indicator function. Electronic indicators are suitable for preventive maintenance because they indicate the instantaneous pressure drop across the element *(Fig. 91)*.

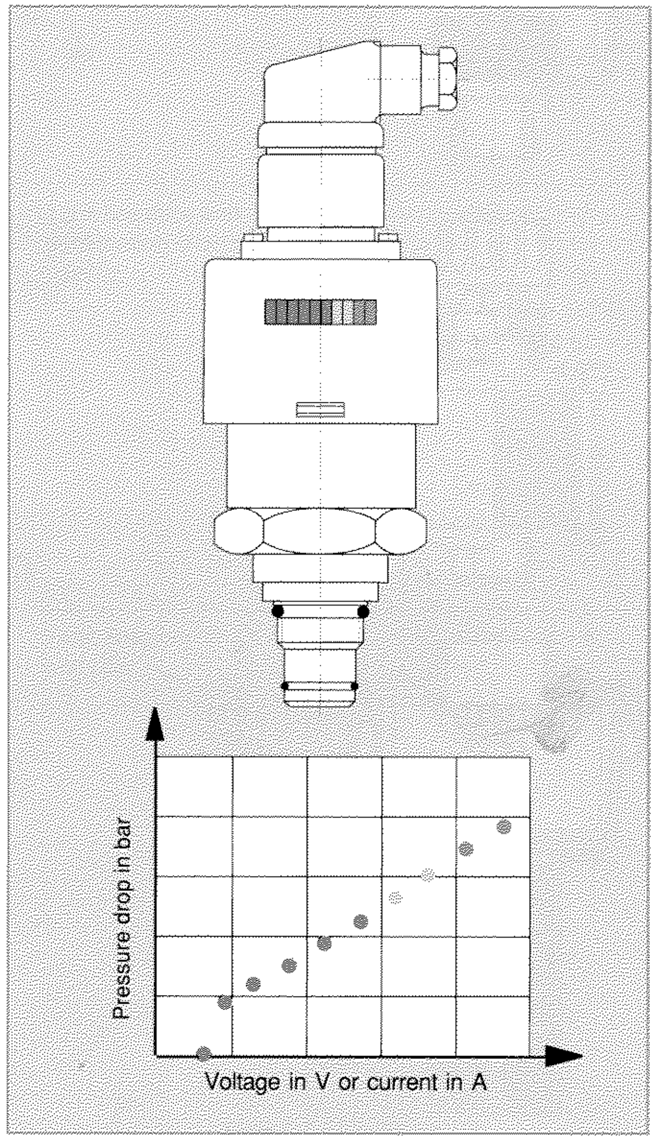


Fig. 91: *Electronic pressure drop indicator*

Contacts for electric clogging indicators

N/C contacts

With this type the circuit is broken when the clogging indicator is triggered. This contact arrangement is preferable because interference with the system is made more difficult and wiring faults can be detected immediately.

N/O contacts

In this case the circuit is closed when the contacts are operated.

Change-over contacts

The contacts can function in either the N/C mode or the N/O mode according to the terminal connections. This version is usually chosen by filter manufacturers so that either mode of display switching can be provided for the operator.

3.6 Breathers

The level of fluid in the hydraulic tank of a system fluctuates due to the supply and return of fluid and due to variations in temperature. This means that air is constantly being drawn into the tank and expelled from it. Depending on the surroundings this "breathing" can cause badly contaminated air to be drawn into the tank and particles find their way into the fluid.

A breather must be fitted to the tank in order to prevent the ingress of this contamination. The filtration rating of the breather must be matched to the filtration rating of the finest filter installed in the system.

Thus, if a system contains a filter with a filtration rating of 3 µm the breather fitted to the tank must also have a filtration rating of 3 µm for air. This is the specification recommended by the Cetop RP 98 H standard.

Smaller tanks are filled through a filling breather, although this arrangement should be avoided whenever possible. A better arrangement is for the system to be filled through a separate connection on the tank or upstream of the return line filter. The filling should be carried out with a mobile filter unit so that the fluid is of the prescribed quality as it enters the system.

Requirements for breathers

The filter element used in the breather must be renewable and have a large filter area.

The retention rate (or filtration rating) chosen for the element must be matched to the main filter.

The intake apertures must be as high as possible above the tank cover so that no dirt which collects on top of the tank can be sucked in.

Monitoring of the condition of the element by means of a clogging indicator is sensible and the cover fitted should offer adequate protection against splashing water.

Types of breather

Oil-wetted air breathers

These breathers contain oil-wetted knitted fabric for separating the particles of dirt from the flow of air. The filter can be cleaned by washing. The retention rate is over 40 µm and so is no longer adequate for the demands of modern fluids.

Oil-bath air breathers

With these breathers the incoming flow of air passes over a bath of oil and picks up minute droplets of oil in the process which bind the particles of dirt together. These dirt-laden droplets of oil are then trapped in knitted fabric and eventually drip back into the oil bath.

There must be a specific value of air flow velocity present in order for an oil-bath air breather to function properly. This is not the case with tank breathing so such breathers are basically unsuitable for hydraulic systems.

Breathers with oil-bath immersed elements

This type of breather is also known as the "pseudo oil-bath air breather". In fact, although the filter element dips into an oil bath, the bath itself plays no part in the function of the filter. Consequently, the combination of oil bath and paper or foam element brings no improvement in filtration capacity which is determined solely by the retention capacity of the paper or foam element.

The dipping of the filter element into the oil bath occupies some of the free area of the filter which shortens its service life.

As mentioned earlier, the filtration capacity of the element is the governing factor for the cleanliness of the air. The extra hydraulic fluid put into the filter only serves to shorten the service life of the element. They are unsuitable for use in hydraulic systems.

Special designs of breather

With back pressure valve

These breathers are used where the overflow of fluid is to be prevented or the inlet or outlet of air is only to take place at predetermined values of positive or negative pressure in order to improve the suction capacity of the hydraulic pump. The use of back pressure valves is intended to prevent the interchange of air in the tank with the atmosphere or to reduce it to a low value.

With dehydrating device

Hydraulic tanks are sometimes used in extremes of weather and climate with the attendant risk of air-borne water gaining ingress to the tank. Under some circumstances substantial quantities of water can get into the hydraulic system which the fluid cannot emulsify and so cause malfunctions. For this situation there are breathers incorporating a dehydrating chamber filled with silica gel.

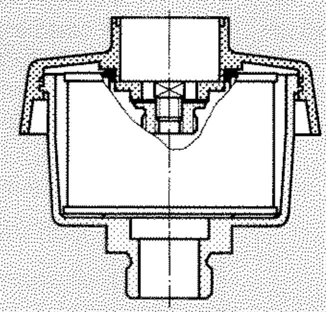
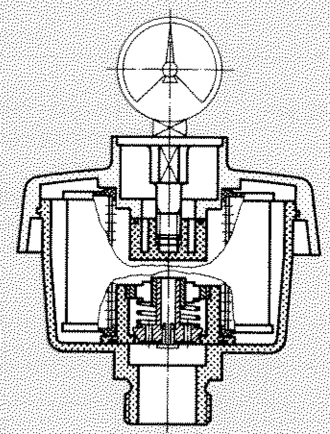
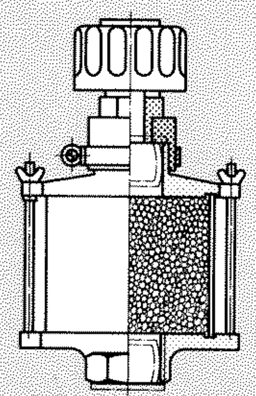
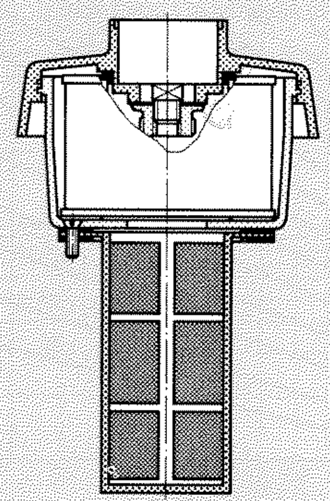
Type	Illustration	Filtration rating	Remarks
Tank breather		3 µm 5 µm 10 µm 20 µm	With replaceable filter element. Design to CETOP RP 98 H possible. Connection for clogging indicator.
Tank breather with check valve		3 µm 5 µm 10 µm 20 µm	With replaceable filter element. Design to CETOP RP 98 H possible Connection for clogging indicator. Fitted with check valve to minimize breathing. To improve pump suction.
Tank breather with dehydrating chamber		3 µm 5 µm 10 µm 20 µm	With replaceable filter element. Design to CETOP RP 98 H possible Connection for clogging indicator The moisture is removed from the incoming air.
Tank breather and filler		3 µm 5 µm 10 µm 20 µm	With replaceable filter element. Design to CETOP RP 98 H possible. Connection for clogging indicator Suitable for tank filling. Check valve fitted if required.

Table 23 (Part 1)

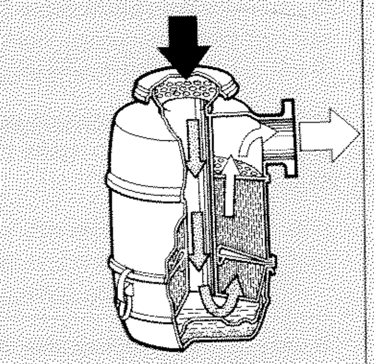
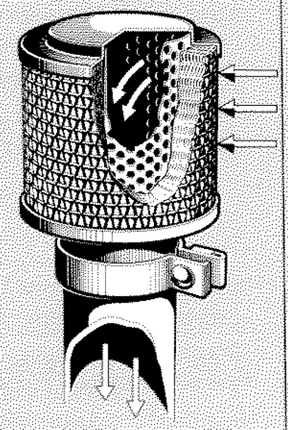
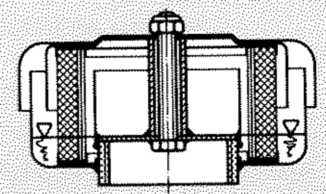
Type	Illustration	Filtration rating	Remarks
Oil-bath air breather		40 µm	For tank breathing, unsuitable for hydraulic systems.
Oil-wetted air breather		40 µm	Poor retention rate. Retention rate very dependent on maintenance. Unsuitable for hydraulic systems.
Breather with oil-bath immersed element		3 µm 5 µm 10 µm 20 µm	No improvement in filter efficiency through oil bath. Reduced service life due to oil bath taking up filter area. Not suitable for hydraulic systems.

Table 23 (Part 2) *(top and centre illustrations by courtesy of Mann und Hummel of Ludwigsburg)*

4 The hydraulic fluid

4.1 General

Hydraulic power systems can be operated with fluids produced from a number of different base fluids.

The different classifications are as follows:

- Mineral oil-based hydraulic fluid
- Vegetable oil-based hydraulic fluid
- Fully-synthetic hydraulic fluid
- Fire-resistant hydraulic fluid
- Pure water.

From the filtration point of view a hydraulic fluid should satisfy the following requirements:

- low solid particle contamination in as-delivered state
- good filtration
- good viscosity/temperature characteristic, i.e. flat
- neutral behaviour towards materials.

Viscosity characteristic

In the design and operation of hydraulic filters the viscosity of the fluid is an important factor so that the whole installation can be operated trouble-free.

The method of determining the viscosity index is dealt with in DIN ISO 2909.

The viscosity/temperature characteristics for fluid lubricants needed for designing hydraulic filters will be found in DIN 51 519 and are reproduced in *Diagram 44.*

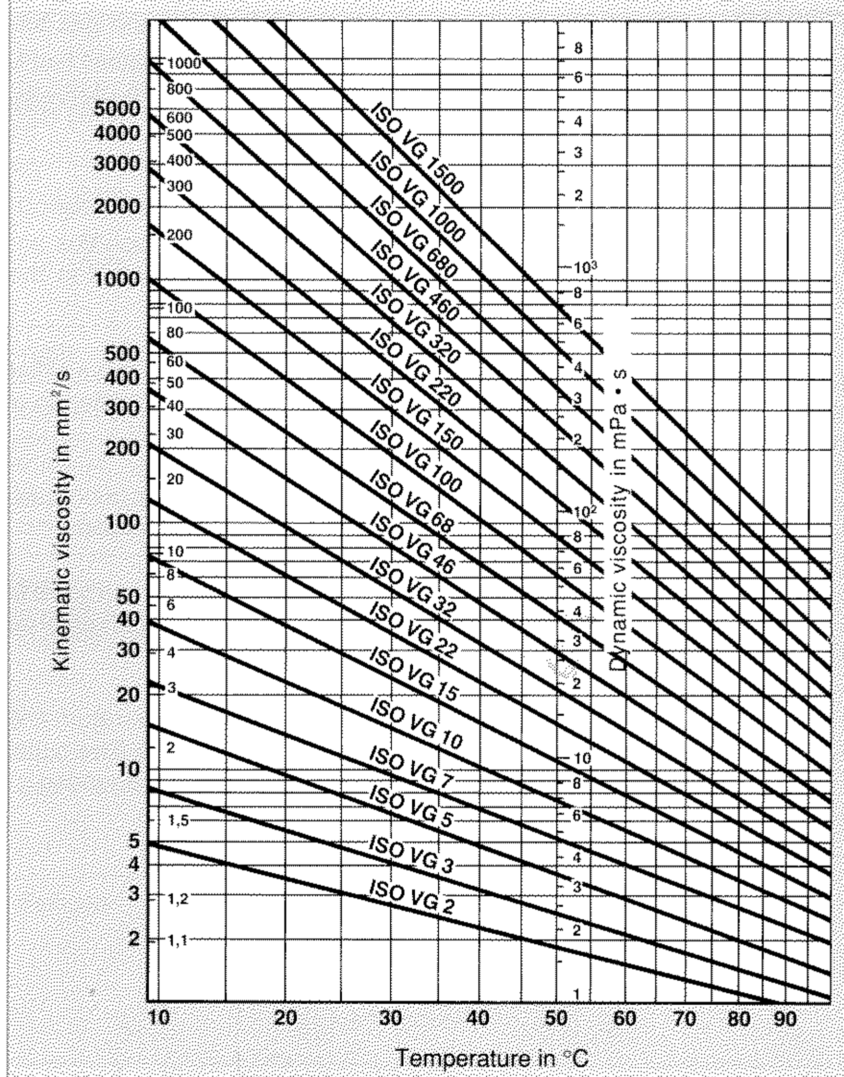


Diagram 44:
Viscosity/temperature characteristic for mineral oils to DIN 51 519

Filterability of hydraulic fluids

Additives in hydraulic fluids, or the adding of fluids other than that originally used in a system, can sometimes cause rapid clogging of filter elements. It results in the pressure drop across the element increasing quikkly and shortening the service life of the element.
The suitability of hydraulic fluids for filtration is ascertained by means of test filter discs. Short through-flow times are desirable.

4.2 Mineral oil-based hydraulic fluids

Most hydraulic power systems use this type of fluid. It is described in detail in *Section 3* of the chapter on *"Hydraulic Fluids"*.

Due to their frequent and widespread use in hydraulic systems the basic version of the hydraulic filter has been designed to filter HL, HLP and HV fluids. This means that when filtering other hydraulic fluids it might be necessary to modify the filter housing, filter element, accessories or seals.

The design of hydraulic filters described in *Section 5* refers to the filtration of HL, HLP and HV fluids.

The filters will have to be designed to different requirements if the properties of the fluid used differ from those of these mineral oils, e.g. dirt settling capacity, filtration, viscosity/temperature characteristic, etc.

For the filtering of HLP-D fluids it is advisable to calculate the size of the filter at approximately 0.2 bar for return line filters and approximately 0.5 bar for pressure-line filters because of their poorer dirt settling capacity in the tank. Also, the filtration rating ascertained from the diagrams must be selected at least one step finer.

4.3 Vegetable oil-based hydraulic fluids

These fluids are bio-degradable and so are being used more frequently in installations that are subject to strict anti-pollution regulations.

The filters are designed in the same way as for the HL, HLP and HV fluids. When operating a system with this type of fluid it is very important to ensure that no mineral oil gets into the fluid otherwise it will be more difficult to filter and the anti-pollution aspect will be compromised.

4.4 Fully-synthetic hydraulic fluids

These fluids are most commonly used in systems where there are special demands on the hydraulic fluid. It is impossible to make any general statement on the use of standard filters so filter makers must be approached as and when necessary. It is possible that a compatibility test to DIN ISO 2943 will be performed in order to check the compatibility of the materials used for the filter housings and elements and make any modifications found necessary.

4.5 Fire-resistant hydraulic fluids

These fluids are used when there is a fire or explosion hazard.

The most common applications are in:

- mining
- die-casting machines
- hydraulic presses for hot working
- governing systems on steam turbines and gas turbines
- various manufacturing processes in the automotive industry, e.g. upholstery
- installations in the chemical industry

The designations of the fluids and their properties are listed in *Table 24* and described in *Section 4* in the chapter on *"Hydraulic Fluids"*.

Modifications must be made to the standard filters so filter makers must be approached for advice on the filtering of these fluids.

Generally speaking, parts made of aluminium, zinc, cadmium and magnesium cannot be used in the filters.

Air carrying traces of these fluids can be very corrosive to steel parts and castings. The formation of a cushion of air in filters must be avoided.

The fitting of check valves downstream of the return line filter or to pressure line filters from which the filter outlet line runs to the hydraulic tank is to be recommended.

Filtration in Hydraulic Systems

All filter housings which come into contact with air bearing traces of these fluids (e.g. return line filters) should be given suitable surface protection.

When determining the size of filter and filtration rating it is most important to consider the poor dirt settling characteristics and the soapy residue of these fluids.

The design of filters for fire-resistant fluids is dealt with in *Section 5.7*.

4.6 Pure water

Pure water is seldom used as the fluid in hydraulic systems due to the disadvantages described in the chapter on *"Hydraulic Fluids"*.

Standard filters cannot be used.

Fluid designation to DIN 51 502 and ISO DIS 6071	**HFA**	**HFB**	**HFC**	**HFD**
Composition	Oil-in-water emulsion or synthetic polymer solution	Water-in-oil emulsionn	Aqueous polyglycol solution	Synthetic anhydrous phosphate esters or chlorinated hydrocarbons
Water content	over 80%	over 40%	over 35%	under 0.1%
Operating temperature	+ 5 °C to + 55 °C	+ 5 °C to 60 °C	- 20 °C to + 60 °C	- 20 °C to + 150 °C
Kinematic viscosity	under 1.6	46 to 100	22 to 68	15 to 100
Density	0.998	0.92 to 1.05	1.04 to 1.09	1.1 to 1.9
pH value	7 to 10	7 to 10	7.5 to 10	7.5 to 10
Materials attacked	Zinc aluminium	Zinc aluminium	Zinc aluminium cadmium and magnesium alloys	
Seal materials	NBR	NBR	NBR EPDM SBR	FPM (Viton) EPDM
Remarks	Susceptible to microbial attack, high mechanical wear due to low viscosity.	Widely used in mining applications in English speaking countries.	Poor dirt separation capacity. Sensitive to mineral oil contamination.	May not be mixed with water. Sensitive to moisture contamination.

Table 24: *Properties of fire-resistant fluids*

4.7 Solid contamination

The contamination of hydraulic fluids by solid particles is dealt with by a number of different classification systems.

There are 5 in all at present:

- SAE 749 D
- ISO DIS 4406
- CETOP RP 70 H
- NAS 1638
- MIL STD 1246 A

Table 25 compares the classification systems with each other. The different classes of contamination define the quantity of particles of a certain size in a 100 ml sample of fluid.

A classification is determined by counting and sizing the contaminating solid particle. It is done either under a microscope or by means of an electronic particle counter. The electronic counter method is more objective than using the microscope. Above a dirt concentration of about 20 mg per litre, or if the fluid is very turbid, the contamination can only ascertained by weight, i.e. by gravimetric analysis. However, with this method the individual dirt particles cannot be classified.

ISO DIS 4406 or Cetop RP 70 H	Particles per ml > 10 µm	ACFTD solids content mg/L	MIL STD 1246 A (1967)	NAS 1638 (1964)	SAE 749 D (1963)
26/23	140000	1000			
25/23	85000		1000		
23/20	14000	100	700		
21/18	4500			12	
20/18	2400		500		
20/17	2300			11	
20/16	1400	10			
19/16	1200			10	
18/15	580			9	6
17/14	280		300	8	5
16/13	140	1		7	4
15/12	70			6	3
14/12	40		200		
14/11	35			5	2
13/10	14	0,1		4	1
12/9	9			3	0
18/8	5			2	
10/8	3		100		
10/7	2,3			1	
10/6	1,4	0,01			
9/6	1,2			0	
8/5	0,6			00	
7/5	0,3		50		
6/3	0,14	0,001			
5/2	0,04		25		

Table 25: *Comparison of contamination classifications*

Using ISO DIS 4406

Diagram 45 shows how the particle size is plotted along the ordinate and the number of particles and classification code along the abscissa.

In the case of ISO DIS 4406 the degree of contamination of the fluid is defined by a two-digit code. One is the number of solid particles over 5 µm in size and the other the number of particles over 15 µm in size in a 100 ml sample of fluid.

In order to determine the degree of contamination to ISO DIS 4406, first count all particles larger than 5 µm in the 100 ml sample and give it a code number. Then count all particles larger than 15 µm and give them another code number *(see example in Diagram 45)*.

These code numbers form the designation of the sample. *Table 26* lists the contamination figures and their coding.

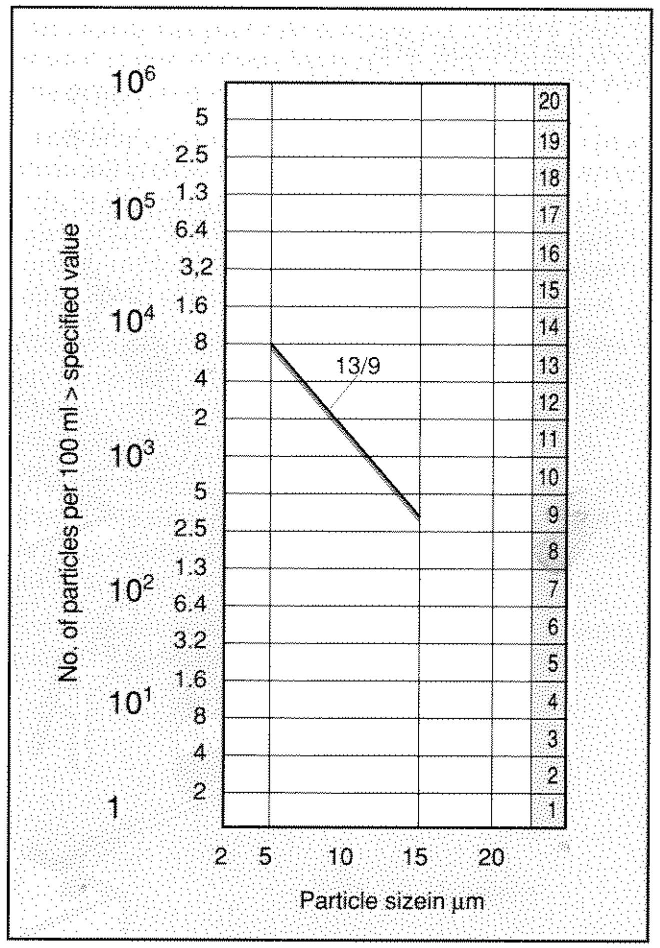


Diagram 45: *Contamination coding to ISO DIS 4406*

Code	No. of particles per 100 ml			
	over 5 µm		over 15 µm	
	more than	and up to	more than	and up to
20/17	500 k	1M	64 k	130 k
20/16	500 k	1M	32 k	64 k
20/15	500 k	1 M	16 k	32 k
20/14	500 k	1 M	8 k	16 k
19/16	250 k	500 k	32 k	64 k
19/15	250 k	500 k	16 k	32 k
19/14	250 k	500 k	8 k	16 k
19/13	250 k	500 k	4 k	8 k
18/15	130 k	250 k	16 k	32 k
18/14	130 k	250 k	8 k	16 k
18/13	130 k	250 k	4 k	8 k
18/12	130 k	250 k	2 k	4 k
17/14	64 k	130 k	8 k	16 k
17/13	64 k	130 k	4 k	8 k
17/12	64 k	130 k	2 k	4 k
17/11	64 k	130 k	1 k	2 k
16/13	32 k	64 k	4 k	8 k
16/12	32 k	64 k	2 k	4 k
16/11	32 k	64 k	1 k	2 k
16/10	32 k	64 k	500	1 k
15/12	16 k	32 k	2 k	4 k
15/11	16 k	32 k	1 k	2 k
15/10	16 k	32 k	500	1 k
15/9	16 k	32 k	250	500
14/11	8 k	16 k	1 k	2 k
14/10	8 k	16 k	500	1 k
14/9	8 k	16 k	250	500
14/8	8 k	16 k	130	250
13/10	4 k	8 k	500	1 k
13/9	4 k	8 k	250	500
13/8	4 k	8 k	130	250
12/9	2 k	4 k	250	500
12/8	2 k	4 k	130	250
11/8	1 k	2 k	130	250

Table 26: *Contamination data and short form coding*

Using NAS 1638

In this standard the individual particle sizes are divided into 5 ranges. A maximum number of particles is allowed for each range in each class.

Class	5-15 µm	15-25 µm	25-50 µm	50-100 µm	> 100 µm
00	125	22	4	1	0
0	250	44	8	2	0
1	500	89	16	3	1
2	1000	178	32	6	1
3	2000	356	63	11	2
4	4000	712	126	22	4
5	8000	1425	253	45	8
6	16000	2850	506	90	16
7	32000	5700	1012	180	32
8	64000	11400	2025	360	64
9	128000	22800	4050	720	128
10	256000	45600	8100	1440	256
11	512000	91200	16200	2880	512
12	1024000	182400	32400	5760	1024

Table 27: *Contamination classes to NAS 1638.*
Maximum number of dirt particles in 100 ml of fluid.

Contamination classification to SAE 749 D

This classification system is hardly ever used due to the relatively small number of gradings (9 particles/ml to 580 particles/ml).

Contamination classification to MIL STD 1246 A

This standard is only used in special cases and is of practically of no importance in industry.

4.8 Measuring the contamination of a system

Solid particle contamination is measured by taking a sample of fluid from the hydraulic system and analyzing it.

The analysis can reveal:

- the solid particle contamination of fluid delivered by suppliers
- the effectiveness of the system filters
- the flushing time when commissioning a system
- the state of the system and any possible damage to components when regular checks are made.

Methods of taking samples *(Fig. 92)*

- Taking a sample from a moving fluid (dynamic sampling)

Sampling point:
Within a system which is operating (there must be turbulent flow). *See ISO 4021.*

- Taking a sample from a stagnant fluid (static sampling)

Sampling point:
From the hydraulic tank.
See CETOP RP 95 H, Section 3.

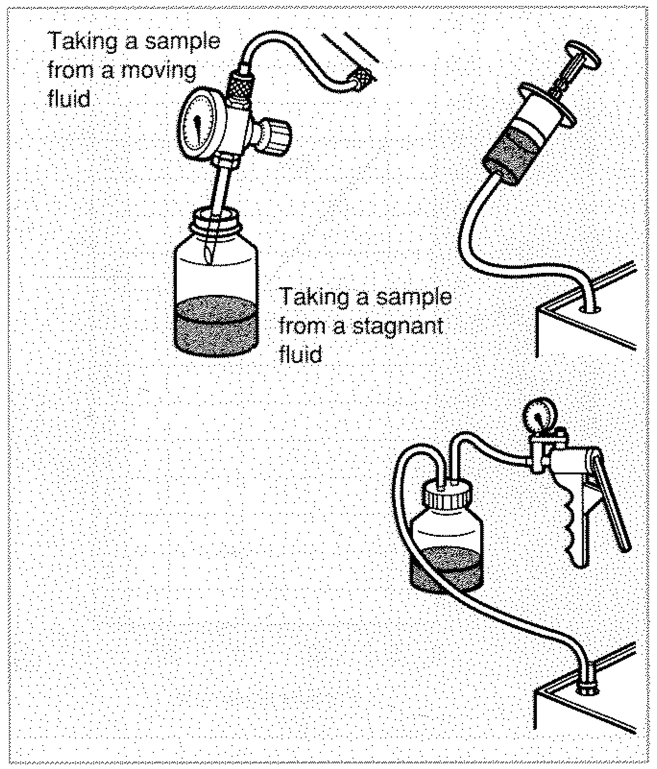


Fig. 92: *Methods of taking samples*

Fluid sampling procedure

General

- Before taking the sample the sampling device must be carefully flushed out with clean solvent
- Only use sample bottles that have been cleaned with fresh solvent
- Remove any remaining solvent before taking the sample
- Allow at least 2 litres of system fluid to flush through the sampling device before the actual sample is taken
- Take a zero sample. This is not used for the analysis because it is not representative of the system contamination
- Put the fluid to be analyzed into a new, clean sample bottle. The protective foil on the bottle should only be lifted.

Analysing the samples

The samples are usually analysed by means of an electronic particle counter.

Due to their high cost and constant attention needed, they will only be found in use by major users of hydraulics, the manufacturers of hydraulic filters and various institutes. This means that your fluid samples will probably have to be sent to one of these institutes. A direct check at the time of sampling is out of the question *(Fig. 94)*.

This is why a system of monitors or test charts has been produced in order to allow a rough but quick assessment of the fluid samples to be made on the spot. A microscope is employed to make an estimate of the solid particle contamination and from that it is possible to assess the state of the system.

What the fluid sample is able to tell depends very much on the person who took the sample. Therefore, only properly trained and experienced persons should be employed for the sampling.

Errors in sampling procedures can have a very great effect in the case of contamination classes below NAS 6 so it is advisable for particle counting to be carried out on site in order to eliminate any errors in sampling.

For such situations there is a mobile laboratory service available to perform the measurements for customers *(Fig. 93)*.

Fig. 93: *Mobile laboratory for on-site analysis*

Fig. 94: *Fluid sample analysis in the laboratory*

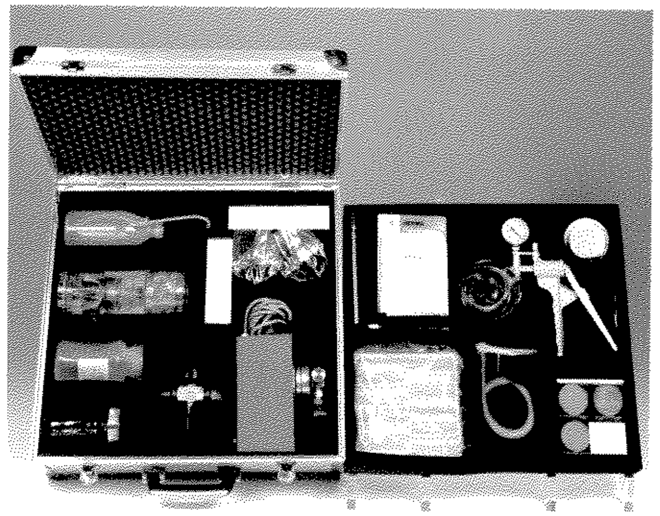


Fig. 95: *Sampling kit*

5 Designing hydraulic filters

5.1 General

Any filter used in a hydraulic system causes a pressure drop which increases steadily with time. The magnitude of this pressure drop is representative of the functional efficiency of the filter. Selecting a filter and positioning it properly in a system requires just as much care and experience as selecting any of the other components. Hydraulic filters should always be fitted with a clogging indicator to monitor the pressure drop across the filter element.

The following criteria govern the selection of a suitable hydraulic filter:

- Specific filtration rating
- Operating pressure
- Number of work cycles
- Filtration efficiency
- Dirt holding capacity of the element
- Place of installation of the filter.

The benefits of a correctly and generously sized filter are:

- greater reliability for the system
- longer service life for both machine and fluid
- less downtime and fewer spares replacements.

Table 28 lists the advantages and disadvantages of the different types of filter.

Type of filter	Advantages	Disadvantages
Pressure-line filters	Filtration takes place directly before the components requiring protection. Required cleanliness of fluid assured.	Expensive filter housings and elements. Complex element construction due to the pressure drop strength required. Pump is not protected. With single filters the system must be shut down to change the element.
Return line filters	The whole return flow of fluid is filtered. No contamination gets into the tank. Cheap filter housings and elements. Generous sizing of the filter possible.	A pressure-line filter must also be fitted when there are precision components such as servo valves installed. A by-pass valve must be fitted to the filter. Elements with low pressure drop strength can be damaged by flow pulsation. With single filters the system must be shut down to change the element.
By-pass filters	Uniform filtration independent of the work cycle. Optimum use of the dirt holding capacity of the filter element. Cheap filter housings and elements. System does not have to be shut down to change the element. Retro-fitting possible.	A pressure-line filter must also be fitted when there are precision components such as servo valves installed. Installing an extra pump increases the power consumption of the total system. Higher capital investment required in plant. More filtration required if contamination occurs regularly.
Suction filters	The fluid drawn in by the pump is filtered	Very fine filtering impossible. Poor cleaning facility. Pump must be protected against cavitation.

Table 28: *Types of hydraulic filter, their advantages and disadvantages*

5.2 Positioning of filters in hydraulic systems

The position of a filter in a fluid circuit depends on the task which that filter is expected to perform *(Fig. 96)*.

Protecting the fluid against contamination

This task is performed by return line filters or complete by-pass filter units in the hydraulic installation. The filter must be selected appropriately for the specified class of fluid cleanliness.

Protecting components sensitive to contamination

In order to offer as much protection as possible to the components in question, the filter must be positioned as close as possible to them. The filter must be selected for the appropriate operating pressure and the filtration rating specified by the component manufacturer.

Protecting the system against environmental contamination

The task of these filters or breathers is to prevent any environmental contamination from reaching the hydraulic fluid. Selection of the appropriate breathers must take into account the pulsating flow of air and the amount of contamination in it.

Protecting the system against component failure

These filters protect the system against major contamination in the event of component failure. They are intended to avoid high repair costs and knock-on costs.

When choosing the position for a filter in a hydraulic system the important points to watch are that the filter is easily accessible, the element can be changed easily and the clogging indicator can be clearly seen at all times. Badly positioned filters have an adverse effect on maintenance since they cannot perform the tasks allocated to them in the best possible manner.

Filters with by-pass valves

By-pass valves fitted to filters perform the following functions:

- They protect the filter element from damage due to excessive pressure drop across it.

A high pressure drop can arise due to the clogging of the element with contamination or high viscosity of the fluid during a cold start.

- They prevent the malfunctioning of components in the system.

With return line filters in particular, an excessive pressure drop across the filter element can result in malfunctioning of valves, uncontrolled operation of cylinders and damage to seals.

The following points must be noted when installing by-pass valves:

- The filtering action is reduced when the by-pass valve is open. When the valve is fully open there is no filtering at all and, therefore, no protection for the components in the system.
- Clogging indicators are absolutely essential so that filter maintenance can be carried out promptly.
- The filter element must be changed immediately when the clogging indicator is triggered.

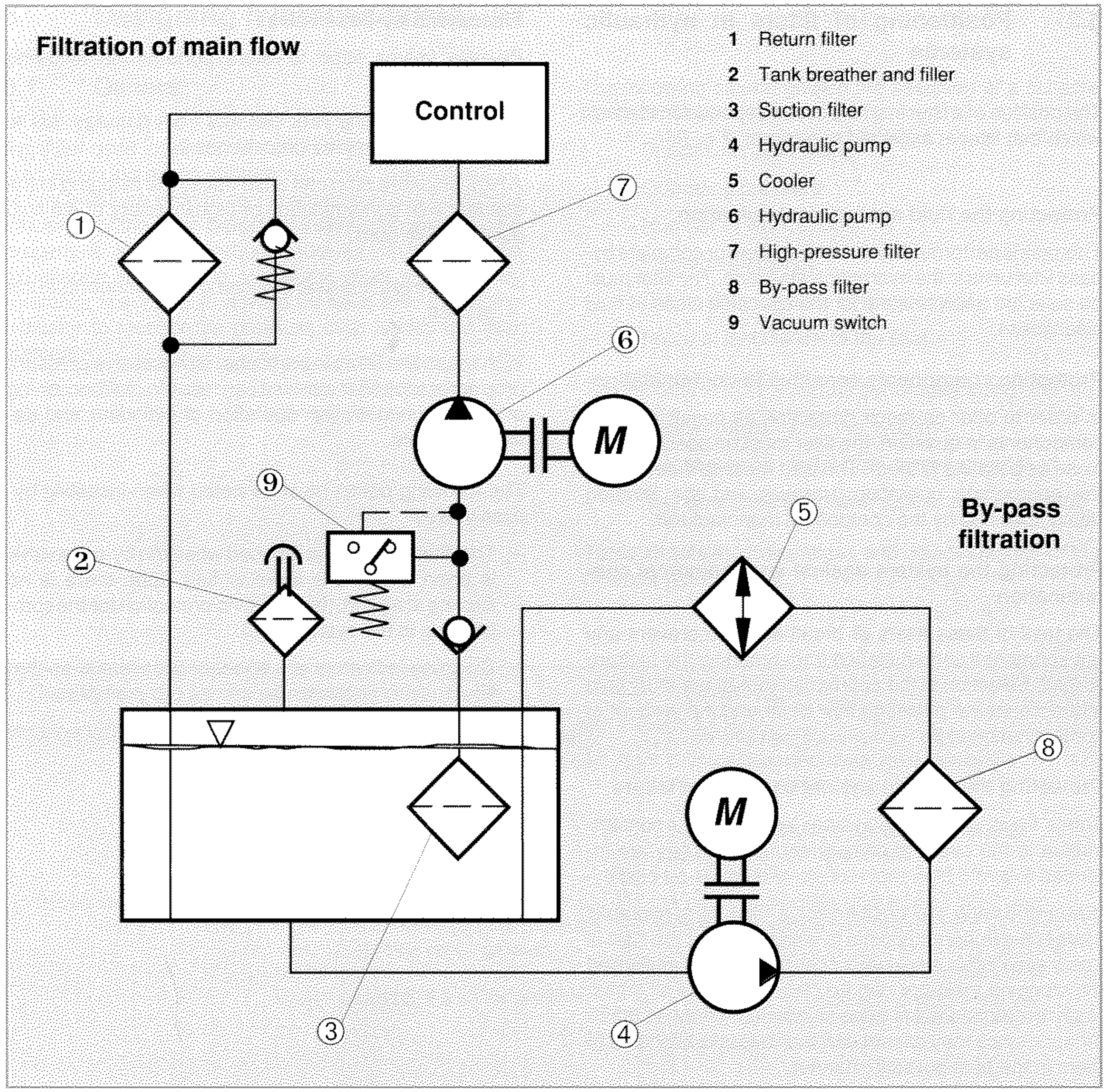


Fig. 96: *Diagram of filter positioning in hydraulic systems*

5.2.1 Main filters

These filter the fluid flowing in the actual main circuit of the system.

The following types of filter are used:

Suction filters

These are fitted between tank and pump and their task is to prevent any severe contamination from reaching the pump. In order to avoid cavitation damage to the pump such filters can only be fitted with coarse strainer elements. A vacuum switch must also be fitted between pump and filter in order to stop the pump if the vacuum falls below a certain figure. The low pressure drop means that fine filtration cannot be achieved with suction filters.

Pressure-line filters

These filters are fitted between the pump and the components of the system. In order to offer complete protection to the components they should not have a by-pass valve. Their task is to ensure the required cleanliness of the fluid fed to the hydraulic components such as servo valves.

Return line filters

The purpose of these filters is to filter the flow of fluid returning to the system tank. The filter must be capable of handling the total volumetric return flow which, when single-rod cylinders or accumulators are included, can be substantially higher than the installed volumetric flow of the pump.

Breathers

Their task is to filter the air drawn into the tank as the fluid volume "breathes".

The first law of filter design

The specified value of filtration rating must be applied to all the filters installed in a hydraulic system, e.g. pressure-line filters, return line filters and breathers.

Task assignment of filters

For the purpose of economy in hydraulic system design the filters are divided into two groups, working filters and protective filters *(Fig. 97)*.

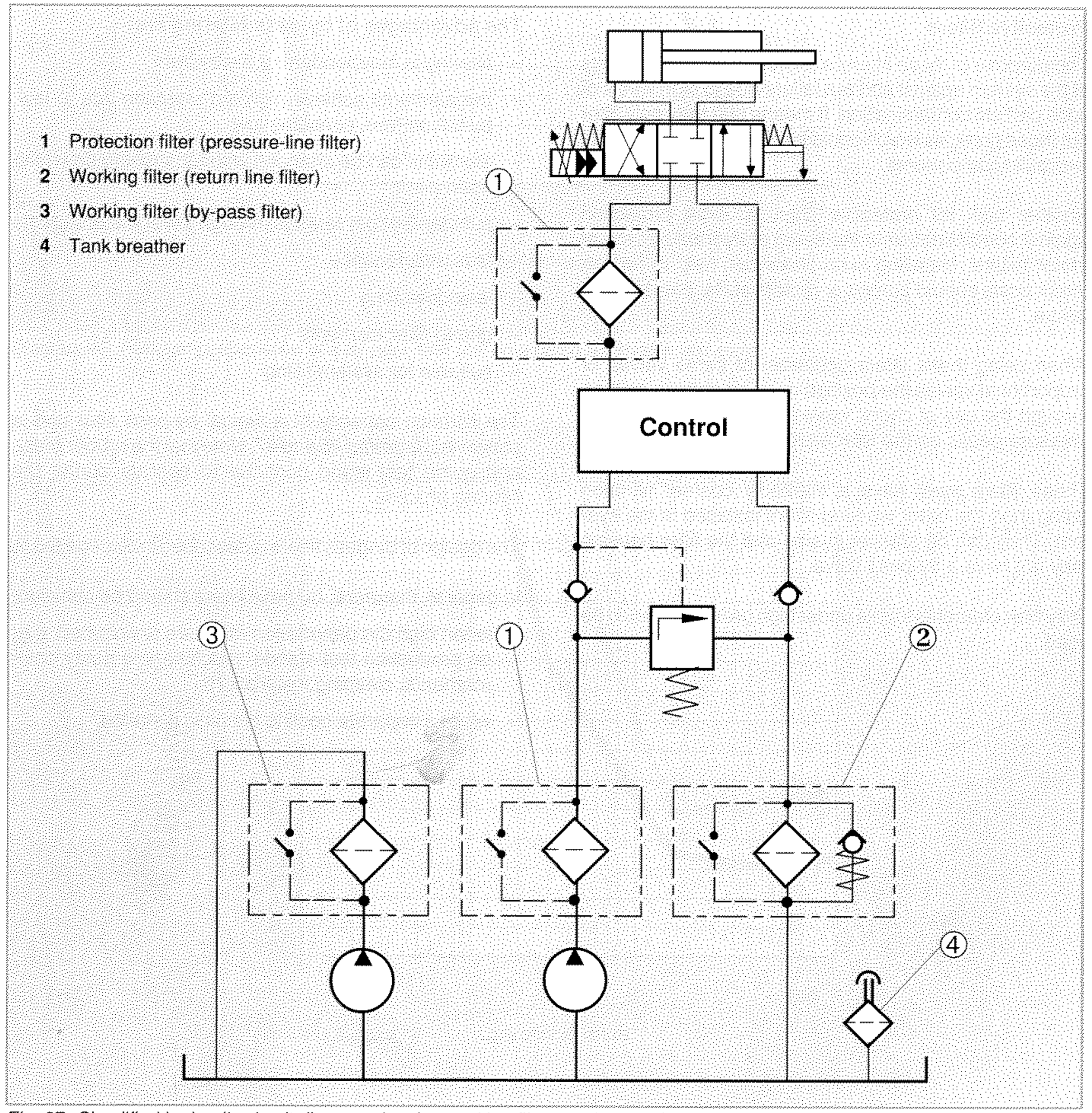


Fig. 97: *Simplified hydraulic circuit diagram showing working filters and protective filters*

Working filters

These include return line filters and pressure-line filters with by-pass valves and also by-pass filters.

Working filters are equipped with elements with good low pressure stability which allow them to have a large filter area and so a high dirt holding capacity.

In order to perform efficiently, return line filters and pressure-line filters used as working filters must be generously sized and positioned in the maximum volumetric flow of the system. If necessary, such filters can also be installed in the leakage fluid lines.

Protective filters

The purpose of these filters is to protect the components of the system against sudden failure due to high levels of solid particle contamination. It means that they only filter out those particles which could lead to sudden seizure of hydraulic components.

Another task for protective filters is extra protection against contamination in the event of hydraulic pump or motor failure. Installing such filters can help to reduce repair costs should pumps or motors suffer catastrophic failure.

When using these filters upstream of servo valves or proportional valves the position must be chosen so that, through the use of check valves, there are no negative pressure peaks on the filter element.

These filters must have a markedly coarser filtration rating than the other working filters installed in the system. They can be smaller in size and the filter housing may not have a by-pass valve.

Only filter elements stable under high pressure should be used.

5.2.2 By-pass filters

The purpose of these filters is to filter the tank fluid continuously in a by-pass circuit. The normal practice is to use a complete by-pass filter unit comprising pump, filter and cooler.

The advantage of by-pass filtering is that the filter can work independently of the operating cycles of the hydraulic system and the flow of fluid through the filter element is constant.

Ageing of the fluid is retarded and a clear improvement in the service life of the fluid is achieved.

The advantages of by-pass filtering are:

- filtering is independent of the system
- the elements achieve high dirt retention due to low, pulsation-free, constant flow
- elements can be changed without shutting down the main plant
- substantial cost savings through lower material costs
- less maintenance
- less downtime
- cheap filter elements
- suitable for system filling

The filtration capacity of a typical by-pass filter unit is shown in *Diagrams 46 and 47*. Note that the rubber press and pump test stand continue to operate during the filtering process.

The design of by-pass filters is described in *Section 5.6.2*.

In general, therefore, by-pass filters should be installed:

- when high dirt penetration rates are anticipated, e.g. on production test stands, machinery in dusty environments, cleaning installations
- when a separate cooling circuit is installed.

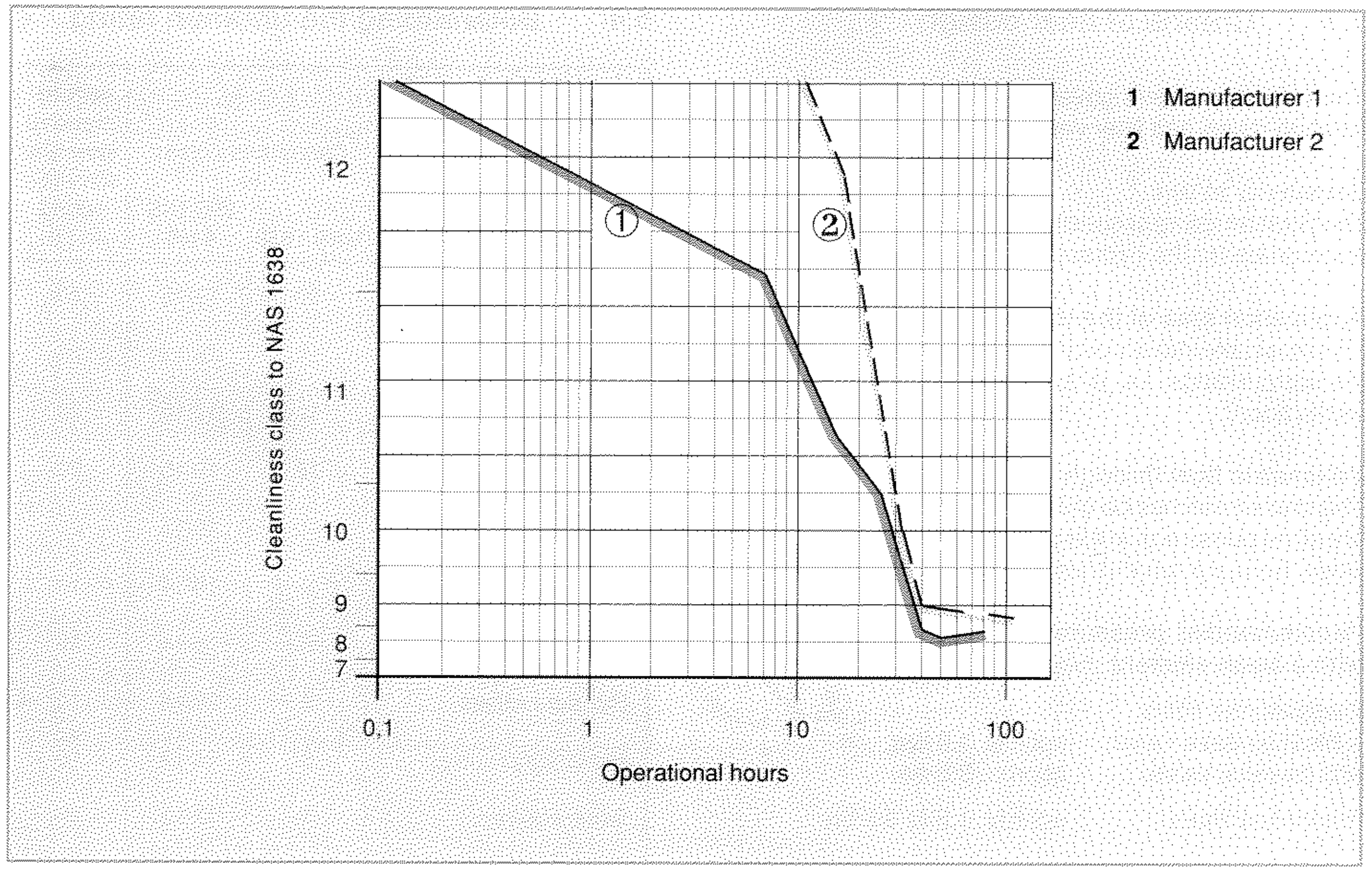


Diagram 46: *Filtration performance of a by-pass filter unit on a pump production test stand*

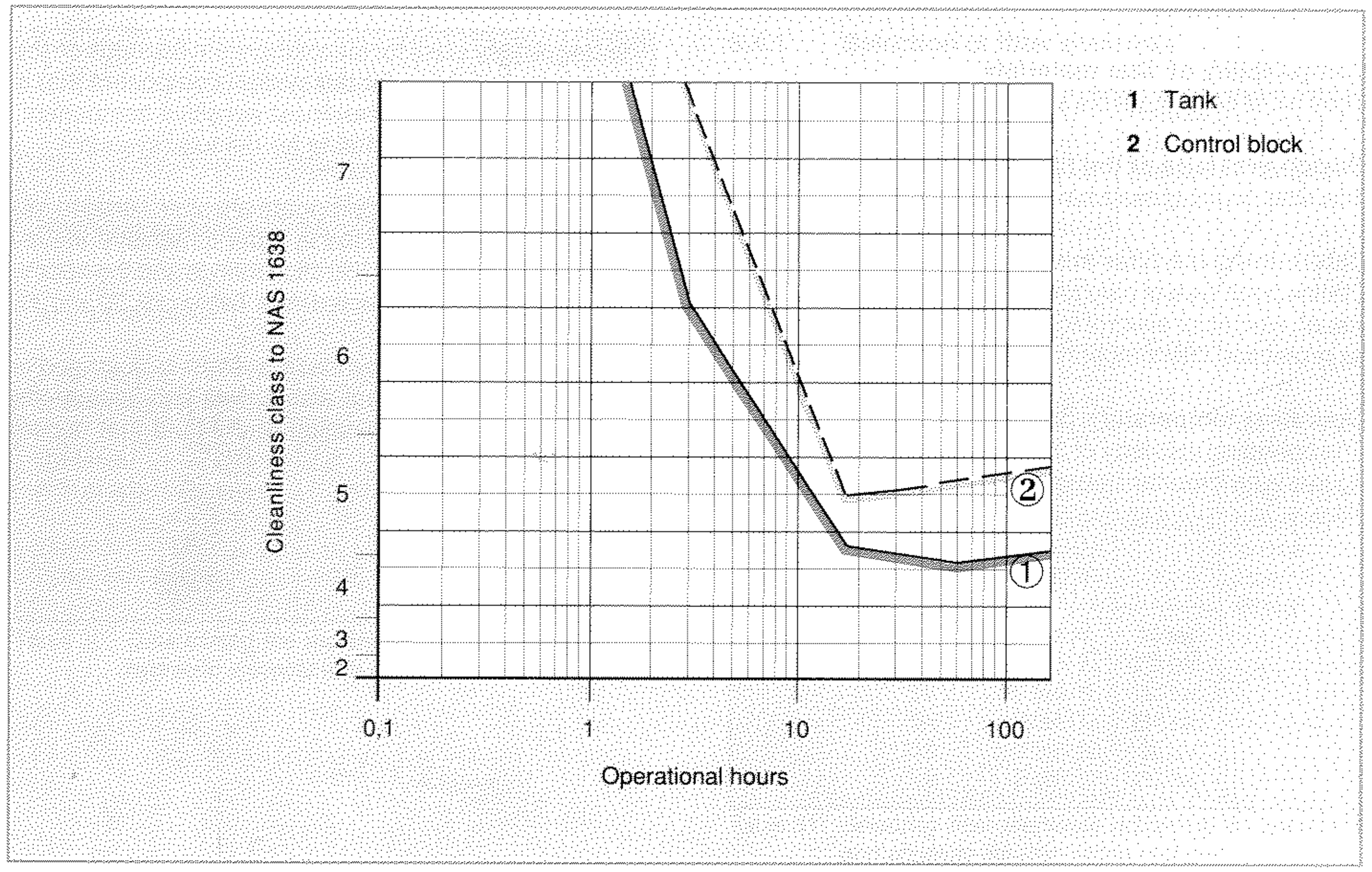


Diagram 47: *Filtration performance of a by-pass filter unit on a rubber press*

5.3 Criteria of filter design

The following criteria govern the selection of filter size, filtration rating and filter version:

Sensitivity to dirt
The filtration rating or specified cleanliness class must be appropriate for the hydraulic components.

Application area of the total system
This must take into account the possible contamination of the surroundings, i.e. is it a laboratory or on the steelworks floor?

Volumetric flow through the filter
This can sometimes be greater than the maximum delivery of the pump, e.g. in the case of single-rod cylinders or when there are return lines from several circuits.

Recommended pressure drop at normal viscosity with a clean element *(Housing and element)*

Pressure-line filters,
without by-pass valve: approx. 1.0 bar
with by-pass valve: approx. 0.5 bar

Return line filters: approx. 0.3 to 0.5 bar

Permissible maximum pressure drop
The maximum pressure drop across the filter element must be appropriate for the system conditions at the place of installation.

Compatibility of filter materials
They must be compatible with the hydraulic fluid.

Design pressure of the filter housing
The filter housing must have adequate fatigue strength.

Determining the filter model
Decide what type of clogging indicator is to be fitted, e.g. visual, electric or electronic. Pressure-line filters working as protection filters must have no by-pass valve.

Operating temperature or design temperature
The operating viscosity of the fluid calculated from these figures is an important factor in determining the size of the filter.

5.4 Determining the filtration rating

Hydraulic components	Cleanliness class to NAS 1638	ISO DIS 4406	recommended absolute filtration rating in μm
Gear pumps	10	19/15	20
Cylinders	10	19/15	20
Directional control valves	10	19/15	20
Relief valves	10	19/15	20
Throttle valves	10	19/15	20
Piston pumps	9	18/14	10
Vane pumps	9	18/14	10
Pressure valves	9	18/14	10
Proportional valves	9	18/14	10
Servo valves	7	17/13	5
Servo cylinders	7	17/13	5

Table 29: *Recommended values of absolute filtration rating for various hydraulic components*

The cleanliness class for the total system depends on the required classification for the system component that is most sensitive to dirt. This "most sensitive component" determines the filtration rating for the total system.

Filter elements with an appropriate absolute filtration rating ($\beta_x \geq 100$) must be used in order to achieve the required cleanliness class. Filtration ratings and the necessary elements can be selected from *Tables 29, 30 and 31*.

The filtering action in a hydraulic system is illustrated in *Diagram 48*. This diagram also shows very clearly the rapid rise in the contamination of the fluid that occurs when no filter is fitted.

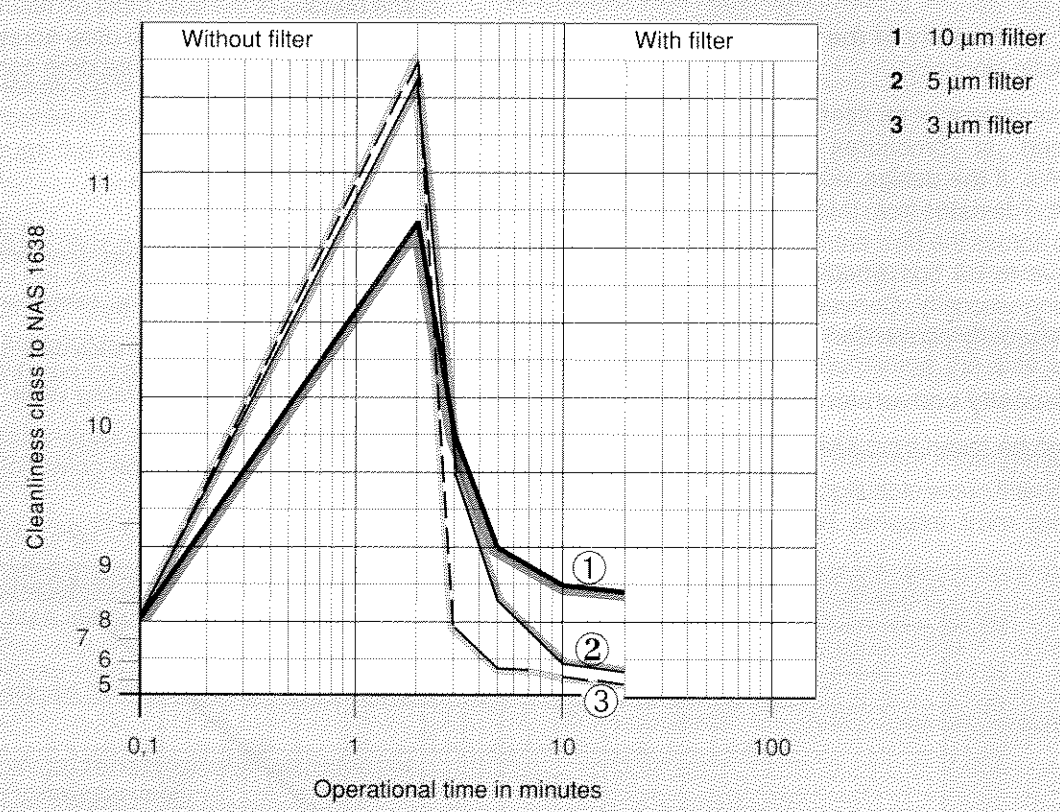


Diagram 48: *Cleanliness classes that can be achieved using the recommended absolute filtration rating*

Hydraulic system	Recommended absolute filtration rating ($\beta_x \geq 100$)	Attainable cleanliness class to NAS 1638 with particles > 5 μm	ISO DIS 4406
Systems with servo valves	5	7	17/13
Systems with precision proportional valves	5	7 to 8	17/13
Systems with proportional valves	10	9	18/14
General hydraulic systems	10 to 20	9 to 10	18/14

Table 30: *Determining the recommended filtration rating for hydraulic systems with Rexroth components*

5.4.1 Selection of filter elements

Application	Filtration rating μm	Rexroth element designation	Pressure drop strength	Remarks
Working filters, by-pass filters, Return line filters, pressure line filters with by-pass valve	3 3 5 5 10 10 20 20	... R 003 BN/HC ... D 003 BN/HC ... R 005 BN/HC ... D 005 BN/HC ... R 010 BN/HC ... D 010 BN/HC ... R 020 BN/HC ... D 020 BN/HC	30 bar	
Protection filters, pressure line filter without by-pass valve	3 5 10 20	... D 003 BH/HC ... D 005 BH/HC ... D 010 BH/HC ... D 020 BH/HC	210 bar	Enquire from manufacturer for other values of filtration
	25 25 50 50 100 100	... D 025 W ... D 025 T ... D 050 W ... D 050 T ... D 100 W ... D 100 T	30 bar 210 bar 30 bar 210 bar 30 bar 210 bar	

Table 31: *Selecting filter elements according to application and absolute filtration rating required*

5.5 How the fluid affects filter design

5.5.1 Viscosity of the fluid

(kinematic viscosity)

The characteristics for filter housings and filter elements published in brochures refer to a fluid viscosity of 30 mm²/s. If the design viscosity (usually the operating viscosity) deviates from this reference value, the pressure drop across the filter element (taken from the diagram) will have to be converted to the appropriate value at operating viscosity. Conversion is performed by means of the viscosity conversion factor f_1.

The viscosity conversion factor f_1

This can be taken from *Diagram 49.*

5.5.2 Density of the fluid

The density of the fluid must be taken into account when determining the pressure drop across the filter housing. The filter housing pressure drop can be calculated with the following equation

$$\Delta p_{HF} = \Delta p_{HR} \cdot \frac{\rho_O}{\rho_R}$$

Δp_{HF} = Housing pressure drop, operating fluid

Δp_{HR} = Housing pressure drop, reference data (from catalogue)

ρ_R = Density of fluid, reference data (from catalogue)

ρ_O = Density of fluid, operating value

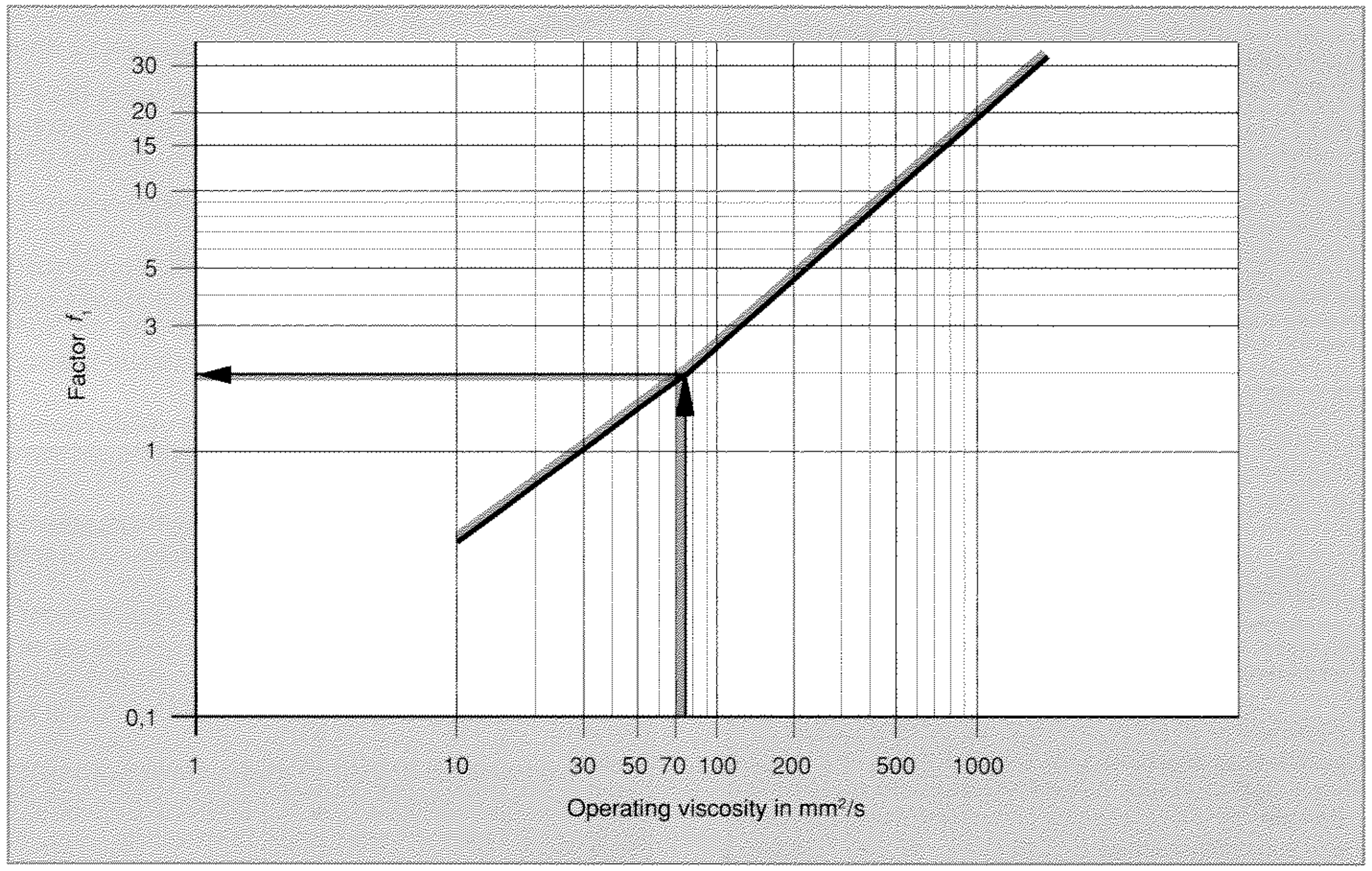


Diagram 49: *Graphic illustration of viscosity conversion factor f_1*

5.6 Determining the filter size

5.6.1 Main filters

The objective in determining the size of the filter is to establish a balance between the dirt gain of the system and the dirt loss through the filter. Filter service life must also be economical.

Therefore, the size of the filter must take into account the amount of dirt around the machine and the maintenance and care provided for the hydraulic systems. The environmental conditions are allowed for by factor f_2. Individual values of factor f_2 can be taken from *Table 32.*

The permitted pressure drop across the filter can be calculated from the following equation:

$$\Delta p_{tot} = (\Delta p_{HF} + f_1 \cdot \Delta p_E) \cdot f_2$$

Δp_{tot} = Total pressure drop across the filter at operating temperature with a clean element and effective volumetric flow

Δp_{HF} = Pressure drop across the filter housing with operating fluid

Δp_E = Pressure drop across the clean element with effective volumetric flow (catalogue data)

f_1 = Viscosity conversion factor

f_2 = Environmental factor

The pressure drop across the filter must be calculated for the effective volumetric flow passing through the filter.

The equation is:

$$Q_W = Q_R \cdot Ad$$

Q_W = Effective volumetric flow

Q_R = Flow of the pump

Ad = Additional flow due to accumulators or cylinders.

The maximum initial pressure drops given in *Table 33* must not be exceeded in determining the size of filter required.

The data refers to a new filter element filtering hydraulic oil. Other design criteria are applicable when filtering fire-resistant fluids or engine oil (enquire from the filter supplier as necessary).

Maintenance and care of hydraulic systems	Contamination of machine surroundings		
	[1] low	[2] average	[3] high
– regular checking of filter – immediate changing of filter element – low dirt ingress – good sealing of tank	1,0	1,0	1,3
– irregular checking of filter – few cylinders used	1,0	1,5	1,7
– minimal or total absence of filter checking – numerous unprotected cylinders – high dirt ingress into the system	1,3	2,0	2,3

Table 32: *Environmental factor f_2*

Notes on *Table 32*

[1] low: e.g. testing machines in closed, air-conditioned rooms

[2] average: e.g. machine tools in heated workshops

[3] high: e.g. presses in foundries, ceramics machinery, potash mining machinery, agricultural and mobile machinery, rolling mills, woodworking machinery

Size of filter required

The total pressure drop across the filter can be determined:

– using the individual diagrams for filter housing and element

Ascertain the individual pressure drops across the filter housing and element at the effective volumetric flow Q_E and operating viscosity. *Diagram 50* shows the pressure drop across the filter housing when filtering hydraulic fluid. *Diagram 51* shows the pressure drop across the clean filter element with a fluid viscosity of 30 mm²/s.

In calculating the required size of filter the total pressure drop must be multiplied by the factor f_2 in order to allow for the environmental conditions.

If the total pressure drop across the filter ascertained in this way is greater than the maximum value given in *Table 33* the whole calculation will have to be repeated for a larger size of filter.

Only when the calculated total pressure drop of the filter is equal to or less than the maximum permitted total pressure drop has the filter been correctly sized.

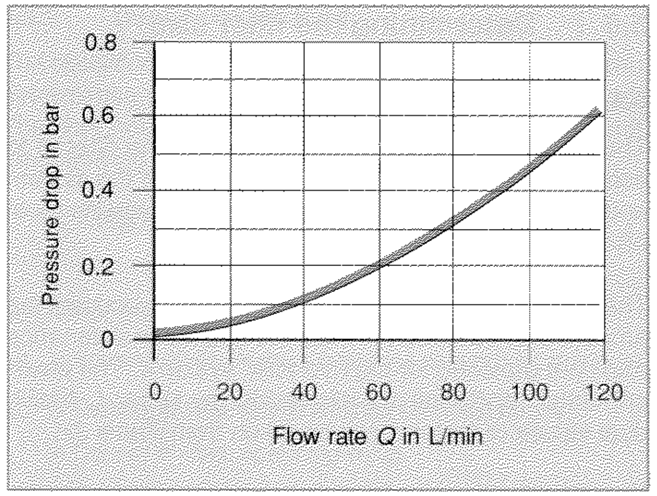


Diagram 50: *Pressure drop across a filter housing*

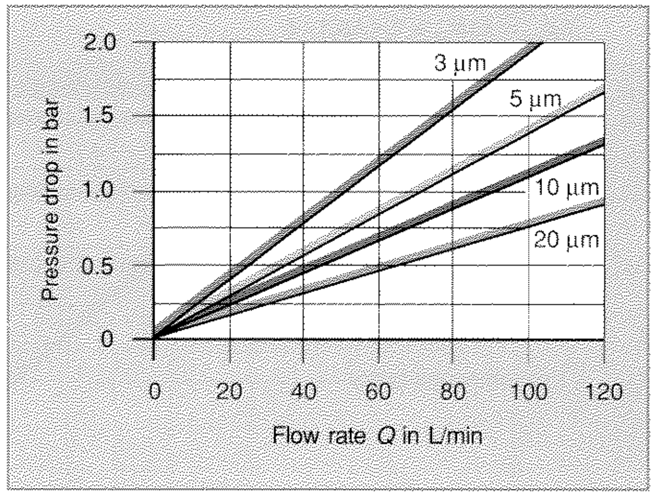


Diagram 51: *Pressure drop across a filter element*

Filter arrangement in Hydraulic systems	Type of filter	Total pressure drop across filter with new element: Using the individual diagrams for filter housing and filter element	Using the design diagrams
Working filters	Return line filters Pressure line filters with by-pass valve	$f_2\,(\Delta p_{housing} + f_1 \cdot \Delta p_{element}) \leq 0{,}5$	$Q_{design} = Q_{system} \cdot f_1 \cdot f_2$
Protection filters	Pressure line filters without by-pass valve	$f_2\,(\Delta p_{housing} + f_1 \cdot \Delta p_{element}) \leq 1{,}0$	$Q_{design} = Q_{system} \cdot f_1 \cdot f_2$
By-pass filters	Line filters separate unit	—	—
Suction filters		$f_2\,(\Delta p_{housing} + f_1 \cdot \Delta p_{element}) \leq 0{,}01$	$Q_{design} = 5 \text{ to } 10 \cdot Q_{pump} \cdot f_2$

Table 33: *Determining the size of filter*

– Using the filter design diagrams

The filter design diagrams *(Diagrams 52 and 53)* were originally developed in order to reduce and simplify the relatively complex procedure of filter sizing. The diagrams refer to a fluid viscosity of 30 mm²/s.

Higher operating viscosities and different environmental conditions are allowed for in determining the volumetric flow for the filter.

The volumetric flow can be calculated from the equation:

$$Q_D = Q_W \cdot f_1 \cdot f_2$$

Q_D = Volumetric flow for filter design
Q_W = Effective volumetric flow
f_1 = Viscosity conversion factor
f_2 = Environmental factor

The required size of filter can be read off from the point of intersection between the volumetric flow Q_D and the filtration rating.

5.6.2 By-pass filters

The cleaning of the fluid circulating in a hydraulic system can be greatly improved by installing a by-pass filter. Furthermore, the solid particle contamination of fluids in existing systems can be reduced at any time and without major modifications by using a by-pass filter.

The by-pass filter should work for longer than the hydraulic system itself so it is better for the filter to be independent of the system and filtration can then continue while the system is shut down, e.g. during meal breaks, tea breaks, weekends, etc.

The sizing of a by-pass filter is based on

– the volumetric flow through the filter
– the filter area.

Volumetric flow required in a by-pass filter

The maximum volumetric flow required can be calculated from the following equation:

$$Q_N = \frac{Q_D \cdot T_{TP} \cdot T_{PW} \cdot f_2}{T_{TB} \cdot T_{BW}}$$

Q_N = Volumetric flow for by-pass filter
Q_D = Total volumetric flow of pumps in power unit
T_{TP} = Operating time of power unit per day
T_{PW} = Operating time of power unit per week
T_{TB} = Operating time of by-pass filter per day
T_{BW} = Operating time of by-pass filter per week
f_1 = Environmental factor *(Table 32)*

When there are minor differences between the period of operation of the power unit and by-pass filter the flow rate through the by-pass filter will be similar to the installed pump delivery in the power unit.

However, this is uneconomical so it is advisable in such cases to design the by-pass filter as follows:

– Fix the flow rate through the by-pass filter so that the contents of a 1000 litre tank are circulated once at least every 30 minutes. With larger tanks the circulation cycle should be at least 120 minutes.
– The cleaning action must be increased so the by-pass filter should be selected one step finer than the filter in the power unit.
– The required filter area must be determined according to the specific area loading for the volumetric flow required.

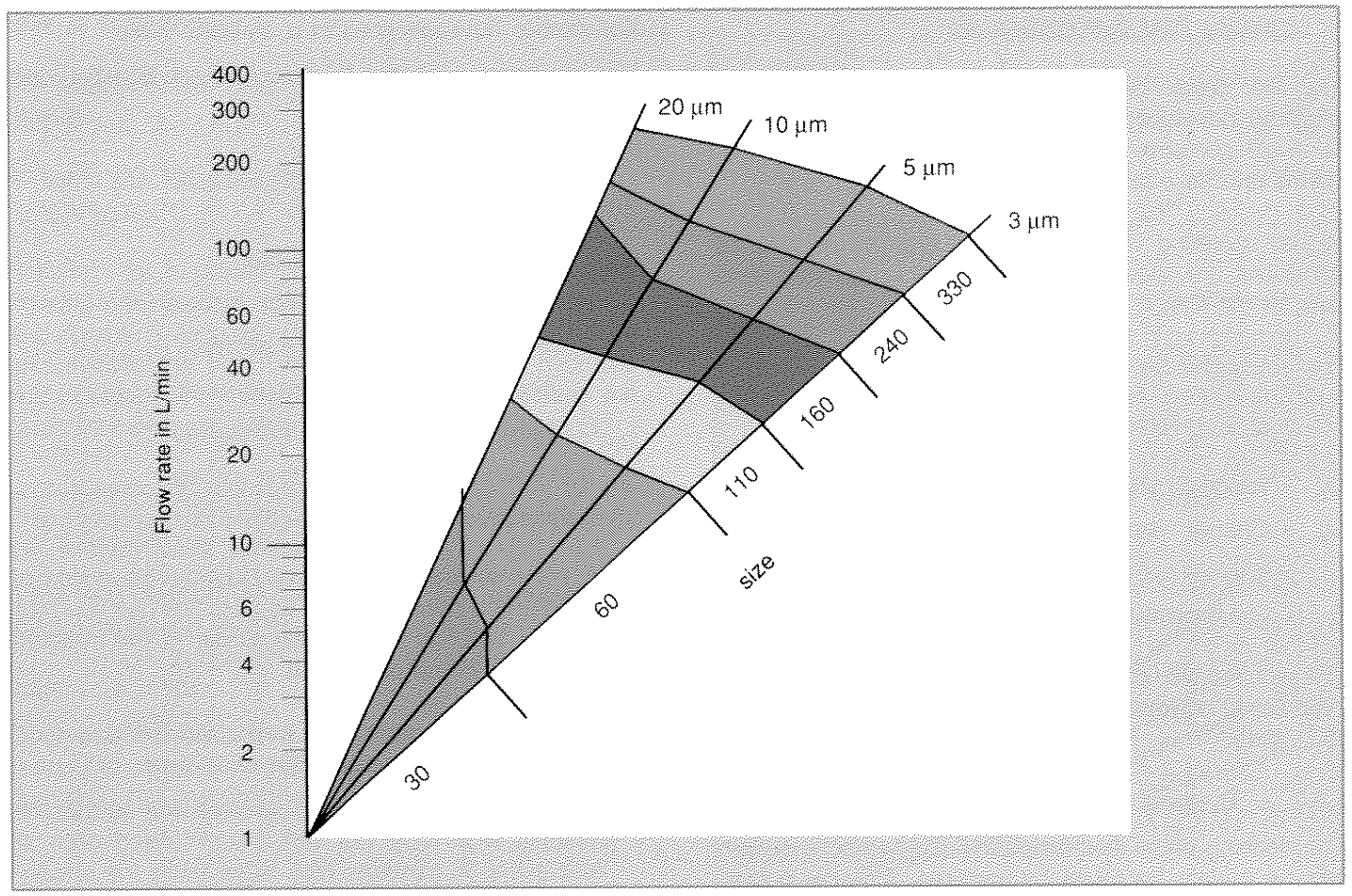


Diagram 52: *Filter design diagram for return line filters*

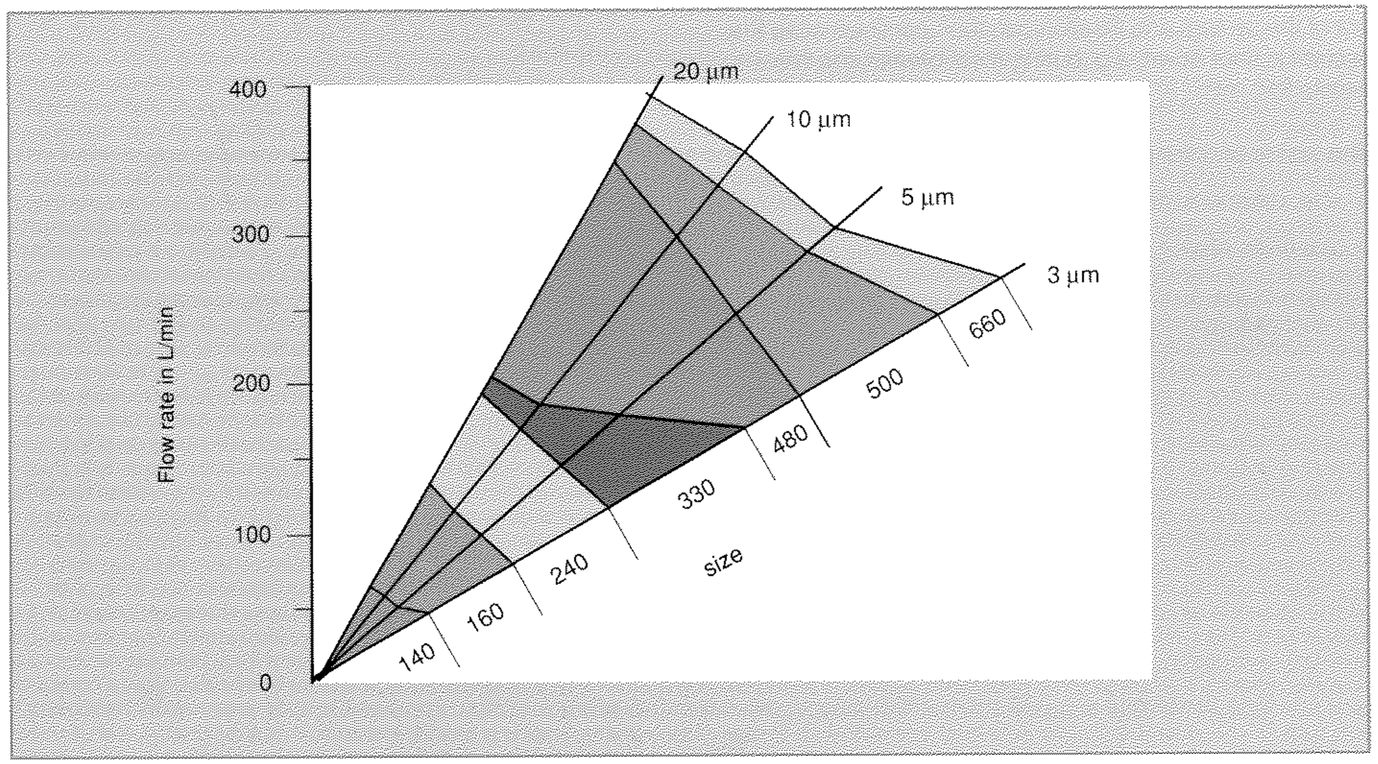


Diagram 53: *Filter design diagram for pressure filters*

Determining the filter area required

The filtration rating must be determined first before the required filter area can be calculated. If possible, it should be one step finer than the system filter.

The filtration rating depends on the hydraulic components used in the power unit. The procedure is described in *Section 5.4.*

The minimum required filter area can then be calculated taking into account the specific area loadings given in *Table 34.*

The maximum required filter area can be calculated with the following equation:

$$A = \frac{Q_N \cdot f_1}{q}$$

A = Required filter area
Q_N = Volumetric flow through by-pass filter
q = Specific area loading *(see Table 34)*
f_1 = Viscosity conversion factor

Filtration rating $\beta_x \geq 100$	Specific area loading L/min/cm²
3 µm	0.0025
5 µm	0.0035
10 µm	0.005
20 µm	0.005

Table 34: *Specific area loading for the design of by-pass filters with elements of glass fibre non-woven*

5.6.3 Tank breathers

The dirt penetration rate has a major effect on the contamination of the system and the tank breathing system is very important in combating the problem. The function of a tank breather is to prevent dirt from the environment penetrating the system while at the same time allowing the tank to "breathe" air. Wrongly or carelessly designed tank breathing can place a substantial extra load on the filter circuit and so shorten the service life of the elements. The performance data of the breathers should be matched to that of the system filters.

The design of breathers should take into account the following data:

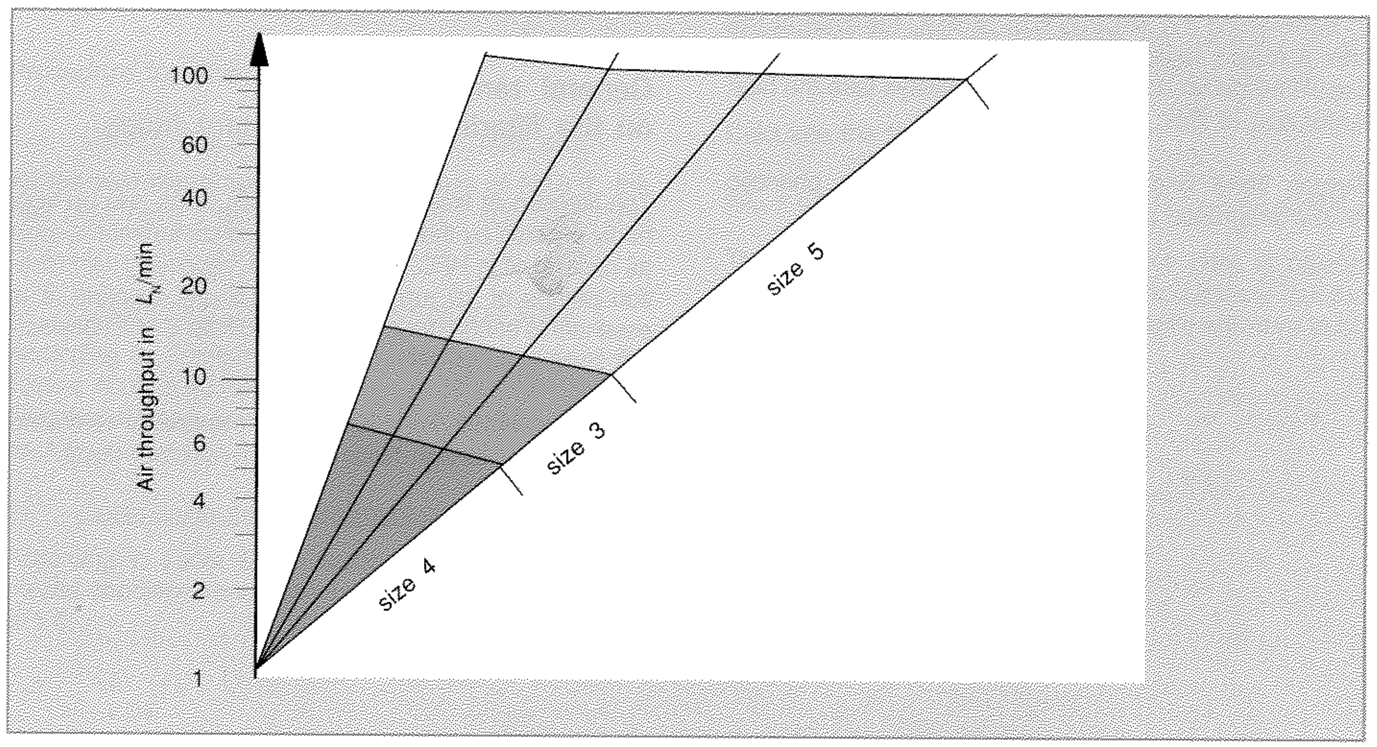
Filtration rating $\beta_x \geq 100$
(must be matched to system filters).

Design flow rate for breather:
5 to 10 times the maximum pump capacity.

Design pressure drop 0.01 bar
(with clean element and at design flow rate).

The breather size can be determined with the aid of *Diagram 54.*

Diagram 54: *Determining the size of tank breathers*



5.7 Filter design for fire-resistant fluids

The filtering of these fluids requires special attention to compatibility with the materials of the filter elements and housings.
Past experience has shown the following materials to be suitable:

Filter elements: glass fibre non-woven, metal non-woven, stainless steel wire mesh

Filter housings: steel, cast iron with surface protection, phosphated or electro-less nickel plated.

Filter housings can also be protected with a suitable paint.

Filters for fire-resistant fluids must have a larger area than those for mineral oil-based fluids because of the greater wear of components, the soapy residue, the growth of micro-organisms and the different dirt settling characteristics.

The filter area can be calculated with the following equations:

For pressure-line filters: $A = 30 \cdot f_1 \cdot f_2 \cdot f_3 \cdot Q_W$

For return line filters: $A = 60 \cdot f_1 \cdot f_2 \cdot f_3 \cdot Q_W$

A = Required filter area

f_1 = Viscosity conversion factor
(f_1 = 1 for HFA and HFB fluids)

f_2 = Environmental factor

f_3 = Fluid density conversion factor

Q_W = Effective volumetric flow

In determining the required size of filter, the area of the filter selected must be equal or greater than the area calculated above. In cases of doubt always take the next larger size of filter.

Suction filters and filter elements containing phenolic resin-impregnated paper must never be used.

Fluid density conversion factor f_3

Fluid designation	Factor f_3
HFA	1.16
HFB	1.16
HFC	1.27
HFD	2.21

Table 35

Determination of filter pore size

The filter pore for different types of hydraulic system are as follows:

General purpose systems:
10 or 20 μm absolute

Systems containing proportional valves:
10 μm absolute

Systems containing servo valves or control valves:
5 μm absolute

The filter size ascertained must be doubled in order to obtain economic operation of systems containing servo valves or control valves.

Design of by-pass filters

The procedure for the design of by-pass filters is the same as that described in *Section 5.6.2.*

General remarks on the filtering of fire-resistant fluids

The filtration of fire-resistant hydraulic fluids can be very badly affected by the presence of fluid contaminants such as mineral oil in HFC. Consequently, especially with filtration ratings of 10 μm absolute or 5 μm absolute, it is essential to ensure that the operating fluid is in a satisfactory state.

It may be necessary to use filters capable of removing any traces of foreign fluids from fire-resistant fluids.

6 Practical examples of filter design

The procedure for filter design will be explained by means of a number of examples.

Example 1

System data:

Pump: 1 PV2 V5-3X/16 RE 01 ML 70 A1

Max. operating pressure: 70 bar

Volumetric flow of pump Q_p = 27.5 L/min
at a motor speed of 1450 rev/min

Hydraulic fluid: ISO VG 46

Operating temperature: 40 °C

Continuous monitoring of the filter is provided and the environmental contamination can be regarded as average.

A Type 4 WS 2 EM 10/4X/5B... servo valve is employed in the power unit control.

Volumetric flow through the servo valve: 5 L/min

According to the brochure for the servo valve, a fluid cleanliness of NAS 7 is specified.

The power unit also drives a hydraulic cylinder having a full bore/annulus area ratio of 2:1.

A working filter is to be installed in the hydraulic power unit and a protective filter upstream of the servo valve.

Procedure for filter selection

1. Determining the required filtration pore size

As the fluid cleanliness required is class of NAS 7, the filter must have a filtration rating of ß5 = 100 *(see Table 30.)*

2. Determining the viscosity conversion factor f_1

According to *Diagram 44* the operating viscosity of the fluid at 40°C is 46 mm²/s.
Diagram 49 then gives the viscosity conversion factor f_1 as 1.5.

3. Determining the environmental factor f_2

According to *Table 32* the factor f_2 for average environmental conditions and continuous monitoring of the filter is 1.0.

4. Determining the nominal filter size

In designing the hydraulic system it has been established that a return line filter will be the working filter and a pressure-line filter upstream of the servo valve the protective filter.

4.1 Determining the nominal size of return line filter (working filter)

First calculate the effective volumetric flow Q_W:

$Q_W = Q_P \cdot Ad$ = 27.5 L/min • 2 = 55 L/min

Determining the filter size from the individual diagrams for housing and element

First select a size from past experience. If the total pressure drop calculated for this size is greater than the figure of maximum pressure drop given in *Table 33* the whole calculation will have to be repeated for a larger size of filter. Only when the calculated figure of total pressure drop is less than the prescribed maximum total pressure drop has the filter size been correctly chosen and can be incorporated into the system.

In the case of this example a return line filter of Type RF BN/HC 110 G 005 C 1.X has been selected.

According to *Diagram 50* the pressure drop across the Type RF 110 filter housing at an effective volumetric flow rate of 55 L/min is 0.18 bar.

According to *Diagram 51* the pressure drop across the Type 0110 R 005 BN/HC clean filter element at an effective volumetric flow rate of 55 L/min is 0.7 bar.

Calculate the total pressure drop:

$\Delta p_{tot} = (\Delta p_H + f_1 \cdot \Delta p_E) \cdot f_2 = (0.18 + 1.5 \cdot 0.7) \cdot 1.0 = 1.23$ bar

The calculated figure of total pressure drop is greater than the permitted figure of 0.5 bar which means that the selected filter is too small. The calculation will have to be repeated for a larger size of filter.

Determining the filter size using the filter design diagram

Calculate the volumetric flow for the filter:

$Q_P = Q_W \cdot f_1 \cdot f_2 = 55$ L/min • 1.5 • 1.0 = 82.5 L/min

The required size of filter can now be read from *Diagram 52*. The point of intersection between Q_P (82.5 L/min) and the 5 μm line is close to Size 240.

Therefore, a return line filter of Type RF BN/HC 240 G 005 C 1.X is used as the working filter.

4.2 Determining the size of protection filter

This filter is installed directly upstream of the servo valve and must have no by-pass valve. The filter is fitted with an electric clogging indicator for monitoring the state of the element. The filtration rating is to be $\beta_5 \geq 100$.

Determining the filter size using the filter design diagram

Calculate the volumetric flow for the filter:

$Q_P = Q_W \cdot f_1 \cdot f_2 = 5$ L/min • 1.5 • 1.0 = 7.5 L/min

In *Diagram 53* the point of intersection between Q_P (7.5 L/min) and the 5 μm line is close to Size 30.

Therefore, a Type LF BN 30 G 005 C 1.X filter must be fitted upstream of the servo valve as the protection filter.

Example 2

The procedure for by-pass filters is explained by the following example:

Power unit data

Tank capacity: approx. 1000 litres
Hydraulic fluid: ISO VG 46
Operating temperature: 50°C
The unit incorporates 2 pumps each with a delivery of 100 L/min
The system is equipped with proportional valves.
The environmental conditions are average.
Continuous monitoring of the filter is provided.
The unit is operated for 7 hours a day and 5 days a week.

By-pass filter data

The by-pass filter can be operated continuously 24 hours a day, 7 days a week.

Procedure for filter selection

1. Determining the volumetric flow

$$Q_N = \frac{Q_P \cdot T_{TP} \cdot T_{PW} \cdot f_2}{T_{TB} \cdot T_{BW}} = \frac{2 \cdot 100 \cdot 7 \cdot 5 \cdot 1}{24 \cdot 7} = 41.6 \text{ L/min}$$

Use: 40 L/min

2. Determining the filtration pore size

According to *Section 5.6.2* the filtration rating of the by-pass filter should be one step finer than that given in *Table 30*. This means that a rating of 5 μm absolute must be used.

3. Determining the filter area

$$A = \frac{Q_N \cdot f_1}{q} = \frac{40 \cdot 1}{0.0035} = 11428 \text{ cm}^2$$

Results

The by-pass filter must carry a flow of 40 L/min and its area must be at least 11,428 cm².

The filtration rating must be 5 μm absolute.

Example 3

For power unit data see Example 2

By-pass filter data

For safety reasons it will only be possible to operate the by-pass filter unit while the power unit is working.

Procedure for filter selection

The contents of the tank must be circulated at least once every 30 minutes. For a tank capacity of 1000 litres, this means that the flow rate through the by-pass filter must be:

$$Q = \frac{1000}{30} = 33.3 \text{ L/min}$$

Use 40 L/min

Filtration pore size required

A filtration rating one step finer means that the rating must be 3 μm absolute.

Determining the filter area

$$A = \frac{Q_N \cdot f_1}{q} = \frac{40 \cdot 1}{0.0025} = 16000 \text{ cm}^2$$

Results

Although the flow rate through the by-pass filter in this example is the same as that calculated in Example 2, the filter efficiency of the element is better and the filter area greater.

Example 4

The procedure for designing filters for filtering fire-resistant hydraulic fluids is explained by the following example:

Power unit data

Fluid: HFC 46

Hydraulic power unit for a die-casting machine

Environmental contamination around the machine: High

Intermittent monitoring of the filter

Tank capacity: approx. 1000 litres

Operating temperature: 50°C

Effective volumetric flow: 80 L/min

The system is equipped with proportional valves.

The power unit is to be fitted with return line filters.

Determining the filter pore size

The use of proportional valves means that a filtration rating of $\beta_{10} \geq 100$ is needed.

Determining the filter area

$$A = 60 \cdot f_1 \cdot f_2 \cdot f_3 \cdot Q_W$$

Viscosity conversion factor f_1

For 46 mm²/s the factor f_1 is 1.5 *(see Diagram 49)*

Environmental factor f_2

According to *Table 32* factor f_2 is 1.7

Fluid density conversion factor f_3

According to *Table 35* factor f_3 is 1.27

Required filter area

A = 60 • 1.5 • 1.7 • 1.27 • 80
A = 15.544 cm²

Results

Filter RF BN/HC 1300 F 010 A 1.1/SO105 must be used.

7 Instructions for operation and maintenance

7.1 Operation

Temperature limits for hydraulic filters

Most hydraulic filters may normally be used at operating temperatures between -10 and +100°C although temperatures up to 120°C for short periods will cause no harm. Higher operating temperatures can damage the filter elements and seals and proper filtering can no longer be assured. When temperatures are low the materials of the housings and seals must be checked for suitability. Filter elements can be stored at temperatures down to -50°C.

Fire-resistant fluids

A higher concentration of contaminants must be anticipated with these fluids. Also, galvanizing is not allowed. Consequently, the filters used for these fluids require special attention such as larger sizes and different kinds of surface protection. See also *Section 5.7.*

Changing of filtration rating or element materials in existing systems

In such cases it must be remembered that the filter elements will clog more quickly due to the particles of dirt already in the circuit. Therefore, a shorter element service life must be anticipated when using finer filters. The use of mobile by-pass units is recommended when carrying out such conversions.

Suggested intervals for element changing

The elements used in hydraulic filters should be changed at the following intervals:

- whenever the clogging indicator incorporated in the filter is triggered
- after 1000 hours of service or 1 year
- whenever the fluid in the whole system is changed.

7.2 Notes for the manufacturers of hydraulic power units

Satisfactory operation of hydraulic power units requires attention to the following points when installing the filters:

- Allow adequate space for changing the elements so that the work can be performed more easily, more quickly, and without damaging the elements.
- Run the pipes on the power unit so that they do not obstruct element changing
- Position the filters in easily accessible places on the unit. Correct positioning encourages proper maintenance. Allow adequate height for removal.
- Note the direction of flow marked on the filter housing.
- Use the N/C switching mode for the electric clogging indicator if possible. It makes interference with the system more difficult, e.g. removal of plugs, cutting of cables.
- Provide filling connections on the tank or upstream of the return line filter to make filling and topping-up of the tank easier.
- Provide Minimess connections for taking fluid samples. In order to protect the filter element, a pulsation damper is also advisable if there are high pressure peaks or fluctuations in flow .
- Avoid negative pressure peaks on the filters because they damage the elements. They can be prevented by fitting a check valve between filter and valve or accumulator.

7.3 Maintenance of hydraulic filters

Filter elements are protected in sealed plastic bags to prevent contamination during storage and handling. The plastic should not be removed until immediately before the element is to be inserted into the filter housing.

Only elements made of wire mesh, braided mesh or metal non-woven material can be cleaned. Filter elements made of non-woven paper or non-woven glass fibre cannot be cleaned.

Procedure for changing a filter element

- When the clogging indicator is triggered, depressurize the filter or the half of the filter containing the clogged element.
- Unscrew the filter housing or remove the cover, ensuring that the thread is not made dirty. Rotating the cover of a return line filter by about 45° makes lifting it off easier.
- Remove the clogged element and examine the residue on its surface. This might give a clue to any damage which has occurred to components. In the case of return line filters ensure that the element is removed together with the catchment basket.
- The fluid left in the housing must be removed into a suitable container. It is very contaminated and must never be allowed to get back into the system.
- Clean the filter housing with a clean, lint-free cloth.
- Examine the seal on the filter housing or cover and change it if necessary.
- Smear the thread and sealing surfaces of the filter and the seal of the element with clean hydraulic fluid.
- Insert the new element after first checking the filtration rating.
- Screw on the filter housing or cover.
- Switch on the system or fill the filter housing with fluid and examine it for leaks.

Maintenance instructions for tank breathers

It is advisable to change the breather element every time the fluid is changed. In some cases the whole breather has to be changed. Others are fitted with a renewable element or cartridge.

7.4 Flushing the whole system

Flushing of the system is recommended:

- when commissioning a new system
- after repair work
- after the system has been opened up in any way, e.g. for fitting a new pump or new valve.

Procedure for flushing a system

Fill the system with cleaned hydraulic fluid.

For this it is better to use a fluid service unit incorporating a filter. It allows both the tank to be filled and its contents to be filtered continuously in a by-pass operation.

High-precision valves such as servo valves or proportional valves must be replaced by flushing plates or flushing valves before flushing takes place.

The system filters should be fitted with elements of the filtration rating specified for operation of the system.

If necessary, use elements with the same filtration rating as the system filters but which will only withstand a lower pressure differential than the working filters.

After the total quantity of fluid has been circulated between 150 and 300 times, examine it for solid particle contamination and either stop or continue flushing.

Throughout flushing pay extra close attention to the clogging indicators on the filters. The elements must be changed immediately if the indicators are triggered so have sufficient spare elements to hand.

8 Symbols and subscripts

Symbols

Symbols	Units	Quantity
Q	L/min, m^3/s	Volumetric flow
A	m^2, cm^2, mm^2	Area, Filter area
p	bar, N/m^2	Pressure
α	g	Dirt retention
ρ	kg/dm^3	Densitiy
τ	n	Operational time
q	$L/min/cm^2$	Specific area loading
n	rpm	Speed
ν	mm^2/s	Kinematic viscosity

Dimensionless symbols

Symbols	Quantity
f	Correction factor, Conversion factor
%	Per cent
t	Time, Flushing time
β	Beta value, Degree of separation
n	Number
M	Million
K	Thousand

Subscripts

Symbols	Quantity
1, 2, 3	Element number, Factor No.
X	Particle size
HF	Housing, with operating fluid
HR	Housing, reference data
R	Data sheet, Pump
O	Operating conditions
W	Effective (working)
Ad	Mechanical ratio
tot	Total
E	Element
H	Housing
D	Design, Area, System
N	Nominal flow, Nominal
TP	Hydraulic system per day
PW	Hydraulic system per week
TB	By-pass filter per day
BW	By-pass filter per week

Prefixes

Symbols	Quantity
Δ	Difference, drop

9 International standards

ISO 228	Pipe threads where pressure-tight joints are not made on the threads - Designation, dimensions and tolerances
ISO 1000	SI units and recommendations for the use of their multiples and of certain other units
ISO 3722	Hydraulic fluid power - Fluid sample containers - Qualifying and controlling cleaning methods
ISO 4021	Hydraulic fluid power - Particulate contamination analysis - Extraction of fluid samples from lines of an operating system
ISO 4402	Hydraulic fluid power - Calibration of liquid automatic particle-count instruments - Method using Air Cleaner Fine Test Dust contaminant
ISODIS 4405	Hydraulic fluid power - Fluid contamination - Determination of particulate contaminants by the gravimetric method
ISO 4406	Hydraulic fluid power - Fluids - Method for coding level of contamination by solid particles
ISODIS 6162	Hydraulic fluid power - Flange connections - Four-bolt split flanges rated for normal duty applications - PN 35 to PN 415 bar (PN 3,5 to PN 41,5 MPa) - Dimensions
DIN ISO 2941	Hydraulic fluid power; filter elements; verification of collapse/burst resistance
DIN ISO 2942	Hydraulic fluid power; filter elements; verification of fabrication integrity; identical with ISO 2942, edition 1985
DIN ISO 2943	Hydraulic fluid power; filter elements; verification of material compatibility with fluids
DIN ISO 3723	Hydraulic fluid power; filter elements; method for end load test; identical with ISO 3723, edition 1976
ISO 3724	Hydraulic fluid power - Filter elements - Verification of flow fatigue characteristics
ISO 3968	Hydraulic fluid power - Filters - Evaluation of pressure drop versus flow characteristics
ISO 4572	Hydraulic fluid power - Filters - Multi-pass method for evaluation filtration performance
ISO 5598	Fluid power systems and components - Vocabulary
DIN ISO 2909	Petroleum Products; Calculation of Viscosity Index from Kinematic Viscosity
DIN 24312	Fluid power systems and components; pressure; quantities, terms
DIN 24550	Fluid power; hydraulic filters; definitions, nominal pressures, nominal sizes, fitting dimensions
DIN 51519	Lubricants; ISO viscosity classification for industrial liquid lubricants
DIN 51562	Viscosimetry; measurement of kinematic viscosity by means of the Ubbelohde viscosimeter; micro Ubbelohde viscosimeter
DIN 51757	Testing of petroleum and related materials; determination of density
DIN 51777	Testing of mineral oil hydrocarbons and solvents; determination of water content according to Karl Fischer; direct method
CETOP RP 91 H	Fluids for Hydraulic Transmission - Mineral Oils Specifications
CETOP RP 92 H	Statement of requirements for filters in hydraulic systems
CETOP RP 94 H	Determination of particulate matter in hydraulic fluids using an automatic particle size analyser employing the light interruption principle
CETOP RP 95 H	Recommended method for the bottle sampling of hydraulic fluids for particle counting
NAS 1638	Cleanliness Requirements of Parts used in Hydraulic Systems

Steelwork Design for Power Units

Hans H. Faatz

1 Introduction

A hydraulic power unit comprises an unpressurized fluid tank, a motor-driven pump, control gear, accessories and interconnecting pipework. The various sub-assemblies of the unit can be arranged either separately or together It is common practice to mount the motor-driven pump, control gear and accessories such as coolers, filters and accumulators on top of the tank or on its sides. The supporting structures required are generally made of weldable materials, usually steel, more seldom aluminium. Plastics have not yet come into use for the supporting structures of hydraulic power units. Although it might not actually be made of steel, the supporting structure made of weldable materials is usually known as the "steelwork". The same basic principles that are applicable to the construction of ordinary steelwork are also applicable to the steelwork for hydraulic power units so generalities have been avoided in this chapter.

Instead, close attention is given to the design of steelwork suitable for welding because of the considerable importance of this subject in connection with hydraulic power units. The special aspects of the design of steelwork for hydraulic power units referring to the individual sub-assemblies which form the unit are also dealt with.

2 Design of steelwork suitable for welding

Suitability for welding is an important factor even at the initial design stage of steelwork. This also includes the selection of steels suitable for the type of welding to be employed. Form, dimensions, manufacturing conditions and operating conditions of the steelwork must also be taken into account.

2.1 Welding instructions on the drawings

Drawings should show clearly the nature of the finished structure. The standardized symbols listed in DIN 1910 to 1912 allow the designer to impart the necessary information in a form of shorthand. Due to the strain on the weld seams the design must specify in each case the form of joint, the method of welding and, if necessary, the filler metal to be used. In the case of fillet welds, the thickness must also be stated.

The symbols for the weld must be entered against each seam. The thickness of fillet welds must also be shown. Methods and soundness for welds and also the classes of weld quality can be tabulated on the drawings.

2.2 Weldability

The materials that are to be welded must be suitable for welding. Most steelwork for hydraulic power units is fabricated from steels of quality RST 37.2 to DIN 17100.

Stainless steel tanks are made of X5CRNI189 or X10RNITI189, Material Nos. 1.4301 or 1.4541 to DIN 17440.

When special acceptance of materials is necessary the procedures must be agreed between contractor and principal.

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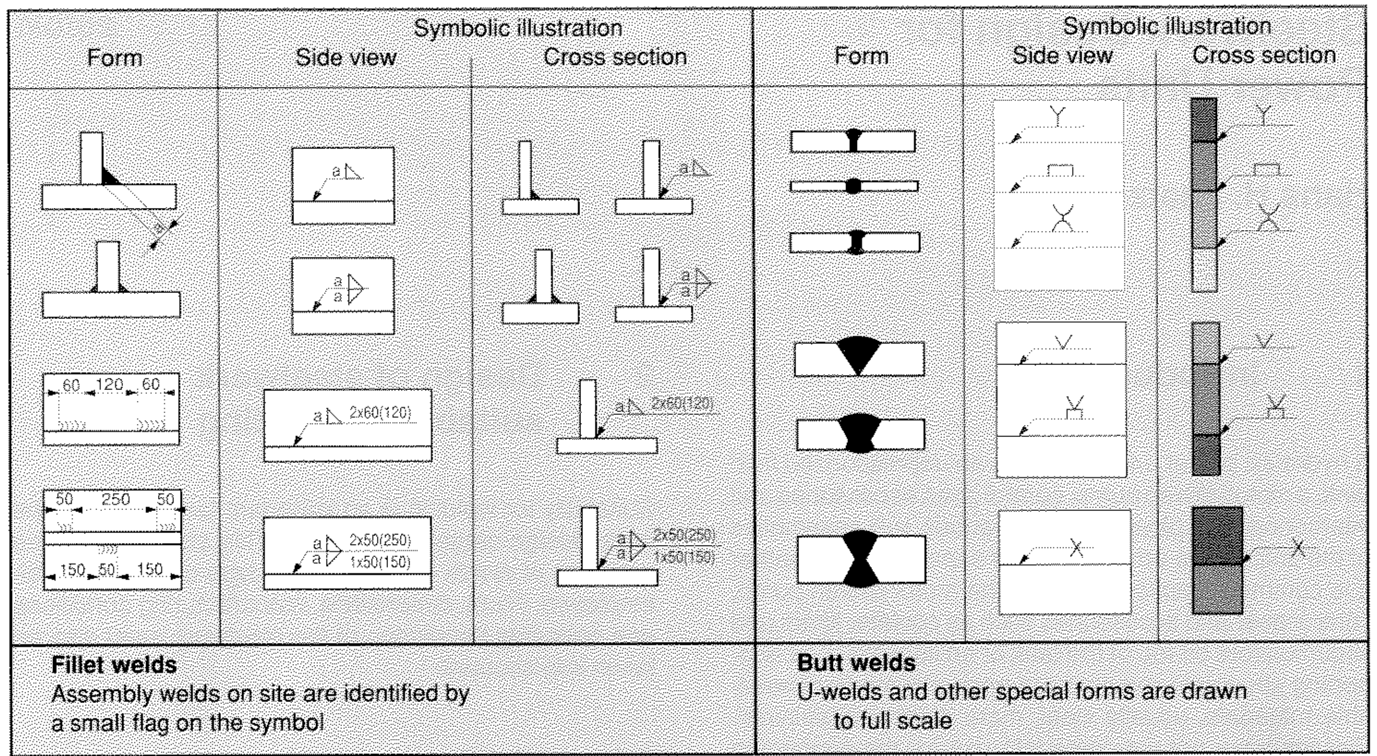
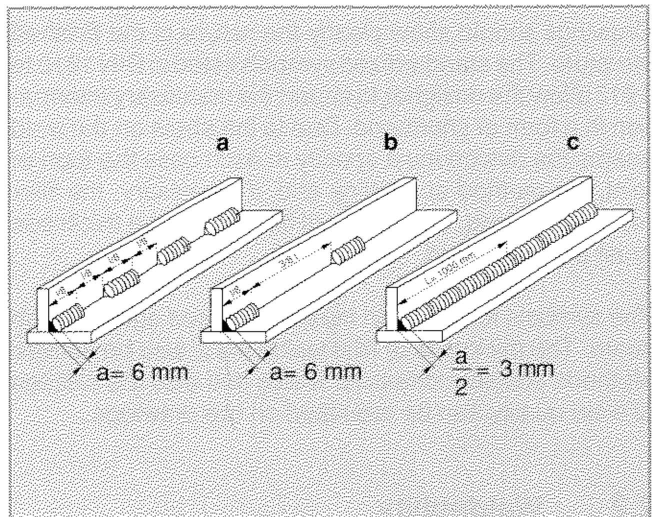


Fig. 98: *Symbols for workshop drawings for the principal forms of welds (further details will be found in DIN 1912)*



	Area under stress	Amount of weld material
a	$\frac{a\,l}{2} = 3000\ \text{mm}^2$	$\frac{a^2\,l}{2} = 18000\ \text{mm}^2$
b	$\frac{a\,l}{4} = 1500\ \text{mm}^2$	$\frac{a^2\,l}{4} = 9000\ \text{mm}^2$
c	$\frac{a\,l}{2} = 3000\ \text{mm}^2$	$\frac{a^2\,l}{4} = 9000\ \text{mm}^2$

Fig. 99: *Comparison of intermittent and continuous fillet welds*

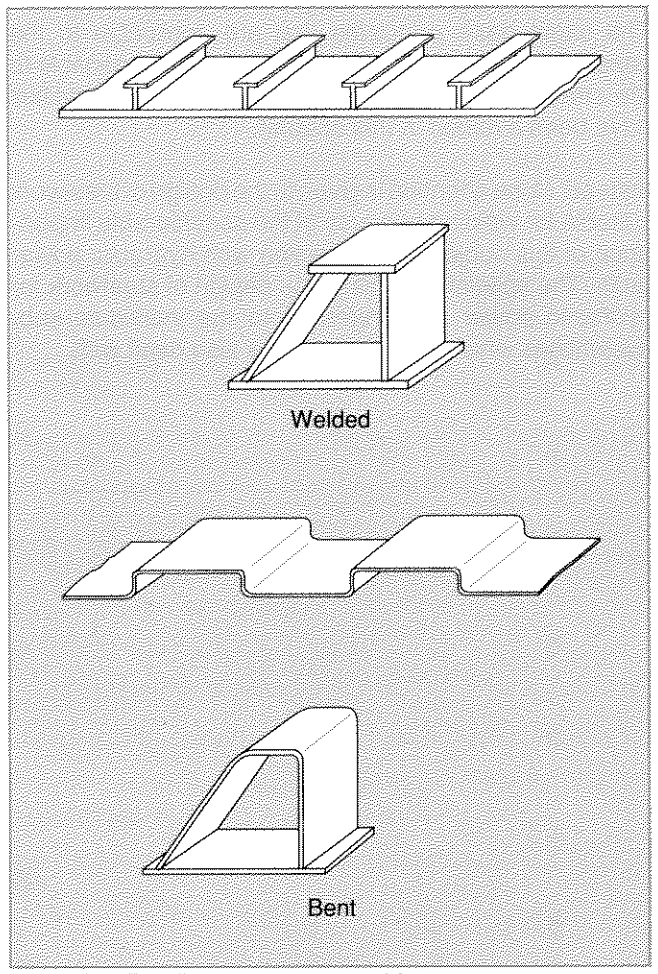


Fig. 100: *Bending can save on welds*

Thickness of workpiece	Form	Description	Symbol [1]	Section through groove	Dimensions: Degrees [2] α, β	Dimensions: Gap [3] b	Dimensions: Root face c	Throat depth h	Welding method [4]	Remarks
< 4	one side	Plain butt weld	||		=	≈ s	=	=	G, E, TIG [5]	=
					=	0 < s	=	=	MIG, MAG	Up to 8 mm with backing-up.
< 8	both sides [6]				=	≈ s/2	=	=	E, TIG [5]	=
					=	0 < s/2	=	=	MIG, MAG	=
3 < 10	one side	Single-V butt weld	V		≈ 60	0 < 3	=	=	G	With back-up seam and
3 < 40	or				≈ 60				E, TIG [5]	larger gap if necessary.
	both sides				40 < 60				MIG, MAG	
10 <	both sides [6]	Single-V butt weld with root faces	Y		≈ 60	0 < 3	2 < 4	=	E, TIG [5]	Also for thinner work and
					40 < 60				MIG, MAG	G in special cases.
12 <	one side or both sides [6]	Single-U butt weld			≈ 8	0 < 3	≈ 3	=	E, TIG 5) MIG, MAG	Root also with e= 4.6 + 0.14 s, when c= 4 mm and β= 8°.
30 <	both sides [6]	Double-U butt weld			≈ 8	0 < 3	≈ 3	≈ s/2	E, TIG 5) MIG, MAG	This type of groove can also have different throat thicknesses like the 2/3 double-V butt weld. e= 5 + 0.1 s, when c= 3 mm and β= 8°.

[1] For any supplementary symbols see DIN 1912, Part 5
[2] For welding in position q (horizontal in vertical plane) also larger and/or asymmetrical welds
[3] The given dimensions refer to the tacked state. The best gap depends on the welding position and method of welding
[4] E = Manual arc welding, G = Gas welding, MIG = Shielded-arc welding, MAG = Metal active gas welding, TIG = Inert-gas tungsten arc welding
[5] With TIG welding, protection against oxidation and backing-up with an inert gas such as anti-slag gas may be necessary (see DIN 32 526 in preparation)
[6] Root cut-out and backing run deposited if necessary

Table 36: *Types of grooves and chamfers for butt welds in steelwork (extract from DIN 8551, Part 1)*

2.3 Economy in the use of welding

Designers should always attempt to keep the number of seams, the seam cross section and seam length to a minimum in order to save unnecessary welding work. Cutting away corners in places where there are no forces to be transmitted can also save work and, therefore, money.

For butt weld preparation the type of groove must be shown on the drawing. The actual shape of the groove depends on the type and size of the workpiece, the welding process, the number of passes and the equipment. Shielded-arc welding (MIG, MAG) and manual arc welding are most common in the manufacture of steelwork for hydraulic power units.

In the case of fillet welds it can often be sensible to employ intermittent seams, although the operating conditions of the finished item must be taken into account. Intermittent fillet welds are not allowed on hydraulic power units used in civil engineering or inside tanks. In order to give an intermittent fillet weld the same fracture area as a continuous weld, the seam must be twice as thick if the interruptions are as long as the bead. But this is not an economical solution to the problem. Therefore, intermittent seams should only be used where there are no large forces to be transmitted, such as with stiffeners. In this case the intermittent welding has the advantage that shrinkage and distortion is less.

As calculations for strength of steelwork are seldom carried out for hydraulic power units, the designer must specify the minimum weld thickness from experience. Weld seams should always be easily accessible. Special care must be taken at the design stage when welding together bent plate.

Weld seams should also always be positioned at points of minimum stress and variations in cross section between the parts should be avoided.

Box sections are to be preferred over flat sections because of their greater static and dynamic stiffness. In the case of hydraulic power units, however, it is important for the box to remain totally closed so that no corrosion can occur inside.

The designer must see whether welds can be saved through bending or folding *(Fig. 100)*. Bending is generally cheaper and stronger than welding.

2.4 Ensuring soundness of welding

The soundness of welds can be assessed according to DIN 8563, Part 3. This DIN standard divides butt welds and fillet welds into qualification classes, giving details of how permissible external and internal faults are to be assessed. Qualification classes DS and CK are sufficient for hydraulic steelwork.

3 Design of hydraulic power units

The design of hydraulic power units must take into account the special needs of fluid technology. The arrangement of the equipment must promote easy maintenance and there must be good accessibility to threaded pipe fittings. The relevant standards such as DIN 24 346, special company specifications and component manufacturers' maintenance instructions must also be adhered to.

In order to allow economic manufacture, sub-assemblies or parts or parts of sub-assemblies should be standardized in company specifications.

Hydraulic power unit design should be based on the model hydraulic circuit diagram and parts lists to DIN 24 347 .

The hydraulic circuit diagram to DIN 24 347 indicates:

- the energy flow of the hydraulic fluid
- the pressure settings
- the pipe sizes.

The parts list must contain all the components in the hydraulic circuit diagram with precise details of type, supplier or manufacturer.

3.1 Procedure

The basic guidelines of design theory also apply to hydraulic power units. The method described in VDI 2221 can therefore, for example, also be applied *(Fig. 101)*.

There are often numerous aids available to the designer such as stick-on symbols for pumps, motors and valves and template drawings for tanks and motor/pump units. The gradual spread of CAD systems is also proving of great benefit to hydraulics designers, especially systems that will run on medium-size computers or personal computers.

3.2 Special aspects of hydraulic power unit design

In addition to the standard rules of design theory there are also a number of special aspects to be taken into account in the design of hydraulic power units. They vary according to the application and type of equipment and are dealt with separately in each case.

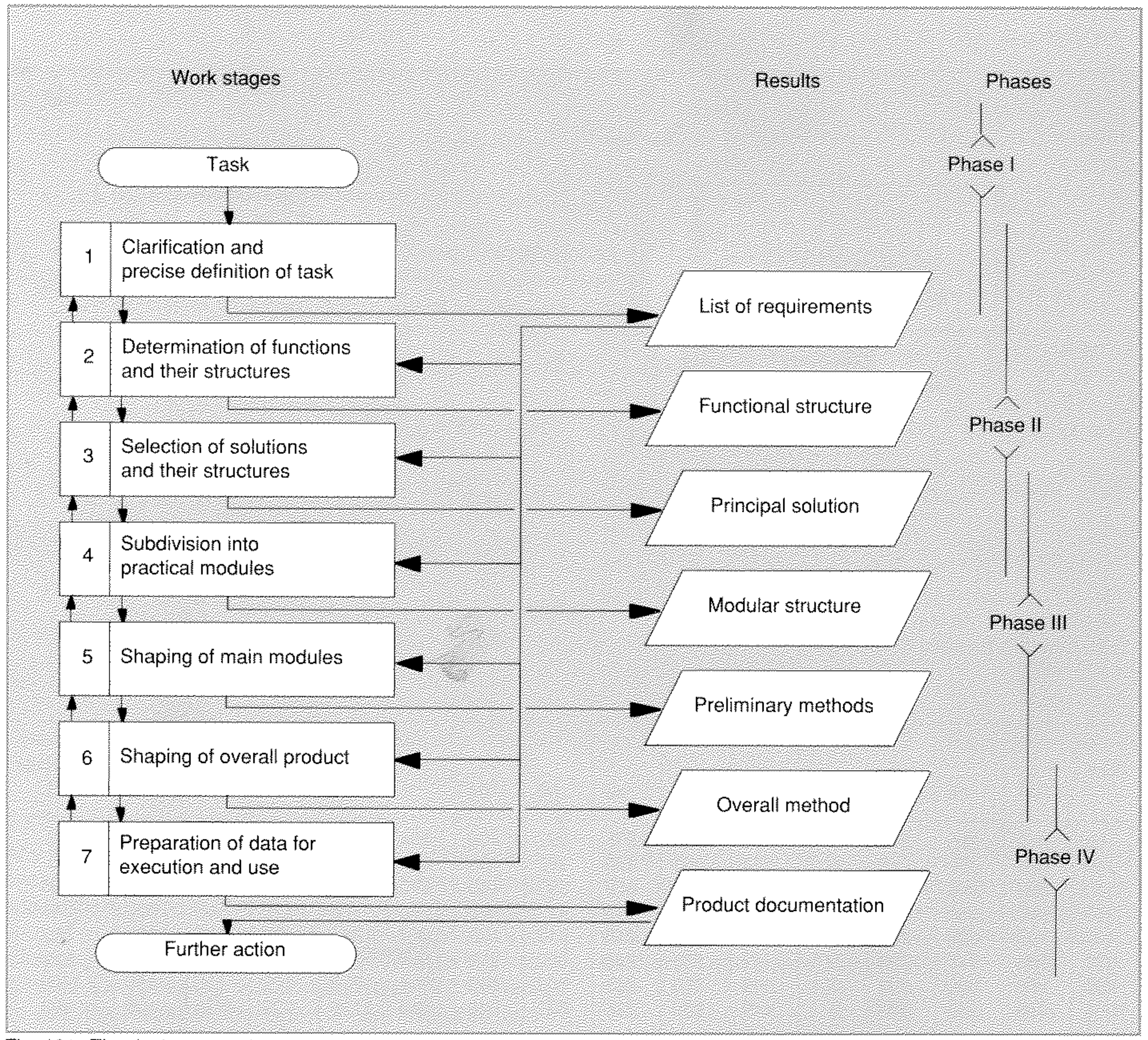


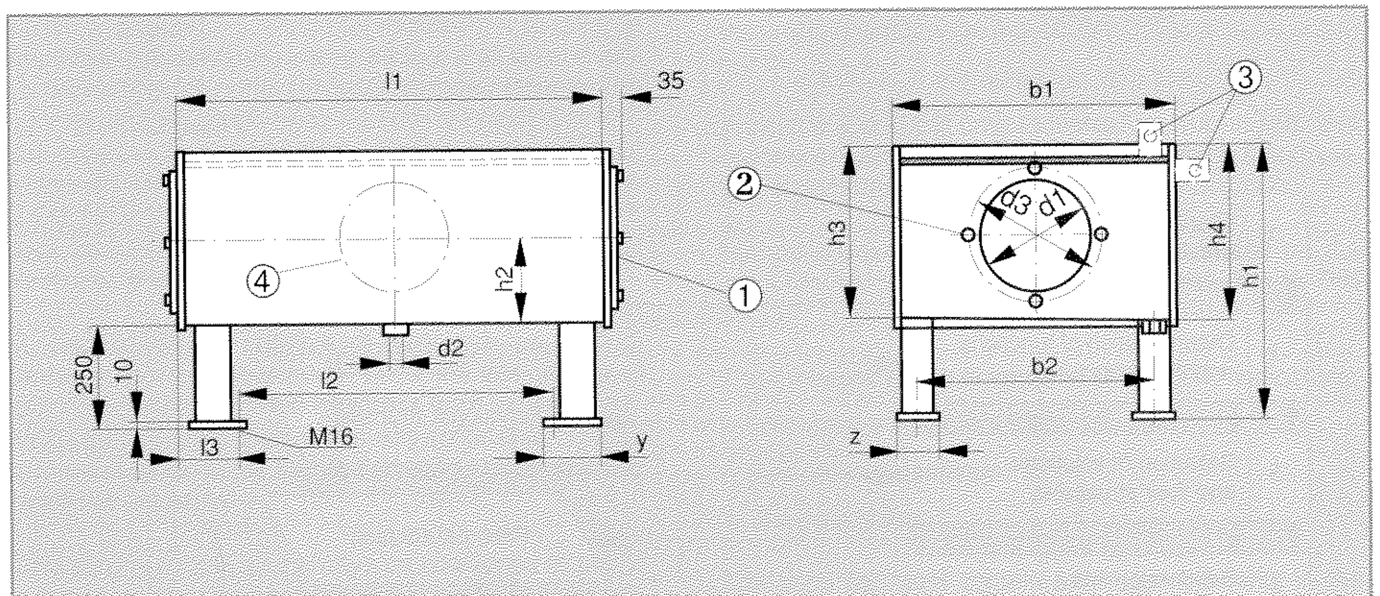
Fig. 101: *The design procedure*

4 Sub-assemblies

4.1 Oil tanks

Unpressurized oil tanks (generally termed oil tanks although often containing other operating fluids) are normally used in hydraulic systems. They should be of sufficient size to accommodate all the fluid in the system when there are no devices in the circuit, such as automatic check valves, which prevent back-flow of the fluid to the tank. The tank volume should be at least equal to or greater than three times the delivery/minute of the hydraulic pumps. This applies to systems operated with mineral oil-based fluids. If a different type of fluid is being used, such as a fire-resistant fluid, the tank volume must be 5 to 8 times the flow/minute of the hydraulic pump, depending on the air-separation and dirt-settling characteristics.

Aluminium is normally used for tanks of up to about 63 L capacity. Above that size they are made of steel plate. For machine tools the steel tanks are rectangular and comply to DIN 24 339 which also standardizes the tank connections. Rectangular tanks with flat reinforced or corrugated sides are used on presses and foundry machinery. Steelworks and rolling mill installations normally use cylindrical tanks.



1 Two for sizes 160 to 1000
2 6 Fixing bolts from size 630
3 Lifting lugs, 50 dia.
4 One only for sizes 63 and 100

Size	b1	b2	d1	d2	d3	h1	h2	h3	h4	l1	l2	l3	y	z	Effective capacity in L
63	365	285	248	G 1	324	660	205	397	409	508	308	100	115	80	60
100	460	360	248	G 1	324	660	205	397	409	633	393	120	135	100	95
160	590	490	248	G 1	324	660	205	397	409	810	570	120	135	100	156
250	690	590	248	G 1	324	680	215	417	429	1010	770	120	135	100	241
400	735	635	248	G 1	324	680	215	417	429	1514	1274	120	135	100	387
630	945	845	383	G 1	449	770	265	502	519	1514	1274	120	135	100	622
800	900	800	383	G 1	449	770	265	502	519	2014	1774	120	135	100	790
1000	1065	965	383	G 1	449	800	285	527	549	2014	1774	120	135	100	993

Fig. 102: *Principal dimensions of rectangular oil tanks in steel similar to DIN 24 339*

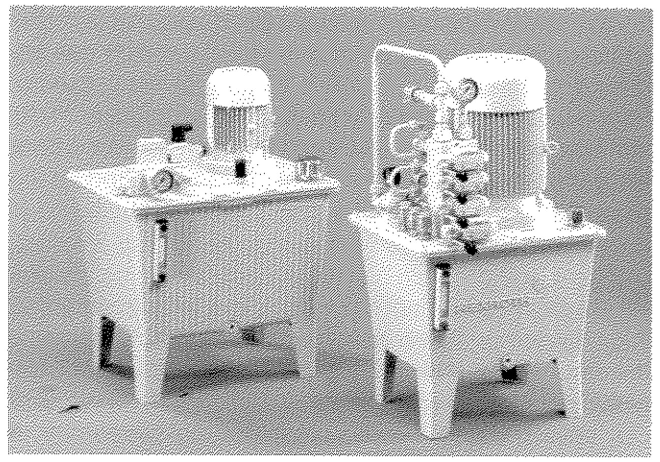


Fig. 103: *Aluminium tanks*

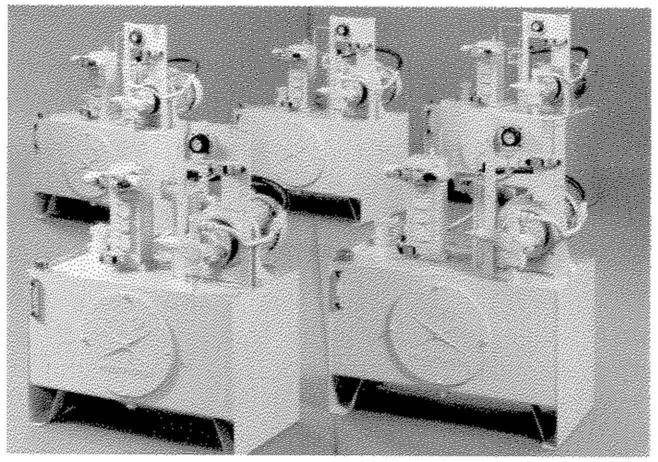


Fig. 104: *Steel tanks*

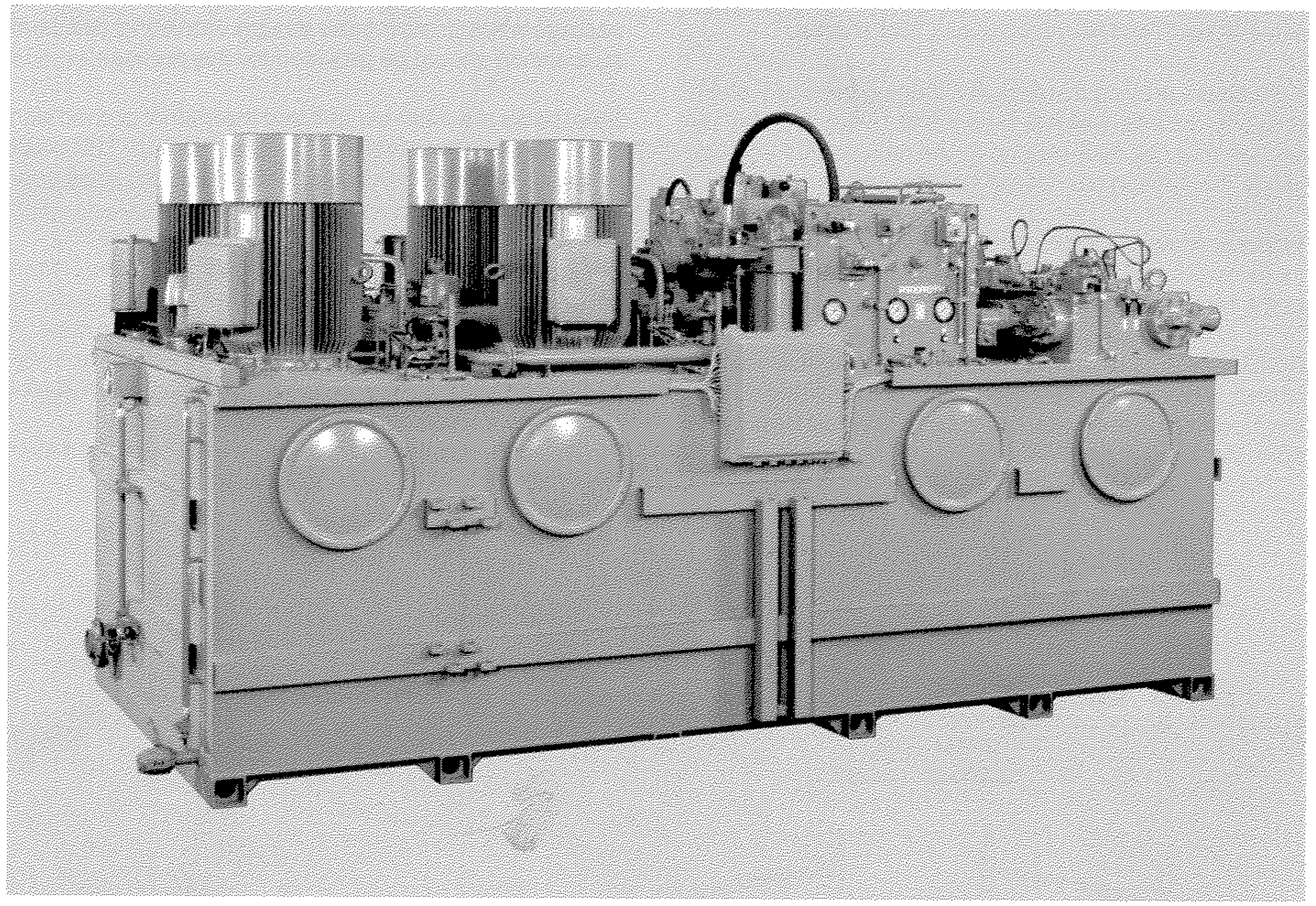


Fig. 105: *16 000 litre tank*

Any air entrained in the hydraulic fluid should be able to escape from it within the tank. Suction and return lines should therefore be kept as far apart as possible. The flow velocity in the tank itself should be as low as possible to allow any dirt suspended in the fluid to settle out. Access holes of adequate size must therefore also be provided, so that the inside of the tank can be inspected and cleaned from time to time.

In the case of tanks with a capacity over 1000 L it is a good idea to have a fully welded separator plate with an overflow between the suction and return chambers. In such cases, of course, the two chambers would have to be drained and cleaned separately.

Instead of this type of separator plate it is also possible to have a permeable separator which can be made of expanded metal. Placing this type of separator at an angle improves the air separation. Even with this design there should at least be a fully welded separator plate in the lower part of the tank in order to prevent dirt being carried from the return chamber along to the suction chamber.

The tank itself is often used as the supporting structure for other components such as the complete motor-driven pump set and/or the control gear. This sometimes makes it necessary to stiffen the sides of the tank. The simplest way of doing this is by means of pressed corrugations which have the advantage of needing no welding.

Sometimes it is necessary to weld in stiffening plates. In their simplest form they are flat plates welded to the tank sides at each end with continuous fillet seams.

If flat plates provide insufficient stiffening the usual alternative is to use welded U-sections. Here too the U-section must be seal-welded to the tank so that any dirt inside the section cannot get into the hydraulic fluid.

When welding in stiffening ribs take care to avoid accumulations of seams and dirt traps in corners. The interior of the tank must always be easy to clean thoroughly.

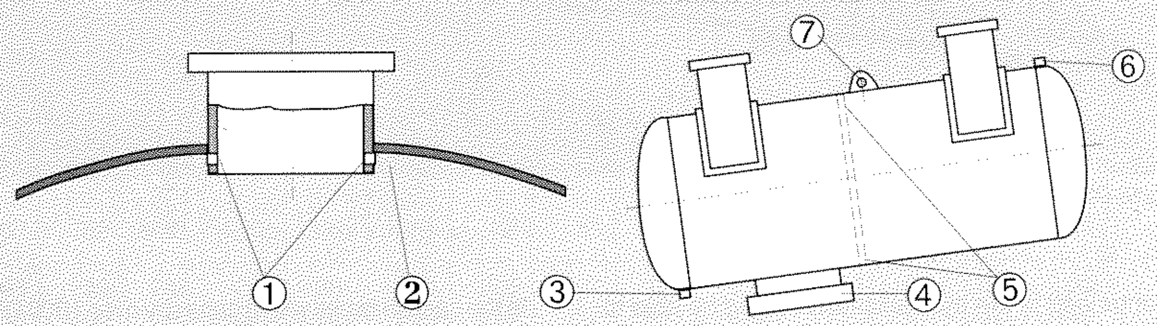
If tanks are to be hot-galvanized, good accessibility to the interior is essential. The interior must be ventilated during the process so vent holes and drain holes must be provided in the various parts.

Spray-galvanizing has only limited application to hydraulic tanks. Its advantage is that tanks can still be galvanized although they will not fit into a galvanizing bath. However, the zinc can only be sprayed on to bright metal surfaces which means that parts to which access is poor will generally receive insufficient galvanizing and, at the other end of the scale, the zinc peels off easily if it is applied too thickly.

When galvanizing hydraulic tanks in a bath ensure that the bath temperature is approximately 480°C because this will relieve any welding stresses in the steel plate which could otherwise cause distortion of the tank. For this reason alone it can sometimes be necessary to stiffen the tank sides. The zinc in the tank and in other voids must be able to run away quickly and without any major drop in temperature. The size of galvanizing bath available and the method of dipping must be taken into account when finalizing the design. Additional points of suspension should be provided for this purpose.

Steel tanks are sand-blasted and the interior primed with zinc-rich paint before the cover is welded on. This paint is resistant to most hydraulic fluids and also offers adequate protection against corrosion.

For systems containing servo valves the tanks are often specified in stainless steel. The same design principles apply to these tanks as to those made of carbon steel. A careful check should be made to see whether the thickness of the sides can be reduced. Welds between austenitic steels and carbon steels should be avoided.
Tanks made of stainless materials should be pickled after manufacture and no painting is necessary.



1 Drain holes
2 Critical zone
3 Extra drain
4 Main drain
5 Apertures in separator plate
6 Vent
7 Lifting lug

Fig. 106: *Additional features for galvanizing a cylindrical fluid tank in a bath*

With larger tanks the design must also take into account the fact that their length will change with variations in temperature. Therefore, cylindrical tanks must have one fixed support and one sliding support *(see Fig. 107)*.

Hydraulic tanks must not only allow none of their contents to escape outwards, they must also prevent any dirt gaining access to them. This is particularly important in the case of rectangular tanks having the top part in the form of a drip tray. Any necessary openings through the tank, e.g. pipe connections, filter connections and pump connections, must be carefully sealed and tightened. Rubber-cork compounds are suitable as sealing materials. They must be flexible and/or thick enough to compensate for any small unevenness in the structure and must be resistant to the particular fluid being used. Sealing surfaces must be flat and clean and the pitch of the fixing bolts must be close enough to ensure a good seal.

Capacity L	d	l1	l2	b1	b2	a1	a2
1000	1000	1510	765	200	150	750	600
1500	1000	2050	1400	200	150	750	600
2000	1250	1830	1100	200	150	950	800
3000	1250	2740	1920	200	150	950	800
4000	1250	3490	2740	200	150	950	800
4000	1600	2230	1280	350	300	1200	1050
5000	1600	2820	1770	350	300	1200	1050
6000	1600	3260	2250	350	300	1200	1050
7000	1600	3740	2770	350	300	1200	1050
10000	1600	5350	4290	350	300	1200	1050
13000	1600	6960	5625	525	475	1150	1000
16000	2000	5550	4210	600	550	1750	1600
20000	2000	6960	5395	600	550	1750	1600

1 Fluid level gauge
2 Filler and breather
3 Overflow
4 Lifting lug
5 Inside diameter 500
6 Drain cock G2
7 Fixed support
8 Sliding support

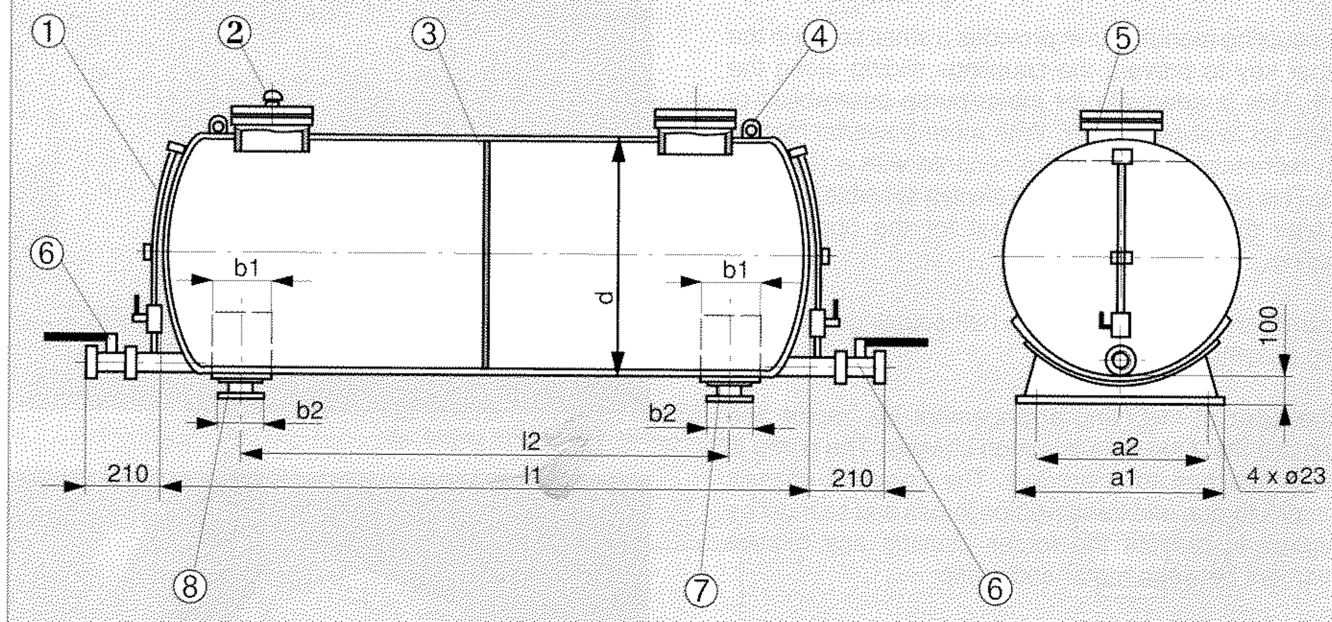


Fig. 107: *Principal dimensions of steel cylindrical fluid tanks*

4.2 Motor-pump sets

The motor-pump set of a hydraulic power unit converts electrical energy into hydraulic energy. Other common names for motor-pump sets are pump unit, pump set, motor-pump unit and others.

The designations for the mounting arrangement of the electric motor is also used for the motor-pump set itself, e.g. V1, B3, B5 or B3/B5.

In the case of types of mounting arrangements V1, B5 and B3/B5 the torque is taken by the bellhousing, which means that the torque forces are kept as short as possible.

Structure-borne noise is isolated by damping rings immediately behind the pump. The rings must be specifically designed for particular pumps.

The various components such as bellhousings with or without damping ring, foot brackets, couplings and mounting plates are normal commercial items.

Types of construction V1, B5 and B3/B5 are preferred because the inherent alignment of pump and motor shaft prevents any misalignment at the coupling and the tedious process of coupling alignment is eliminated.

The coupling manufacturer provides a small screw in the feather key to prevent the couplings sliding along the shaft. This is adequate for motor-pump assemblies of ratings up to about 15 kW. With more powerful motors both half-couplings should be secured with a retaining disc. This type of attachment is always sensible with the V1 type of construction, at least for the upper half-coupling.

Ensure that the gap between the half-couplings is correct when assembling the unit.

In the case of mounting arrangement B3, motor and pump are mounted on a common chassis. The pump is usually attached to a bracket. The common chassis transmits torque and transmission forces and so must be designed to prevent the forces causing any relative movement between motor and pump.

A simple frame made of flat steel plate is sufficient for electric motors up to Size 180. The mounting bracket can be welded to the frame.

For motor Sizes 200 to 315 the simple construction will have to be stiffened by additional welded angle sections or flat sections.

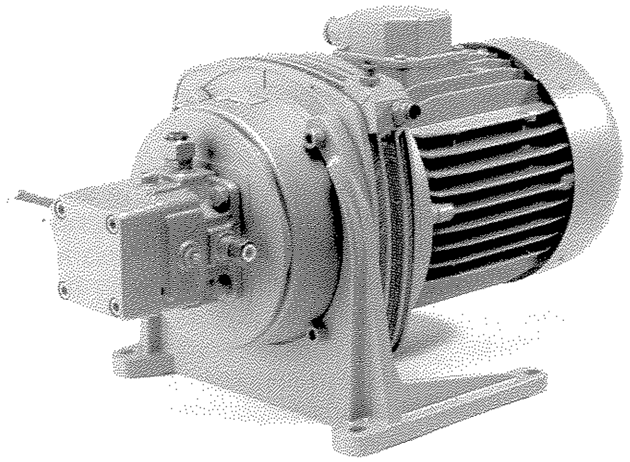


Fig. 108: *Motor-pump set*

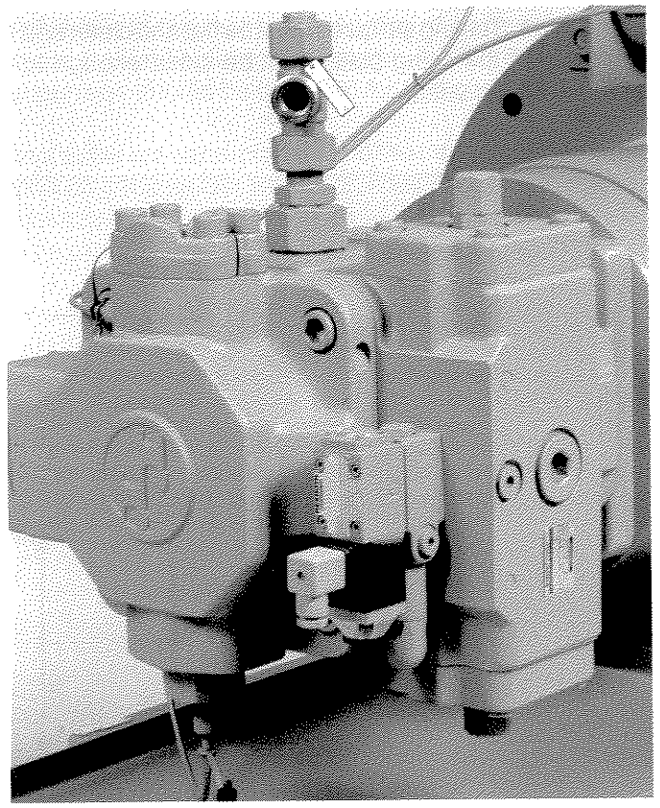


Fig. 109: *Motor-pump set*

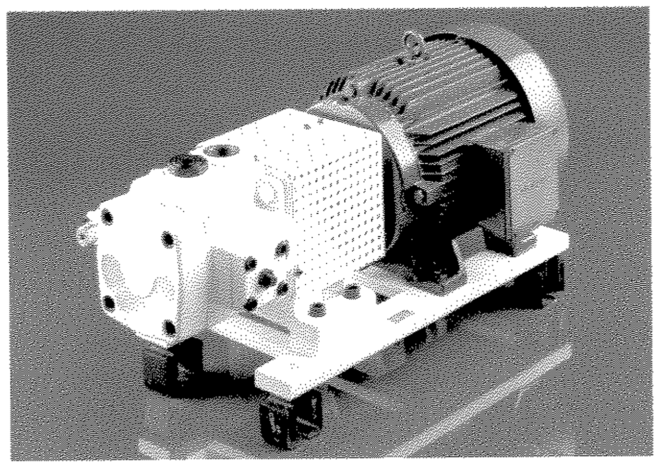


Fig. 110: *Motor-pump set*

Motors of Size 355 and larger need rigid base-frames fabricated from flat steel plate and U-sections. In this case it is better to bolt the mounting bracket for the pump on to the base-frame and not to weld it.

Shims can be used under the motor to adjust the height. Suitable lugs should be provided on the base-frame to prevent any lateral movement, at least over Size 200. The motor-pump sets often carry the pump control devices and/or pressure relief valves.

With the B5 and B3/B5 mounting arrangements, a mounting plate for attaching these devices can be secured to the motor fixing bolts.

With the V1 type of construction the devices can be placed on the mounting plate.

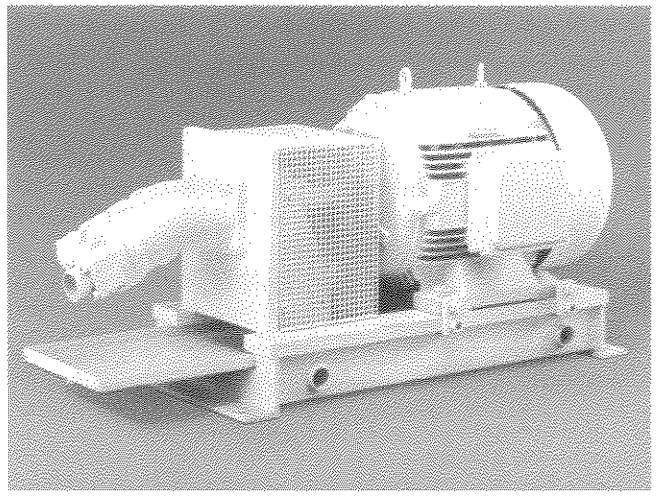


Fig. 111: *Motor-pump set*

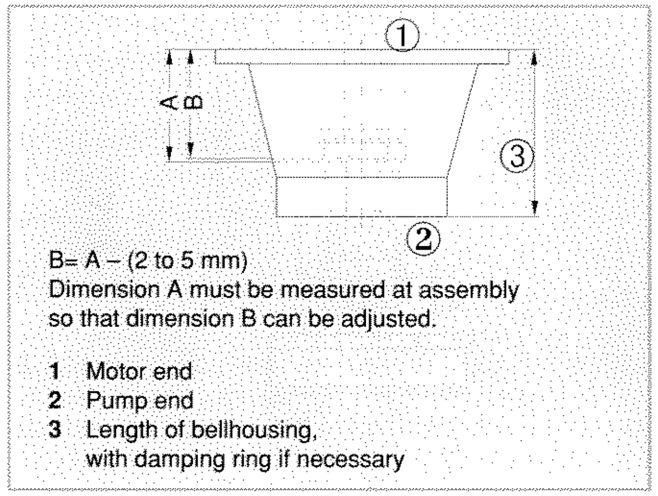


Fig. 112: *Bell housing*

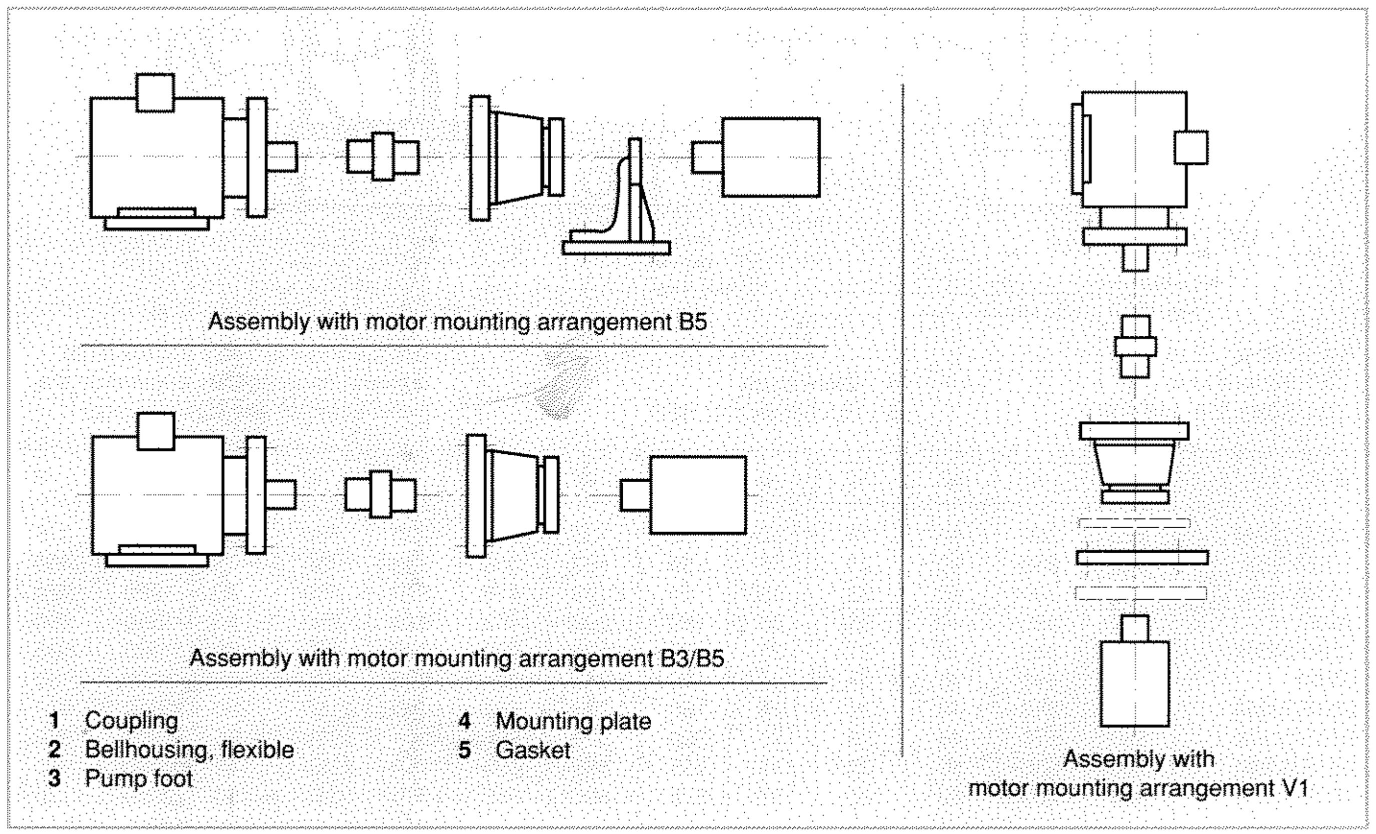
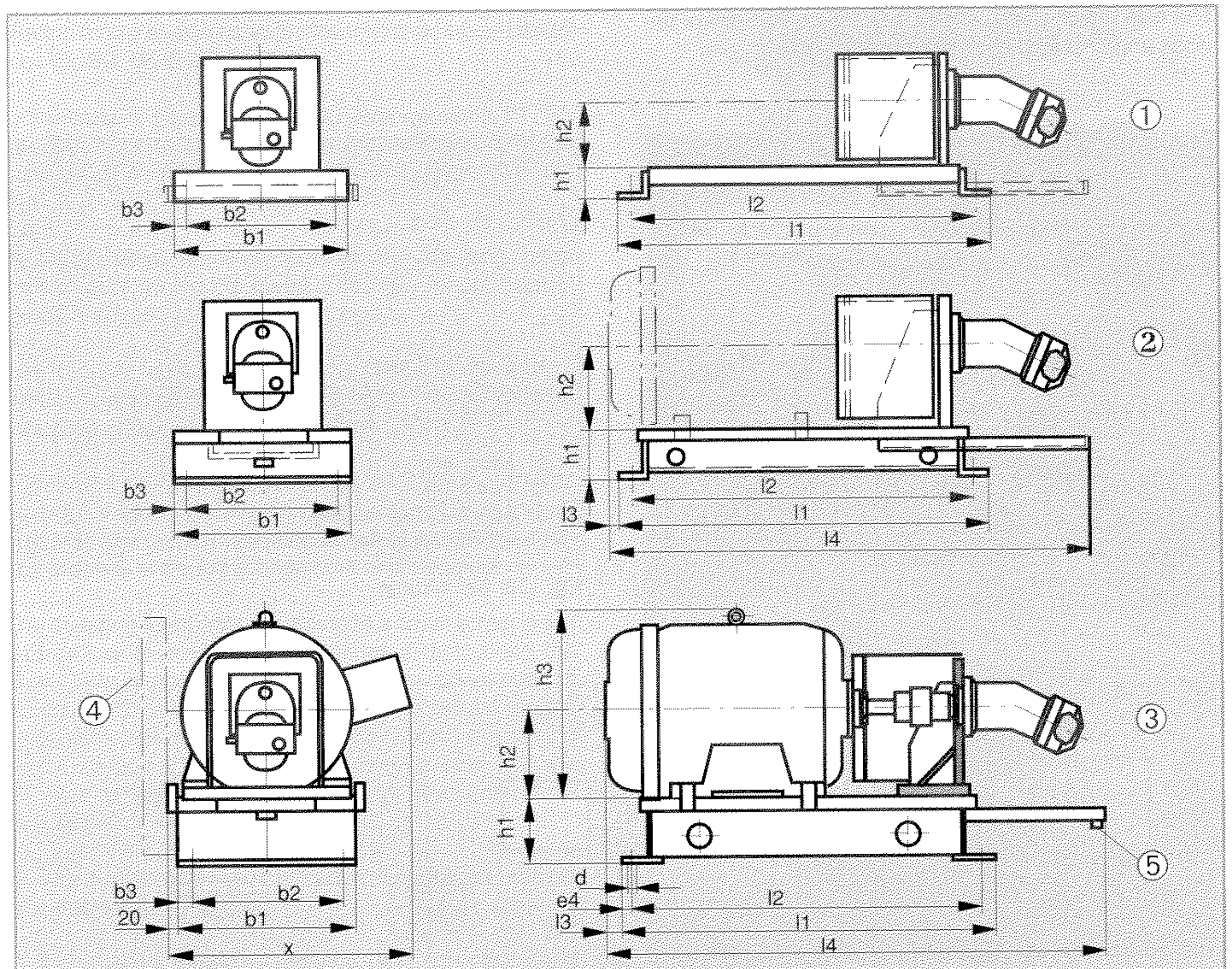


Fig. 113: *Assembly of motor-pump sets*

For mounting arrangement B3, a valve stack must be mounted on the base-frame.

Study each application individually to see whether there is any relative movement between pump and pipework which will make hoses necessary in suction and delivery lines and/or an expansion piece in the suction line.

Hoses are a good idea in the delivery and leakage-fluid lines in order to isolate any structure-borne noise. An expansion piece in the suction line is advisable.



1 Sizes 132 to 180
2 Sizes 200 to 315
3 Sizes 355 and 400

4 Valve stack
5 Fluid drain 1/4" BSP

(L3, L4 and x are only approximate dimensions since they depend on the make of motor and size of pump)

Size	b1	b2	b3	d	e	h1	h2	h3	l1	l2	l3	l4	x
132S/132M	280	200	40	18	30	55	135	308	720	660	65	900	375
160M/160L	330	250	40	23	35	55	163	368	820	750	75	1050	450
180M/180L	370	290	40	23	35	55	183	418	870	800	75	1200	520
200L	410	330	40	23	35	150	203	463	920	850	80	1350	590
225S/225M	460	380	40	23	35	150	228	520	1020	950	80	1450	630
250M	520	440	40	23	45	190	253	584	1090	1000	100	1520	730
280S/280M	620	540	40	23	45	190	283	664	1240	1150	100	1650	810
315S/315M	650	570	40	23	40	220	318	700	1360	1280	90	1800	870
355S/355M	750	650	50	23	35	255	355	795	1450	1380	140	2000	950
400S/400M	860	760	50	26	50	325	400	895	1750	1650	160	2250	1120

Fig. 114: *Principal dimensions of motor-pump sets with motor mounting arrangement type B3*

Steelwork Design for Power Units

4.3 Device mounting frames and front panels

There must be facilities for the permanent and secure mounting of the devices, valves and manifold blocks used for controlling hydraulic systems.

With hydraulic systems for machine tools, the control gear is often mounted on the tank together with the motor-pump assembly. In order to keep noise emission to a minimum, the noise-emitting surface should be as small as possible. It is therefore advisable for the control gear to be incorporated into a manifold block which is better mounted straight on to the tank without an extra mounting panel.

However, space reasons often prevent this arrangement. In order, therefore, to keep the noise-emitting area as small as possible mounting frames are utilised. These are basically a standardized frame made of square tube into which standard elements to accept components can be welded.

The size of the square tube depends on the size and weight of equipment to be supported:

- Square tube of 20 x 30 x 2 is sufficient for valves of Size 6
- Square tube of 30 x 60 x 3 is used for valves of Sizes 10 to 16
- Square tube of 60 x 60 x 4 must be used for valves of Size 25 and larger.

Angled struts must be used to brace the frame if the equipment is heavy.

If it is necessary to reduce the transmission of structure-borne noise and hence lower the noise level of the unit, the elements can be bolted to the frame with a damping material such as a rubber-cork composition between the two. The same applies to the fitting of the mounting frame on the tank.

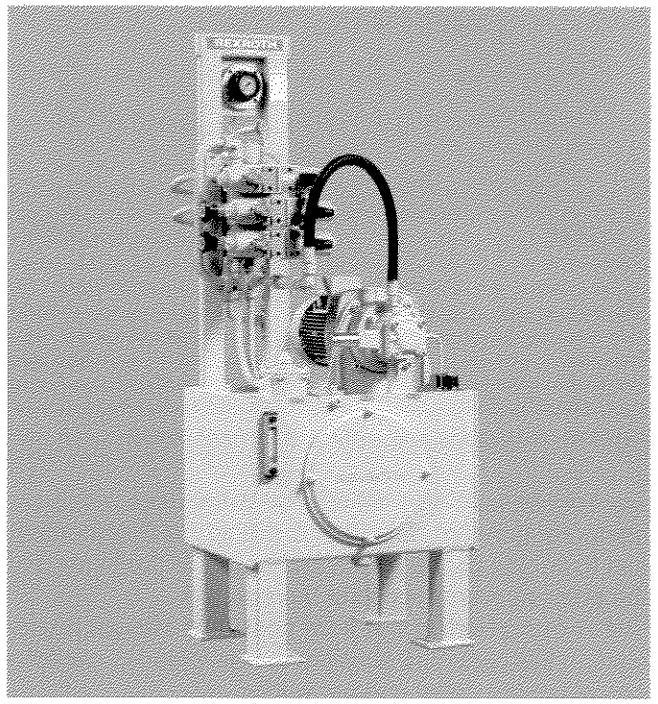
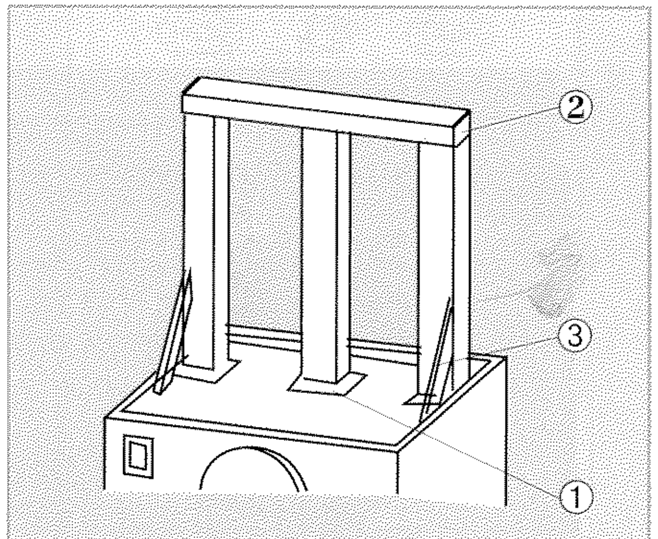


Fig. 115



1 Continuously welded round the circumference, a = 3 mm
2 Closed with plastic caps
3 Optional struts

Fig. 116: *The principle of the mounting frame*

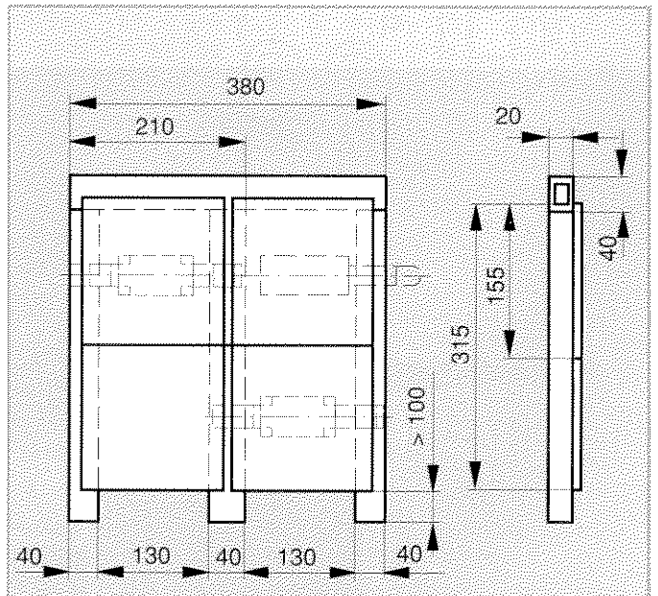


Fig. 117: *Twin-section mounting frame to accept Size 6 elements*

Access to pipe fittings is an important factor with hydraulic power units employing mounting frames. The elements should be placed behind the frame so that the devices lie between the square tubes. This can obstruct accessibility to the adjusting devices so the position of elements must be examined closely from case to case.

The mounting frame arrangement takes up more room than the front panel arrangement. The latter will have to be used if there is insufficient space to mount the equipment on a mounting frame.

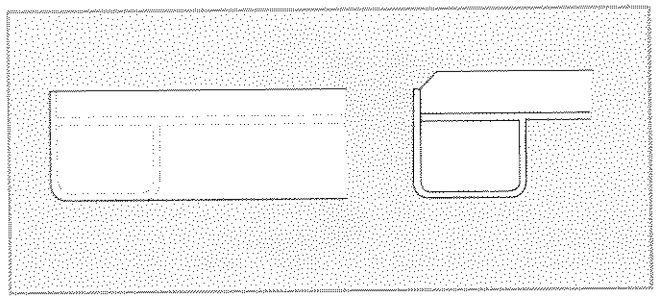


Fig. 118: *C-sections and stiffeners for front panels*

For front panels, also called mounting plates, there are company standards for cut-outs so that the designer only has to decide on the length and height of the panel and the position and direction of the cut-outs.
The rails supporting the front panel are also standardized. In this case it is sensible to use folded C-sections into which the panel can be welded.

Stiffeners may be needed for the mounting plate and braces for the tank depending on the size of front panel and weight of equipment mounted.

It is a good idea to provide lifting holes in the top part of the C-sections.

4.4 Valve stands

With hydraulic systems for large machines and installations the tanks, motor-pump sets and control gear are mounted separately because of their size. Whereas the tanks and motor-pump sets are mostly housed in specially built rooms (often basements), the control gear is best mounted as close as possible to the actuators, e.g. hydraulic motors or cylinders. In many cases the control gear is mounted and piped up on valve stands.

The size of a valve stand depends on the type and weight of the equipment to be mounted so only the supporting parts of it, such as the feet, drip tray and supporting elements can be standardized.

Once again, as with front panels, stiffeners and braces may be necessary.

Tubes welded into the side members are a good idea for lifting, as lifting from below is impossible.
The design must allow sufficient clearance between the components mounted on the stand so that they can be piped with U-bends.

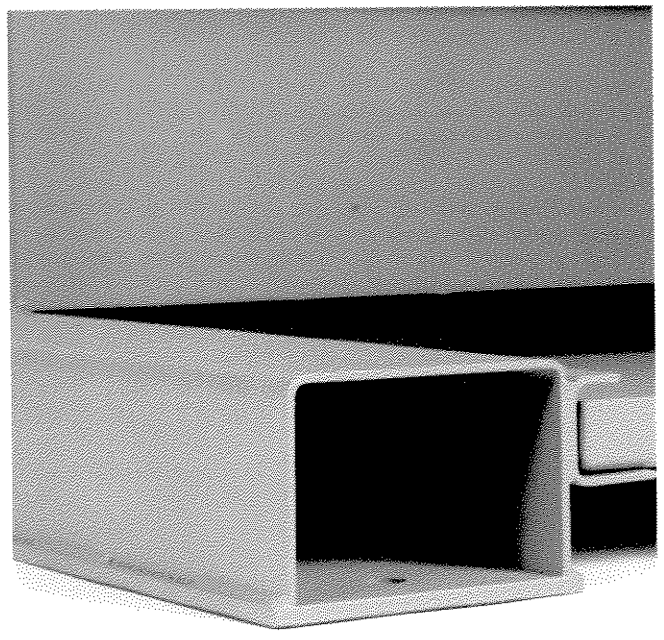


Fig. 119:

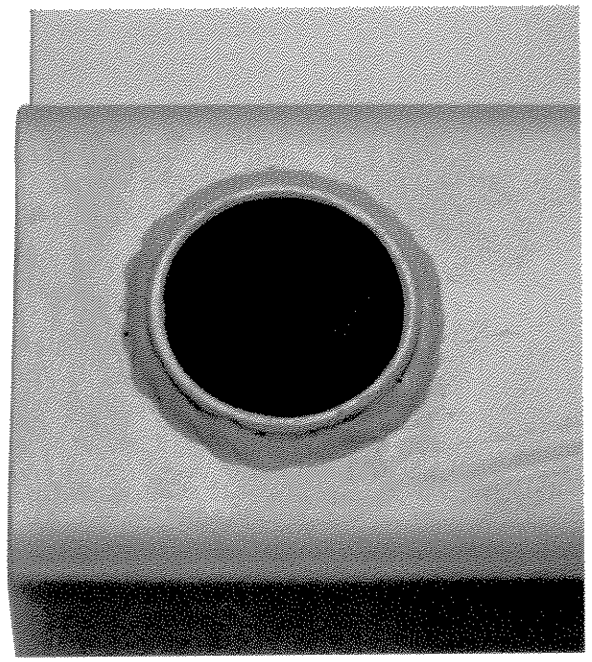


Fig. 120:

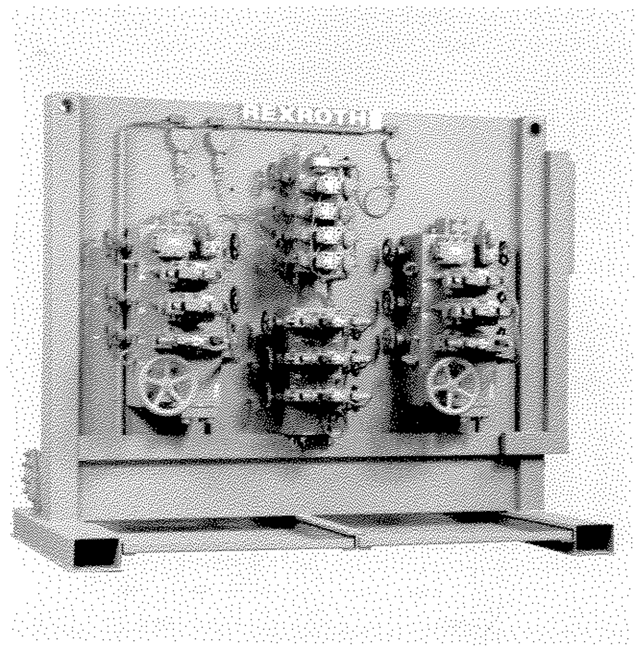


Fig. 121:

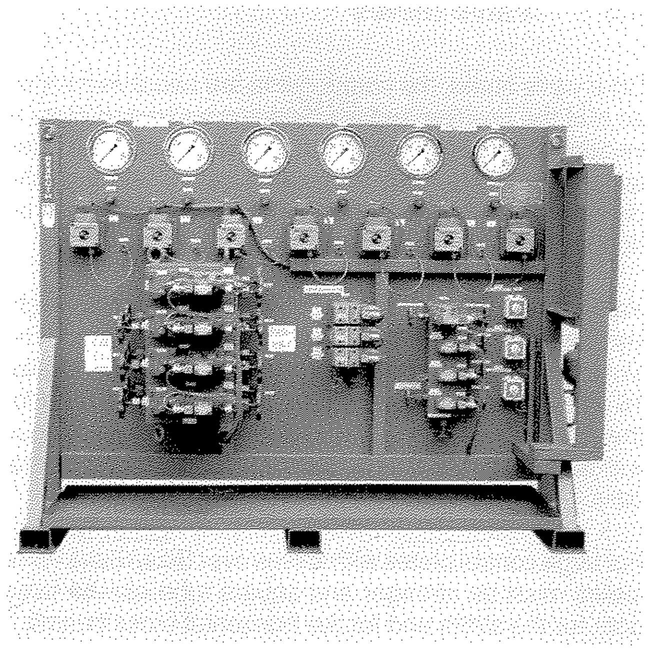
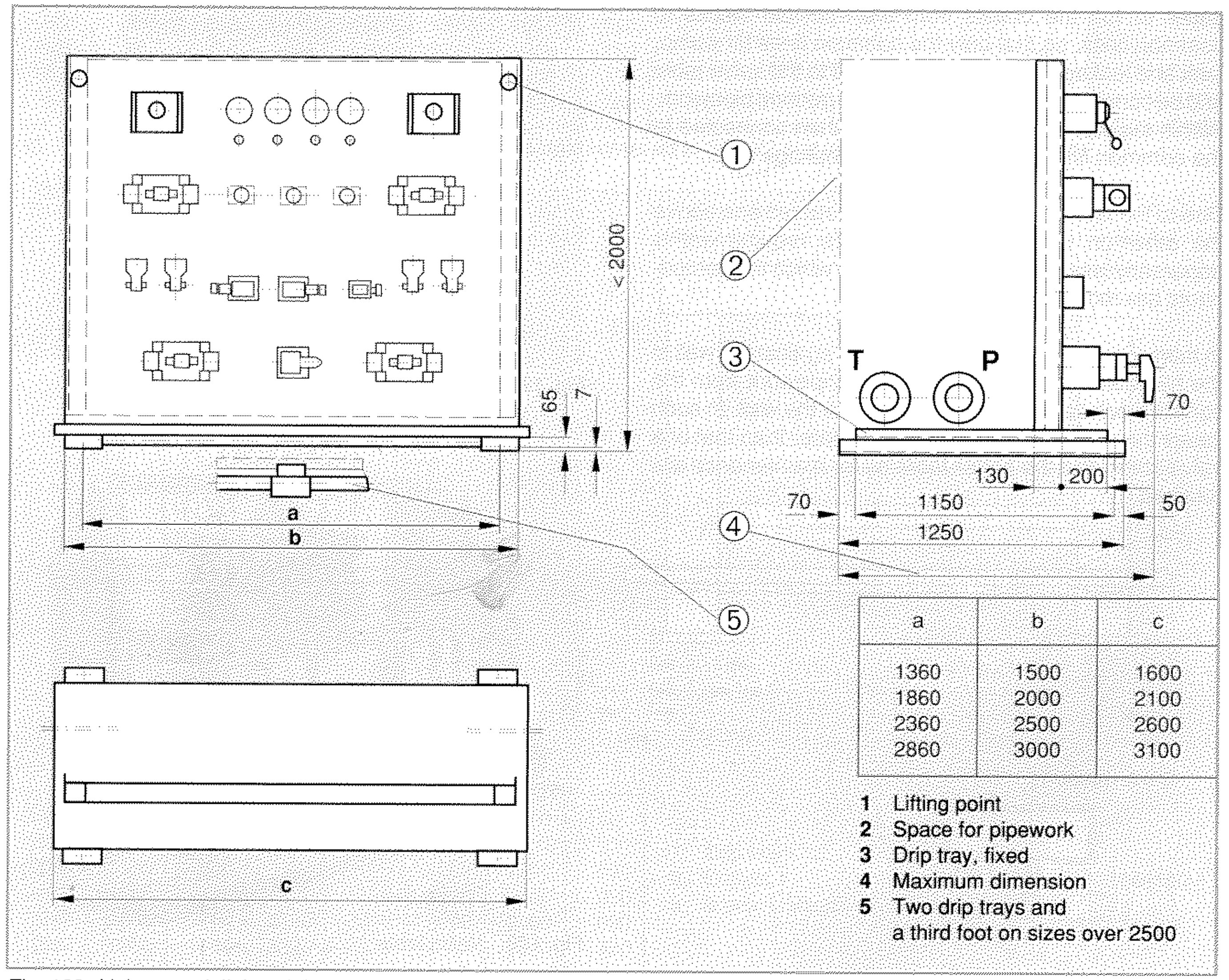


Fig. 122:



a	b	c
1360	1500	1600
1860	2000	2100
2360	2500	2600
2860	3000	3100

1 Lifting point
2 Space for pipework
3 Drip tray, fixed
4 Maximum dimension
5 Two drip trays and a third foot on sizes over 2500

Fig. 123: *Valve stand, lightweight type*

4.5 Valve tables

On a valve stand, the components are usually arranged so that the dividing surface between valve and plate is vertical. This is not always best for maintenance so valve tables are often used as an alternative. Most of the dividing surfaces between valves and plates can then be horizontal which makes maintenance easier.

Valve tables also come in lightweight or heavyweight versions depending on the weight to be supported. The simplest version for control plates with valves up to Size 16 uses angle sections both for the vertical feet and for the horizontal bearers. The table top can be continuous with suitable cut-outs or platforms between the horizontal bars.

For large control plates with valves over Size 16 the valve tables must be stronger and square tube can be used both for the struts and for the horizontal cross beams.

Valve stands are often designed complete with connections for outlet piping. In the case of valve tables it is better to run the outlets from the manifold blocks to common lines, so that only the common lines for pump, tank and leakage fluid have to be fitted to the valve table and be connected up later on site to the general pipework. Within the valve table the blocks are then connected to the common line for tank, pump and leakage fluid.

Due to the fact that components are mounted one above the other on valve stands the space requirement is apparently less than for valve tables in which the components are mounted side by side.

In many cases valve tables are placed against the wall in the hydraulics basement so that the pipework from the control block can run immediately on to the wall. In such cases the space requirement is no greater than that of a valve stand and the better facilities for maintenance are retained.

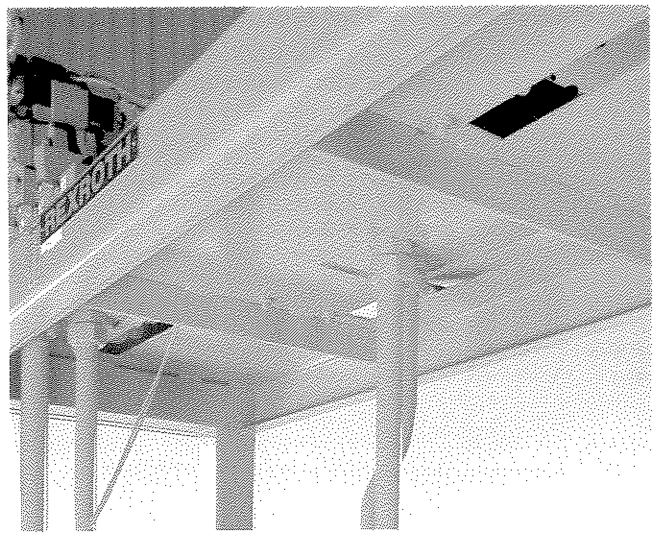


Fig. 124: Valve table

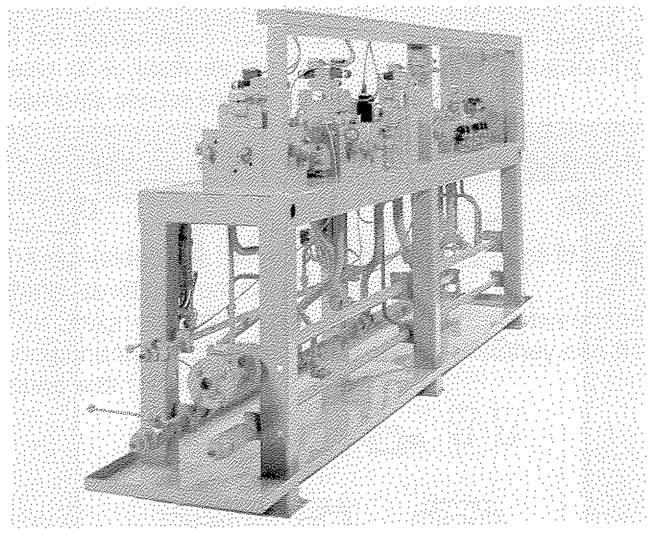


Fig. 125: Valve table

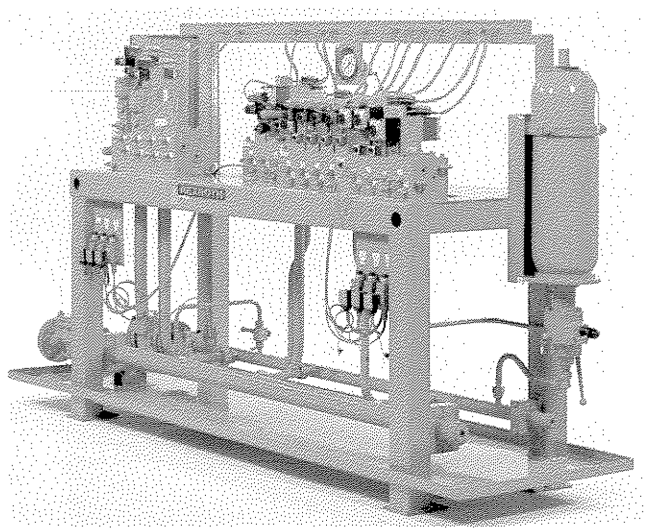
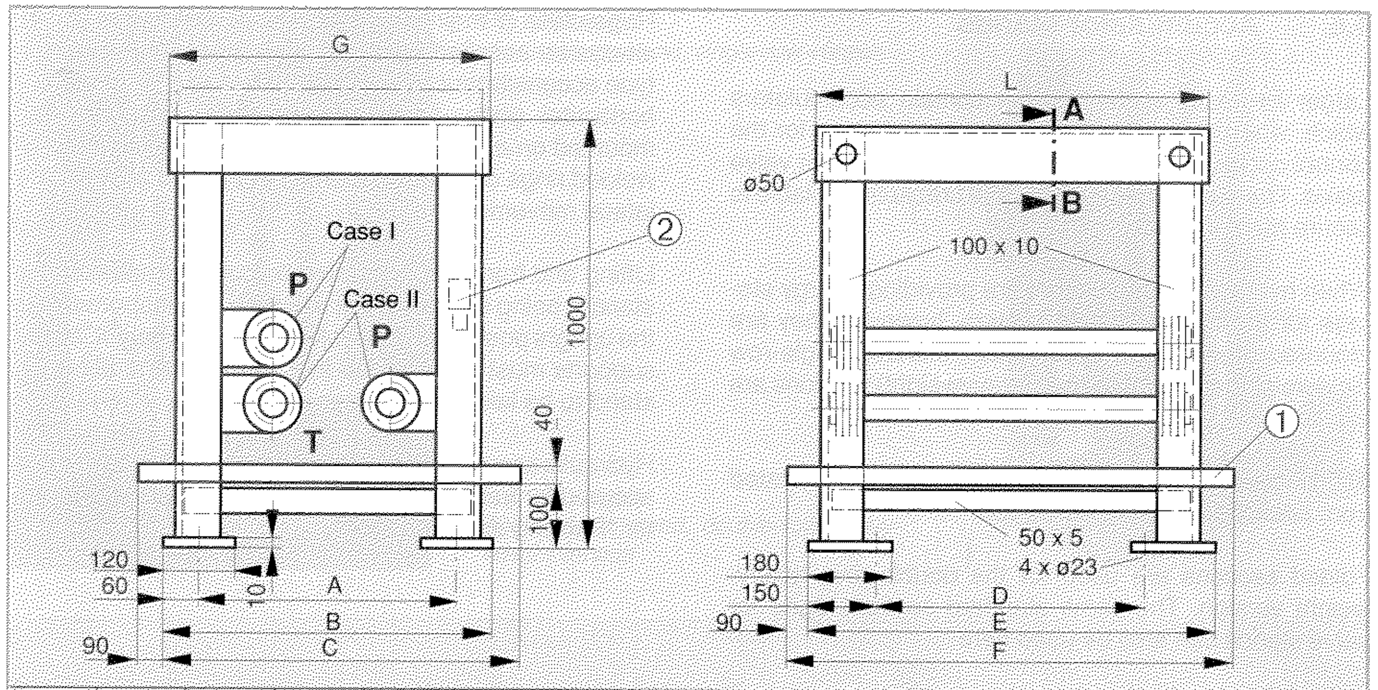


Fig. 126: Valve table



A	B	C	D	E	F	L	G
630	750	930	720	1020	1200	1000	750
630	750	930	1220	1520	1700	1500	750
630	750	930	1720	2020	2200	2000	750
630	750	930	2220	2520	2700	2500	750

1 Drip tray, 3 mm plate
2 Outlets

Section A—B

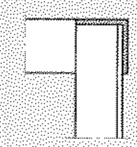
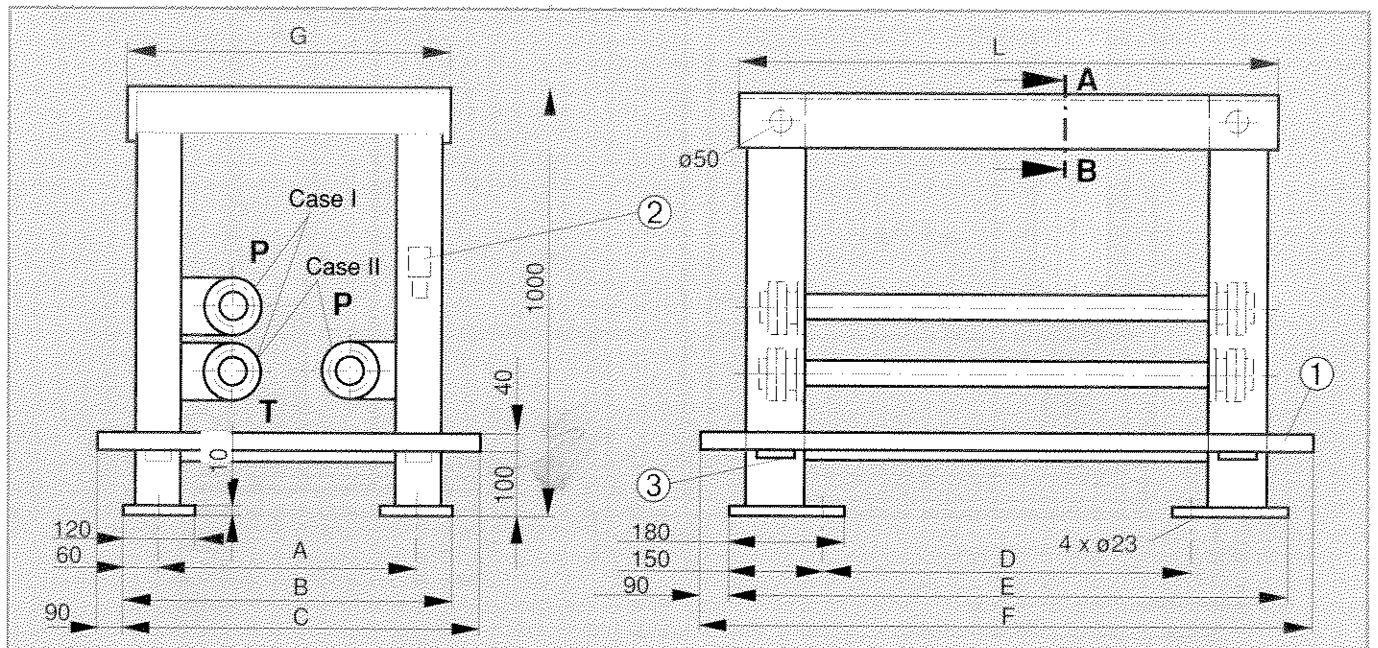
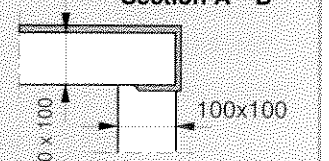


Fig. 127: *Valve table, lightweight version, for mounting single valves up to Size 22 and control plates up to Size 16*



Section A—B



A	B	C	D	E	F	L	G
880	1000	1180	720	1020	1200	1000	990
880	1000	1180	1220	1520	1700	1500	990
880	1000	1180	1720	2020	2200	2000	990
880	1000	1180	2220	2520	2700	2500	990

1 Drip tray
2 Outlets
3 Brace, U50x25

Fig. 128: *Valve table, heavyweight version*

4.6 Accumulator stands

Larger hydraulic systems often contain accumulators which, when they are providing energy storage for the control circuit, are mounted on the valve stands or valve tables. There are standard clips and brackets available to secure them in place.

When hydraulic accumulators are providing energy storage for the main pumping system, however, they are mounted on separate accumulator stands. Single-row (lightweight) and double-row (heavyweight) versions are available.

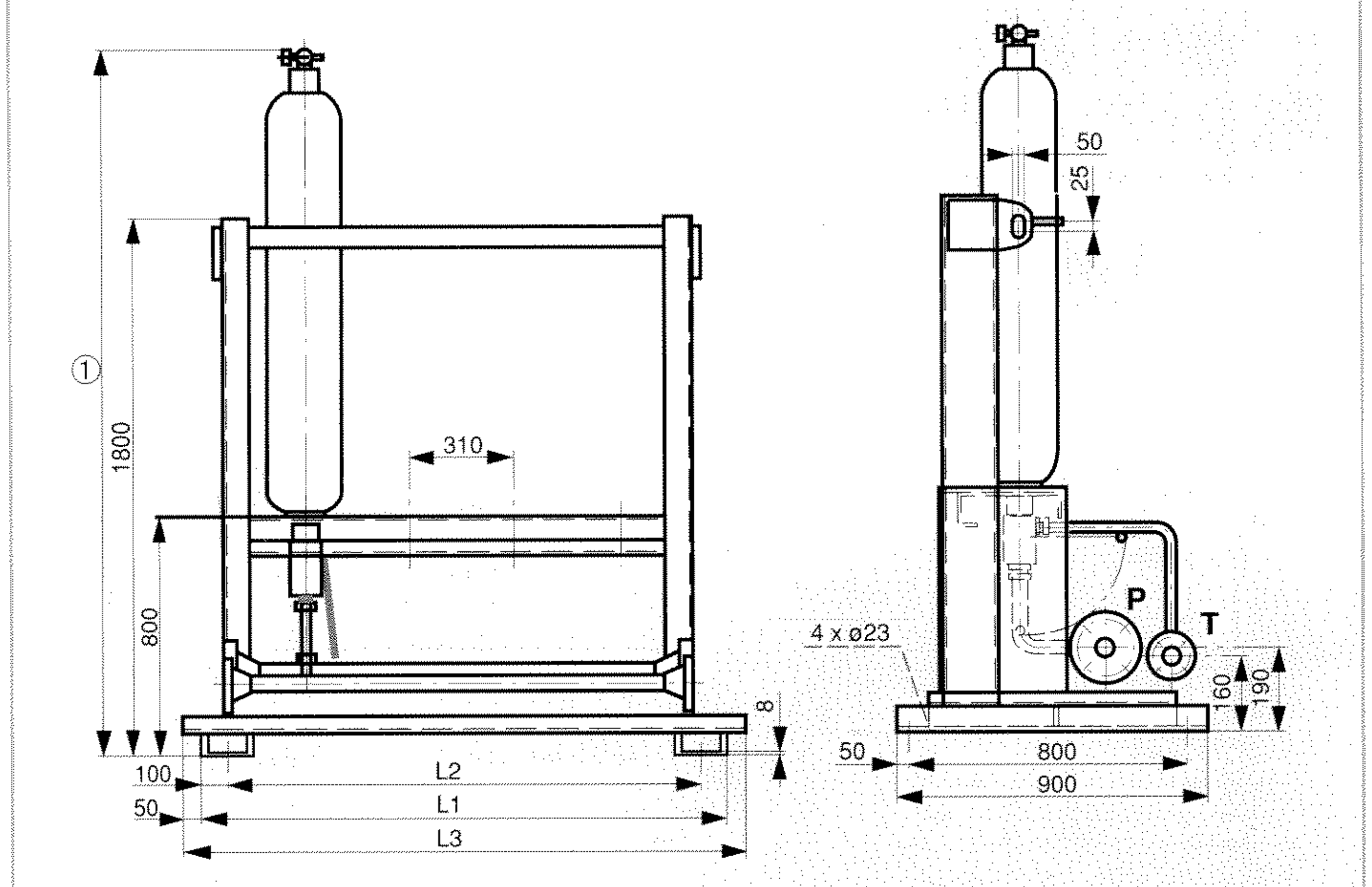
The single-row, lightweight version is usually fabricated from sectional steel.

Folded sections are used for the double-row, heavyweight version. The same standard sections used for valve stands can also be used here.

Regardless of their type, hydraulic accumulators should always be installed vertically. This means that bladder-type accumulator can rest in a rubber ring in a hole in a horizontal plate. Piston-type accumulators are also mounted on a horizontal plate; they must be secured by welded rings or other restraining devices to prevent any lateral movement.

Lifting and transportation must be considered when the stand is being designed. For example, if transportation in the upright position is impossible due to the height, bolted supports must be provided from the outset to allow the stand to be laid down.

Hydraulic accumulators must be specifically marked and secured for transportation. Details are given in the chapter "Packing and Transport".



Version	No. of accumulators		
single-row	2	3	4
L1	950	1250	1550
L2	750	1050	1350
L3	1050	1550	1650

1 approx. 2120 with 32 L accumulators, approx. 2650 with 50 L accumulators

Fig. 129: *Accumulator stand, single-row version*

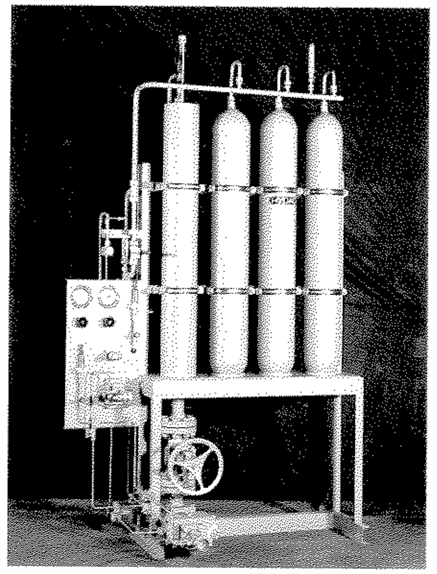


Fig. 130: (left)
Accumulator stand, single-row version

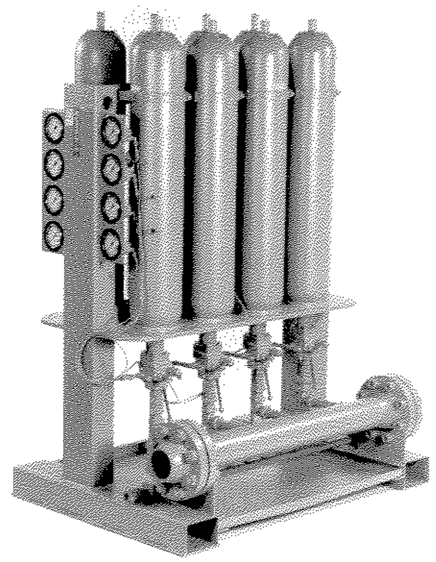
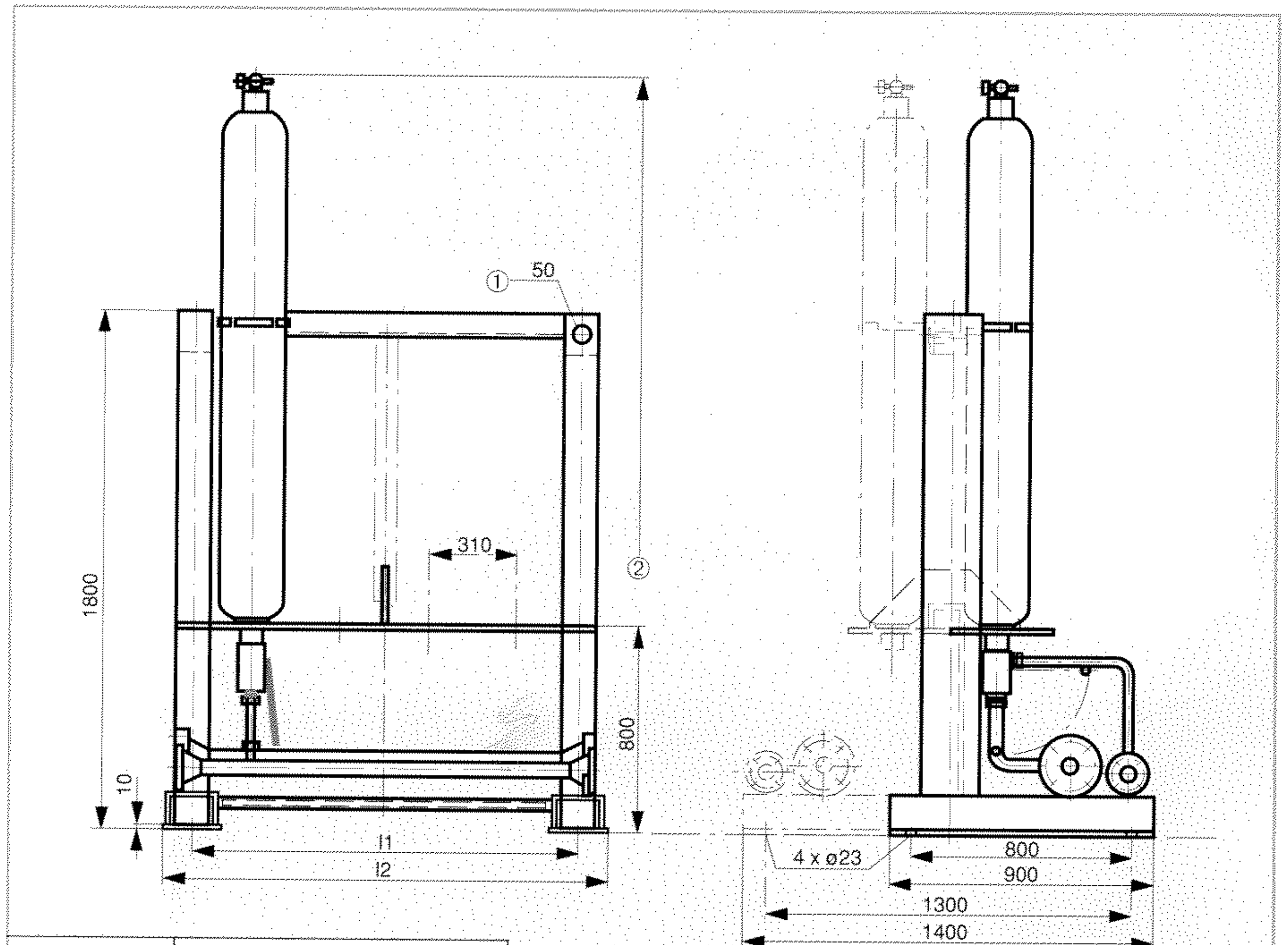


Fig. 131: (right)
Accumulator stand, double-row heavyweight version



Version	No. of accumulators		
single-row			5
double-row	6	7 and 8	9 and 10
L1	1250	1550	1850
L2	1050	1350	1650

1 Lifting tube
2 approx. 2220 with 32 L accumulators, approx. 2750 with 50 L accumulators

Fig. 132: *Accumulator stand, double-row heavyweight version*

5 References

[1] Jahrbuch Schweißtechnik DVS

[2] Dr.-Ing. Malisius
Wirtschaftlichkeitsfragen
der praktischen Schweißtechnik

[3] VDI-Richtlinie VDI 2221
Methodik zum Entwickeln und Konstruieren
technischer Systeme und Produkte

Notes

Noise Reduction

Dr. Joachim Morlok

1 The task

Probably the most outstanding feature of hydraulic drives and control gear is their extraordinarily high power density which is reflected in the physical volume and weight of the components. However, one of the inevitable side-effects of high power being packed into a small space is a high noise level.

Many industrial countries have already introduced legislation to restrict noise pollution. Such laws state that the mean noise level at a workplace must not exceed 90 dB(A) or, in many cases, even 85 dB(A). These limits have arisen from the discovery that continuous exposure to sound levels over 90 dB(A) can damage a person's hearing.

Each country (and indeed perhaps each individual industry) has its own special requirements which must be carefully checked at the project and design stages.

Nearly all workplaces are exposed to the sound emissions of a number of machines and equipments together with reflections from walls, floors and ceilings. Consequently, the specification of a maximum sound level of 80 dB(A) from each individual machine is no longer a rarity.
Therefore, the manufacturers of machines are coming under increasing pressure to design their products with noise in mind at the outset and to optimize their performance as much as possible.

Misunderstandings constantly arise when the values obtained by the manufacturer under standard conditions, i.e. with no environmental effects, cannot be repeated on the user's premises. The obvious cause of this situation is the nature of the surroundings at the place where the machine or equipment is finally installed. It is often overlooked that some additional provisions will be necessary in such cases in order to satisfy the requirements for low sound emissions.

From this it can be seen that it is not only necessary to consider the causes and effects of noise and measures for its reduction but also appropriate methods of measurement and interpretation of these measurements.

2 First a little physics

Sound is the sensation caused in the human ear by the vibration of an elastic medium. Depending on the nature of the medium we speak of either air-borne sound, structure-borne sound or fluid-borne sound.

All three are encountered with hydraulic equipment. Pumps and motors vibrate and the motion is transmitted to the mounting points as structure-borne noise. Positive displacement pumps produce pressure pulsations which can be detected through the whole circuit as fluid-borne noise and can excite mechanical vibration anywhere in the circuit. Air-borne noise is produced anywhere where a component excited by structure-borne noise or fluid-borne noise transfers its vibration to the adjacent mass of air.

2.1 Air-borne sound

The number of oscillations per second, i.e. the frequency, is measured in Hertz. It determines the pitch. The human ear has a pitch range of 16 to 20,000 Hz, although the upper limit decreases with advancing age. A healthy twenty-year-old should be able to hear a 17,000 Hz note whereas senior citizens often have difficulty at only 10,000 Hz.

In addition to high and low pitch our ears also distinguish between high volume and low volume, or loud and quiet. The governing factor of this is the sound pressure. It is an alternating pressure superimposed on the static atmospheric pressure. A sound pressure of 1 μbar (mean-square value) corresponds approximately to the normal volume of speech, i.e. the millionth part of the pressure of the atmosphere.

The sensitivity of the ear as a "measuring device" is astounding. It can detect variations in pressure of an order of magnitude of only a few 10^{-10} bar. Another remarkable quality is its wide dynamic range. At a frequency of 1 kHz the limit of audibility is 0.0002 μbar. The range of audibility extends to about 200 μbar without overloading the ear in any way (the threshold of pain is at approximately 300 μbar). Hence, the dynamic range of the human ear is 1:1 000 000.

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Nearly all workplaces are exposed to the sound emissions of a number of machines and equipments together with reflections from walls, floors and ceilings. Consequently, the specification of a maximum sound level of 80 dB(A) from each individual machine is no longer a rarity.
Therefore, the manufacturers of machines are coming under increasing pressure to design their products with noise in mind at the outset and to optimize their performance as much as possible.

Misunderstandings constantly arise when the values obtained by the manufacturer under standard conditions, i.e. with no environmental effects, cannot be repeated on the user's premises. The obvious cause of this situation is the nature of the surroundings at the place where the machine or equipment is finally installed. It is often overlooked that some additional provisions will be necessary in such cases in order to satisfy the requirements for low sound emissions.

From this it can be seen that it is not only necessary to consider the causes and effects of noise and measures for its reduction but also appropriate methods of measurement and interpretation of these measurements.

2 First a little physics

Sound is the sensation caused in the human ear by the vibration of an elastic medium. Depending on the nature of the medium we speak of either air-borne sound, structure-borne sound or fluid-borne sound.

All three are encountered with hydraulic equipment. Pumps and motors vibrate and the motion is transmitted to the mounting points as structure-borne noise. Positive displacement pumps produce pressure pulsations which can be detected through the whole circuit as fluid-borne noise and can excite mechanical vibration anywhere in the circuit. Air-borne noise is produced anywhere where a component excited by structure-borne noise or fluid-borne noise transfers its vibration to the adjacent mass of air.

2.1 Air-borne sound

The number of oscillations per second, i.e. the frequency, is measured in Hertz. It determines the pitch. The human ear has a pitch range of 16 to 20,000 Hz, although the upper limit decreases with advancing age. A healthy twenty-year-old should be able to hear a 17,000 Hz note whereas senior citizens often have difficulty at only 10,000 Hz.

In addition to high and low pitch our ears also distinguish between high volume and low volume, or loud and quiet. The governing factor of this is the sound pressure. It is an alternating pressure superimposed on the static atmospheric pressure. A sound pressure of 1 μbar (mean-square value) corresponds approximately to the normal volume of speech, i.e. the millionth part of the pressure of the atmosphere.

The sensitivity of the ear as a "measuring device" is astounding. It can detect variations in pressure of an order of magnitude of only a few 10^{-10} bar. Another remarkable quality is its wide dynamic range. At a frequency of 1 kHz the limit of audibility is 0.0002 μbar. The range of audibility extends to about 200 μbar without overloading the ear in any way (the threshold of pain is at approximately 300 μbar). Hence, the dynamic range of the human ear is 1:1 000 000.

In order, on the one hand, to cover this exceptionally large range and, on the other hand, to achieve a high "resolution", our hearing employs a logarithmic law in sensing the energy (or power) of the sound information. Thus, an increase in the sound energy (or sound power) by a factor of ten corresponds approximately to a doubling of the volume.

2.2 Sound energy (or power)

From what has been said so far it can be seen that sound power is the quantity to use for identifying different amounts of noise. The sound power transmitted to the air from the sources of sound varies over several orders of magnitude. For example, speech radiates a power between microwatts and milliwatts. It takes a jet aircraft to get up into the kilowatt range - *see Table 37.*

This means that the acoustic power P_{ac} of machines is usually very low. The most important relationship is:

$$P_{ac} \sim P_{mean}{}^2 \cdot S$$

Where S is the area of an imaginary envelope around the machine and p_{mean} the mean-square value of values of sound pressure measured at various points on the envelope.

Sound power P in W	Sound power level L_W in dB (10^{-12} W)	Sound sources
40000000	195	Saturn rocket
100000	170	Jet engine
10000	160	Turbojet aircraft
1000	150	4-propeller aircraft
100	140	Threshold of pain
10	130	Large orchestra
1	120	Motor horn
0.1	110	Loud radio
0.01	100	Car on motorway
0.001	90	Inside underground train
0.0001	80	Loud conversation
0.00001	70	Normal conversation
0.000001	60	In an office
0.0000001	50	Quiet conversation
0.00000001	40	Whispering

Table 37: *Relation between sound power and sound power level*

Fig. 133 shows clearly how the value of "sound pressure" (or sound pressure level) measured with instruments depends on the particular points at which the measurement is taken. A value of sound pressure or sound pressure level quoted without its distance from the source is worthless.

On the other hand, the sound power of the subject is independent of the distance from the source. Greater distance only changes the distribution of the energy. This is termed a decreasing sound intensity (sound power per unit area).

Therefore, the sound power of the subject can be positively identified by establishing a theoretical enveloping surface around the source and measuring the sound pressure or sound pressure level at several points on that surface.

There are two conditions to take into account in choosing the form of this theoretical enveloping surface and its distance from the sound source:

- The minimum distance from the sound source must be maintained. It is usually 1 m.
- All sound waves emanating from the sound source must be able to penetrate the enveloping surface (perpendicularly if possible). Only then will it be possible to measure the full transmitted power.

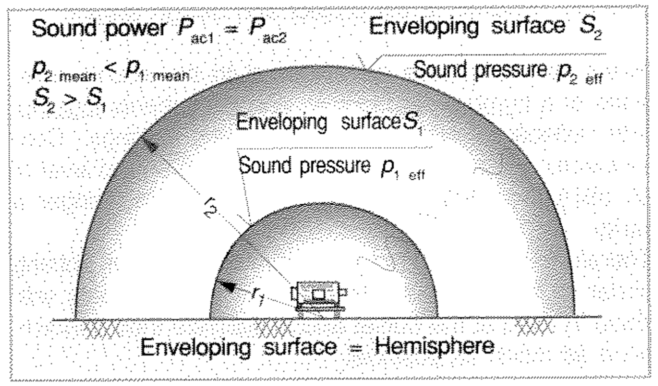


Fig. 133: *Relation between sound pressure, measuring surface and sound power*

In German standards it is common to use a theoretical enveloping surface of rectangular form, especially for hydraulic power units.

Measuring points are then placed in the plane of this theoretical measuring surface. In addition to other points, measuring points should be set at the corners, in the middle of each side and the top. A hard base is not assumed - sound waves are reflected from it and emerge at the sides. The distance from the outer contour of the machine is usually fixed at 1 m.

2.3 Measuring sound power level and sound pressure level

The absolute value of sound power in Watts is not very suitable for practical purposes due to the wide range of numerical values *(Table 37)*. Also, as mentioned earlier, the sensitivity of the human ear to volume is not linear but linked logarithmically to the sound power and sound pressure. It is for these two reasons that a logarithmic reference system was introduced. The actual value of sound power P_{ac} is referred to a fixed basic value P_{ac0}. Using logarithms for this ratio gives a level designated L. Therefore, the level is the log ratio of two energy values:

$$L_W = \lg (P_{ac}/P_{ac\,0}) \text{ in Bel (B)}$$

The added note or symbol Bel indicates that common logarithms have been used, although Bel are not units in the strict sense of the word.
In order to obtain numerical values that are easier to handle, the practice is to convert to the smaller units decibels (dB).

$$\textbf{Sound power level } L_W = 10 \lg \frac{P_{ac}}{P_{ac\,0}} \textbf{ in dB}$$

Substituting the following

$$P_{ac\,0} \sim p_0^2 \cdot S_0 \text{ and } P_{ac} \sim p^2 \cdot S$$

gives the following relationship when solved:

$$L_W = \underbrace{10 \lg \frac{p^2}{p_0^2}}_{\text{Sound pressure level } L_p} + \underbrace{10 \lg \frac{S}{S_0}}_{\text{Measuring-surface level } L_S} \text{ in dB}$$

Thus, the sound power level L_W comprises two quantities - the sound pressure level L_p and the measuring-surface level L_s.

The sound pressure level L_p is indicated directly in dB by measuring instruments. The chosen reference value or base value p_0 is $p_{0\ mean} = 2 \cdot 10^{-4}$ µbar which is the threshold of audibility of the human ear. Thus, a measured sound pressure of $p_{mean} = 2 \cdot 10^{-4}$ µbar is precise sound pressure level of $L_p = 0$ dB, the starting point of the scale.

The measuring instruments measure the actual sound pressure directly and convert it internally, following the above relationships, to give decibels. The reading of sound pressure level is then given in dB.

The measuring-surface level L_s is calculated from the type of enveloping surface chosen (e.g. the surface area of the theoretical envelope in m²) and the reference surface $S_0 = 1$ m².

2.4 Averaging level values

Determining the sound power level requires a number of measuring points to be fixed on an enveloping surface. Measurements of sound pressure level are then taken at those points. Calculation of the sound power level then requires an average value, the measuring-surface sound pressure level. If the difference between the maximum value and the minimum value is less than 3 dB, the arithmetic average can be taken. Otherwise the averaging will have to be undertaken according to the following equation (n = No. of measuring points):

$$\bar{L}_p = 10 \lg \left[\frac{1}{n}\left(10^{0{,}1 \cdot L_{p1}} + 10^{0{,}1 \cdot L_{p2}} + \ldots + 10^{0{,}1 \cdot L_{pn}}\right)\right]$$

Therefore, for the sound power level:

$$L_W = \bar{L}_p + L_S \text{ in dB}$$

2.5 Auditory weighting

Another problem with the measurement of noise emissions arises from the property of the human ear to perceive sounds of different frequency as of different volume or different degrees of unpleasantness.

We are particularly sensitive to the 500 to 5000 Hz pitch range. For example, a 3000 Hz whistle requires only one hundredth of the sound power of a 20 Hz hum in order for it to be perceived to be equally loud.

In the measuring instrument this effect is taken into account by the physically-correct measured value of sound pressure level being corrected or "weighted" according to the frequency by a weighting network with standard values.

There are a number of different weighting methods or curves. The most popular is the A-curve. When the weighting is switched on the measuring instrument displays a value of sound pressure level in dB(A) weighted for human perception.

As the human ear responds very sensitively in the frequency range between 500 and 5000 Hz it is often important to know where predominant levels occur in the range because they can then be reduced by specifically designed silencers.

Electronic filter networks are used for this "frequency analysis". Octave filters do not give particularly good results; one-third octave filters are better; but the best results are obtained by narrow-band analysis.

2.6 Standards and measuring procedures

There is a comprehensive set of standards dealing with the emission of noise from machines and equipment.

The base standard is DIN 45 635, Part 1 which is the outline work dealing with the general conditions for the measurement of noise emissions.

Individual standards, such as DIN 45 635, Part 26 for hydraulic pumps and DIN 45 635, Part 41 for hydraulic power units, then provide details of enveloping surfaces, measuring points and operating conditions.

The main feature of these measurements is that it is only the noise emissions of the machine itself that are determined. All environmental influences such as reflection of noise are eliminated.

Procedure

The hydraulic power unit is mounted in a place where the background level is low and the environmental influences are low or known (method described in DIN 45 635, Part 1).

The enveloping surface is taken as a theoretical rectangular hollow form following the outer contour of the unit at a distance of 1 m.

The prescribed measuring points are the corners , the mid-points of the sides and of the top of this envelope - *see Fig. 134.*

At these points the values of A-weighted sound pressure level are measured in dB(A) under prescribed operating conditions. The values are then averaged in accordance with *Section 2.4.*

Environmental influences are allowed for if necessary by correcting this value.
The sound power level is obtained by employing the measuring-surface level L_s calculated from the enveloping surface - *see Section 2.3.*

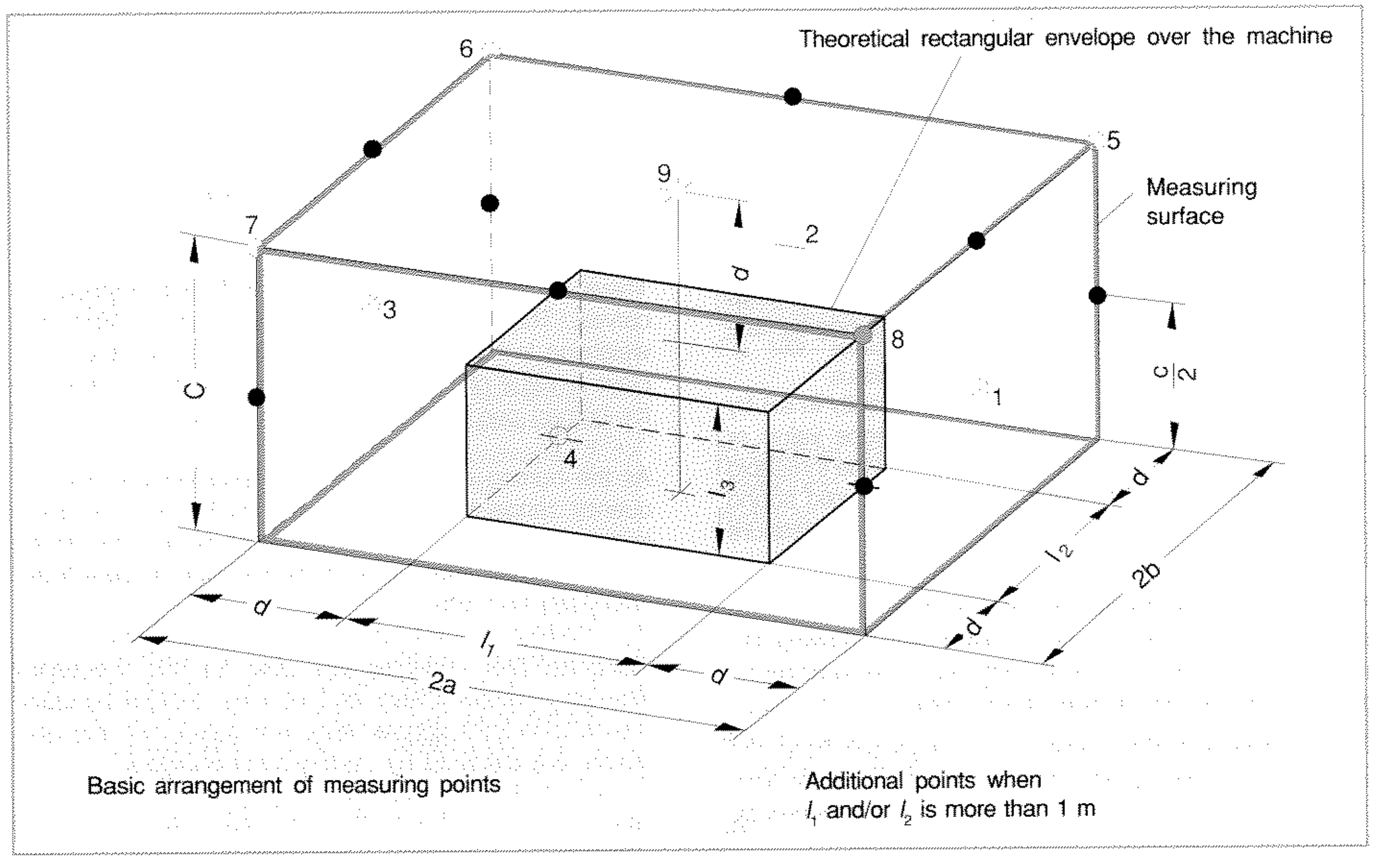


Fig. 134: *Measuring surface (theoretical) and arrangement of measuring points to DIN 45 635, Part 41*

2.7 Emission and Immission

Emission: The radiation of sound waves from a source
Immission: The effect of a number of sound sources on a particular point (e.g. the operating position of a machine).

The theory so far has dealt with the quantities and methods associated with the measurement of emission. DIN 45 635 and the sound power level provide a firm base from which to work.

Ascertaining the characteristic values in accordance with the requirements of the relevant standards is the task of the manufacturer of the machine. He can influence the values by appropriate design. Another benefit is that the data can be compared with other manufacturers' data obtained under the same conditions.

The user of the machine faces a completely different situation.

When considering the anticipated immission at the operating position of a machine there are several additive influences to take into account:

a) Emissions from the machine being operated

b) Environmental influences, such as reflection from floors, walls and ceilings

c) Noise emissions from neighbouring machines

d) Background noise at the place of installation

The manufacturers of machines know practically nothing about influences **b)** to **d)**.

Therefore, specifications in contracts for adherence to poorly defined "noise levels" sometimes lead to misunderstanding and subsequent claims.

The most common reason is that the environmental influences at the final place of installation increase the values measured by the manufacturer under standard conditions. Multiple reflection from nearby walls and ceilings together with reflection from neighbouring machines, for example, can result in levels being increased by up to 5 to 10 dB(A). This is particularly true when the installation is in a small room, against a wall or close to other machines.

Environmental influences can be determined either by calculation with empirical values (DIN 45 635, Part 1) or by precise methods of measurement using reference sound sources for example.

Basically, there should be agreement on the noise level to be maintained based on DIN 45 635 and manufacturers should test their machines under standard conditions.

If agreements differ from this basic form the user must provide very comprehensive and very precise data on the environmental influences involved.

2.8 Estimating noise levels

A measuring point is often exposed to the simultaneous emissions from a number of different sound sources. The rule for calculating the immission at that point is:

When the sound fields of several sources overlap, the values of sound power must be added together.

Performing this addition - and various conversions and rearrangements - gives the following relationship:

$$L_\Sigma = 10 \lg (10^{0,1 \cdot L1} + 10^{0,1 \cdot L2} + \ldots + 10^{0,1 \cdot Li}) \text{ in dB} \qquad (1)$$

where
L_Σ = Overall level
$L_1 \ldots L_i$ = Level of individual source

It makes no difference whether the sum required is a sound pressure level from individual values L_p or a sound power level from individual values L_W.

The case of several sound sources producing the same sound pressures or sound pressure level L_p at a measuring point is described by the following simple equation:

$$L_{p\Sigma} = 10 \lg (k \cdot 10^{0,1 \cdot Lp}) = L_p + 10 \lg k \qquad (2)$$

k = No. of sources

An extreme example:
When there are two sound sources, each with a sound pressure level of L_p = 0 dB, acting on a measuring point, according to Equation (2):

$$L_{p\Sigma} = 0 \text{ dB} + 10 \lg 2 = 3 \text{ dB}$$

This is probably the only time that you will ever write 0 + 0 = 3!

Remember that values of sound power are being added together here. At 0 dB there is still sound power present but it is so low that it cannot be perceived by the average human ear.

Table 38 shows the relationship when the number of sound sources all producing the same level at the measuring point is increased.

Conversely, it is easy to see how many sources must be removed in order to obtain a certain level.

No. of sources k	Increase in level ΔL in dB	Sound power factor	Perceived volume
1	0	1	
2	3	2	doubled
4	6		
10	10	10	
25	14		doubled
50	17		
100	20	100	

Table 38: *Relation between sound level, perceived volume and number of identical sound sources*

For example, 90% of the sources (or machines) must be removed in order to achieve a reduction of 10 dB in the level. Equation (2) now shows very clearly indeed that the same effect can be achieved by making each sound source itself 10 dB quieter.
This relationship illustrates how important it is for the noise emissions from individual machines to be reduced by design measures or other means of insulation or muffling.

3 Source and effect

Measures for reducing noise are particularly successful and economic when it is possible to influence the sources of structure-borne noise and fluid-borne noise directly. For this it is useful to prepare a so-called "sound flow plan". This notes all the possible sources of sound, the paths of transmission and the likely places of air-borne sound emission.

3.1 Flow of sound

There are many sources of sound in hydraulic drives; it is also transmitted in a number of different ways and finally radiated from a number of different surfaces.

It is common practice for the individual sub-assemblies, such as motor and pump and control valves, to be combined and mounted on the oil tank.

These hydraulic power units are usually installed separately from the machine being driven and connection to the hydraulic cylinders or motors is by rigid pipes and/or flexible hoses - *see Fig. 135.*

From a sound flow plan for such a power unit it can be seen that it is primarily the pump which radiates air-borne sound. It also excites structure-borne sound and fluid-borne sound. The transmission and propagation of structure-borne sound are made easier by the mechanical link between the motor-driven pump and the oil tank. Direct pipe connections between the pump and the valves provide another bridge for structure-borne sound.

Due to the principle of operation of the pump, fluid-borne sound is also continuously excited in the form of periodic pressure pulsations. The fluid-borne sound propagates throughout the whole piping system. Receivers of the oscillations are the pump itself, the driven machine and all downstream components including the hydraulic power unit. Linked structures, such as floors and walls to which the pipes are clipped, are also affected. Each of these receivers radiates air-borne sound whose original source can be traced back to the principle by which the pump operates.

Hydraulic valves also excite air-borne, structure-borne and fluid-borne sound. The operation of a directional control valve causes a flow of fluid to be either slowed down or speeded up. This causes oscillations in pressure which propagate through the system as fluid-borne sound.

Directional control valves, pressure valves and flow valves can cause high-frequency hissing noises due to turbulence and cavitation at points of throttling.

Beginning from this explanation it is now intended to show, in a general form, how there are a number of "transfer elements" in the path between excitation and radiation to be taken into account with the object of determining where noise-reducing measures can be successfully included.

Fig. 136 shows a schematic illustration of a mounting panel for a hydraulic power unit.

It is required to mount a valve on a sheet-metal panel the input of which is exposed to a periodically changing force *F(t)*. This force can be calculated, for example, from the fluctuations in pressure of the fluid arising from the pump. From the variation in operating force with time it is possible to calculate the frequency spectrum which we will now call the "excitation spectrum".

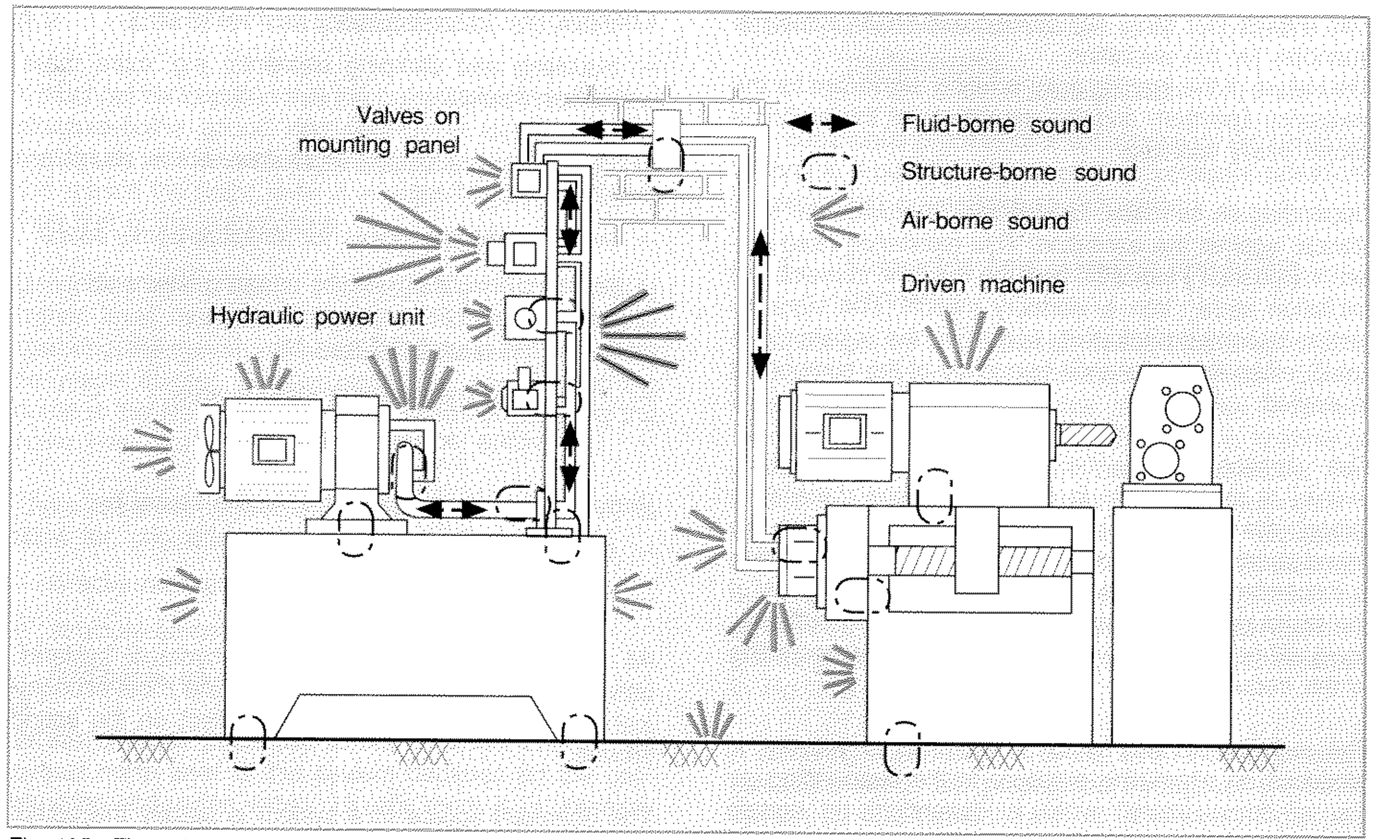


Fig. 135: *Flow of sound from hydraulic drives and control gear*

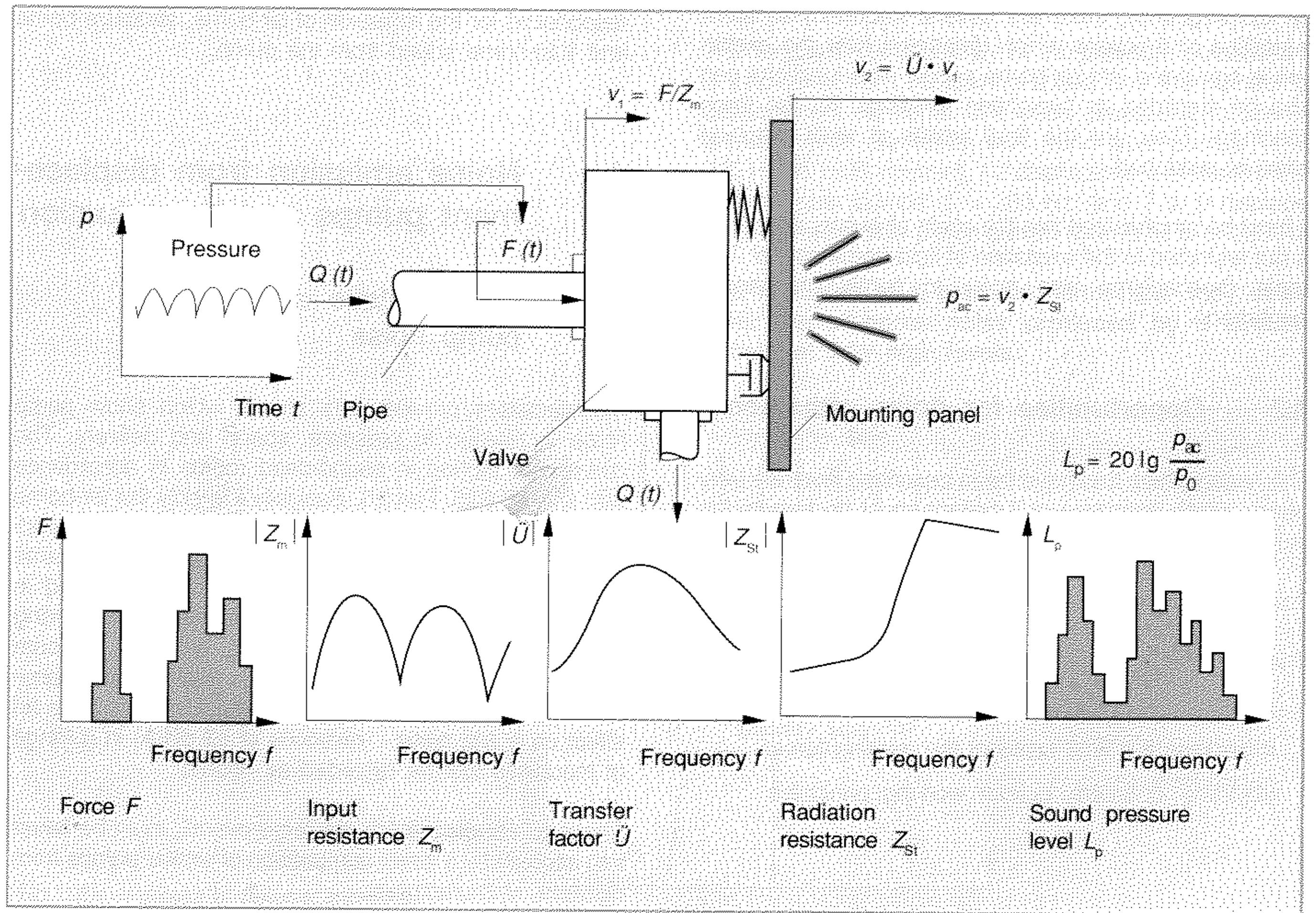


Fig. 136: *Excitation spectrum and frequency-sensitive transfer characteristics*

This means that there are defined and measurable frequencies for the excitation.
The nature of the starting point is highly significant for the frequency spectrum and the magnitude of the air-borne sound finally radiated.

Depending on its mass, for example, it provides a "resistance" against the excitation. This resistance (also called "impedance") indicates how the machine converts the excitation force into a vibratory motion. The vibratory motion becomes particularly large (i.e. the resistance is small) when the resonance frequency of the component is excited.

The vibration velocity, i.e. the structure-borne sound, excited at the machine in this way is then transmitted through the immediate structure and propagates into the linked surrounding structure. The process of propagation is influenced by the insulating or damping properties of the structure. This means that, at the radiating surfaces, there will be a velocity v_2 which differs from v_1 according to the frequency.

The radiating surface which is vibrating at velocity v_2 produces an alternating pressure in the surrounding air (i.e. air-borne sound) which is dependent on certain properties of the surface such as size, shape, material and thickness. This process is also related to the frequency. Consequently, there are several frequency-sensitive quantities or functions which determine how the excitation is converted into air-borne sound. The functions can also be regarded as "weighting functions" for the force spectrum. They are the final arbiter of what machine noise is produced.

It must be emphasized that the chain of events described is not restricted to the excitation of fluid-borne sound. The same mechanisms also apply to purely mechanical excitation.

3.2 Opportunities for noise reduction

The relationships that have been described illustrate very clearly which quantities or functions must be targeted in order to reduce noise.

Possible starting points are:

Changing the exciting force *F(f)* by:

- reducing the amplitude
- expanding the time scale of the exciting forces
- selecting inherently quieter working principles

Changing the input resistance by means of:

- insulating and damping elements, e.g. extra mass, rubber insulators

Changing the propagation of structure-borne sound and fluid-borne sound by:

- avoiding bridging points for structure-borne sound
- providing insulating and/or damping elements

Reducing the sound radiation by:

- reducing the radiating surface area
- providing acoustic "short-circuits"

Reducing the propagation of air-borne sound by:

- enclosure
- mufflers.

All of these measures are used with hydraulic systems, although the most effective and often the most economical are those which have a direct effect on the sound source.

The dominant source in a system is usually the pump. The most effective way of reducing the noise level of an installation is to concentrate on the mechanisms that are responsible for the emission of air-borne, structure-borne and fluid-borne sound from the pump.

4 Noise emission from displacement pumps and its reduction

4.1 Characteristic values for pump noise

Manufacturers provide data sheets which show pump noise in relation to pressure, speed and delivery. The curves are plotted in sound-dead rooms in accordance with DIN 45 635, Part 26. Basically it is only the air-borne sound radiated directly from the pump that is measured. It has become apparent that the noise level of the same pump can vary by up to 5 dB(A) when measured in different test rooms. The causes are different mounting arrangements, methods of clamping, types of pressure lines and suction lines and different load valves. Manufacturing tolerances and setting tolerances also have an effect on pump noise - variations of ±2 dB(A) in a range of machines can be regarded as normal.

These tolerances, which are large overall, must be taken into account by a direct comparison of similar pumps from different manufacturers.

4.2 Effect of the pump in the system

It must not be forgotten that the pump is also transmitting structure-borne sound and fluid-borne sound into the system as well as radiating air-borne sound directly. For this reason the noise level of a system is always higher than that of the pump alone.

Depending on the design of the system a 5 to 10 dB(A) higher level must be anticipated. However, generally speaking, the system will be quieter if the pump alone can be made quieter. Consequently, the selection of the pump is the first important step towards establishing the noise level of a system.

There are three fundamental influences on the noise to allow for, originating from the motor-driven pump:

- directly radiated air-borne sound
- structure-borne sound transferred to the system
- fluid-borne sound transferred to the system

Which of these influences is the dominant one for the overall noise level of the system depends on the construction and varies widely. Usually there is an interplay of all three.

4.3 Fluid-borne noise and pressure pulsations

The continuous process of a succession of displacement actions in a pump produces periodic pressure pulsations involving two superimposed processes.

4.3.1 Volumetric flow pulsation due to the geometry

The principle by which pumps deliver fluid is identical in all positive displacement-type machines. Piston pumps, vane pumps and gear pumps all possess a specific number of delivery chambers which, in a continuous cycle, are first opened to be filled (i.e. suction), then closed to prevent backflow, and finally opened again in order to expel the contents - *see Fig. 137*. Obviously, therefore, it is not a truly continuous process as there are a number of separate flows superimposed on one another which must be added together. The end result is a pulsating delivery such as that shown diagrammatically in *Fig. 138*.

A pulsating flow causes a similar sequence of pressure pulsations which are propagated throughout the whole system as fluid-borne sound. All downstream components in the system and also the machine being driven are therefore excited to vibration.

The fundamental frequency of the pressure pulsations f_0 is calculated from the driving speed and the number of displacement elements. The usual practice is to use 11 displacement elements with vane-type pumps (f_0 = 275 Hz at 1500 rev/min), 12 displacement elements with external gear pumps (f_0 ≈ 300 Hz) and 7 or 9 displacement elements with piston pumps (f_0 = 175 Hz or 225 Hz)

The multiples of f_0 produce harmonics; they determine the air-borne noise produced by a system, usually at very high levels.

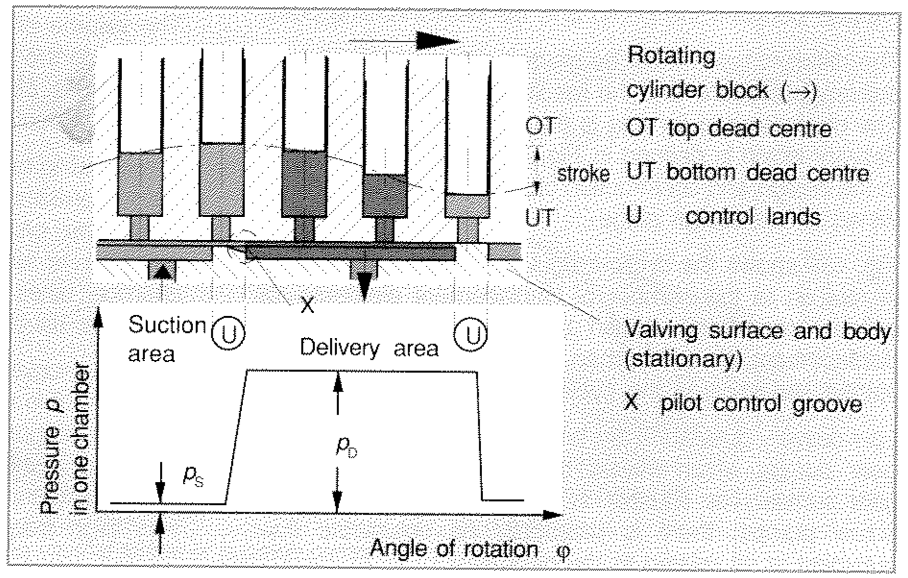


Fig. 137: *Process of delivery and pressure generation*

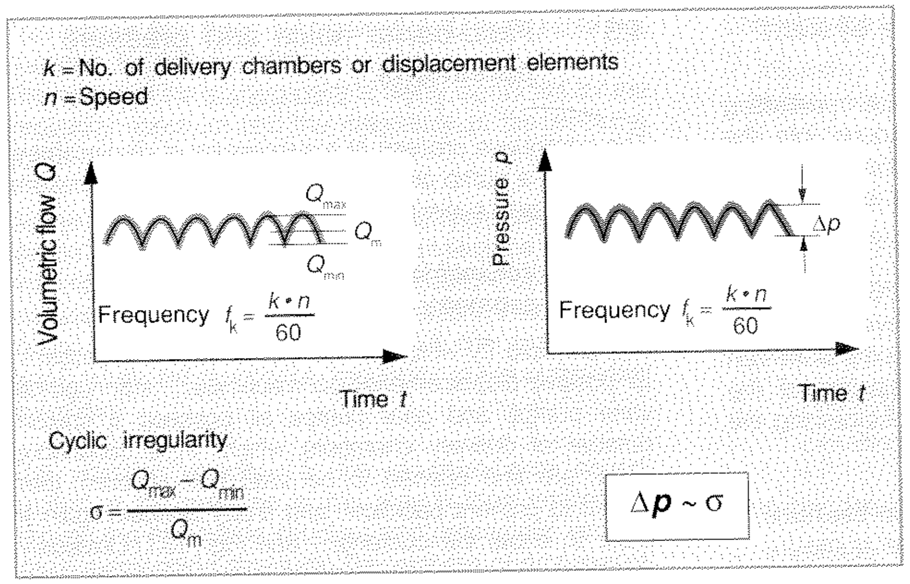


Fig. 138:
Delivery and pressure pulsations from positive displacement pumps (diagrammatic)

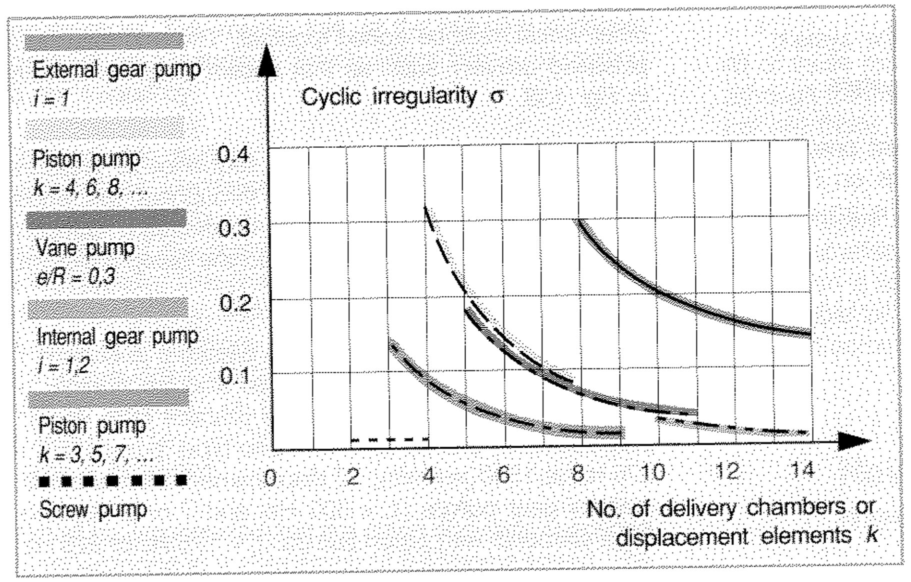


Fig. 139:
A comparison of cyclic irregularity of delivery for different types of pump

These frequencies are exceptionally pronounced when they match the resonance range of the column of fluid trapped between the pump and the next throttling point.

The different types must be graded differently in terms of delivery pulsations and the resulting pressure pulsations. The characteristic for this is the cyclic irregularity of the delivery which is defined in *Fig. 138.* It is directly related to the design and construction of the pump and so can be varied at the design stage.

Fig. 139 shows the cyclic irregularity of different designs in relation to the number of displacement elements (or delivery chambers). It can be seen that having more displacement elements than normal produces no substantial reduction in the cyclic irregularity.

Low delivery pulsations are an inherent feature of internal gear pumps which makes them quieter than external gear pumps. The situation is even more favourable with screw pumps; there is practically no volumetric flow pulsation at all related to the geometry.

4.3.2 Pressure matching in the delivery chamber - compression pulsation

The volume of a delivery chamber is constantly increasing during the suction stroke. At the transition stage the chamber is closed and during the delivery stroke the volume is decreasing - *see Fig. 137.*

The deciding factor is the time scale of the matching of the suction pressure and delivery pressure which takes place inside the chamber. If the exposure to the delivery pressure is not gradual, the rise in pressure in the chamber takes place at very high speed; figures of up to 10^8 bar/s have been measured.

From the resulting periodic variations in the forces inside the pump it is to be expected that there will be considerable consequences for the external noise. The pattern of force has a shock-like character; the excitation spectrum of the deforming forces exhibits several components of very high frequency with a high excitation capability.

Since the force loop of the pump is closed through the cylinder block the latter is excited to flexural vibration at these frequencies.

It must also be remembered that a reduction in volume accompanies the compression of the fluid in the delivery chamber. The "missing volume" must be offset by backflow from the delivery side. If the rise in pressure is very rapid, brief periods of high-velocity backflow occur leading to transient expansion in the pump outlet, i.e. to a drop in pressure.

Fig. 140 shows how important pressure matching can be for the noise emission from a pump. Internal forces are generated, i.e. deforming forces, but there are also external forces which induce the pump body to vibrate as a compact mass.

This process excites the pump itself to radiate air-borne sound; it excites structure-borne sound in the mounting bracket and causes periodic pressure fluctuations in the connected pipework which propagate throughout the whole system as fluid-borne sound. Such fluctuations in pressure are called "compression pulsations". They are superimposed on the geometry-related volumetric flow pulsations described in *Section 4.1.*

A simple practical means of influencing the process that has been described is "pre-opening". It means employing grooves, notches or small holes to provide a link between the delivery side of the system and the delivery chamber of the pump before the main opening to the delivery side takes place - *see Figs. 137 and 140.* In terms of the rotating chamber the groove behaves like a gradually opening throttle so that the compression of the fluid in the chamber is determined by the higher pressure fluid which is admitted according to a specific time function. The result is a rise in pressure and force inside the chamber considerably slower than for an arrangement with no groove - *Fig. 140 (B).* The reduction in noise is considerable.

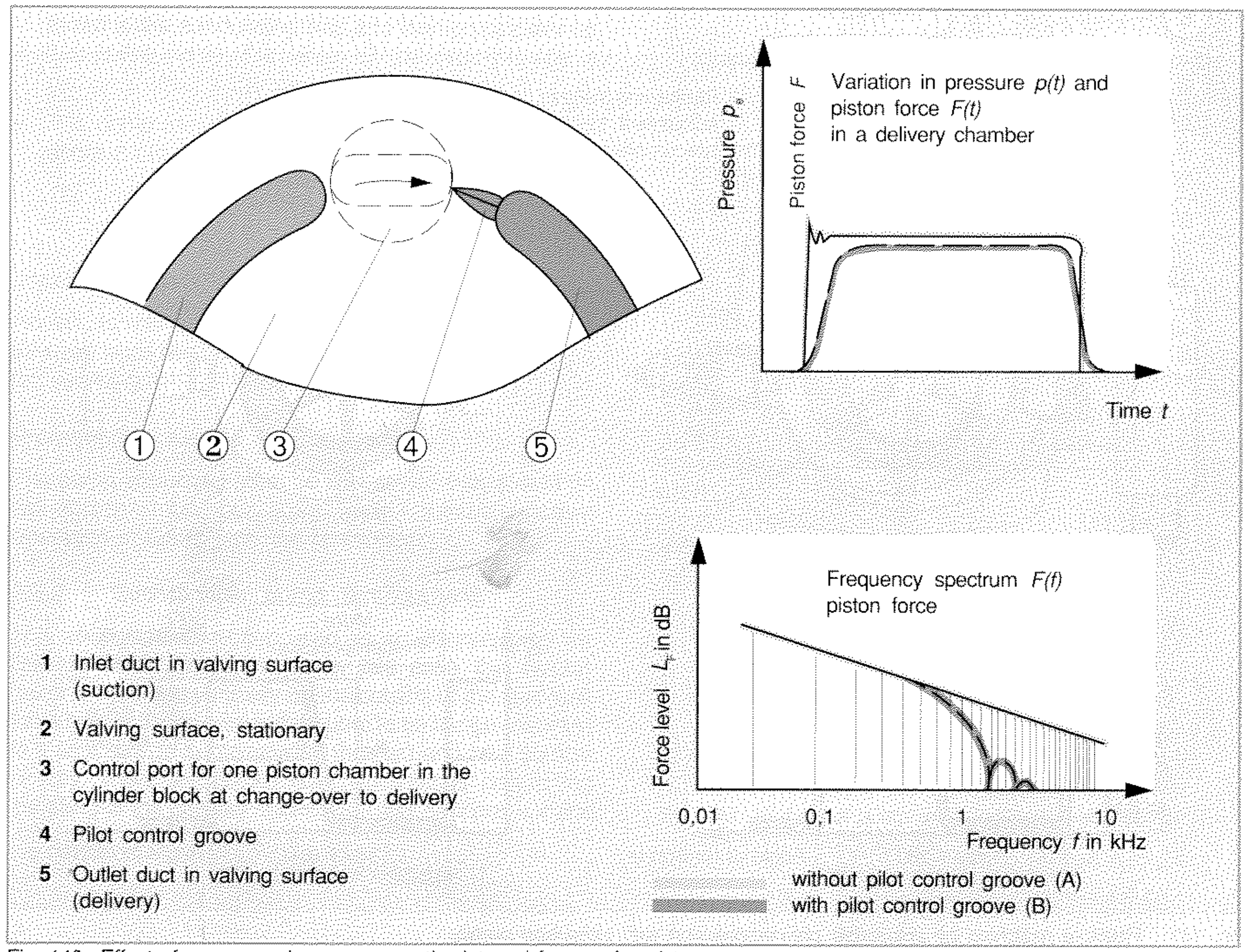


Fig. 140: *Effect of a pre-opening groove on the internal forces of a piston pump*

All designs are carefully optimized with regard to rise in pressure. However, one disadvantage of this fixed pre-opening is that it can only really be optimized for one value of operating pressure - *see also Fig. 141.*

Therefore, a different system is employed with vane-type pumps. In this case there is an opportunity to initiate the compression process before the delivery chamber has been opened up to the delivery side. This means that the contents of the chamber are "pre-compressed" by the displacement elements. The difficulty lies in timing the opening-up to the delivery side precisely when the pre-compression has reached the operating pressure. Missing the "change-over point" results in incorrect matching. With this method the pressure build-up, and hence the noise, of this type of pump can be optimized for any required operating point. Another major advantage of the method is that the optimizing can be performed on site.

Fig. 141 shows how the pre-compression of a vane-type pump can be adjusted by moving the stroke ring at right angles to the direction of the stroke motion.

4.4 Effect of the suction process

It must be pointed out that the process of pressure matching also depends to a large extent on the amount of filling and amount of air in the delivery chambers. If the chambers are not completely full, air will be drawn in and cause compression pulsations of considerable amplitude and also cavitation knock.

In order for the suction fluid to be free from turbulence and air bubbles the hydraulic power unit must exhibit features such as:

- a large tank volume
- a return line as far as possible from the suction line and separator plates in the tank to provide a settling area
- a suction line completely free from leaks

Flow resistance in the suction must be avoided through the use of short suction lines of large diameter and the absence of constrictions and kinks in the pipes. If high flow resistance in the suction line cannot be avoided it will be necessary to employ feed pumps or booster pumps.

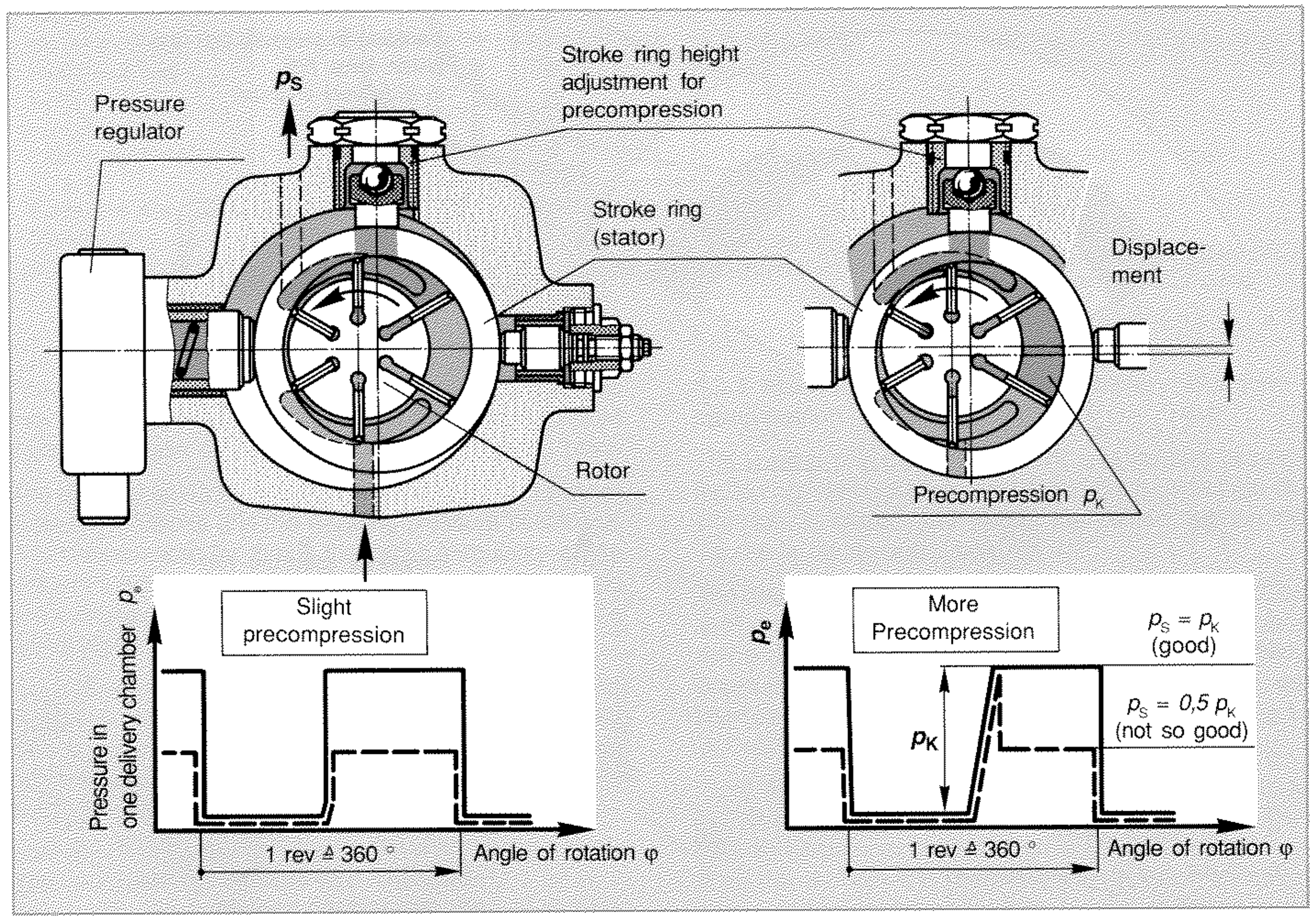


Fig. 141: *Pressure matching by adjustable pre-compression with vane-type pumps (diagrammatic - the number of chambers has been reduced from 11 to 6 for illustration only)*

The maximum permitted values of pressure at the suction stated by the manufacturer must be adhered to. Suction lines and return lines should not be placed too close to the side of the tank; the clearance should be at least 10 cm.

4.5 Arrangement of pump and motor in the tank

In order to reduce the amount of air-borne noise radiated directly it can be useful to mount the pump inside the tank. There are two alternative methods.

4.5.1 Fluid-immersed arrangement

In this case the whole unit, i.e. the pump and an electric motor of the oil-immersed type, is placed inside the closed tank. The motor-pump unit is usually suspended from the lid on anti-vibration mountings and is completely immersed in the fluid - *see Fig. 152.*

Advantages

- There is no directly-radiated air-borne noise. Although the vibration of the pump and motor is transferred to the surrounding fluid and then propagates as fluid-borne noise, the sides of the tank provide a damping effect if the clearance is sufficient.
- The pump can take suction directly at a pressure of about 1 bar.
- The problem of passing the suction pipe through the lid is eliminated.

Disadvantages

Access to the pump for maintenance and servicing is severely restricted. The whole tank lid has to be removed for servicing.

Prerequisites for full effectiveness

The pump and motor must be suspended from the lid on anti-vibration mountings - not rigidly - with a minimum clearance of 0.5 m from the tank sides.

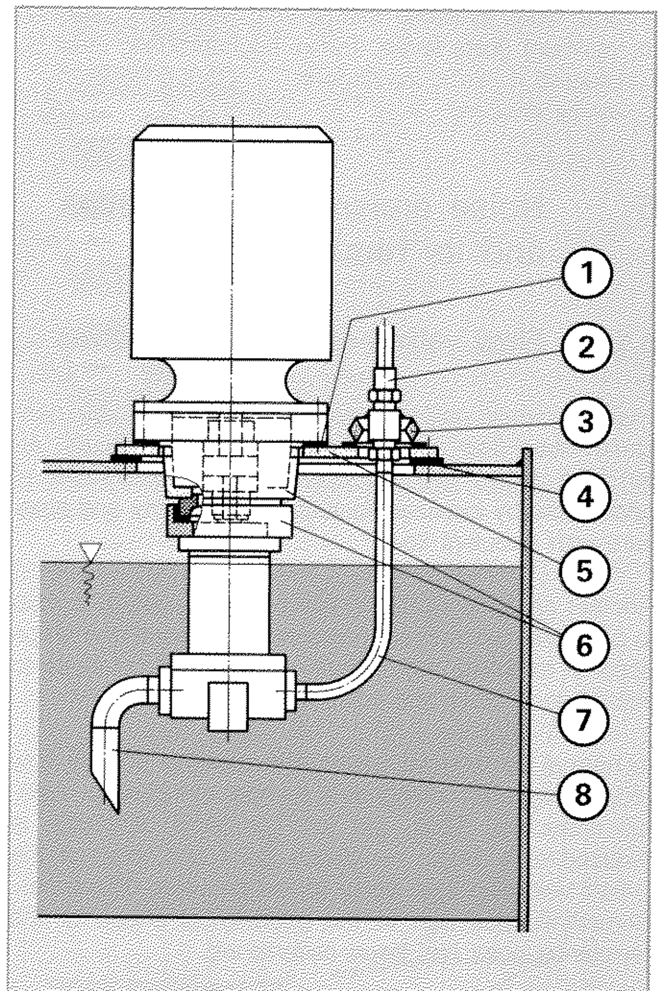
4.5.2 Submersible pump arrangement

This arrangement is often used as an alternative to the fluid-immersed arrangement. In this case the electric motor is mounted vertically on the outside of the fluid tank lid. A hole in the lid allows the pump to be flange-mounted to it so it can be submerged in the fluid and take suction directly. There are considerable advantages concerning the filling of the delivery chambers - *see Sections 4.4 and 4.5.1.*

Prerequisites for full effectiveness

There must be a clearance of at least 0.5 m from the tank sides and the mounting flanges must be of the type providing insulation of structure-borne sound - *see Figs. 142 and 146.*

These measures are particularly effective when the air-borne sound radiated by the power unit is the governing factor for the overall noise level.
The disadvantage, again, is that access to the pump is restricted which makes maintenance and servicing difficult.



1 Gasket
2 Hose
3 Elastic pipe bushing
4 Gasket
5 Mounting plate
6 Bellhousing with damping ring
7 Pipe or hose with large bending radius
8 Short suction pipe

Fig. 142: *V1 type of pump unit - low-noise version*

4.6 Electric motors

In the submersible pump arrangement the directly-radiated air-borne sound from the electric motor can propagate freely as before. It should be noted that the motors generate a similarly high noise level to the pumps that they are driving, although with the advantage of a broad-band frequency distribution in the spectrum which is "less noticeable".

In considering the reduction of noise this means that the electric motor also has to be included and there are suitable types available with bearings and fans designed for optimum noise levels. Partial soundproofing must be considered if this is insufficient.

5 Noise emission from valves and its reduction

5.1 Flow valves and pressure valves

High-frequency noise is generated in valves by non-steady-state flow at points where throttling occurs. Alternating forces are produced by the detachment of eddies and by localized fluctuations in velocity and pressure with turbulent flow.

Another cause is cavitation which is produced by a drop in pressure. Since these pressure fluctuations and drops occur in turbulent, high-velocity flow, the effects seem to initiate each other and influence each other.

In normal commercial valves it is usual to employ flow resistances which comprise a number of individual variable restrictors. Thus, the total pressure difference is reduced at a single resistance point and most of the critical velocities then occur at low pressure differences. Some influence due to the geometrical shape of the resistance point has been ascertained.

If noise is to be reduced the critical flow velocity or pressure difference must not be exceeded. This requirement can be satisfied by connecting several throttles together in either parallel or series.

With parallel-connected resistances the total flow cross section is divided into a number of smaller areas - *see Fig. 143*. This allows the total area, and therefore the flow velocity, to remain approximately the same. However, the critical Reynolds number, i.e. the change-over to turbulent, cavitation flow, is only attained at much greater pressure differences when there is a group of resistances. This also causes non-steady-state flow at greater pressure differences or volumetric flows. *Fig. 143* shows how the pressure difference critical to sound is shifted when the number of resistances in a throttling cylinder is increased while retaining the same cross area.

An effective reduction in noise can also be achieved by reducing the total pressure difference in stages by a number of resistances connected in series. A combination of resistances comprising 3 pressure stages (the pressure compensator of a flow control valve) has been tested - *see Fig. 144*. The lower the inlet pressure and the pressure difference (flow velocity) at the final stage, the quieter this combination becomes. *Fig. 144* shows a diagram of the pressure compensator of a flow control valve of this type, its circuitry and the reduction in noise achieved.

Such designs are somewhat cumbersome, complicated to manufacture and therefore expensive so their use has so far been restricted to special applications only.

5.2 Directional control valves (reversing)

When valves perform control actions, the flow of fluid is suddenly accelerated or decelerated, producing fluctuations in pressure which eventually propagate through the system as fluid-borne noise. The valve bodies are also excited to vibration when their moving parts are accelerated or decelerated.

The behaviour of a hydraulic system is fundamentally different when a pulse-like noise is superimposed on the steady-state noise. The former can, in fact, determine the overall noise of a system if operation of the valves is frequent.

Operating shocks can be avoided by extending the actuating times of the valves. This also makes a substantial reduction in the speed of pressure rise or fall in valve chambers, pipes or hydraulic cylinders. It means that the shock loads or sudden unloading are changed to relatively slow processes.

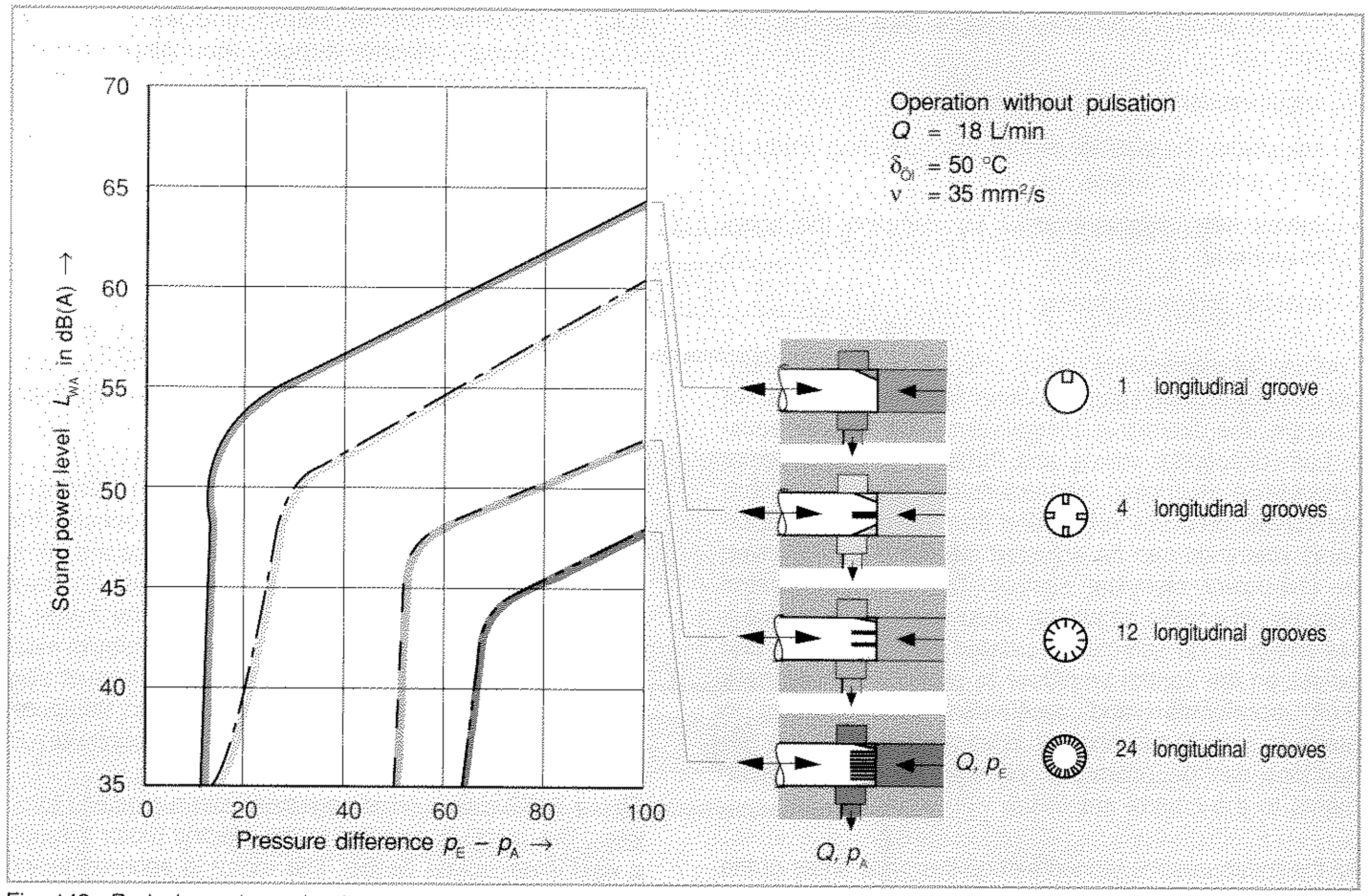


Fig. 143: *Reducing valve noise by connecting resistances in parallel (according to G. Schmid)*

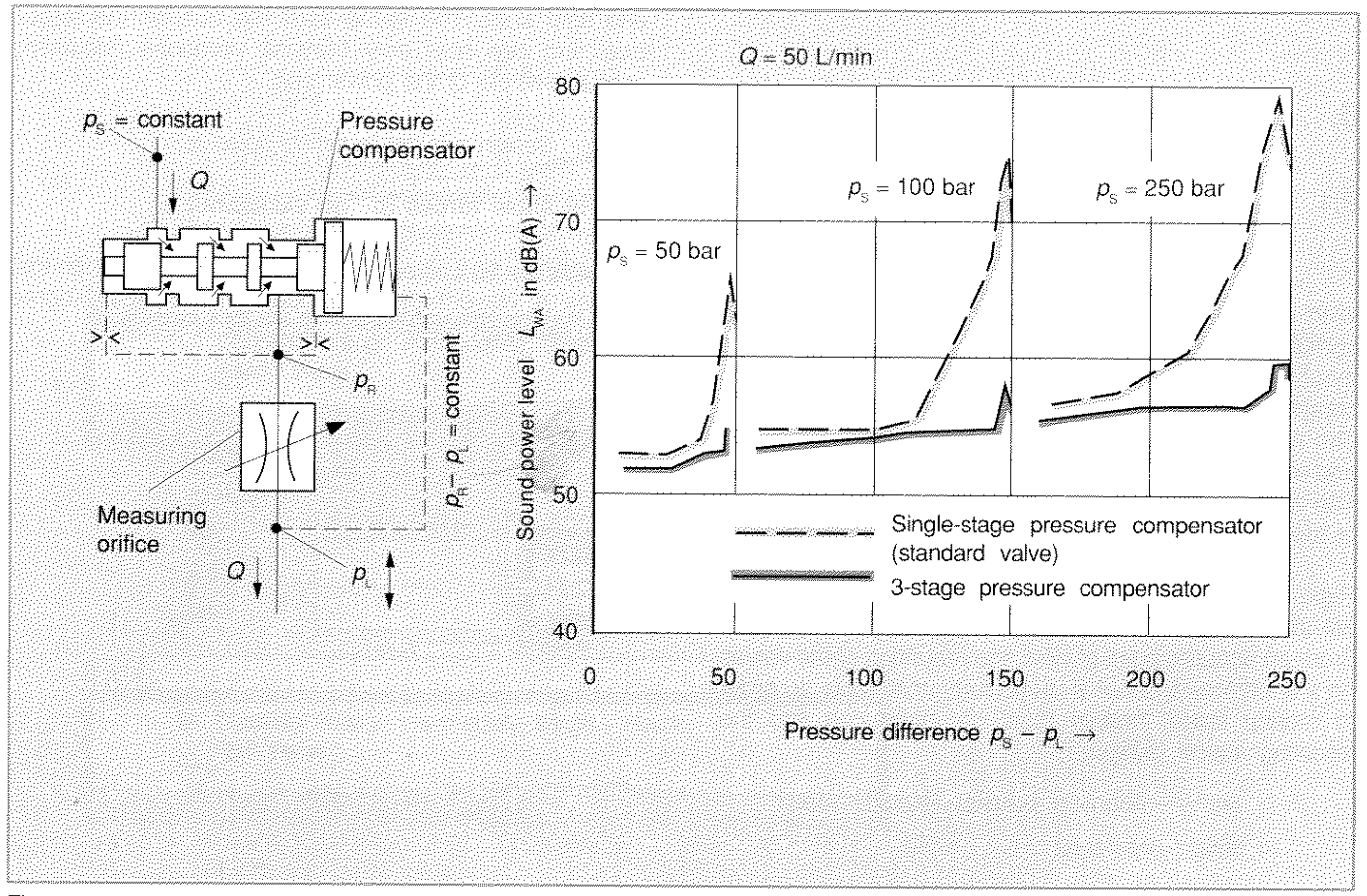


Fig. 144: *Reducing valve noise by connecting resistances in series (according to O. Eich)*

There are several alternative designs that can be employed:

- spool valves instead of seated valves
- adjustable throttles to control the operating times of pilot-operated valves
- valves with integral throttles for damping the spool movement
- special control solenoids.

The use of proportional valves offers a particularly elegant means of regulating actuating time. In this case, for example, the directional control valves are simultaneously both directional valves and throttles with a linear or progressive restriction characteristics. Directional control valves, pressure valves and flow valves with proportional solenoids can be adjusted steplessly when driven by low-power electrical signals. The electrical control enables alternative actuating sequences with appropriate timing to be selected. So-called ramp functions can be set on the control amplifiers to produce exceptionally "soft" acceleration and deceleration and changes in pressure or speed. Another advantage is that the sequences can be optimized "on site" by simple adjustments to the control amplifiers.

All the measures mentioned extend the cycle times of the processes and a compromise between a fast cycle and low noise emissions must always be sought after.

6 Reducing system noise

6.1 Reducing the transmission of structure-borne noise

With drives and control gear it is common practice to combine the various devices into sub-assemblies and to mount them on the oil tank. Pipes and hoses link the components together and to the machine being driven. The motor-pump set comprises an electric motor, pump and mounting bracket which is bolted to the lid of the tank. The vibration from the unit is transmitted to the large area of the lid and tank sides as structure-borne sound.

Rigid pipes connecting the motor-driven pump to the valves provide another bridge for structure-borne sound. From an acoustic point of view, the mounting of the valves on a sheet-metal panel is particularly unsuitable. In order to reduce noise, breaks must be introduced into the following sound bridges:

→ pump—mounting bracket—electric motor

→ power unit—tank

→ power unit—valve unit

6.1.1 Structure-borne sound bridges pump - electric motor - mounting bracket and power unit - tank

Fig. 145 shows several methods by which the electric motor is coupled to the pump.

In *figure 145A* the torque loop between pump and motor is completed through the common base-frame. The vibrations from pump and motor are transmitted through the frame and its rigid coupling to the fluid tank as structure-borne sound. Due to the large radiating area and the strong excitation, the tank becomes the main source of noise.

Figure 145B shows a method of decoupling. Mounting the frame on rubber blocks considerably reduces the excitation of the tank by structure-borne sound. It is particularly important for the flexible supports to have three degrees of freedom. For example, if the frame is bolted to the tank lid right through the rubber blocks most of the desired decoupling effect will be lost. It must also be ensured that the tank is not exposed to structure-borne sound from any other routes - *see Section 6.1.2.*

Figure 145C shows an arrangement with a modified force loop. The pump is coupled to the electric motor through a rigid intermediate flange. The motor is secured to the frame by a mounting bracket. In this case the torque loop between pump and motor only passes through a small and rigid area. The frame itself lies outside the torque loop and can now only transmit vibrations originating from the whole of the power unit.

A basic design rule for reducing noise is:

Do not allow forces to "go walkabout".

The arrangement shown in *figure 145C* is better than that in case *A*. Nevertheless, decoupling the structure-borne sound between base-frame and tank is sensible.

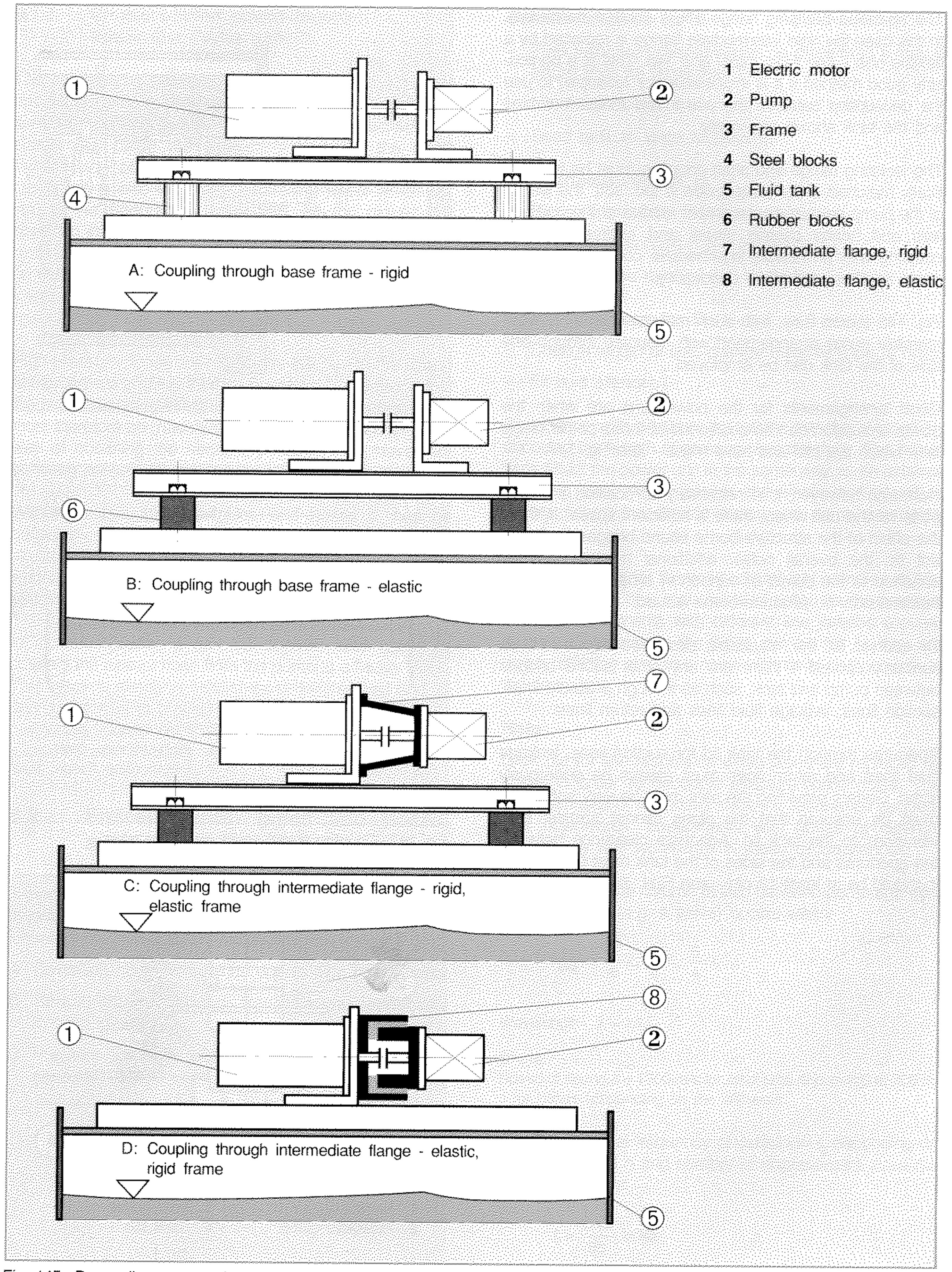


Fig. 145: *Decoupling structure-borne sound between pump - electric motor - mounting bracket - tank*

The so-called damping flange offers another alternative. In this case the rigid intermediate flange is replaced by a flexible one - *see Figs. 145D and 146.* This is a particularly good method if it is impossible (for example) to use the decoupling between the pump and motor mounting and the tank shown previously.

The isolating flange and shaft coupling must be kinematically matched. If the suspension is too "soft" the weight of the pump causes displacement relative to the coupling axis and hence causes stress and excites noise. A coupling with an flexible intermediate component represents another method of decoupling.

Fig. 146 shows how, with such measures, even a "submersible pump arrangement" with mounted through one side of the tank can be employed.

Good arrangements for the power unit are when the points of support can be positioned at those points of the tank which are stiff and have mass - *see Fig. 147.*

Normally, however, it is necessary to position the supports for the unit where there is sufficient space. If, then, the effect of the structure-borne sound bridges is dominant on the overall noise, additional masses can be attached to the points of support in order to reduce the transmission of structure-borne sound.

Of course, all the measures described above will be seriously spoiled if there are any other sound bridges between pump and tank, such as through pressure lines, suction lines, leakage fluid lines and return lines.

Apertures through the tank lid for suction lines, leakage fluid lines and return lines must always be generously sized. Gaiters providing the seal must be very soft and it must be ensured that the pipes cannot transfer their vibrations to the tank lid. Adequate clearance between the apertures and the sides of the tank must be allowed; it should be at least 10 cm - *see Figs. 142 and 148.*

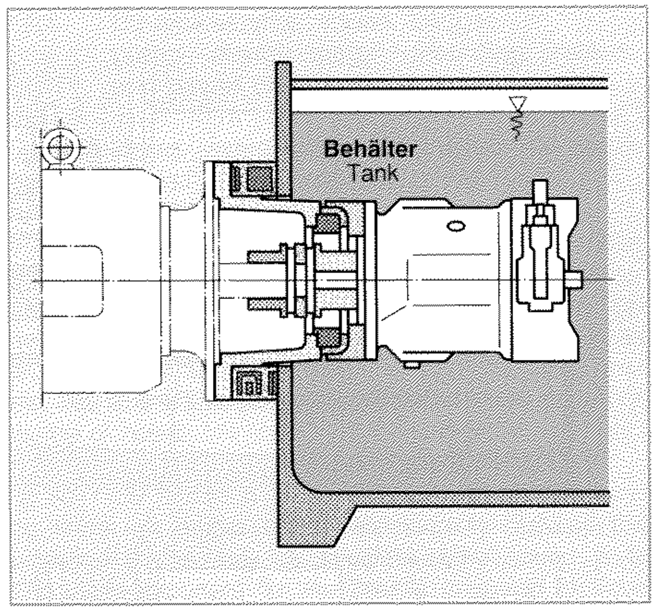
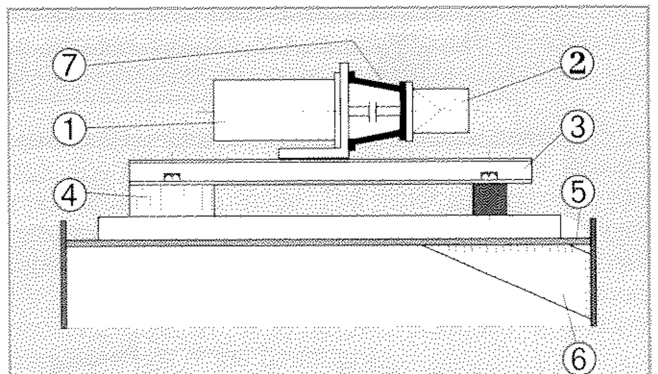
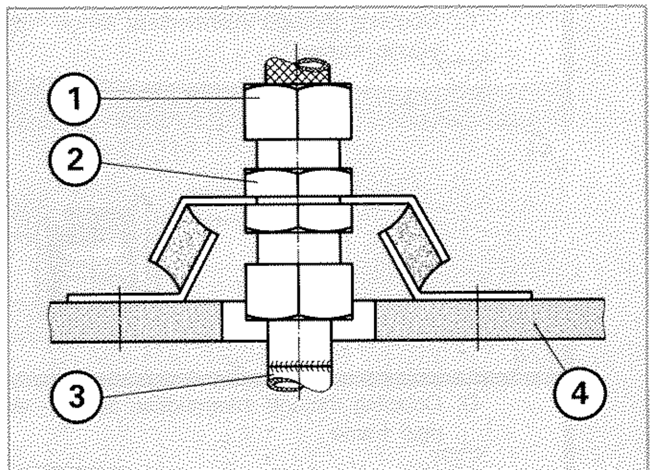


Fig. 146: *Damping flange*



1 Electric motor
2 Pump
3 Frame
4 High-mass element
5 Fluid tank
6 High-stiffness point
7 Intermediate flange, rigid

Fig. 147: *Coupling to stiff points with extra mass*



1 Hose
2 Bulkhead gland
3 from pump
4 Tank top

Fig. 148: *Flexible bulkhead adaptor in tank lid*

6.1.2 Structure-borne sound bridges Power unit - valve unit

The connections between pumps, valves and actuators are sound bridges that need special attention. Connections to mounting panels (with the valves attached to a sheet-metal plate) are particularly prone to the direct transmission of structure-borne sound. Direct mechanical coupling also introduces a bracing effect which obstructs the freedom of movement for the power unit specified in *Section 6.1.1.* Hoses are the answer, especially for the low and medium pressure ranges. They must be fitted so that the flexible support for the power unit has freedom of movement in all axes. The fluid velocity in the hose fittings must also be restricted to 4 m/s.

High-pressure hoses are really very stiff due to the strong reinforcement and the internal pressure. This means that the sound damping effect is less for high pressures. Tests must therefore be made with different runs to find the best way of obtaining the necessary freedom of movement between power unit and valve unit.

With rigid and inflexible pipes and hoses it may be necessary to resort to extra mass in order to isolate structure-borne sound. Such masses are heavy weights whose inertia reduces the transmission of vibration at bridging points.

In individual cases it can be sensible and useful to connect the pipe or hose from the pump to a heavy block. This block is simply a through-point for the fluid and its only function is to provide inertia. The same function is performed ideally by those types of equipment with subplates and vertically stacked valves.

6.2 Emission from pipes and hoses, avoiding line resonance

If pipes are found to be emitting clearly perceived and measurable sound it will be necessary to investigate the following mechanisms of excitation:

A Cavitation noise, turbulence due to high flow velocity

B Structure-borne sound transmission, especially through pipes

C Pressure pulsations of high frequency and amplitude, especially with line resonance.

Countermeasures for A

- Flow velocity in pressure lines no more than 4 m/s, in return lines no more than 2 m/s
- No sharp bends or changes in cross section
- Bend radii as large as possible

Countermeasures for B

- Pipes replaced by hoses
- Extra masses fitted - *see Section 6.1.2*
- Modified pipe runs

Countermeasures for C

- Change length of line, change material (pipe to hose)
- Fit fluid silencer.

Resonance with pressure pulsations can be avoided by choosing the length of pressure line carefully. The pulsations effect the power unit in the same way as an external pulsating force which excites the whole power unit to vibrate (when the electric motor and pump are rigidly coupled).

The calculations for this method are very complex so trial and error is the only real course of action for the practical person. Experiments with different line lengths are necessary if vibration in the pipes can be felt or they are clearly emitting air-borne noise with tonal components in the frequency.

Note

Look out for line resonance when pipes are replaced by hoses in order to decouple structure-borne noise. The lower stiffness of the hoses shifts the resonance frequencies downwards. Experiments with hoses of different lengths are well worthwhile in these circumstances.

Resonance of the column of fluid between pump and next throttling point (e.g. a valve) occurs when:

$$l_{line} = \frac{1}{4}\lambda, \frac{3}{4}\lambda, \frac{5}{4}\lambda \ldots$$

Wavelength $\lambda = c/f$

where
Speed of sound $c \approx 1300$ m/s for steel tube and c=400 to 700 m/s for hoses (depending on the stiffness)

The fundamental frequency of excitation f_0 depending on the drive speed n and number of displacement elements k in the pump is:

$$f_0 = \frac{n \cdot k}{60} \text{ in Hz}$$

6.3 Reducing the propagation of fluid-borne noise by silencers

As already emphasized several times before, there are three principal mechanisms governing the amount of noise produced by a hydrostatic drive:

- the air-borne noise emitted directly from the pump, electric motor and pipework
- the structure-borne noise generated by the pump and electric motor and transmitted through bridging points
- the pressure pulsations, i.e. fluid-borne noise, imposed at a specific frequency on the system by the pump.

Which of these three mechanisms governs the noise depends on the construction, the power and many other parameters of the hydraulic system.

The pressure pulsations provide a back-effect on the motor-driven pump acting as an external force and causing vibration which, as structure-borne noise, is impressed on to the system. The pressure pulsations reach all parts of the system and can excite structure-borne noise and air-borne noise everywhere.

Hence, providing a break in sound bridges is an effective means of reducing noise from hydraulic power units. Sometimes, of course, reducing the fluid-borne noise is just as important.

The opportunities for influencing the delivery pulsations or pressure pulsations by design, construction and type of displacement pump have already been dealt with in *Section 4.3.*

Most common types of pump always produce a pulsating flow and therefore a pulsating pressure. Fluid silencers can be employed in order to reduce, or at least obstruct, the propagation of this fluid-borne noise. Such devices must satisfy the following requirements:

- reduction of pressure pulsations across as wide a frequency band as possible with low pressure losses
- simple construction, no maintenance, no air bubbles, no contamination of the operating fluid.

A variety of terms and quantities are used to assess the effectiveness of silencers.

The insertion damping value is the reduction in sound pressure level ascertained by comparison measurements with and without silencers in the system. This value can only be determined experimentally.

The throughput damping value is the ratio of input sound energy to output sound energy. Level notation is used here similarly to air-borne sound.

$$D_d = 10 \lg \frac{\Delta p_{e\,mean}^{\;2}}{\Delta p_{a\,mean}^{\;2}} \text{ in dB}$$

e = Input
a = Output

The value is frequency-sensitive, characteristic to the silencer, independent of the overall system and describes the frequencies at which the silencer reduces pressure pulsations. However, the actual reduction in the air-borne sound level of a system that can be attained cannot be calculated.

6.3.1 Absorption silencers

Absorption means destroying the sound energy by converting it into heat. It can be done by friction in the transfer medium in or at an absorbent layer (e.g. layers of mineral fibre in air-borne noise silencers) and/or by compression and expansion of a volume of gas.

6.3.2 Reflection silencers

Since adequate absorption silencing is difficult to achieve in fluids, more use is made of reflection silencers which employ an interference effect. Hybrid types are used less often, i.e. reflection silencers with additional absorption capacity.

With interference or reflection silencers the unwanted primary sound wave is eliminated by superimposing on it a second wave of the same amplitude and frequency. This second wave is obtained by reflection of the primary wave at resistances (e.g. sudden change in cross section, branch point, etc.) and is displaced in phase by 180°. Typical examples of reflection silencers, hybrid types and their characteristics are shown in *Fig. 149.*

Pure reflection silencers have the advantage over absorption silencers or hybrid types in that they contain no material to become dirty and/or deteriorate in the course of time.

The sizing depends on various parameters such as mounting position (length of upstream and downstream lines), the type of connection of the downstream throttling point, and many others. The natural frequencies of the silencer with an upstream column of fluid must be noted. As a first approximation the volume of the silencer should be more than 1500 cm³. This ensures that the natural frequency of the silencer is sufficiently far from the normal fundamental frequency of the pump pulsations.

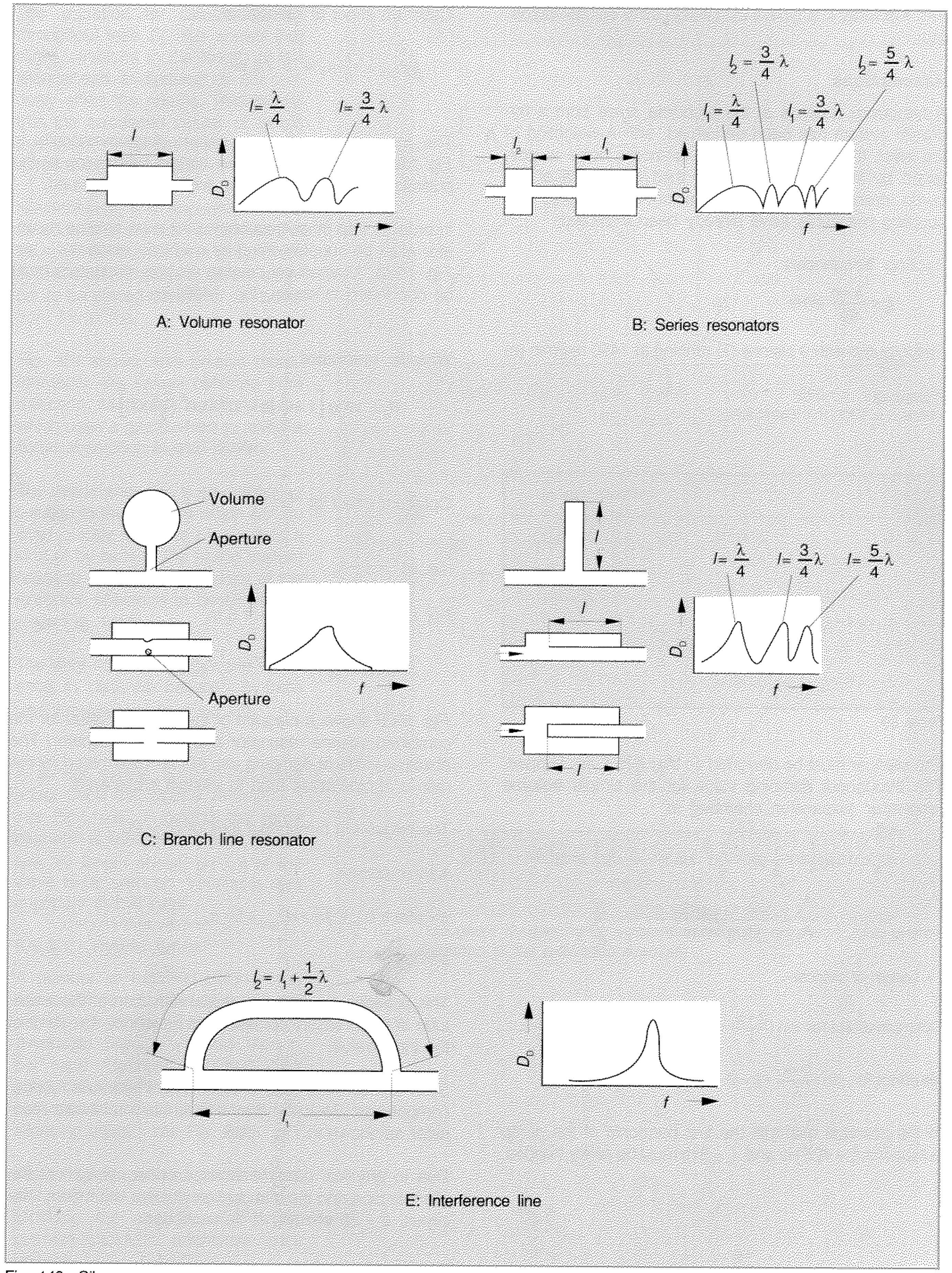


Fig. 149: *Silencers*

The following is a typical calculation for a volume resonator.

Assumptions

A frequency analysis of the air-borne noise from a hydraulic system has been performed, with A-weighting of the level. It is revealed that the air-borne noise is governed by the first, second and third harmonics of the pump pressure pulsations (structure-borne noise decoupling measures have already been instituted).

Exciting frequencies:

$$f_0 = \frac{k \cdot n}{60} \text{ in Hz}$$

Piston pump with 9 pistons (k) running at 1450 rev/min (n)

Wavelength $\lambda = c/f$
where c = 1300 m/s (steel tube)

		Wavelength λ
f_0	218 Hz	6 m
$f_1 = 2 \cdot f_0$	= 435 Hz	3 m
$f_2 = 3 \cdot f_0$	= 652 Hz	2 m
$f_3 = 4 \cdot f_0$	= 870 Hz	1,5 m
$f_4 = 5 \cdot f_0$	= 1088 Hz	1,2 m
$f_5 = 6 \cdot f_0$	= 1305Hz	1 m

Table 39

The silencer must be designed so that f_1 to f_3 are trapped. The throughput damping value for the simple **volume resonator** (expansion chamber) is:

$$D_d = 10 \lg \left[1 + \frac{1}{4}\left(q - \frac{1}{q}\right)^2 \sin^2\left(\frac{2 \cdot \pi}{\lambda} \cdot l\right)\right] \text{ in dB}$$

$$\text{where } q = \frac{A_2}{A_1} = \frac{\text{Silencer cross area}}{\text{Pipe cross area}} = \frac{D^2}{d^2}$$

l = Length of silencer

Optimum effect for $\sin\left(\frac{2 \cdot \pi}{\lambda} \cdot l\right) = 1$, i.e. when $l = \frac{1}{4}\lambda, \frac{3}{2}\lambda, \ldots$

No effect for $\sin\left(\frac{2 \cdot \pi}{\lambda} \cdot l\right) = 0$, i.e. when $l = \frac{1}{2}\lambda, \frac{3}{2}\lambda, \ldots$

In the practical example the first harmonic of the pump pulsations f_1 = 425 Hz and l_1 = 3 m must be safely trapped.

Optimum effect of silencer at:

$$l_{Sil} = \frac{1}{4}\lambda_1 = 0.75 \text{ m}$$

The form of the effect is shown diagrammatically in *Fig. 150.* It can be seen how certain harmonics of the pulsations are not trapped or only slightly damped.

In order to trap other frequencies the pure volume resonator must be supplemented by a whistle resonator - *see Fig. 150B.* Alternatively several volume resonators must be connected in series *(Fig. 149B).*

Whistle resonator

$$D_d = 10 \lg \left[1 + \frac{1}{4}(q - 1)^2 \cdot \tan^2\left(\frac{2\pi}{\lambda} \cdot l\right)\right]$$

Optimum effect at $\tan\left(\frac{2\pi}{\lambda} \cdot l\right) \cdot \rightarrow \infty$

i.e. at $l = \frac{1}{4}\lambda_x$

For f_3 = 840 Hz and λ_3 = 1.5 m

$$l_{Whistle} = \frac{1}{4}\lambda_3 = 0.375 = \frac{1}{2} l_{Silencer}$$

Fig. 150B shows a diagram of the damping value for the combined volume resonator and whistle resonator. The absolute value of damping can still be determined by the ratio q of pipe cross area to silencer cross area.

The target is a throughput loss of $D_d \approx 20$

Assumptions

q = where $D_{Sil} = \sqrt{q} \cdot d_{Pipe} = 5 \cdot d_{Pipe\,(inside\,diameter)}$

giving $D_d \approx 22$.

This provides a purely theoretical reduction of the amplitude of the pressure pulsations of frequency f_1 to 1/12 of the input value.

Such silencers are easy to make out of hydraulic cylinder components. Basically, they should have a whistle resonator as shown in *Fig. 150B.*

Due to practical considerations, i.e. the strength of the silencer pipe and the introduction of extra volume into the circuit, q = 25 should not be exceeded.

The equation for calculating the throughput loss D_d also shows that slight variations in the length of the silencer and its diameter do not seriously affect the results. This means that the calculated values of length and diameter do not have to be adhered to with any degree of accuracy.

Nevertheless, it is important for the air content of the fluid to be low and for no air bubbles to form in the silencer. It is also important for the silencer to be placed directly at the outlet from the pump.

Fig. 151 shows, with precise measurements, the values obtained from reflection silencers. A hybrid type which uses a gas cushion to achieve some absorption is also shown.

The interference line is an interesting solution to the problem which can be fitted quickly in some cases. It is a kind of by-pass in the piping system enabling one single frequency of the excitation spectrum to be eliminated - *see Fig. 149E.*

The only condition is that the difference in distance between by-pass and through-pipe must be $\lambda/2$.

A similar simple solution of equal effect uses the whistle resonator - *see Fig. 149D.* A pipe of the same diameter is fitted as a "dead" branch line. Its length should be $\lambda/4$ of the wave to be silenced. Venting is also important in this case.

6.3.3 Hybrid types

A number of modified bladder-type accumulators are offered under such names as "Pulse Tone Silencer", "Hydraulic Silencer", "Shock Absorber" and "Suction Stabilizer". The latter should be regarded as branch line resonators with a characteristic silencing effect - *see Fig. 149.*

The elasticity of the volume of gas also provides an absorption effect. Similarly, the elasticity of the gas cushion provides additional extra volume.

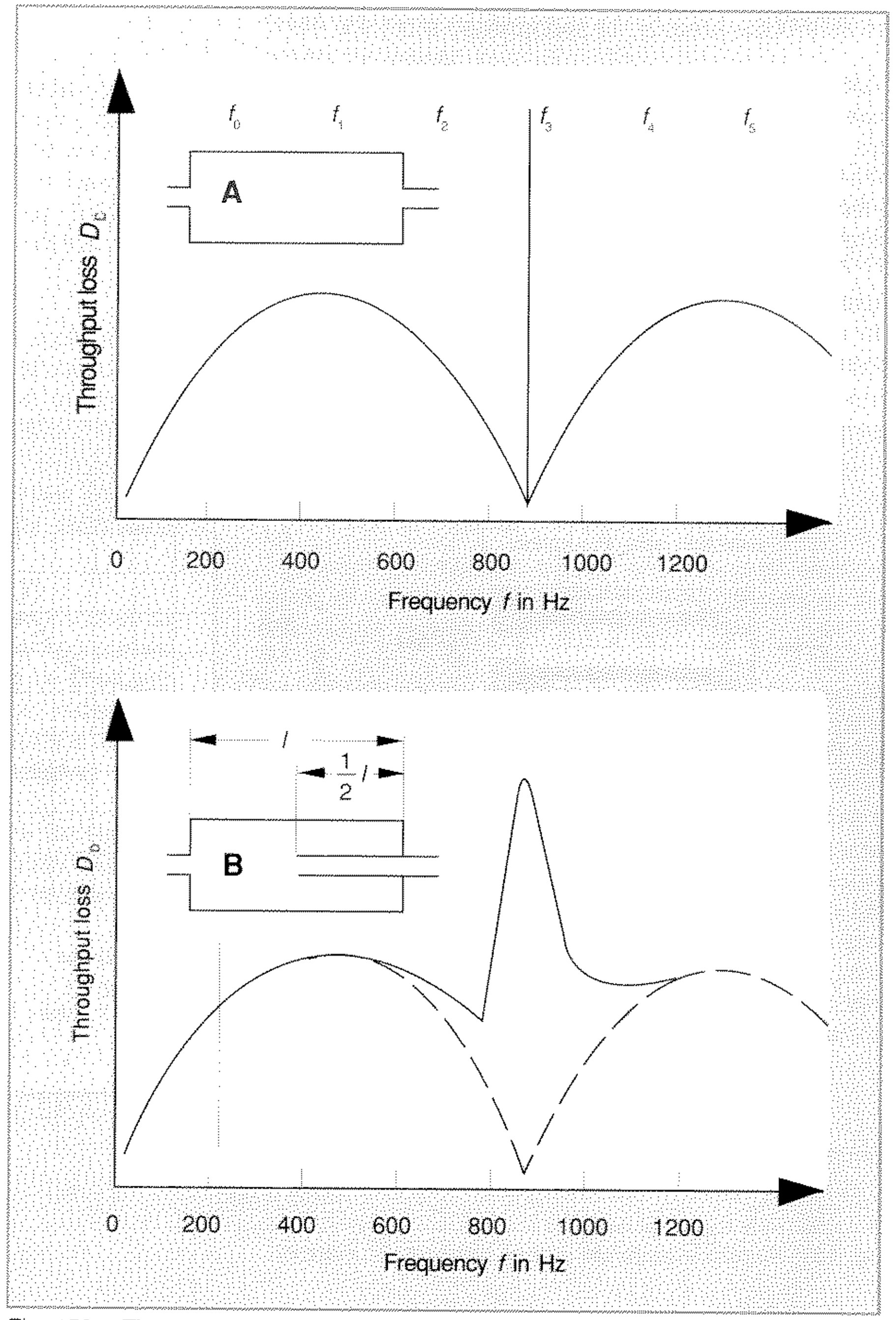


Fig. 150: *Throughput damping value versus frequency for resonators (diagrammatic illustration of the calculation example)*

Summary

Fluid silencers are a well-tried means of reducing pressure pulsations and the associated noise-inducing forces. They should be fitted when the measures for structure-borne sound decoupling produce an insufficient effect or no effect at all. They should also be used in large, extensive or subdivided systems.

Apart from noise reduction they also improve the service life and safety of the system through the reduced mechanical stresses.

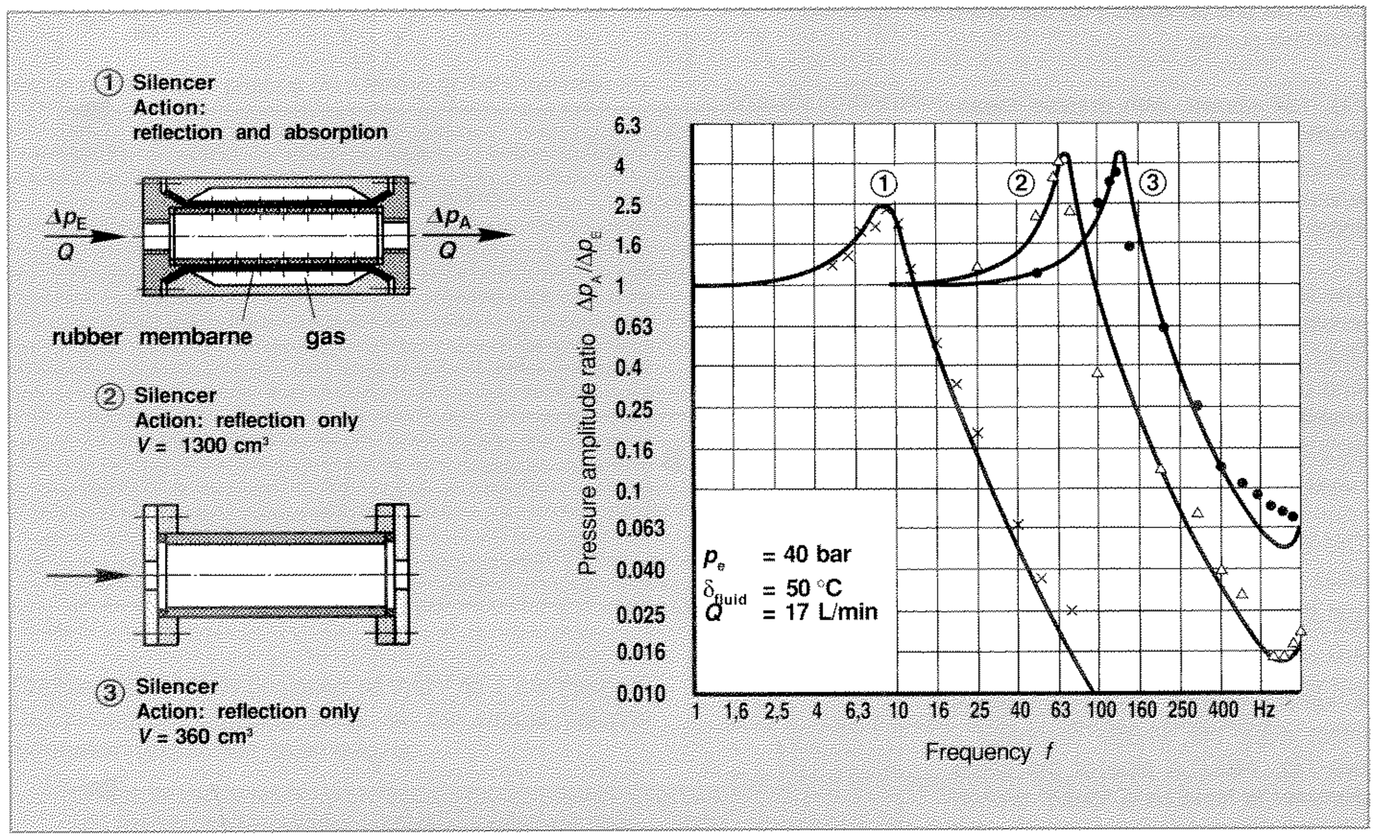


Fig. 151: *Amplitude response of fluid silencers (W. Herzog)*

6.4 Reducing noise radiation

So far this chapter has dealt with the causes of noise emission and how, through suitable measures, its propagation through the system can be prevented or reduced. The sequence in which the various aspects have been considered - pumps, structure-borne noise, fluid-borne noise - is also indicative of the importance of the various countermeasures.

Action to reduce noise radiation is always useful if it has been impossible to decouple most of the structure-borne noise and/or to make a major reduction in pressure pulsations.

Noise radiation is particularly prevalent where there are large surfaces which can be affected by even small exciting forces.

In the case of hydraulic systems such places are often the large-area tanks and the component "mounting panels". These "passive" radiators of noise have to be dealt with. As a first approximation air-borne noise radiation is proportional to the size of the excited surface and inversely proportional to the mass of the radiating object.

Therefore, the objective must be to reduce the radiating surface area and increase its mass at the same time. With tanks this can only be done by using thicker plate, possibly corrugated.

There are more opportunities for success with mounting panels for valves:

– The use of perforated plate

This type of plate produces a kind of "short-circuit" between the front and back which, up to high frequencies, allows the equalization of localized fluctuations in air pressure. This greatly reduces the radiation of air-borne noise.

– The use of framed constructions

with small individual sheet-metal panels for mounting control components. The frame itself should be of heavy construction.

– The use of modular construction

This refers to "manifold blocks" and "stacked" arrangements of equipment. Both are the ideal answer to the demand for reduced area and greater mass

Fig. 152 shows the reduction in sound level obtained by the measures described in a small hydraulic power unit of 3 kW capacity. The original proposed design was of the mounting panel type.
Replacing the mounting panel with a block produced a level reduction of about 6 dB(A). The use of an internal gear pump provided a substantial reduction in pressure pulsation amplitude, reducing the exciting forces. A further reduction in level was achieved by using anti-vibration suspension of the motor-driven pump in the tank fluid. The total reduction of nearly 22 dB(A) is dependent on the basic assumptions for this experimental case but, nevertheless, shows just what can be achieved in the way of noise reduction with hydraulic systems.

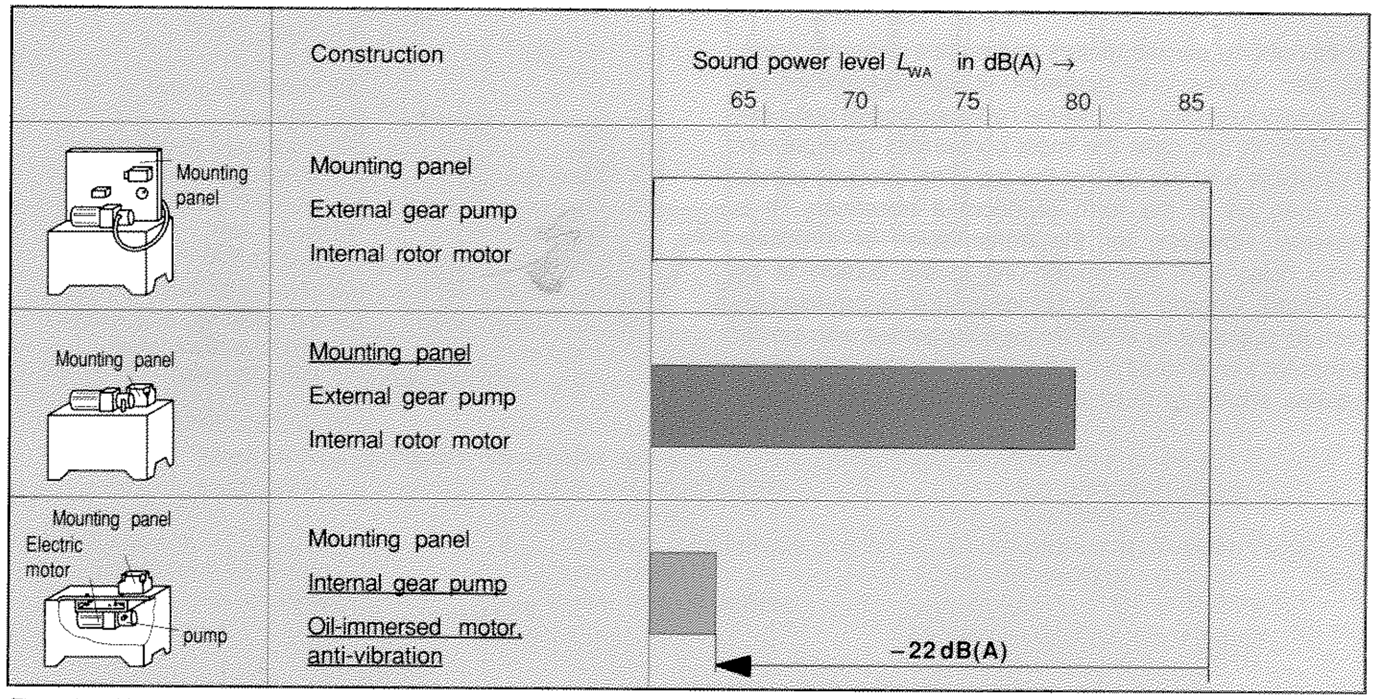


Fig. 152: *Noise-reducing countermeasures for a 3 kW hydraulic power unit*

6.5 Encapsulation

Encapsulation is the next step if the countermeasures against noise radiation have not achieved the required lower values.

It is very effective but also complicated and expensive. Another disadvantage is that accessibility and maintainability of the equipment are compromised and, in some cases, it might be necessary to adopt additional measures for the removal of heat, such as the fitting of additional air/fluid or water/fluid heat exchangers. An enclosure will only reduce the air-borne noise radiated from the piece of equipment inside. Fluid-borne noise is naturally unaffected; appropriate measures are described in *Section 6.3*.
The first step, therefore, is to find out where most of the radiated air-borne noise is coming from. Quite often it is enough to provide partial enclosure, e.g. of the power unit. This course of action is made much easier if the motor-pump unit is mounted separately from the tank - as is usually the case in the larger installations *(see Fig. 154)*.

All acoustic hoods are the same in principle. There is a supporting structure made of sectional material with removable panels attached to it. The panels comprise a supporting outer skin, an absorbent material such as mineral wool and an inner lining whose only function is protection and support - *see Fig. 153*.

Without absorption the enclosure only has an insulating effect which is often insufficient since the level inside the enclosure rises. The absorbent material uses conversion into heat to produce a real reduction in sound energy inside the enclosed space.

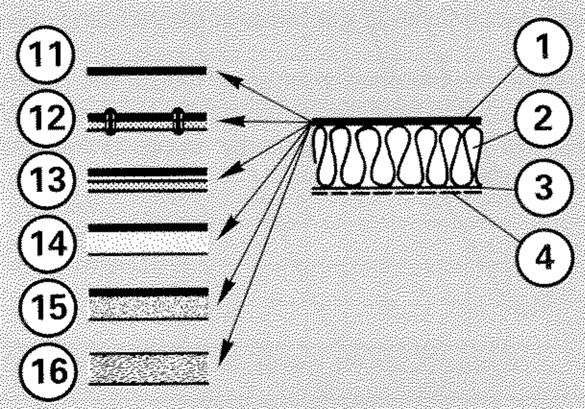
Since not only electric motors and pumps generate noise, but also valves and pipework, it can sometimes be necessary to enclose the whole hydraulic power unit - *see Fig. 155*. In this case the most convenient arrangement is a cubicle with the tank at the top. The acoustic panels can then be in the form of doors if necessary. It is essential, however, for the transmission of structure-borne noise from the power unit to the cubicle to be prevented. Cooling of the enclosed unit is provided by fresh air supplied by a fan through air baffles with similar baffles for the exhaust air. Whether or not the valve stand requires enclosure must be decided from case to case.

6.6 Screening

A full enclosure for a sound source prevents all or very little air-borne noise reaching the outside.

Screening, on the other hand, ensures that air-borne sound from a source can only reach a certain point by an indirect route. The screening takes the form of insulating and possibly absorbent movable panels and curtains. The latter especially are an often-used method of "on-site" improvements. They are usually heavy rubber or plastic mats containing lead; they are flexible and, when suspended from rails like normal curtains, can be pushed aside. Good designs can achieve level reductions of up to 10 dB(A).

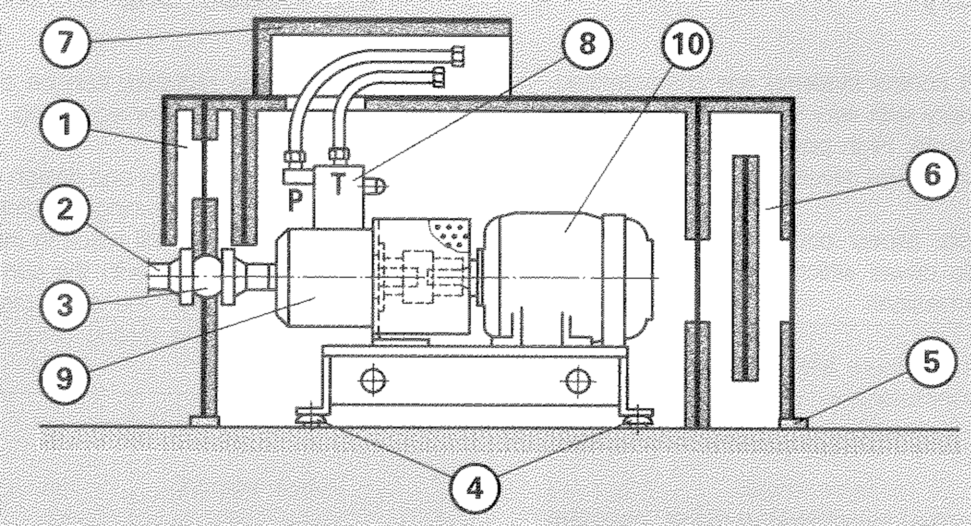
Since there is practically no absorption and air gaps usually cannot be avoided, the reductions are never the same as those achievable by full enclosure. An advantage, however, is that maintainability and accessibility can be made much better.



1 Outer skin
2 Absorbent material (40 to 80 mm)
3 Lightweight protective film ($h \approx 30$ µm)
4 Sound-permeable backing (e.g. perforated plate)
11 Single sheet (1 to 4 mm)
12 Multi-layer sheet (riveted)
13 Composite sheet (sheets bonded around insulating plastic)
14 Sheet with anti-booming covering
15 Sheet with flexible insulating covering (e.g. rubber)
16 Wood particle board (10 to 20 mm)

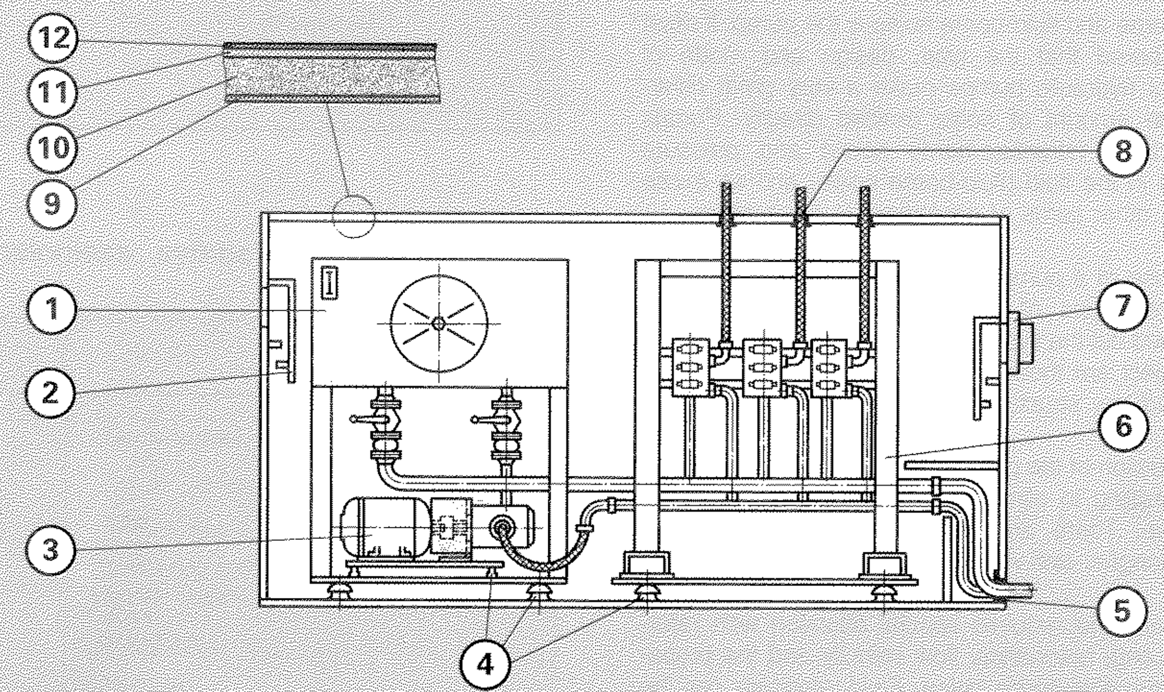
Fig. 153: *Section through an acoustic panel*

Noise Reduction



1 Air-borne sound baffle for ventilation
2 Suction line
3 Rubber expansion fitting
4 Anti-vibration mountings
5 Seal to foundation
6 Air-borne sound baffle for ventilation
7 Air-borne sound damper for discharge hoses (anywhere on the hood)
8 Pressure valve
9 Pump
10 Motor

Fig. 154: *Diagram of an acoustic hood for a motor-driven pump*



1 Tank
2 Air-borne sound baffle for ventilation
3 Pump set
4 Anti-vibration mountings
5 Damping and insulating penetration
6 Valve stand
7 Fan with sound baffles for ventilation
8 Hose from plate to outle from cubicle (better through baffles)
9 Protective covering (e.g. perforated plate)
10 Absorbent material
11 Lead-rubber insulation
12 Supporting structure

Fig. 155: *Cubicle-type hydraulic power unit*

7 Summary

Hydraulic drives and control systems are characterized by a high power density with very small physical dimensions of the components. High power in a small space usually generates high noise levels.

A systematic analysis allows countermeasures to be adopted for hydraulic drives which make a substantial reduction in noise emissions.

The main problem is that there are three mechanisms of air-borne noise transmission:

- noise radiated directly from components (e.g. pumps and valves)
- structure-borne noise transmitted through coupling points between components
- propagation of fluid-borne noise throughout the piping system.

The three mechanisms are interdependent to a certain extent and it can seldom be forecast which of them will be the governing factor for the total air-borne noise radiation from the equipment.

Usually the dominant item in the generation of air-borne, structure-borne and fluid-borne noise is the pump. It must be remembered that the type, design and mode of operation of hydraulic pumps make a very great difference in the amount of noise emitted directly. Pumps cause vibration (structure-borne noise) and pressure fluctuations in the fluid (fluid-borne noise) and so excite vibration in other items of equipment in the system.

Often the use of a pump design optimized for low noise or a different type of pump can vary the excitation spectrum and cut the system noise level. The best pumps in this respect are those with a low cyclic irregularity of delivery and low structure-borne sound amplitudes.

The construction of the pump and motor unit must incorporate decoupling to eliminate any bridges for structure-borne noise. There are numerous methods such as damping flanges, anti-vibration mountings, etc.

The pressure line requires special attention with reference to routing, resonance, flow velocity and the propagation of structure-borne noise. Fluid silencers can be used effectively for reducing pressure fluctuations in the fluid flow.

Minimum noise radiation is an aspect to consider in the design of hydraulic power units. Manifold blocks and vertical stacking satisfy the requirements for a small radiating surface area and high mass.

Air-borne noise, once radiated, can be prevented from spreading by full enclosure or screening.

Close attention at the planning and design stage to all the measures described for reducing structure-borne, fluid-borne and air-borne noise will result in installations with noise levels fully compatible with the current legislation. Remember, of course, that the extra planning, design, equipment, installation and testing costs money.

8 Symbols and subscripts

Symbols

Symbol	Units	Quantity
c	m/s	Velocity of sound
d	cm^2	Diameter (pipe)
D	cm^2	Diameter (silencer)
f	1/s	Frequency
F	N	Force
l	m	Length
n	1/min	Speed
p	bar	Pressure
P	W	Power
S	m^2	Surface area (enveloping)
λ	m	Wavelength
v	m/s	Vibration velocity

Subscripts

Symbol	Quantity
eff	Mean value
p	Sound pressure
W	Power
ac	Acoustic
0	Reference value, Basic value
e	Input
a	Output

Dimensionless symbols

Symbol	Quantity
L_W	Sound power level
L_p	Sound pressure level
L_S	Measuring-surface level
$\bar{L}_S$	Measuring-surface sound pressure level (mean value)
D_d	Throughput loss
k	Number
q	Area ratio

9 References

[1] J. Rébel, G. Schmid
"Geräuschemission von Hydroelementen und Hydrosystemen und Maßnahmen zur Lärmminderung".
Forschungsbericht Nr. 180 der Bundesanstalt für Arbeitsschutz und Unfallforschung, Dortmund
189 Seiten, viele Quellenhinweise
ISBN 3-88314-009-0

[2] VDI-Richtlinie 3720 Blatt 5
Lärmarm Konstruieren
Hydrokomponenten und Systeme
Beuth Verlag, Berlin
24 Seiten, viele Quellenhinweise

Notes

The Sizing of Pipework in Hydraulic Systems

Dr.-Ing. Norbert Achten

1 Introduction

The obvious task of the pipework in hydraulic systems is to carry the hydraulic fluid to and from the various components. In the process, it is subjected to mechanical, thermal and corrosive stresses either individually or all at once. It is these stresses that are the governing factor for the sizing of the pipework.

Mechanical stresses are mostly as a result of the pressure varying with respect to time. The task of the designer is to produce an economical, safe and reliable design appropriate for these circumstances. The procedure for attaining this target is shown diagrammatically in Fig. 156. There are company specifications and standards to be followed as well as all the general rules and recommendations that are applicable.

The procedure for the design and sizing of pipework is based on the original circuit diagram and the principal data such as the medium to be used, the flow per unit time, the pressure and the temperature. As can be seen from *Table 40* there are also a number of other factors of influence affecting the principal parameters which must also be included in the calculations, i.e.

- pipe inside diameter
- wall thickness and
- material.

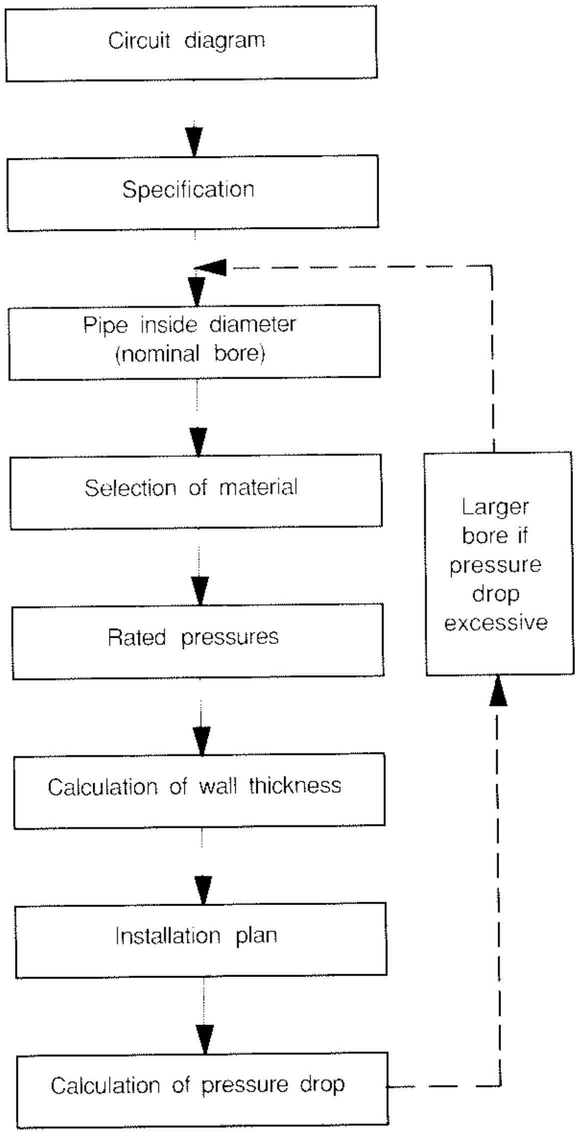


Fig. 156: *Sequence diagram for the design and sizing of hydraulic system pipework*

Parameter	Factors of influence
Pipe inside diameter	Volumetric flow Flow velocity Fluid viscosity Pressure losses
Pipe wall thickness	Operating pressure (and any extra stresses) Safety factors required or specified Reduced wall thickness due to manufacturing Internal and external corrosion Strength of material Operating and ambient temperatures Standard dimensions
Pipe material	Strength parameters Preconditions for use (surface finish, weldability) Effects of corrosion Permitted temperature range

Table 40: *Factors of influence for the sizing of pipework*

2 Determining the inside diameter or bore

Together with the flow and physical properties of the hydraulic fluid, the inside diameter or bore of the pipes also affects the flow resistance. In order to determine the required pump power it is necessary to calculate the total flow losses in the system. If the calculated pressure losses are too high compared with the projected values, the pipework will have to be re-sized choosing a larger bore.

The flow $\dot{V}$ given in the principal data is used for determining the pipe bore d_i. According to Equation (1) it is:

$$d_i = \sqrt{\frac{4 \cdot \dot{V}}{w\,\pi}}\,. \qquad (1)$$

Substituting the flow in L/min and the average velocity w in m/s in Equation (1) gives the pipe bore in mm thus:

$$d_i = 4{,}607\,\sqrt{\frac{\dot{V}}{w}}\,. \qquad (2)$$

The average velocity in Equations (1) and (2) must be determined according to economic and technical factors. The economic aspect includes capital costs and operating costs. On the technical side there are also limits related to the flow, which, if exceeded, lead to noise emission, excessive vibration of the pipework and erosion at points of change in direction. Starting points for the selection of an average velocity [1] can be taken from *Table 41* which shows recommended figures from both German and American sources.

Determining the bore of the pipe also fixes the nominal size (abbreviation DN) of the pipe to DIN 2402 [2] *(see Table 42)*. This has the advantage that all other pipe components used will be of the appropriate size for connection.

3 Selecting the material

The choice of material for the pipework depends mainly on the strength needed, although there are other significant factors such as the method of manufacture - seamless or welded, any subsequent fabrication needed and the suitability of pipe connectors. In view of possible internal or external corrosion the resistance of the material to such attack must also be examined.

In the field of hydraulics, the most widely used material up to DN 32 is precision steel tube to DIN 2391-C [3] made of St 35 supplied normalized (DIN 2391, Part 2). Its popularity is due to its excellent suitability for fabrication (welding, bending and flaring), high resistance to repeated stress and excellent matching of its outside diameter to pipe connectors. For the extra strength required for high-pressure applications St 52 can be used instead of St 35. Seamless steel tube to DIN 2448 [4] and 2445 [5] in St 37.4 or St 52.4 to DIN 1630 [6] is used for sizes over DN 40.

Suction line		Pressure line		Return line
Kinematic viscosity ν in mm²/s	w in m/s	pressure p in bar	w in m/s	w in m/s
150	0,6	25	2,5 to 3	1,7 to 4,5
100	0,75	50	3,5 to 4	
50	1,2	100	4,5 to 5	
30	1,3	200	5 to 6	
		> 200	6	
		When ν= 30 to 150 mm²/s		

Table 41: *Recommended values of flow velocity in hydraulic system pipework*

	10	100
	12	125
	15	150
	20	200
	25	250
3	32	300 350
4	40	400 450
5	50	500
6	65	600 700
8	80	800 900

Table 42: *Nominal sizes (DN) of pipework to DIN 2402 (extract)*

Federal Republic of Germany			United Kingdom		USA		France	
Material designation	Material No.	Standard	Material designation	Standard	Material designation	Standard	Material designation	Standard
St 37.4	1.0255	DIN 1630	CDS 23	BS 3602	A	ASTM A53	—	—
St 52.4	1.0581	DIN 1630	HFS 23	BS 1775	3	ASTM A252	—	—
St 35	1.0308	DIN 2391	CDS 3	BS 980	1010	ASTM A519	Tu 37-b	PRA 49-310
St 37.0	1.0254	DIN 1626	ERW 360	BS 3601	A	ASTM A53	Tu 37-b	A 49.112
X6CrNiMoTi 17122	1.4571	DIN 17 458	320 S17	BS 970 P.4	316Ti	AISI	Z8CNDT 17-12	A 35-572
X6CrNiTi 1810	1.4541	DIN 17 458	321 S12	BS 970 P.4	321	AISI	Z6CNDT 18-11	A 35-572

Table 43: *Preferred pipe materials in German and foreign standards*

Due to the high pressures involved, the only types of welded tube used should be those complying with special quality standards (Group 2) and having a ratio of weld strength to parent metal strength of unity. Welded tube cannot be used with cutting ring fittings or flared joints. Table 43 shows a comparison of the preferred pipe materials for hydraulic systems according to German and foreign standards. It also lists stainless steels for precision tube to DIN 2463 [7]. The foreign materials listed in the table are equivalent to the German ones and must be used accordingly.

It is usually necessary for tube exposed to high pressures to be certified in accordance with DIN 50049-3.1B [8]. So-called "commercial quality" tube should not be used because its pressure range is limited and larger safety factors would have to be allowed.

4 Nominal pressures

The nominal pressure of pipe and pipe components is the name given to the pressure grading of components of identical construction and identical connection dimensions. The standard pressure grades are listed in DIN 2401, Part 1 [9] (Table 44). A plain figure is stated for nominal pressure (abbreviation PN); there are no units such as "bar". The numerical value of nominal pressure is the maximum pressure of use at a reference temperature of 20°C.

1	10	100	1000
1,6	16	160	1600
2,5	25	250	2500
4	40	400	4000
6	63	630	6300

Table 44: *Pressure grading (PN) of pipe to DIN 2401, Part 1*

5 Calculating the wall thickness

Calculation of the required wall thickness of a pipe can generally be performed for a specific load to DIN 2413 [10] or, as part of a pressure vessel subject to certification, according to AD-Merkblatt B1 [11]. These calculation references are applicable to piping systems which are either operated in Germany or are recognized by the competent certification authority when installed in other countries. The overview (Table 45) lists the calculation formulae for determining the theoretical wall thickness according to these regulations. The safety factors S in Formulae (3) to (6) and the weld efficiencies v can be taken from *Tables 46 and 47. Table 46* also lists the strength coefficients K to be used in the individual formulae.

The formulae to DIN 2413 are based on the requirement that the operating pressure must cause no plastic flow of the material at the most highly stressed inner fibre of the pipe.

There are three load cases to take into account:

- Case I

  Primarily steady-state stress up to a maximum temperature of 120°C

- Case II

  Primarily steady-state stress over 120°C (also applicable to temperatures below 120°C under certain circumstances)

- Case III

Repeated stress.

Cases I and II are based on primarily steady-state stress whereby certain maximum numbers of stress cycles must not be exceeded. "Stress cycles" means alternating pressures of large amplitude such as when starting up and shutting down a hydraulic system. *Tables 48 and 49* list the maximum numbers of stress cycles in relation to the tensile strength R_m and the permitted stress K/S for two different types of steel tube. When numbers of stress cycles above the specified limits are anticipated, the theoretical wall thickness for predominantly steady-state stress is calculated first.

Reference	Application limits	Type of stress	Formulae for theoretical wall thickness
DIN 2413	$d_a/d_i \leq 1{,}7$ Temperature ≤ 120 °C	I, primarily steady-state	$s_v = \dfrac{d_i \cdot p}{20\dfrac{K}{S} \cdot v - 2p}$ (3)
DIN 2413	a) $d_a/d_i \leq 1{,}7$ Temperature > 120 °C b) $d_a/d_i \geq 1{,}1$ and $\leq 1{,}7$ Temperature < 120 °C	II, primarily steady-state	$s_v = \dfrac{d_i \cdot p}{(20\dfrac{K}{S} - p) \cdot v}$ (4)
DIN 2413	$d_a/d_i \leq 1{,}7$	III, repeated	a) s_v according to formulae (3) b) $s_v = \dfrac{d_i \cdot (\hat{p} - \check{p})}{20\dfrac{K}{S} - 3 \cdot (\hat{p} - \check{p})}$ (5) Use $s_{v\,max}$ from a) and b)
AD-Merkblatt B1	$d_a/d_i \leq 1{,}2$ or $d_a \leq 200$ mm and $d_a/d_i \leq 1{,}7$	primarily steady-state	$s_v = \dfrac{d_i \cdot p}{20\dfrac{K}{S} \cdot v - p}$ (6) $s_{v\,min} = 2$ mm

Table 45: *Basis of calculation to DIN 2413 and AD-Merkblatt B1*

Reference	Strength coefficient K	Elongation at fracture A_5	Safety factor for pipes with acceptance test certificate to DIN 50 049 S
DIN 2413 Case I	$R_{p\,0,2}$ at 20 °C	≥ 25 % 20 % 15 %	1,5 1,6 1,7
DIN 2413 Case II	a) Minimum value from $R_{p\,0,2}$ *) and $R_{m/2 \cdot 10^5}$ at calculation temperature		1,5
	b) $R_{p\,0,2}$ at 20 °C	≥ 25 % 20 % 15 %	1,6 1,7 1,8
DIN 2413 Case III	σ_{Sch}		1,5
AD-Merkblatt B1	$R_{p\,0,2}$ or $R_{m/10^5}$ at calculation temperature to AD-Merkblatt W4		1,5

**) Rp1 at calculation temperature can be used for pipes of 1.4571 or 1.4541*

Table 46: *Safety factors*

Pipes	Material complying with	Tests	Weld efficiency v
For general use (commercial quality) DIN 1626	DIN 17 100 Quality group 1	without factory certification with factory certification	0,5 0,7
With quality specification DIN 1626	DIN 17 100 Quality group 2	without user's inspection with user's inspection	0,8 0,9
With special quality specification*	At least DIN 17 100 Quality group 2	Special tests, primarily 100% weld seam test	1,0

**As longitudinally seam-welded tube, this is to be preferred for hydraulic applications because of the pressures involved*

Table 47: *Weld efficiency of longitudinally-welded tube to DIN 2413*

The calculation for Case III must also be performed, whereby only stress cycles of the same pressure fluctuation amplitude between maximum pressure $\hat{p}$ and minimum pressure $\check{p}$ are taken into account. The larger wall thickness of the two calculations is then used. In the case of pipes which are exposed to a changing pressure fluctuation amplitude at irregular intervals, calculation of the wall thickness by means of the given formulae is impossible. Special studies are necessary in such cases concerning primarily the checking of damage anticipated during normal operation.

Unlike steady-state stress, dynamic stress also requires the appropriate strength coefficients to be brought in. The simplest way of doing this for a repeated internal pressure stress is with a stress-number diagram (SN diagram). *Diagram 55* shows how the calculation of the pipe wall thickness under dynamic stress can be performed for either the fatigue limit range or fatigue strength range. It should be noted that the fatigue life is related to the appropriate number of stress cycles, i.e. the calculation of pipe wall thickness is only valid for this number of stress cycles.

Permitted stress K/S in N/mm²	Tensile strength R_m in N/mm² ≤ 450	500	550	600	650
160	100 000	> 100 000	> 100 000	> 100 000	> 100 000
180	50 000	90 000	> 100 000	> 100 000	> 100 000
200	30 000	50 000	80 000	> 100 000	> 100 000
250	10 000	17 000	26 000	40 000	56 000
300				16 000	22 000
350					10 000

Table 48: *Maximum number of stress cycles to DIN 2413 (Cases I and II) for seamless and HF-welded steel tube (v = 1) with a stress cycle safety factor S_L = 10*

Permitted stress K/S in N/mm²	tensile strength R_m in N/mm² ≤ 500	550	600	650	700
120	32 000	50 000	80 000	>100 000	> 100 000
140	18 000	26 000	40 000	56 000	80 000
160	10 000	15 000	22 000	30 000	42 000
180	6 000	10 000	13 000	19 000	25 000
200	4 000	6 000	8 000	11 000	16 000
250			3 000	5 000	6 000
300				2 000	3 000

Table 49: *Maximum number of stress cycles to DIN 2413 (Cases I and II) for submerged-arc welded steel tube (v = 1) with a stress cycle safety factor S_L = 10*

In contrast, there is no limiting number of stress cycles for fatigue strength. *Diagrams 56 and 57* show the SN diagrams for seamless, HF-welded and submerged-arc welded steel tube to DIN 2413 subjected to pulsating stress; the strength values can be taken directly from the diagrams.

In the case of pipes that form part of a pressure vessel, the calculation of the tube wall thickness is performed in accordance with AD-Merkblatt B1 *(see Formula 6)*. In this case the governing strength coefficients are those given in Material Sheet AD-W1.

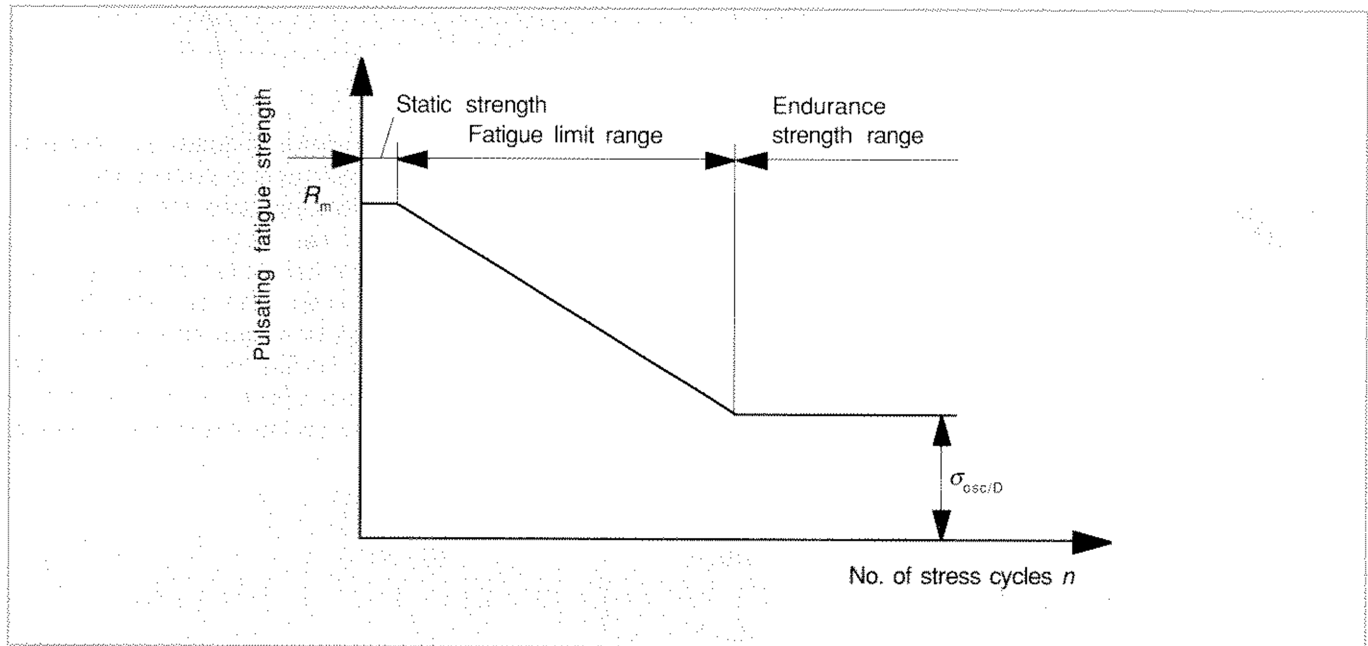


Diagram 55: *SN diagram*

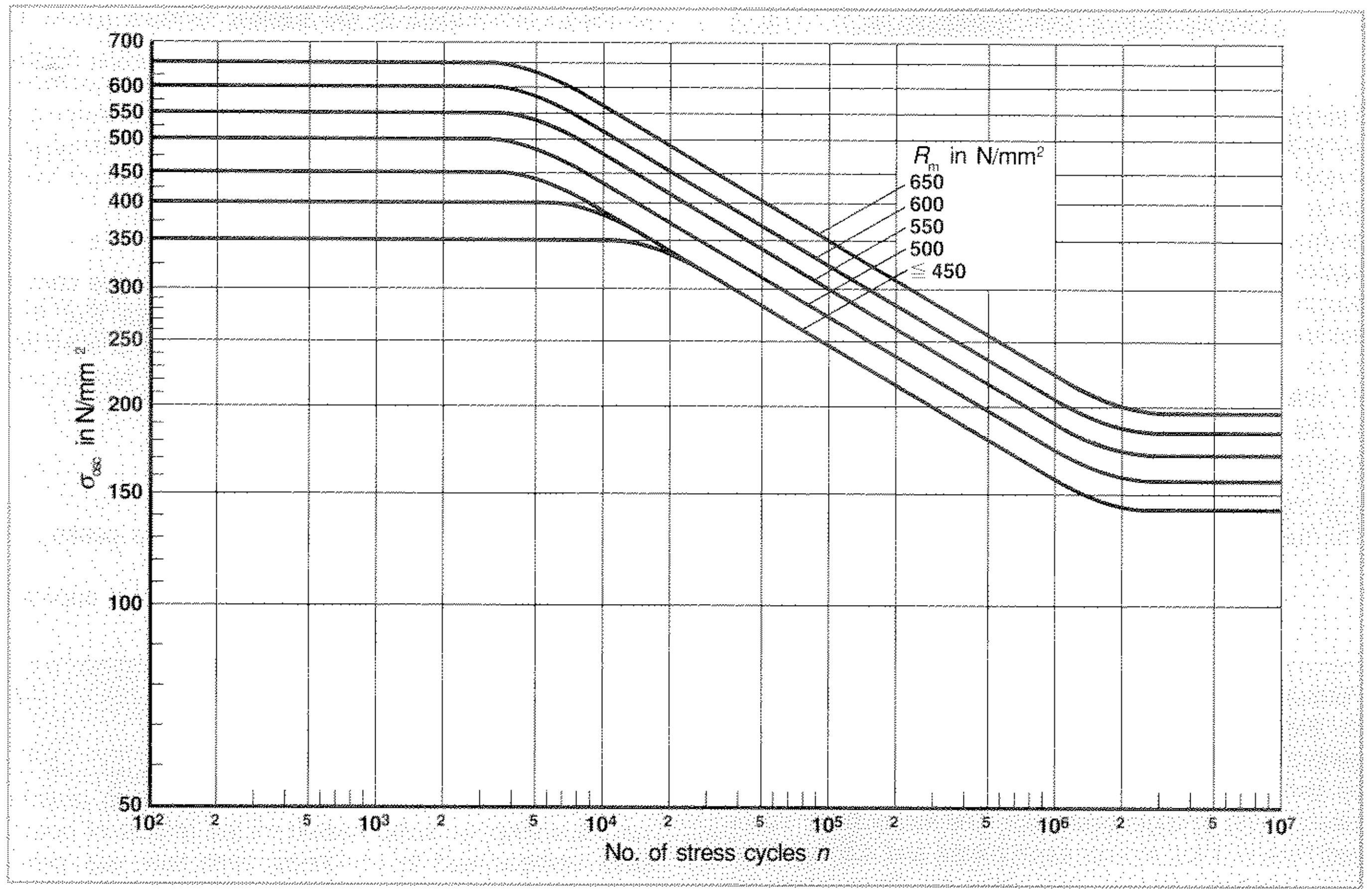


Diagram 56: *Strength under pulsating stress of seamless and HF-welded steel tube (v = 1)*

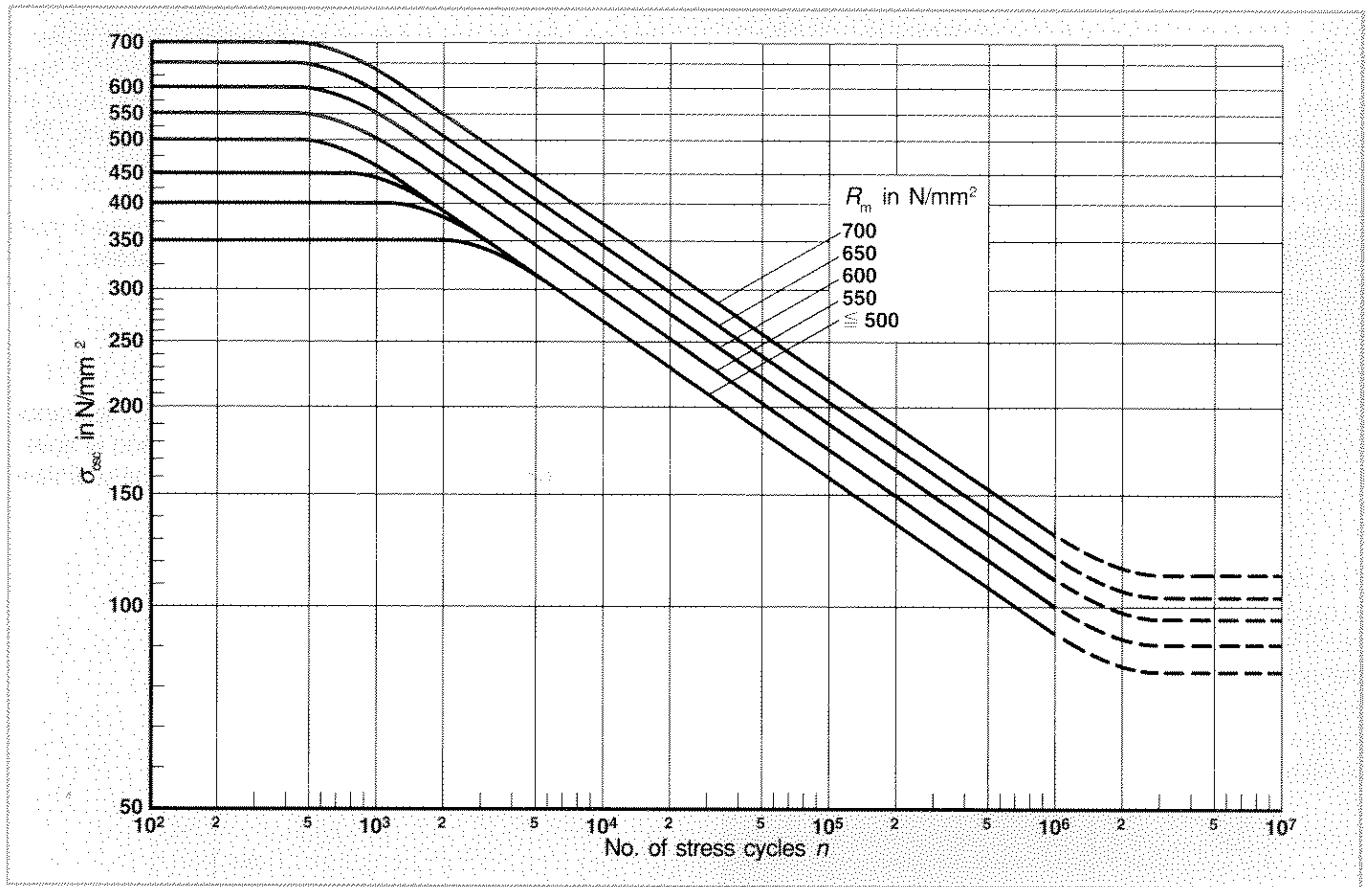


Diagram 57: *Strength under pulsating stress of submerged-arc welded steel tube (v = 1)*

5.1 Calculation of wall thickness to non-German standards

Special calculation formulae taken from the relevant guide-lines of the foreign acceptance authorities must be employed for installations built to non-German regulations. It must be established before the pipework is designed what regulations are to be employed for the calculation. The following formulae are representative of the principal foreign regulations; there is no claim that the list is complete:

– Wall thickness calculation to British Standard BS 778, Appendix A

In the United Kingdom the calculation formula for pipe wall thickness is laid down in BS 778, Appendix A [12]. The formula is:

$$s_v = \frac{d_i \cdot p}{(20 \frac{K}{S} - p)} + x \, . \qquad (7)$$

Formula (7) is identical to DIN 2413/II and AD-B1 except for the wall thickness allowance x. The allowance is generally 12.5% and takes care of any weakness caused by bending. The minimum value of tensile strength is taken as the strength coefficient. The safety factor specified here is 4 for non-corrosive media and 4.5 for water. The minimum bending radii must be three times the outside diameter of the pipe and the eccentricity must not exceed 5% of the outside diameter.

– Pipe calculations to French Standard NF A 49-300

Unlike their German and British counterparts, the French standard for pipe calculations [13] does not specify a direct calculation of the wall thickness but gives basic guide-lines for the calculation of the test pressure and bursting pressure of actual tubes. The information allows the following calculation formulae for determining the pipe wall thickness to be derived:

$$s_v = \frac{d_i \cdot p'}{20\,K - 1{,}2\,p'} \, , \qquad (8)$$

$$s_v = \frac{d_i \cdot p}{\frac{20\,K}{S} - 1{,}2\,p} \, . \qquad (9)$$

The theoretical wall thickness for the test pressure p' is ascertained with Formula (8), with 90% of the $R_{p0.2}$ yield point being used as the strength coefficient. In contrast, the operating pressure and the minimum tensile strength are used in Formula (9). The French standard does not state any required safety factor S. It simply refers to the existing regulations for different fields of application.

– Pipe calculations to the Barlow formula (USA)

Most American regulations for pipe calculations make use of the Barlow formula [14]

$$s_v = \frac{d_i \cdot p}{20 \frac{K}{S} - 2\,p} \qquad (10)$$

Basically, Formula (10) is broadly identical to the calculation formula of DIN 2413, Case I except that the minimum tensile strength of the pipe material is used as the strength coefficient. The following safety factors are applicable to the different applications:

$S = 4$ - normal operating conditions

$S = 6$ - high hydraulic and mechanical stress peaks

$S = 8$ - harsh operating conditions, dangerous applications.

5.2 Influencing variables

The calculation formulae of German regulations contain no influencing variables related to the effects of corrosion, vibration due to non-steady-state flow, e.g. pressure pulsation, or others. However, such factors must not be underestimated because, under some circumstances, they can establish different preconditions for the design data or exceed them by a multiple. In the case of pressure pulsations that can occur as a result of fast-closing valves, the magnitude of the pulsations must be calculated and added to the operating pressure. The relevant formulae for this are given in DIN 2413.

The final determination of the pipe wall thickness s must take into account two other influencing variables:

- pipe wall thickness undersize c_1
- wear due to corrosion c_2.

The pipe wall thickness undersize (minimum tolerance level) arises from manufacturing tolerances and is specified in the conditions of supply and delivery for seamless and welded tube *(see Table 50).*

The allowance for corrosion is normally 1 mm for ferritic steels. This allowance can be dispensed with if media or environmental conditions are involved which cause no corrosive attack at all. It can also be dispensed with for austenitic (i.e. stainless) pipe materials. The actual pipe wall thickness required is then:

$$s = s_V + c_1 + c_2 \qquad (11)$$

If the wall thickness undersize is given in %, s can be calculated from Formula (12)

$$s = (s_V + c_2) \cdot \frac{100}{100 - c_1} \qquad (12)$$

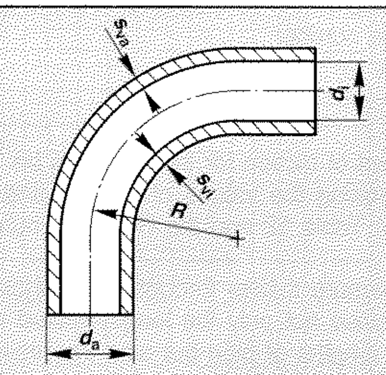
5.3 Calculation of pipe bends

The calculations for the wall thickness of pipe bends are mostly based on the calculations for straight lengths of pipe. However, as a result of bending, greater values of wall thickness undersize and different stressing of the outside and inside of the bend must be anticipated. In order to prevent an adverse effect on the endurance limit at repeated stress, primarily during dynamic stressing of the pipe, there must be no major wall thickness undersize or flattening as a result of bending. Therefore, the bending radii should not be made too small. DIN 5508 [15] recommends suitable bending radii in relation to the outside diameter *(see Table 51).* The ratio of bending radius to outside diameter of the pipe is usually 2.5 to 3 and is given as a minimum dimension for thin-walled tube.

	Outside diameter d_a in mm	Wall thickness s	Wall thickness undersize c_1
Seamless precision steel tube nato ch DIN 2391, Part 1	< 5 $6 \le d_a \le 8$ > 8		20 % 15 % 10 %
Seamless precision steel tube to DIN 1629 (extract)	≤ 130	$< 4\,s_n$ * $> 4\,s_n$	10 % 9 %
Welded steel tube to DIN 1628		$s \le 3$ mm 3 mm $< s \le 10$ mm $s > 10$ mm	0,25 mm 0,35 mm 0,50 mm

* s_n Normal wall thickness to DIN 2448

Table 50: *Permitted wall thickness undersize of seamless and welded steel tube*



d_a in mm	6	8	10	12	14 15 16	18
R in mm	16	20	25	32,5	40	45
d_a in mm	20	22 25	28 30	35 38	42	60
R in mm	55	65	80	100	110	160

Table 51: *Recommended bending radii to DIN 5508 (extract)*

The required wall thickness on the inside and outside of the bend (sv_i and sv_o) is calculated according to DIN 2413 from:

$$s_{vi} = s_v \cdot B_i \text{ and} \quad (13)$$

$$s_{vo} = s_v \cdot B_o. \quad (14)$$

In Formulae (13) and (14) B_i and B_o are coefficients that can be taken from Diagram 58 with reference to the bend radius R/d_i and the parameter s_v/d_i. For thin-walled tubes (s_v/d_i < 0.02) approximate values of the coefficients can also be calculated with the following formulae

$$B_i = \frac{2R - \frac{d_o}{2}}{2R - d_o}, \quad (15)$$

$$B_o = \frac{2R + \frac{d_o}{2}}{2R + d_o}. \quad (16)$$

6 Calculating the pressure losses

The pressure drop Δp_l due to friction between the hydraulic fluid and the inner wall of the tube can be calculated from:

$$\Delta p_\lambda = \lambda \frac{L}{d_i} \cdot \rho \cdot \frac{\overline{w}^2}{2}. \quad (17)$$

Where λ is the coefficient of friction, L the length of the pipe and ρ the density of the fluid. The coefficient of friction depends on the surface finish k of the tube and on the Reynolds number

$$Re = \frac{\overline{w} \cdot d_i}{\nu}. \quad (18)$$

The coefficient of friction for the pipe can be taken from *Diagram 59* using the values of surface finish for steel tube given in Table 52 and the Reynolds number.

The total pressure losses throughout a whole hydraulic system comprise not only the length-related resistances of individual lengths of pipe but also the pressure drops across individual points of resistance as represented by valves, fittings and other similar components. Therefore, it is useful to calculate the total pressure drop Δp_v from the resistance coefficient of all individual resistances. This gives the total pressure drop thus:

$$\Delta p_v = \Delta p_\lambda + \Delta p_\xi \quad (19)$$

and

$$\Delta p_\xi = \Sigma \xi \cdot \rho \frac{\overline{w}^2}{2}. \quad (20)$$

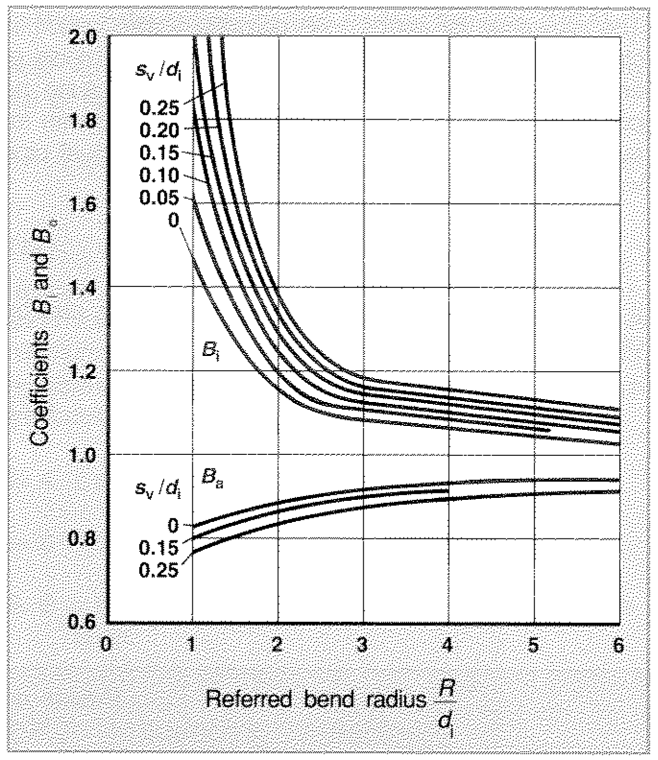


Diagram 58: *Coefficients for calculating the wall thickness of tube bends*

Substituting Formulae (17) and (20) in Formulae (19) gives:

$$\Delta p_v = \left(\frac{\lambda L}{d_i} + \Sigma\xi\right) \cdot \rho \cdot \frac{\overline{w}^2}{2}. \quad (21)$$

The resistance values of the proposed valves and fittings can be taken from the manufacturer's catalogues. In most cases manufacturer's documentation also gives the pressure drop directly in the form of a curve versus flow. This allows the pressure drops of the individual fittings for the particular value of flow to be summated and substituted directly in Formula (19).

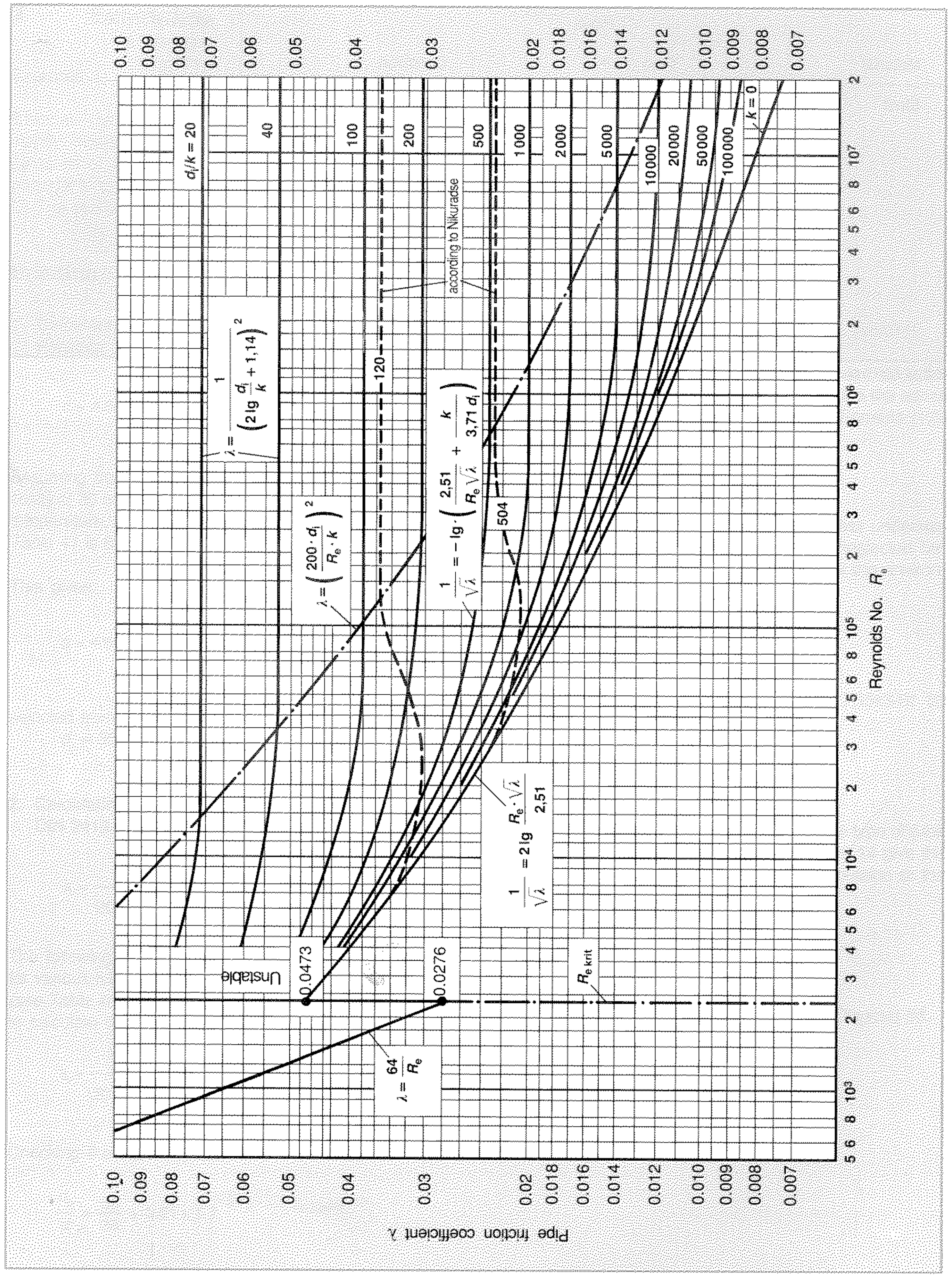


Diagram 59: *Pipe coefficient of friction l versus Reynolds number Re (see [16] (for example))*

Material	Pipes: Type	Pipes: Condition	Absolute roughness k in mm
Steel	Seamless (commercial quality)	new • as-rolled • pickled • galvanized	 0.02 to 0.06 0.03 bis 0.04 0.07 bis 0.10
	Longitudinally seam-welded	new • as-rolled • bitumenized • galvanized	 0.04 to 0.10 0.01 to 0.05 0.008
	Seamless and longitudinally seam-welded	used • medium rusting or slight incrustation	0.1 to 0.2

Table 52: *Internal surface finish of steel tube (see [16](for example))*

7 Examples

Example 1

Calculate the size of a pressure line subjected to predominantly steady-state stress by a maximum operating pressure of 210 bar and a temperature of 50°C. The material is to be precision steel tube St 35 to DIN 2391-C. The pump delivery is 160 L/min.

Solution

1. Calculating the pipe inside diameter using Formula 2

$$d_i = 4{,}607\sqrt{\frac{\dot{V}}{w}}\,.$$

Assuming that the mineral-oil fluid has a kinematic viscosity of 30 mm²/s and a density of 0.3 g/cm³ at operating temperature, the average value of flow velocity given by *Table 41* is 6 m/s.

This gives:

$$d_i = 4.607\sqrt{\frac{160}{6}} = 23.79\text{ mm},$$

selected from Table 55:
$d_o = 35$ mm, $s = 3$ mm, $d_i = 29$ mm

2. Calculating the required wall thickness to DIN 2413, Case I (see Formula 3)

$$s_v = \frac{d_i \cdot p}{20 \cdot \frac{K}{S} \cdot v - 2p}$$

The following *Table 53* lists the mechanical parameters for various tube materials. From $K = 235$ N/mm², $S = 1.5$ *(see Table 46)* and $v = 1$ for seamless tube it is possible to calculate the wall thickness from

$$s_v = \frac{29 \cdot 210}{20 \cdot \frac{235}{1{,}5} - 2 \cdot 210} = 2.25\text{ mm}.$$

Checking the diameter ratio

$$\frac{d_o}{d_i} = \frac{35}{29} = 1.21 < 1.7$$

3. Calculating the actual wall thickness required

As the wall thickness undersize is given in % for the selected type of precision seamless steel tube the actual wall thickness required is calculated from Formula (12).

$$s = (s_v + c_2) \cdot \frac{100}{100 - c_1}$$

Table 50 shows the appropriate wall thickness undersize to be 10%. The wear allowance can be neglected because there is no corrosion from the fluid or the environment.

$$s = 2.25 \cdot \frac{100}{100 - 10} = 2.5\text{ mm} < 3\text{ mm}$$

Therefore, the selected tube DIN 2391-C-35x3-St 35 NBK is adequately sized.

4. Calculating pipe bends

Table 51 shows the bending radius R for the type of tube being used to be 100 mm. Formulae 13 and 14 give the required wall thickness for the inside and outside of the bend as:

$$s_{vi} = s_v \cdot B_i$$
$$s_{vo} = s_v \cdot B_o$$

Coefficients B_i and B_o can be taken from *Diagram 58.*

$$\frac{R}{d_i} = \frac{100}{29} = 3.45 \quad \text{and}$$

$$\frac{s_v}{d_i} = \frac{2.25}{29} = 0.078$$

gives $B_i = 1.15$ and $B_o = 0.92$

$$s_{vi} = 2.25 \cdot 1.15 = 2.59 \text{ mm}$$
$$s_{va} = 2.25 \cdot 0.92 = 2.07 \text{ mm}$$

The pipe must be bent so that there is no undersize of these two values of wall thickness on the inside and outside of the bend.

5. Calculating the pressure losses

The pressure drop per unit length due to pipe friction can be calculated with Formula 17:

$$\frac{\Delta p_\lambda}{L} = \lambda \cdot \frac{1}{d_i} \cdot \rho \cdot \frac{\overline{w}^2}{2}$$

Determining the pipe coefficient of friction

First calculate the Reynolds number with Formula 18:

$$Re = \frac{\overline{w} \cdot d_i}{\nu}$$

The velocity for the type of tube selected is given by:

$$\overline{w} = \frac{\dot{V}}{d_i^2 \cdot \frac{\pi}{4}} = \frac{160 \cdot 10^3}{29^2 \cdot \frac{\pi}{4} \cdot 60} = 4.04 \text{ m/s}.$$

which gives:

$$Re = \frac{4.04 \cdot 29 \cdot 10^3}{30} = 3905.3$$

Table 52 shows the average value of internal roughness of a seamless steel tube with a rolled finish to be 0.04 mm.

Using the ratio: $\frac{d_i}{k} = \frac{29}{0,04} = 725$

it is possible to read off the value of 0.04 from *Diagram 59* for the coefficient of friction. The pressure drop per unit length can now be calculated thus:

$$\frac{\Delta p_\lambda}{L} = 0.04 \cdot \frac{1}{29} \cdot 0.9 \cdot \frac{4.04^2}{2} \cdot 10 = 0.101 \frac{\text{bar}}{\text{m}}.$$

Example 2

For the type of tube used in *Example 1* now determine what maximum value of pulsating pressure can be tolerated continuously.

Solution

The calculation is performed according to Case III in DIN 2413. The lower value of operating pressure is taken as zero, which gives the maximum pressure from Formula 5 as follows:

$$\hat{p} = \frac{20 \cdot \frac{K}{S} \cdot s_v}{d_i + 3\, s_v}.$$

In this case the endurance limit at repeated stress must be used for the strength coefficient *K* in *Table 46*; for this tube material it is 226 N/mm² *(see Table 53)*. The theoretical value of wall thickness to be used for the existing tube is calculated with Formula 12 thus:

$$s_v = s \cdot \frac{100 - c_1}{100} - c_2 = 3\,\frac{100 - 10}{100} = 2.7 \text{ mm}$$

Which gives:

$$\hat{p} = \frac{20 \cdot \frac{226}{1.5} \cdot 2.7}{29 + 3 \cdot 2.7} = 219.3 \text{ bar}.$$

According to Case I the maximum tolerable pressure is:

$$p = \frac{20 \cdot \frac{K}{S}\, v \cdot s_v}{d_i + 2\, s_v}$$

$$= \frac{20 \cdot \frac{235}{1.5} \cdot 1 \cdot 2.7}{29 + 2 \cdot 27} = 245.9 \text{ bar}.$$

Therefore: min $(\hat{p}, p) = 219.3$ bar

Which means that the pipe can permanently withstand a pulsating pressure of 219 bar without suffering damage. This assumes, however, that there are no other stresses act on the pipe.

8 Mechanical properties of tube materials and tube selection tables

Designation	St 37.4	St 52.4	St 37.4	St 37.0	St 35 NBK	X6CrNiMoTi17 122	X6CrNiTi1810
Material No. DIN	1.0255 1630	1.0581 1630	1.0255 1628	1.0254 1626	1.0308 2391	1.4571 17458	1.4541 17458
Tensile strength R_m in N/mm² (min)	340	490	340	340	340	500	500
0,2% yield strength $R_{p\,0,2}$ in N/mm² (min) or upper yield point R_{eH} in N/mm² (min)	235 *	350 *	235 *	235	235	20 °C: 210 50 °C: 202 100 °C: 185	200 190 176
1% yield strength R_{p1} in N/mm² (min)	—	—	—	—	—	20 °C: 245 50 °C: 234 100 °C: 218	235 222 208
Elongation at fracture (min) A_5 in % ($L_o = 5 \cdot d_o$)	25	21	25	25	25	> 30	> 30
Strength coefficient K in N/mm² to AD-Merkblatt W 4 at 20 °C at 100 (120) °C	 235 186	 355 255	 235 186	 235 186	 235 186	—	—
Endurance limit at repeated stress $\sigma_{osc\,/\,D}$ in N/mm² to DIN 2445 supplement to DIN 2413 see sheet 3.1/3.2	226	—	—	—	—	(190) **	(190) **

* For calculations to DIN 2413 the given values can be used up to 120°C
** Not given in DIN 2445 (see [1])

Table 53: *Mechanical properties of different tube materials*

Material St 52.4 to DIN 1630, certification e.g. to DIN 50 049-3.1 B

DN	PN 100 d_o	PN 100 s	PN 100 d_i	PN 160 d_o	PN 160 s	PN 160 d_i	PN 320 d_o	PN 320 s	PN 320 d_i	PN 400 d_o	PN 400 s	PN 400 d_i
40	48,3	3,6	41,1	48,3	4	40,3	48,3	8	32,3	70	14,2	41,6
50	60,3	4,5	51,3	60,3	5	50,3	60,3	10	40,3	88,9	17,5	53,9
65	76,1	4,5	67,1	76,1	6,3	63,5	76,1	12,5	51,1	101,6	20	61,6
80	88,9	6,3	76,7	101,6	8,8	84	101,6	16	69,6	139,7	28	83,7
100	114,3	8,8	96,7	114,3	10	94,3	114,3	17,5	79,3	168,3	32	104
125	139,7	10	119,7	152,4	12,5	127	193,7	30	134	219,1	45	129
150	168,3	12,5	143,3	177,8	16	146	219,1	36	147	244,5	50	144
200	219,1	16	187,1	244,5	20	204	298,5	45	208	323,9	65	194
250	273	20	233	298,5	25	248	355,6	55	246	406,4	75	256
300	355,6	25	305,6	355,6	30	296	—	—	—	—	—	—

Designation of steel tube to DIN 2448 of 76.1 mm outside diameter and 12.5 mm wall thickness made of St 52.4 steel and user's inspection to DIN 1630
Tube DIN 2448-76.1 x 12.5 DIN 1630-St 52.4

Table 54: *Selection table for seamless steel tube for pulsating pressure to DIN 2445, Sheet 1*

Material St 35; as-supplied to DIN 2391, Part 2 July 81; certification e.g. to DIN 50 049-2.2							
Tube 4 to 16 mm				Tube 18 to 42 mm			
d_o	s	d_i	PN	d_o	s	d_i	PN
4	1,0	2	400	18	1,5	15	160
6	1,0	4	320	20	3,0	14	320
6	1,5	3	400	22	2,0	18	160
8	1,5	5	320	25	3,0	19	250
10	1,5	7	320	25	4,0	17	320
10	2,0	6	400	28	3,0	22	160
12	1,5	9	160	30	4,0	22	250
12	2,0	8	320	35	3,0	29	160
12	3,0	6	400	38	4,0	30	160
15	1,5	12	160	38	5,0	28	250
16	2,5	11	320	42	3,0	36	160
Designation of precision steel tube of 30 mm outside diameter and 4 mm wall thickness in St 35, as-supplied to DIN 2391, Part 2, July 81, normalized NBK **Tube DIN 2391-C- 30 x 4-St 35 NBK**							

Table 55: *Selection table for precision seamless steel tube to DIN 2391*

	PN 16			PN 160			PN 320		
	Material St 37.0 to DIN 1629, Oct. 84 certification e.g. to DIN 50 049-2.2			Material St 37.0 to DIN 1629, Oct. 84 certification e.g. to DIN 50 049-3.1 B			Material St 37.4 N to DIN 1630, Oct. 84 certification e.g. to DIN 50 049-3.1 B		
DN	d_o	s	d_i	d_o	s	d_i	d_o	s	d_i
							48,3	8,0	32,3
40	48,3	3,2	41,9	48,3	4,5	39,3	60,3	10,0	40,3
50	60,3	3,6	53,1	60,3	5,6	49,1	76,1	12,5	51,1
63	76,1	3,6	68,9	76,1	7,1	61,9	88,9	14,2	60,5
80	88,9	3,6	81,7	101,6	8,8	84,0	114,3	20,0	74,3
100	114,3	3,6	107,1	114,3	10,0	94,3	152,4	25,0	102,4
125	139,7	4,0	131,7	139,7	12,5	114,7	177,8	30,0	117,8
150	168,3	4,5	159,3	193,7	25,0	143,7	219,1	38,0	143,1
200	219,1	5,9	207,3						
Designation of seamless steel tube to DIN 2448 of 88.9 mm outside diameter and 14.2 mm wall thickness in steel St 37.4 DIN 1630 (Material No. 1.0255), normalized (N) **Tube DIN 2448 - 88.9 x 14.2 DIN 1630-St 37.4 N**									

Table 56: *Selection table for seamless steel tube to DIN 2448*

Note

Tables 54, 55 and 56 use the notation according to DIN 1629 and DIN 1630

9 Symbols and subscripts

Symbols

Symbols	Units	Quantity
A_5	%	Elongation at fracture ($L_0 = 5 \cdot d$)
c_1	mm, %	Allowance for waal thickness undersize
c_2	mm	Corrosion and wear allowance
d	mm	Diameter
K	N/mm²	Strength coefficient
k	mm	Internal roughness of tube
L	mm	Length
p	bar	Design pressure, i.e. maximum permitted peritted internal pressure allowing for all imaginable operating states including pressure shock
p'	bar	Test pressure
Δp	bar	Pressure drop
R	mm	Bending radius
R_{eH}	N/mm²	Upper yield point
R_m	N/mm²	Tensile strength
$R_{m/10^5}$	N/mm²	Fatigue strength for 100,000 hours
$R_{m/2 \cdot 10^5}$	N/mm²	Fatigue strength for 200,000 hours
$R_{p\,0,2}$	N/mm²	0,2% yield strength
$R_{p\,1}$	N/mm²	1% yield strength
s	mm	Actual wall thickness
s_v	mm	Theoretical wall thickness (without allowances)
$\dot{V}$	L/min	Volumetric flow
$\overline{w}$	m/s	Mean flow velocity
x	%	Wall thickness allowance to British Standards
ν	mm²/s	Kinematic viscosity
ρ	g/cm³	Density
σ_{osc}	N/mm²	Fatigue strength under pulsating stress
$\sigma_{osc/D}$	N/mm²	Endurance limit at repeated stress

Dimensionless symbols

Symbols	Quantity
B_o, B_i	Coefficients allowing for reduced stress at outside and inside of pipe bends
n	No. of stress cycles
Re	Reynolds No.
S	Safety factor
S_L	Load cycles safety factor
v	Weld efficiency
λ	Pipe friction coefficient
ξ	Resistance coefficient

Indizes

Symbols	Quantity
o	outside
i	inside
max	maximum
min	minimum
v	loss
λ	referred to pipe friction
ξ	referred to individual resistances

Headers

Symbols	Quantity
^	Maximum value
v	Minimum value

10 References

[1] Fiala O., Köhnlechner R., Ordelheide G.: Berechnung und Dimensionierung von Rohrleitungen in der Hydraulik. Ölhydraulik und Pneumatik 27 (1983) 5, S. 335 - 341

[2] DIN 2402: Nennweiten. Beuth Verlag, Berlin, Febr. 1975

[3] DIN 2391, Teil 1 und 2: Nahtlose Präzisionsstahlrohre mit besonderer Maßgenauigkeit. Juli 1981

[4] DIN 2448: Nahtlose Stahlrohre. Febr. 1981

[5] DIN 2445: Nahtlose Stahlrohre für schwellende Beanspruchung. Nov. 1974

[6] DIN 1630: Nahtlose kreisförmige Rohre aus unlegierten Stählen für besonders hohe Anforderungen. Okt. 1984

[7] DIN 2463: Geschweißte Rohre aus austenitischen nichtrostenden Stählen. März 1981

[8] DIN 50049: Bescheinigung über Materialprüfungen. Juli 1982

[9] DIN 2401, Teil 1: Druck- und Temperaturangaben. Mai 1977

[10] DIN 2413: Stahlrohre. Juni 1972

[11] AD-Merkblatt B1: Zylinder- und Kugelschalen unter innerem Überdruck. Juni 1986

[12] BS 778: Specification for Steel PIPES And JOINTS FOR HYDRAULIC PURPOSES. British Standards Institution, 1966

[13] NF A 49-330: TUBES SANS SOUDURE ETIRES A FROID POUR CIRCUITS HYDRAULIQUES ET PNEUMATIQUES. NORME FRANCAISE ENREGISTREE, 1976

[14] SAE Handbook Supplement HS 150, 1982 Edition

[15] DIN 5508: Biegeradien für Rohre. Nov. 1979

[16] Wagner W.: Rohrleitungstechnik. Vogel Verlag, Würzburg, 2. Auflage 1983 S. 130-131

The Production and Installation of Pipework

Arnold Krielen and Hans H. Faatz

1 Introduction

The pipework of a hydraulic system should be regarded as a major component playing a full part in the function of the system. It transports the hydraulic energy in the form of flow and pressure, sometimes over long distances. The high demands made on hydraulic systems obviously refer just as much to the pipework. It must be able to withstand the high pressures, pulsation and vibration to which all the components are exposed without suffering any leaks or other damage.

Proper installation of pipework needs

- thorough project design
- careful manufacture
- correct assembly
- careful pickling and flushing
- pressure testing.

The same great care with which other components of hydraulic systems are selected must also be taken with the design of the pipework. This applies both to the sizing of the pipes, the type of connections and the routing. Moreover, the connecting pipes between hydraulic power units and actuators require the same careful, skilled attention to design as the pipes within the hydraulic power units.

In this chapter the word "pipework" will usually mean the connecting pipework between hydraulic power units and actuators.

2 Planning

Planning is based on the data obtained in the previous chapter "*The Sizing of Pipework in Hydraulic Systems*" and on the properties and characteristics of existing, commercially-available pipe and pipe connectors. The planning must take account of routing, accessibility and safety. These are factors which have a very important effect on the price.

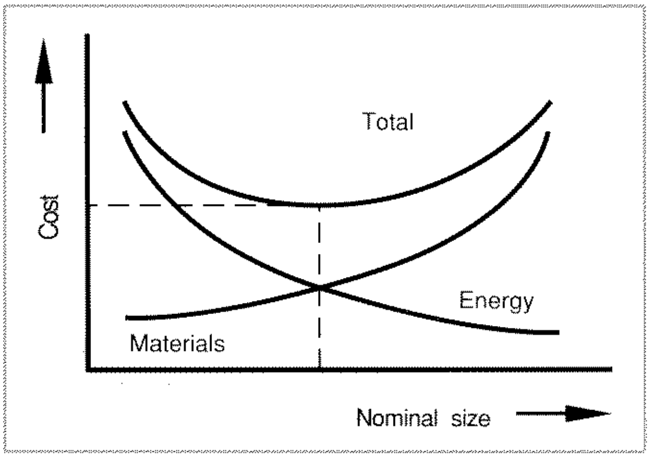


Diagram 60

It should be mentioned once again, before going any further, that the velocity and pressure drop of the fluid bear a relation to each other and, when the pipes are long, the bore must be large enough to keep the pressure drop as small as possible. This is, of course, more expensive. The relation between nominal size (fluid velocity) and cost is illustrated in *Diagram 60.*

Planning criteria

The installation of the pipework for a hydraulic system is almost the final link in a chain of activities which can only be completed on schedule and with satisfactory quality after careful prior planning. The following points in the form of a checklist are the ones requiring most attention in the planning of pipework. No claim is laid to completeness.

- pressures
- velocities
- external forces
- environment
- cleanliness
- assembly and disassembly
- safety against damage
- clarity
- monitoring devices
- maximum pressure drops
- material quality
- external and internal corrosion protection
- fixing and attachment.

3 Pipes

Different types of pipe are used in hydraulic systems depending on the nominal size or bore and pressure.

Precision seamless steel tube to DIN 2391 C St 35.4 NBK is normally used for up to nominal size 32.

Over nominal size 40 and up to 160 bar the usual practice is to use seamless steel tube to DIN 2448 or DIN 2445, St 37.0 or St 52.0 to DIN 1629 certified to DIN 50 049-3.1B. St 37.4 or St 52.4 N to DIN 1630 is used when the pressure is over 161 bar.

Seamless steel tube to DIN 2391, DIN 2448 and DIN 2445 is also available commercially in other materials. It is normal to use Material 1.4571 for stainless steel tube to DIN 2462.

Longitudinally-welded and spiral-welded steel tube are not normally found in hydraulic systems; although they are suitable for secondary purposes such as suction lines and return lines. They need special care with pickling.

Copper alloy tube is only used in hydraulic systems under exceptional circumstances in a corrosive environment.

The earlier chapter *"The Sizing of Pipework in Hydraulic Systems"* contains selection tables for pipes so there is no need to repeat them here.

4 Pipe fittings

4.1 Introduction

Steel tube usually comes in lengths of about 6 m so the use of some form of connector is unavoidable. Basically, these may be classified as

- permanent connections and
- detachable connections.

"Permanent connections" in hydraulic pipework are welded and brazed joints which make the lengths of pipe into an endless run.

Welded joints make use of welding nipples, flanges, collars, bends and other fittings for welding directly together, and the subsequent pickling and flushing must be particularly careful in order to prevent dirt being carried into the system.

Brazed joints are unusual with steel pipe nowadays, although they are used for copper alloy pipe. Once again, bends, sockets and other fittings are employed to produce an "endless" run of pipe.

Detachable connections mean that the lengths of pipe are joined together by means of a threaded connector or flange. There are many different types of such connections:

- threaded connectors
- cutting ring fittings
- compression joints
- flared joints
- welding nipple threaded connectors
- flanged joints.

All pipe joints have to perform two functions - "securing" and "sealing".

It is also necessary to distinguish between the connection between pipe and connector, fitting or joint and between connector, fitting or joint and components such as valves, subplates, control blocks, pumps, etc.

4.2 Threaded Fittings

4.2.1 Connection between pipe and fitting

For the connection between pipe and fitting it makes no difference whether it is a joint between two pipe ends or a joint between a pipe and a component.

Figs. 158 to 165 show the joining of fitting bodies to pipe using DIN 2391 precision steel tube.

The type of joint gives each fitting its name.

4.2.1.1 Threaded connectors *(Fig. 157)*

The main feature of the threaded connector is that the connection between pipe and fitting is made by means of a thread cut on the pipe. The seal is provided by the metal-to-metal joint and the "securing" function is provided by the thread. Such connectors are normally only used for secondary purposes, such as water lines to coolers and, possibly, suction lines to pumps with the thread on the suction side.

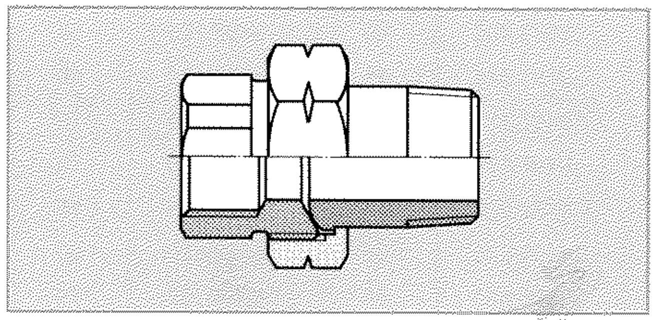


Fig. 157: *Threaded connector, metal-to-metal seal, securing function by thread, for low pressure applications (e.g. water lines)*

4.2.1.2 Cutting ring fittings

Cutting ring fittings are perhaps the best known means of connecting a pipe to a fitting. The cutting ring is fitted to the pipe with a special tool. It cuts into the surface of the pipe in order to perform its "securing" and "sealing" functions. The seal between the pipe and the body of the fitting is also metal-to-metal. A union nut secures the cutting ring to the fitting itself.

All cutting ring fittings require maintenance because the metal-to-metal seal is susceptible to "setting" which can result in leaks. On the other hand, there are also limits on the amount of re-tightening because of surface embrittlement.

The fitting of the cutting ring on to the pipe must be done very carefully because it can slip off when the pressure is applied if it has not cut into the pipe deeply enough.

Ensure that the ends of the pipes have been cut squarely and have been deburred properly.

The single cutting ring *(Fig. 158)* has now been largely superseded by the double cutting ring.

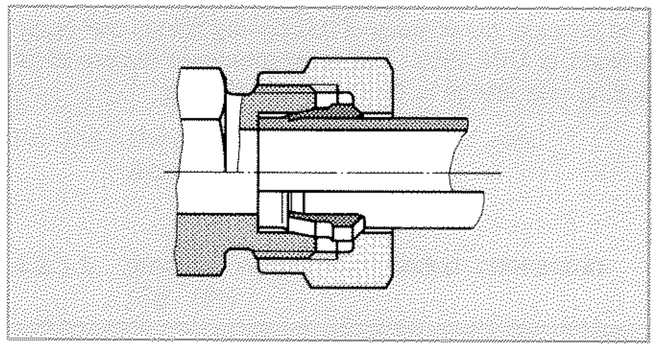


Fig. 158: *Single cutting ring, metal-to-metal seal, securing function by a single edge*

The double cutting ring *(Fig. 159)* improves the securing and sealing functions by having two edges which cut into the pipe.

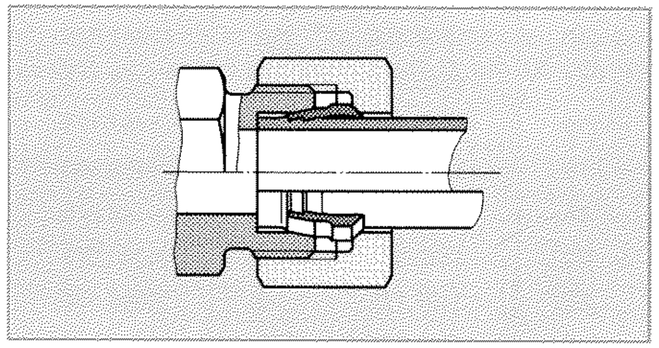


Fig. 159: *Double cutting ring, metal-to-metal seal, securing function by two edges, more fitting force needed.*

The "Walpro" cutting ring shown in *Fig. 160* has a double cutting ring with a reinforced shoulder. It makes the securing and sealing functions safer.

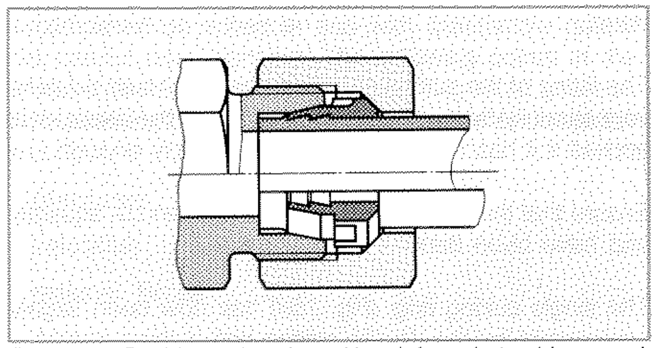


Fig. 160: *Double cutting ring with reinforced shoulder, metal-to-metal seal, securing function by two edges, more fitting force needed than the single cutting ring*

The principal dimensions of cutting ring fittings are standardized in DIN 2353. The most important feature of all cutting ring fittings is the 24° internal taper.

4.2.1.3. Flared joints *(Figs. 161 to 163)*

In the flared joint the securing function is provided by flaring the end of the pipe and clamping it with a special ring. In the case of the Parker Triple Lok and Walterscheid flare joints the flare angle is 37°. With pipe wall thicknesses over 3 mm this can lead to difficulties because of the danger of hairline cracks in the area of the flare. The Parker Triple Lok joint (Fig. 161) has the 37° taper turned on the body of the joint. The metal-to-metal seal is provided with the aid of a ring.

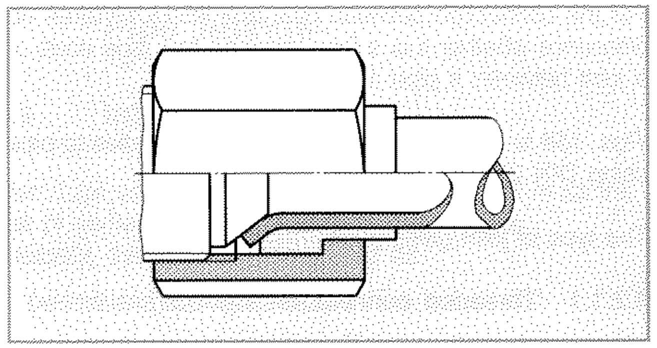


Fig. 161: *Flared joint, metal-to-metal seal, securing function by 37° flare, turned taper on body*

Pipe o.d.	Steel pipe				Stainless steel pipe 1.4571			
	Single edge cutting ring		Double edge cutting ring		Single edge cutting ring		Double edge cutting ring	
	F in kgf	*) *p in bar*	*F* in kgf	*) *p in bar*	*F* in kgf	*) *p in bar*	*F* in kgf	*) *p in bar*
6	1526,040	24	2225,475	35	2543,400	40	3179,250	50
8	1526,040	24	2225,475	35	2543,400	40	3179,250	50
10	1780,380	28	2543,400	40	2543,400	40	3497,175	55
12	1907,550	30	2861,325	45	2861,325	45	3815,100	60
14	2543,400	40	3497,175	55	3497,175	55	4450,950	70
15	2543,400	40	3497,175	55	3497,175	55	4450,950	70
16	3179,250	50	4450,950	70	4133,025	65	5404,725	85
18	3179,250	50	4768,875	75	4133,025	65	5722,650	90
20	4133,025	65	6040,575	95	5086,800	80	6994,350	110
22	3497,175	55	5086,800	80	4450,950	70	6040,575	95
25	4768,875	75	6994,350	110	5722,650	90	7948,125	125
28	4133,025	65	6040,575	95	5086,800	80	6994,350	110
30	6358,500	100	9537,750	150	7630,200	120	10491,525	165
35	6040,575	95	8583,975	135	6994,350	110	9537,750	150
38	8901,900	140	12717,000	200	9855,675	155	13670,775	215
42	8583,975	135	10491,525	165	7630,200	120	11445,300	180

*) For fitting machines with 90 mm diameter piston

Table 57: *Recommended tightening force for cutting ring fittings (average values). Forces can differ from manufacturer to manufacturer.*

The Walterscheid flared joint *(Fig. 162)* can be used with a DIN 2353 connector body with the 24° internal taper. The intermediate ring seals on to the 24° taper with an O-ring. The sealing function between fitting and pipe is also executed with an O-ring on the 37° flared taper. The securing function is performed by a backing ring and the retaining nut.

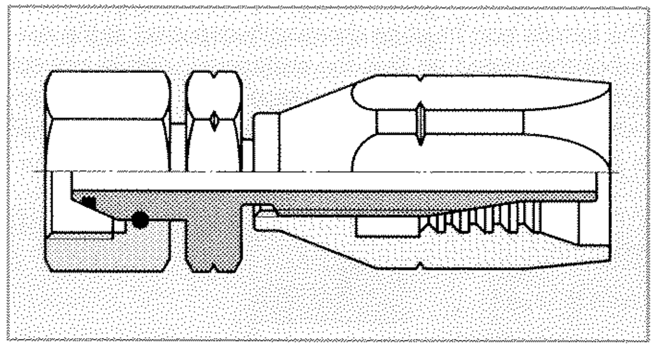


Fig. 162: *Flared joint with intermediate ring, elastic seal, securing function by 37° flare on intermediate ring*

The flared joint marketed by Voss *(Fig. 163)* also has a metal-to-metal seal between the pipe and backing ring although, in this case, the flare is only 10°. The securing function between pipe and ring is effected with the backing ring and the retaining nut.

The smaller flare is better for thicker walled tube. A DIN 2353 connector body is used and an O-ring provides the seal to it.

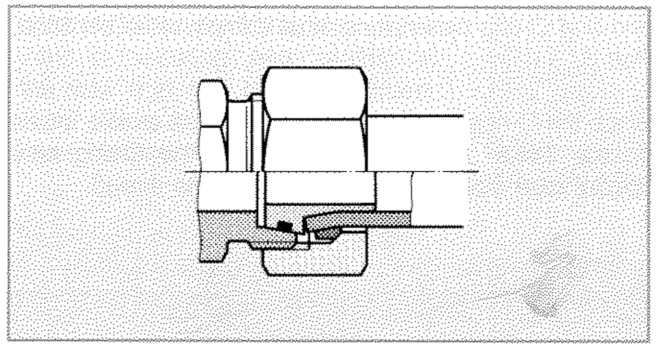


Fig. 163: *Flared joint with intermediate ring, metal-to-metal seal, securing function by 10° flare, better for thicker walled tube, intermediate ring and backing ring needed.*

4.2.1.4 Compression joints *(Fig. 164)*

Compression joints use non-standard connector bodies. They provide a contrast to cutting ring fitting in that the securing function is provided by the clamping of the cutting ring on to the pipe. The seal is metal-to-metal - both to the pipe and to the connector body.

All the connectors, fittings and joints described so far are known as "solderless" fittings.

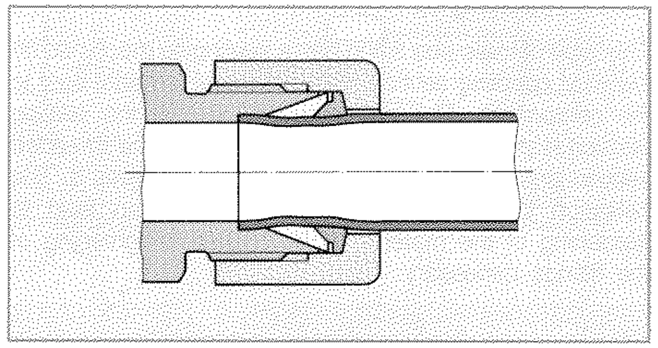


Fig. 164: *Compression joint, metal-to-metal seal, securing function by clamping*

4.2.1.5 Welding nipple threadeded connectors *(Fig. 165)*

With this type of connector a welding nipple is welded to the end of the pipe. All modern methods of welding can be employed as described in *Section 6.2.2.3.2.* This ensures that there are no problems with the seal between pipe and welding nipple

The seal between the welding nipple and the DIN 2353 connector body is elastic so once again there should be no problems with its use. The securing function between welding nipple and connector body is provided by the retaining nut.

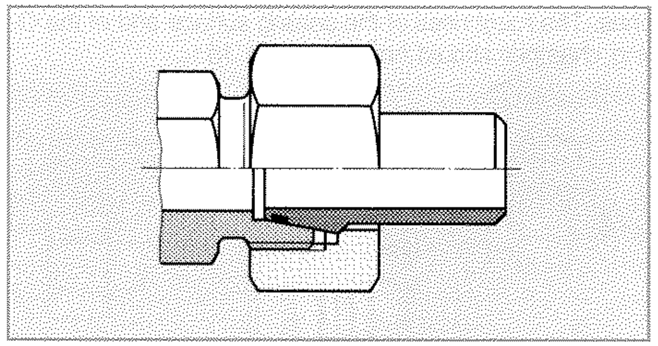


Fig. 165: *Welding-nipple threaded connector with taper to DIN 3865, elastic seal, securing function by welding*

4.2.2 Connection between fittings and components

With all threaded fittings the securing function between fitting and component is provided by the thread. Male port fittings with a parallel thread provide a good joint of this type. The female ports and the male port fittings are standardized in DIN 3852. The only difference between the male fittings is the type of seal between the connector body and the component.

4.2.2.1 Sealing lip *(Fig. 166)*

This type of male port fitting has a sealing lip machined on the stud coupling section, giving a metal-to-metal seal to the component. The prerequisites are that the sealing surface is perpendicular to the centreline of the threaded section and has no scratches or grooves across it. The pulsation common in hydraulic systems can lead to hardening of the material which makes re-tightening of the fittings necessary.

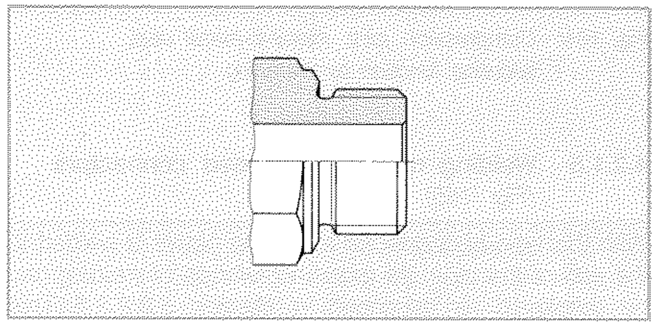


Fig. 166: *Sealing lip, metal-to-metal seal, undercut perpendicular to thread, the counterbore will be damaged by cutting edge*

4.2.2.2 Connector body with O-ring *(Fig. 167)*

The sealing function of this type of fitting is provided by an O-ring. The prerequisites again are that the clamping surface is perpendicular to the centre-line of the threaded section and there are no transverse grooves or scratches. The roughness of the sealing surface and of the O-ring chamber should not exceed Rt 16.

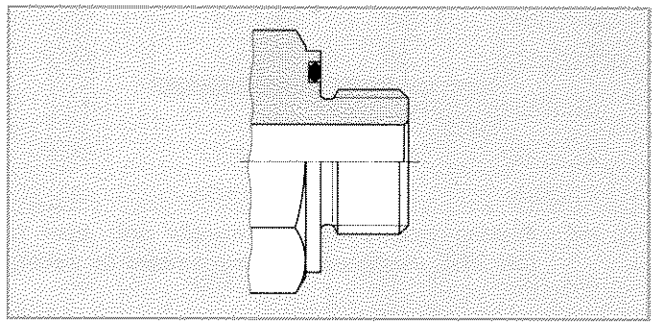


Fig. 167: *O-ring, female port to DIN 3852 wide, elastic seal, undercut larger than cutting edge, high tensile force on thread*

The tensile forces on the thread are relatively high due to the large diameter of the O-ring. The female port must have the "wide" undercut conforming to "DIN 3852-wide"

4.2.2.3 Connector body with profile seal *(Fig. 168)*

With this type of fitting the O-ring is replaced by a rectangular ring of smaller diameter inserted into the connector body in order to provide the sealing function. The female port must be thoroughly deburred at the run-out of the thread so that the rectangular ring is not cut accidentally. The clamping surface must again be perpendicular to the thread. A "normal" undercut to DIN 3852 is sufficient.

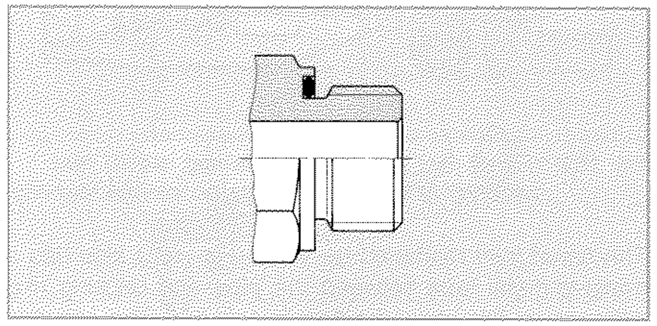


Fig. 168: *Profile seal, female port to DIN 3852, elastic seal, undercut as for sealing lip, tensile force on thread less than with O-ring*

4.2.2.4 Male port fitting with O-ring for female ports to ISO 6149 and DIN 3852, Part 3, Form W *(Fig. 169)*

With this type of fitting the O-ring is fitted in the thread undercut of the stud end. The female port must comply with ISO 6149 and DIN 3852, Part 3, Form W. The stud end and O-ring dimensions must comply with DIN 3852, Part 3.

Since the O-ring is closer to the thread the tensile forces on the thread are less. When fitting the O-ring take special care that it is not damaged by the thread.

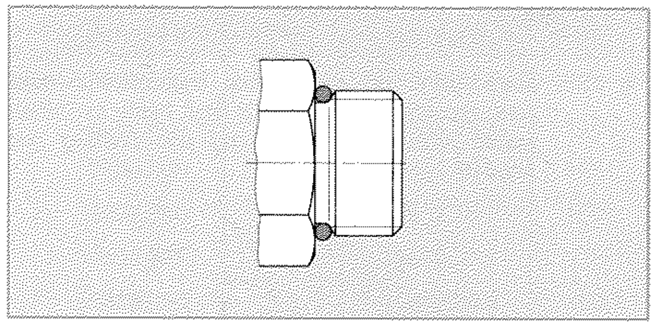


Fig. 169: *Female port to DIN 3852, Part 3, elastic seal, O-ring closer to thread, reduced tensile force*

4.2.2.5 NPT threads *(Fig. 170)*

NPT threads are tapered and are designed to fit into tapered mating threads. The seal is metal-to-metal, although some sealant such as Loctite is usually applied.

The tapering of the thread on a male fitting means that a high bursting force can be exerted when the fitting is screwed into components made of cast iron or aluminium. Nevertheless, fittings with this type of thread are very popular in North America, although encountered much less in Europe.

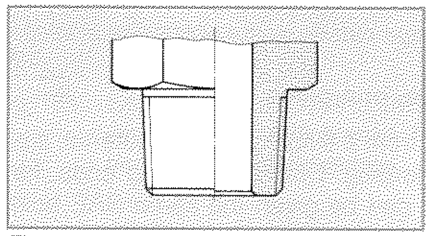


Fig. 170: *NPT thread, metal-to-metal seal, sealant necessary, high bursting forces, undefined position of pipe end.*

Naturally, there are many different versions of the types of fittings described here available on the market but they all perform the basic functions of "securing" and "sealing" in the way that has been described. *Figs. 171 to 182* show some of the more common standard fittings.

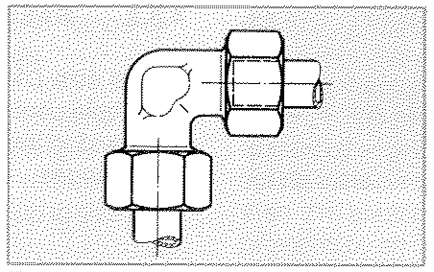


Fig. 171: *Elbow*

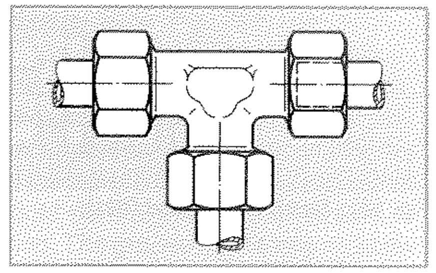


Fig. 172: *Tee-piece*

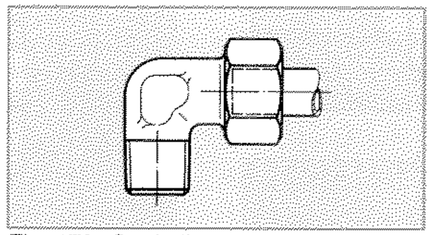


Fig. 173: *Stud elbow, non-adjustable*

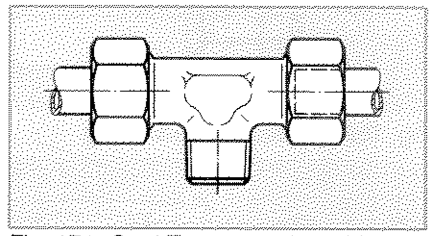


Fig. 174: *Stud Tee, non-adjustable*

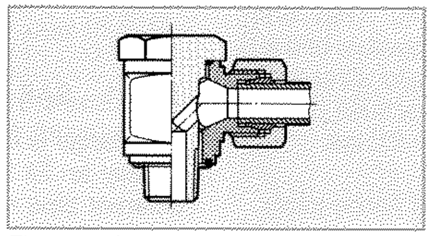


Fig. 175: *Stud banjo fitting*

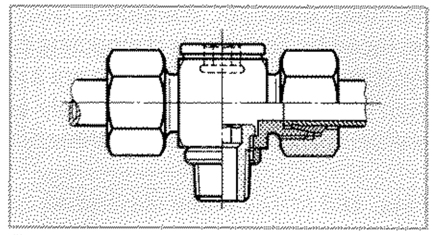


Fig. 176: *Stud banjo-Tee*

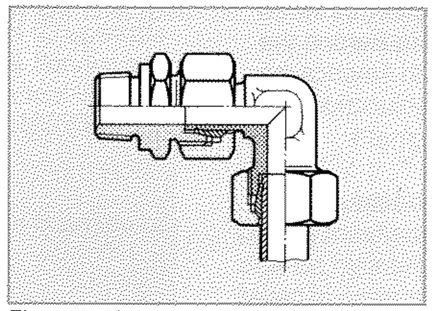


Fig. 177: *Adjustable stud elbow*

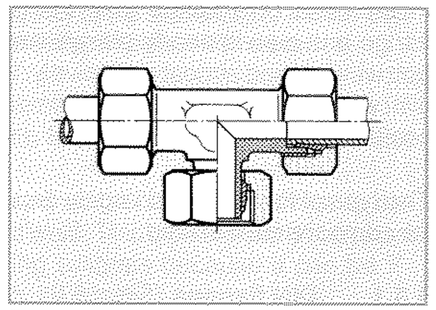


Fig. 178: *Adjustable Tee-piece*

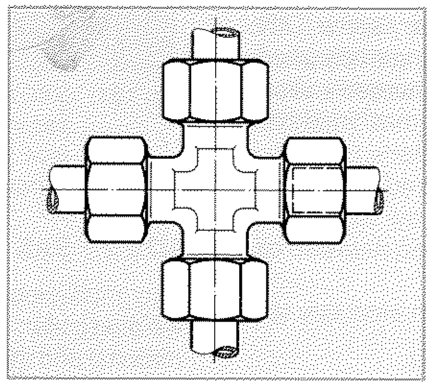


Fig. 179: *Cross fitting*

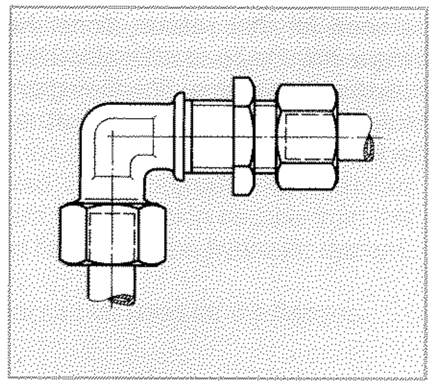


Fig. 180: *Bulkhead elbow fitting*

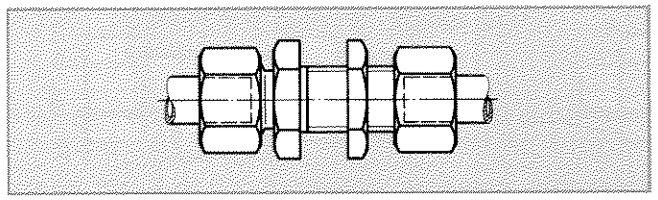


Fig. 181: *Straight bulkhead fitting*

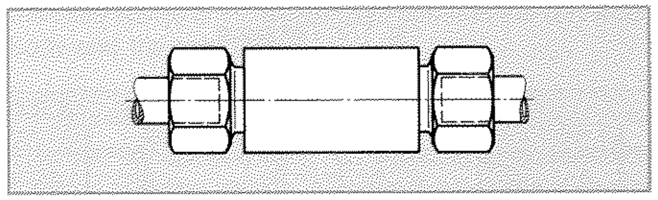


Fig. 182: *Welded bulkhead fitting*

4.3 Flange joints

Threaded-type fittings, joints and connectors are not normally used for diameters over DN 40. They are replaced by flanges for connecting pipes together and to system components. Flanges are available for sizes less than DN 40 but are not so widely used because of the greater cost compared with threaded-type fittings.

Figs. 183 to 187 show the normal types of flange joints for pipes to DIN 2448 and 2445 used in hydraulic systems.

In the case of a flange joint the connection between flange and pipe is nearly always welded. In Europe the butt weld is used almost exclusively. In North America the fillet weld is very common, sometimes in conjunction with sockets.

The type of flanges used largely depends on the particular components involved.

4.3.1 DIN flanges *(Fig. 183)*

Some shut-off cocks, flap valves and other fittings have a hole pattern for DIN flanges so that DIN flanges have to be used for that reason alone. However, some DIN flanges have to be modified for the pipes normally used in hydraulic systems. The O-ring groove normally used for sealing in hydraulic equipment must be machined into the DIN flange.

4.3.2 SAE flanges

Above size DN 40 many items of hydraulic equipment have hole configurations for SAE flanges. These can be sub-divided into one-piece SAE flanges and multi-piece SAE flanges.

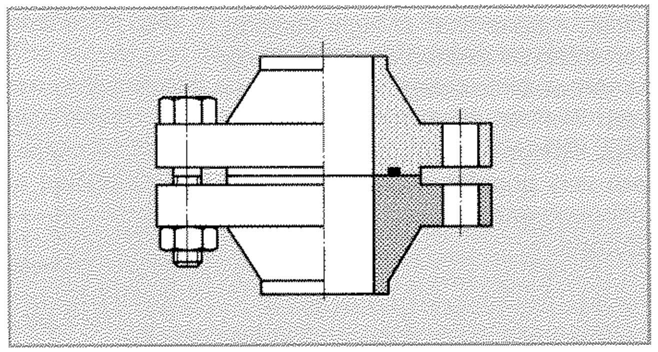


Fig. 183: *Flange with O-ring seal to DIN 2632, 2638 and 2629, securing function by screws, ground sealing surface. The large outside diameter is a disadvantage*

4.3.2.1 SAE flange, one-piece *(Fig. 184)*

This flange is made from a single piece of material; it is usually an St 37.4 forging. Different welding ends, for different pipe sizes, can be given the same SAE hole configuration. This type of flange is comparatively cheap and has the advantage of being relatively compact. Its disadvantage is that it can no longer be "adjusted" (rotated) after the welding.

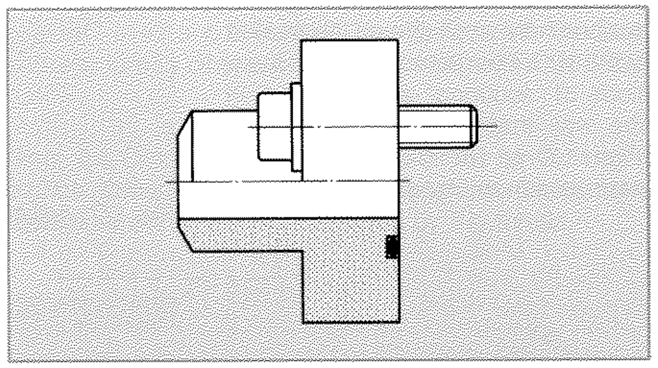


Fig. 184: *SAE flange, one-piece with O-ring to SAE-J 518 C, securing function by screws, ground sealing surface, overall shape an advantage since it is more compact*

4.3.2.2 SAE flange, multi-piece *(Fig. 185)*

The multi-piece SAE flange comprises the so-called welding collar, a turned piece or shaped part of weldable material and the two flange shoulders which perform the securing function. The space needed is slightly greater than the one-piece SAE flange and the split flange shoulders can also be a disadvantage.

The Production and Installation of Pipework

The seal between the base surface and SAE flange or between SAE flanges is provided by O-rings.

SAE flanges can be used for pressures up to about 400 bar (6000 psi). There are multi-piece SAE flanges available for low pressures up to about 16 bar (240 psi) for sizes up to DN 100 and, for high pressures, up to size DN 63.

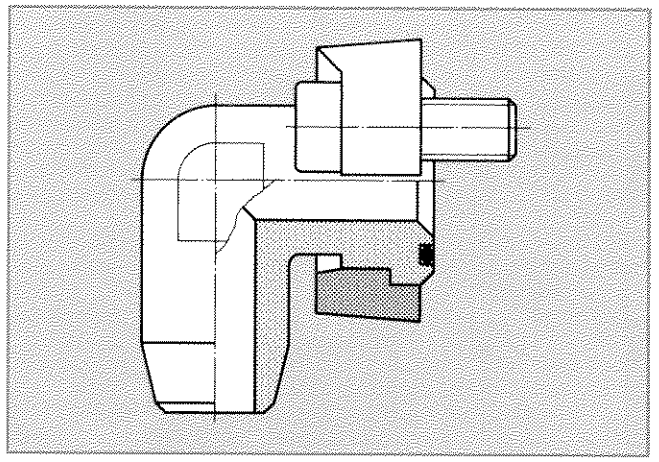


Fig. 185: *SAE flange, multi-piece to SAE-J 518 C, elastic seal, securing function by screws. The split flange shoulders can be a disadvantage*

4.3.3 Square flange joints *(Fig. 186)*

In Europe the so-called "square" flange is the most popular for flange joints from about size DN 63 upwards and for pressures from about 210 bar. The flanges are usually in two pieces and comprise the actual welding collar and a square flange which performs the securing function.

Technically, there is no difference between this type of joint and the SAE flange.

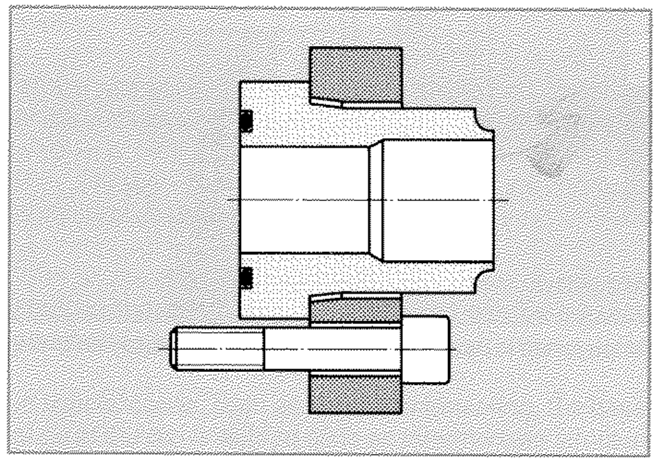


Fig. 186: *Flange joint with square flange, elastic seal (usually an O-ring), securing function by screws*

4.3.4 GS Hydro flange joint *(Fig. 187)*

The main feature of the GS Hydro flange joint is that the securing function between flange and pipe is not performed by a weld but by a spiral spring. The groove for the spring must be cut into the pipe so a thicker wall is needed than the pressure would normally require. The seal is provided by an USIT ring. The groove for the ring must be machined into the end face of the pipe. This type of flange joint is very good if it is impossible to employ a welded joint. The USIT ring is a metal ring with an integral elastometric sealing ring.

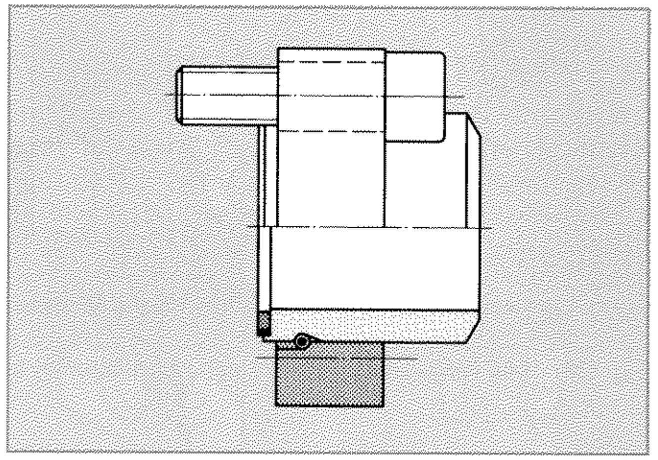


Fig. 187: *GS Hydro flange joint, sealing by USIT ring, securing function by spiral spring and screws, thicker pipe walls and machining of pipe ends needed*

4.3.5 Flanges with fillet welds *(Figs. 188 and 189)*

All the flanges described so far have been joined to the pipe by means of a butt weld. This type of weld is the usual one for hydraulic piping systems because it can be X-rayed in order to verify its quality.

Joining the flange to pipe by means of fillet welds is cheaper. There are suitable flanges on the market. Their form, i.e. hole configuration, depends on the hole configuration of the items of equipment involved. The main disadvantage of fillet welds is that their quality cannot be verified by X-ray. Also, during pickling it is possible for acid to penetrate into the gap between pipe and flange and it cannot be removed. Over a long period of time it can damage the weld seam. Dirt can also escape from the gap between pipe and flange into the hydraulic system.

Only weldable material can be used for the flanges.

When inserting the pipe ensure that it does not sit on the bottom of the flange hole. There must be a gap between them so that the expansion and contraction resulting from changes in temperature do not break the weld.

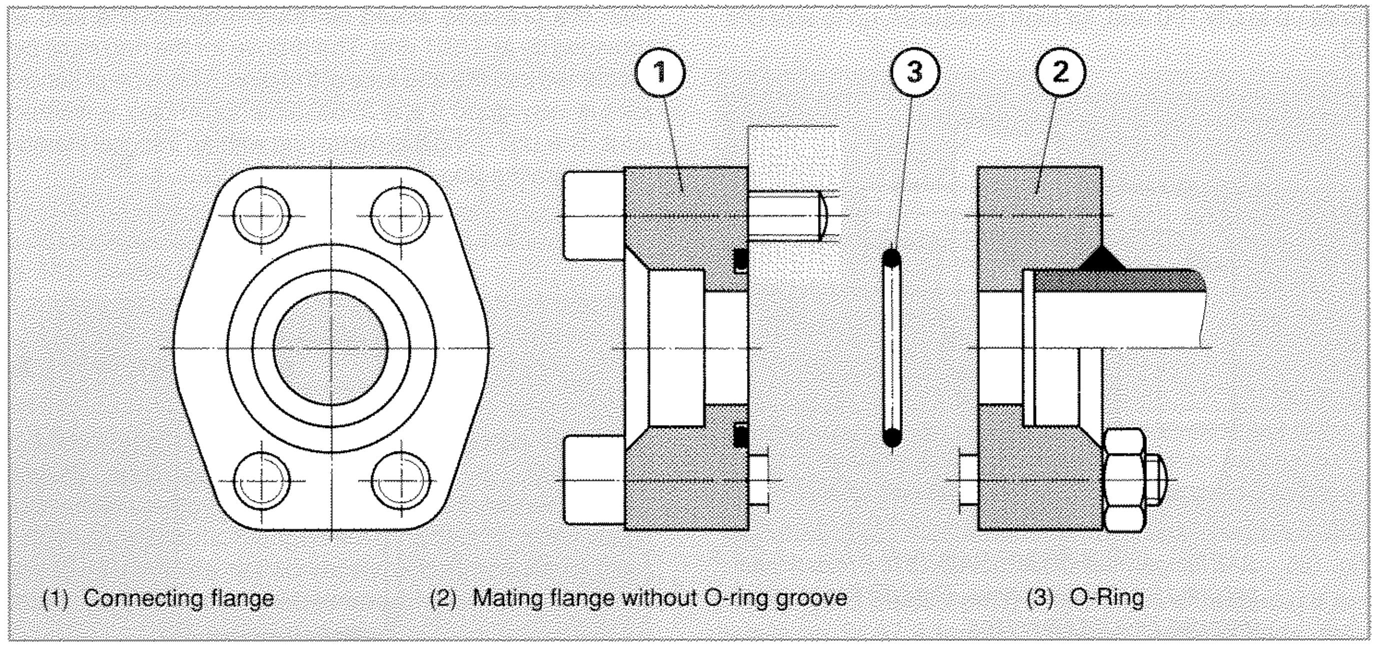


Fig. 188: *SAE flange and flange joint, securing function by screws, flange joined to pipe by fillet weld, elastic seal usually an O-ring*

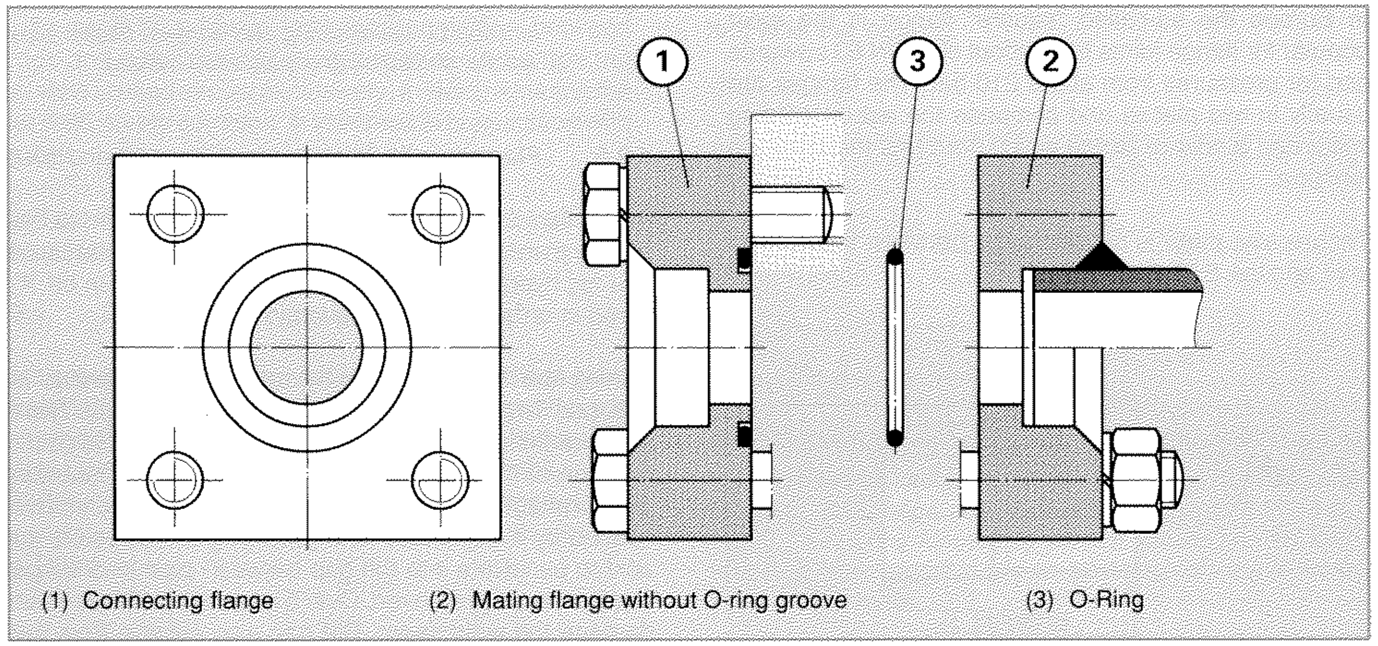


Fig. 189: *Square flange and flange joint, securing function by screws, flange joined to pipe by fillet weld, elastic seal usually an O-ring*

All joints, connectors, fittings and flanges are available in the same materials as the pipes. So it is easy to obtain flange joints in St 37.4 or stainless steel, Material 1.4571.

5 Accessories

The pipework of a system does not just consist of pipes, connectors and flanges. There are also mountings, clips, glands, hoses and expansion pieces which all have their part to play.

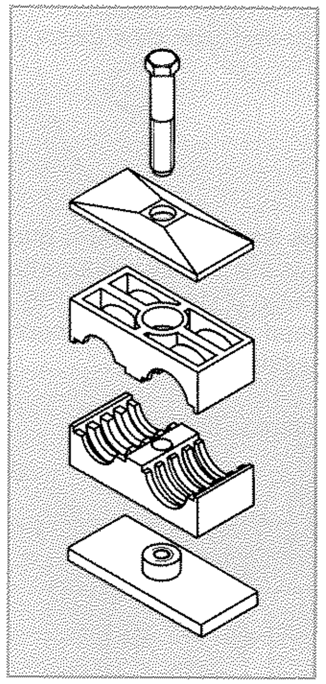


Fig. 190:
Twin pipe clip, standard version

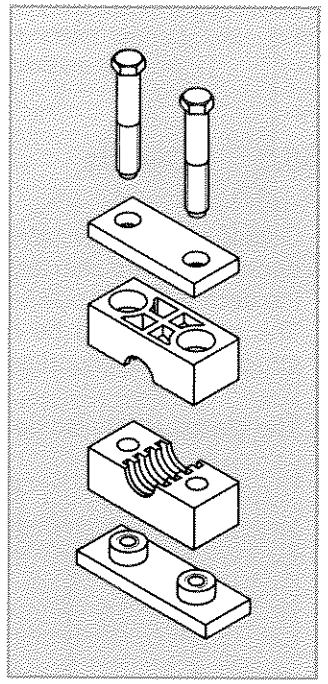


Fig. 191:
Pipe clip, heavy-duty version

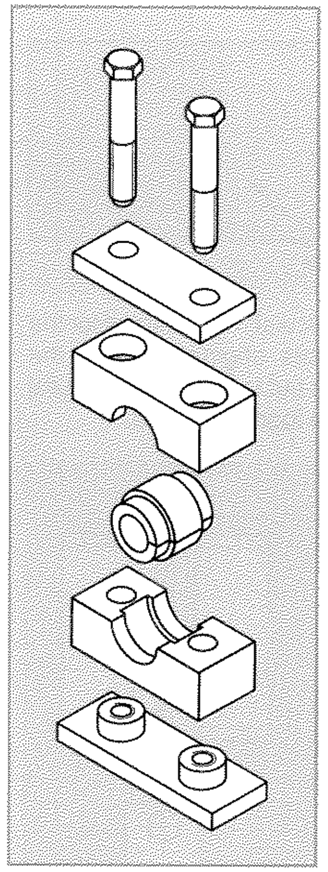


Fig. 192:
Pipe clip, heavy-duty version with elastomer insert

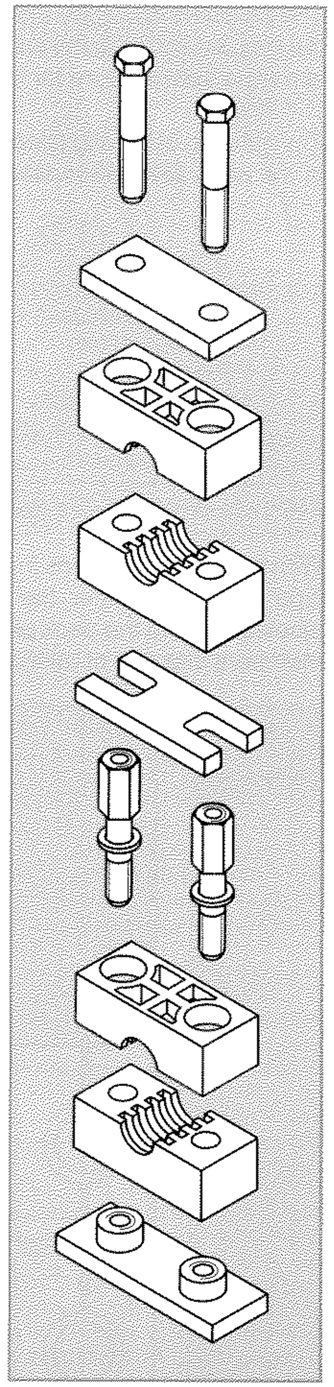


Fig. 193:
Piggyback clip, e.g. for two pipes above one another

5.1 Supports *(Figs. 190 to 193)*

The purpose of supports is to provide a secure mounting for pipework. Normal commercially-available pipe clips are made of aluminium, plastic or plastic with a rubber insert.

Which type of clip is used depends on the circumstances of the installation. For example, aluminium clips are used whenever high temperatures are expected. Plastic clips with a rubber insert are often used to reduce the propagation of structure-borne sound.

With all the clips the baseplates are made of weldable steel so that the baseplate of the clip can be securely attached to a substructure.

In special cases, such as theatre engineering, the baseplate of the clip is welded on to an intermediate structure which can then be attached to the actual supporting structure through anti-vibration mountings. This is another method of reducing the transmission of structure-borne noise *(Fig. 194)*.

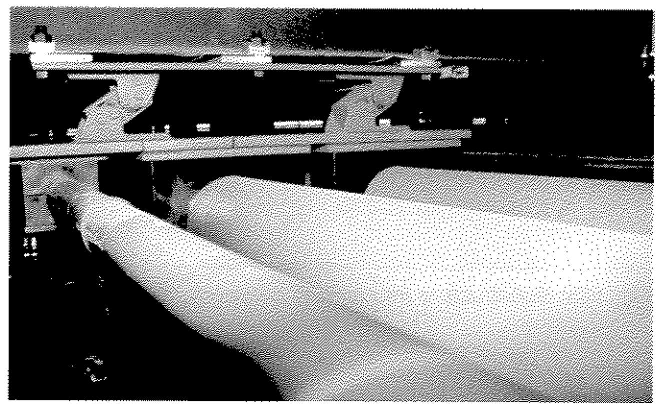


Fig. 194: *Pipe support with flexible mountings for isolating structure-borne sound*

The pipework is usually attached to the main structure of the machinery. In some cases it is necessary to make a separate substructure for the pipework *(Fig. 195)*.

The spacing of the clips depends on the outside diameter of the pipe. DIN 24346 states the following:

Pipe outside diameter in mm	Spacing in m
up to10	1.0
over 10 up to 25	1.5
over 25	2.0

Table 58: *Spacing of pipe supports*

With marine applications in particular it is essential for the pipe clips to transfer the forces originating from the pipework not to the deck plates but to suitable structural members capable of taking the load *(Fig. 196)*.

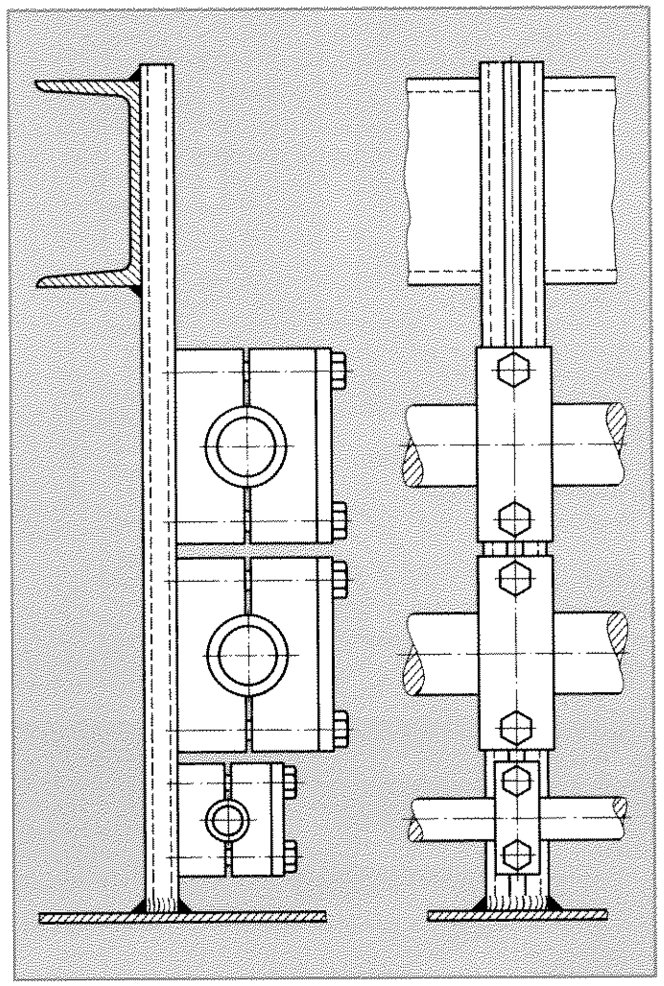


Fig. 195: *Substructure for carrying pipework*

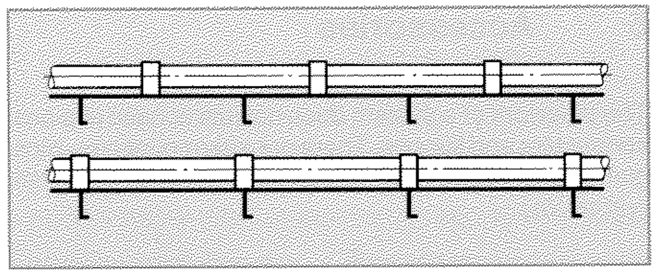


Fig. 196: *Correct method of installing pipe clips (bottom); incorrect method (top)*

5.2 Glands

Glands are used to carry pipes through bulkheads and decks. On board ship the glands are usually welded into the deck plates or bulkhead plates (Figs. 197 and 198). The glands should be designed to insulate as much structure-borne sound as possible *(Fig. 199)*.

The glands must also be resistant to fire.

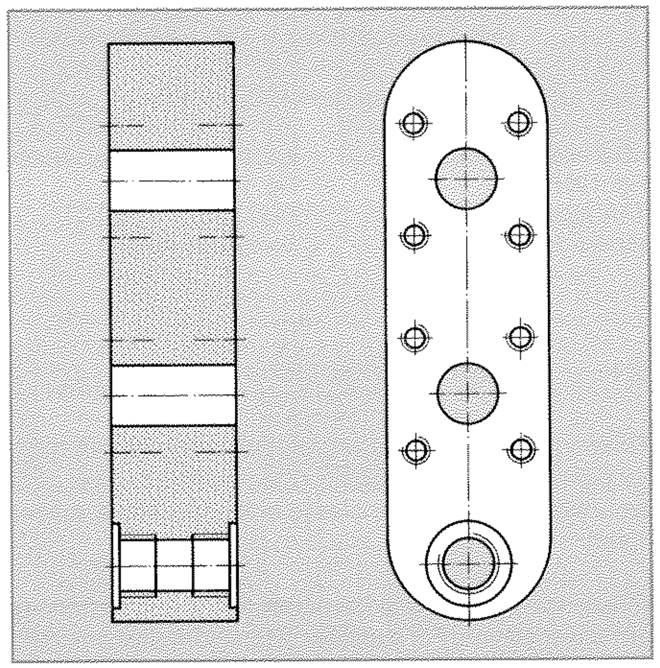


Fig. 197: *Rigid deck gland*

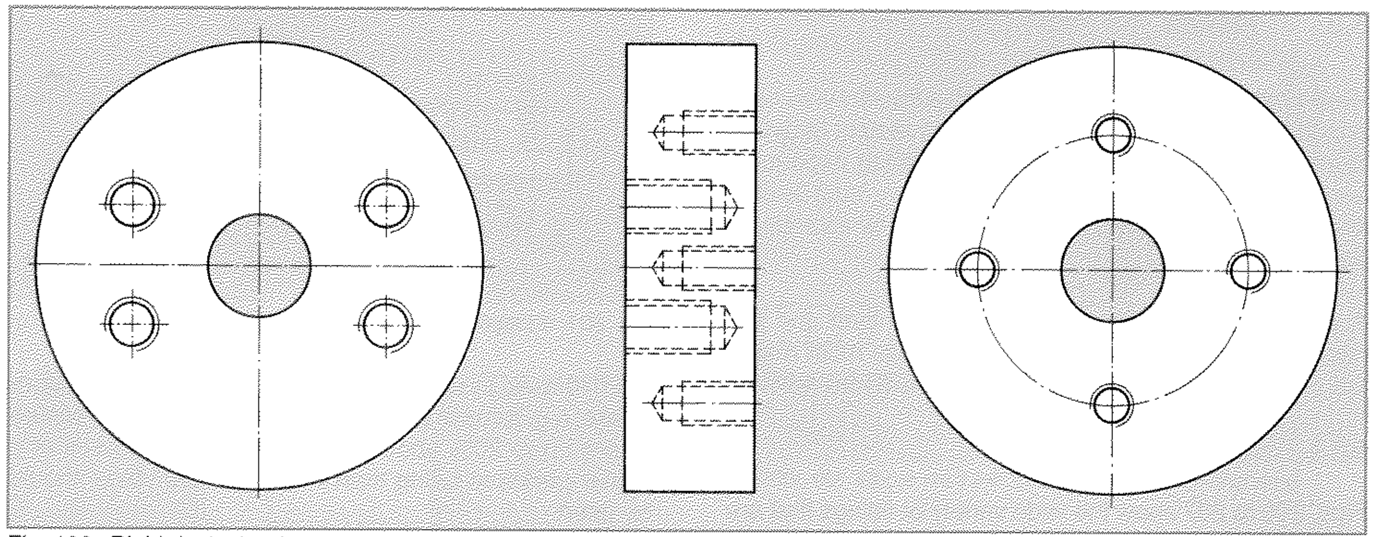


Fig. 198: *Rigid deck gland*

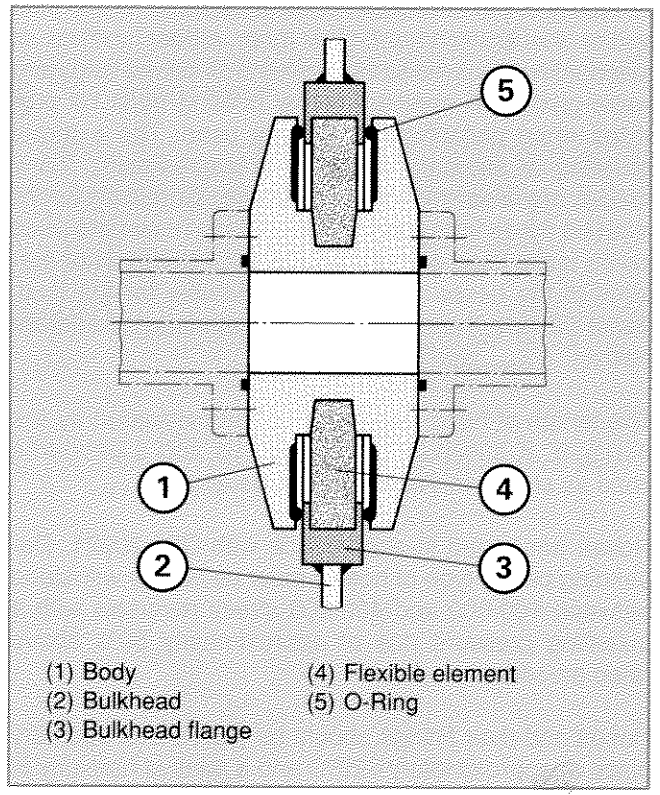


Fig. 199: *Flexible gland*

5.3 Hoses

Hoses can form an important part of hydraulic systems in providing compensation for relative movement between items of equipment and for accommodating expansion and contraction in long lengths of pipe. They must be able to meet all the demands made upon them by a variety of applications.

A hose line comprises the actual hose itself and the necessary fittings at each end. The fittings must match the other fittings used throughout the pipework system.

5.3.1 Press-fit hoses

Figs. 200 to 207 show the wide variety of fittings that are available by crimping to the end of hoses . There are several flare fittings, 24° external taper fittings and others for flanges. The standard versions come in straight form, 45° bends and 90° elbows.

Hose and fitting can be selected separately by the designer.

The choice of fitting depends on the pipework system, e.g. whether flared fittings, cutting ring fittings, welding-end fittings or flange joints are being used. In hydraulic systems the fittings are usually of steel or, in special cases, of stainless steel.

The choice of hose usually has to begin from the nominal size and nominal pressure. These two factors are predetermined by the flow and operating pressure stated in the circuit diagram. Suitability for the fluid, operating temperature and environment must also be taken into account. In pressurized hoses the fluid velocity should not exceed 2 to 3 m/s, also in order to keep noise to a minimum. The same velocity should not be exceeded in return lines either.

The operating pressure should not exceed 25% of the hose bursting pressure. This is usually taken into account in the catalogue data for the permitted values of operating pressure.

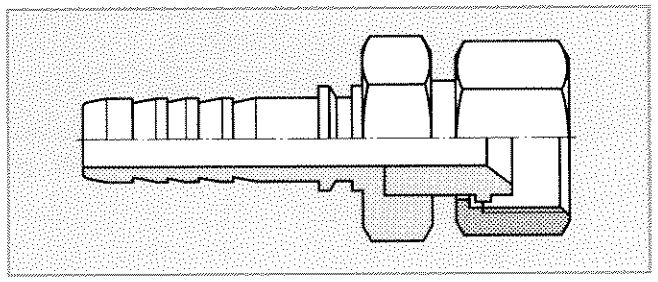


Fig. 200: *Straight hose end fitting for flared joints*

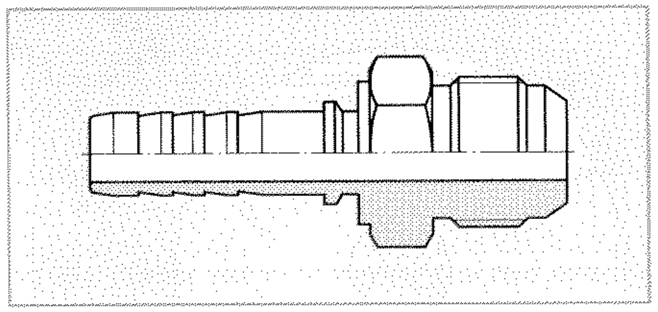


Fig. 201: *Straight male hose end fitting for flare joints*

If a hydraulic system suffers from pronounced pressure peaks, e.g. rates of change over 3000 bar/s and pressure peaks up to 20% above the nominal value, hoses of a higher pressure rating than the theoretical value should be used.

The choice of hose material must also take account of the suitability of the inside and outside surfaces for the fluid and/or the environment. The inner rubber surface of standard hoses is resistant to mineral oil-based fluids and water-glycols. A special internal coating must be provided when phosphate esters are used. The external covering of standard hoses is ozone-resistant; many makes are also mineral oil-resistant. Detailed checks must be made from case to case. When hoses are used inside tanks filled with phosphate ester the exterior material of the hoses must also be suitable for the fluid. *Table 59* lists the resistance of materials to different fluids.

The length of hose required must be calculated by the designer. A minimum bending radius and a bend-free zone as shown in *Table 60* must be allowed.

Hoses are tested in accordance with DIN 20021. Depending on the type of hose it must be able to withstand between 100,000 and 400,000 stress reversals with a pulsating pressure rise of approximately 50% above rated pressure.

Hoses and their fittings should be stored in a cool, dry place at approximately +20°C and a relative humidity of 65%. They must be protected from direct sunlight.

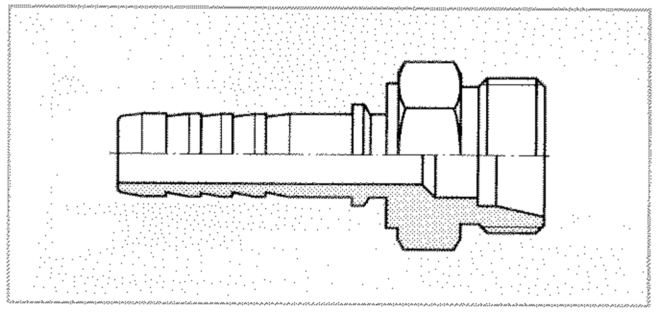


Fig. 202: *Straight hose end fitting for 24° connections*

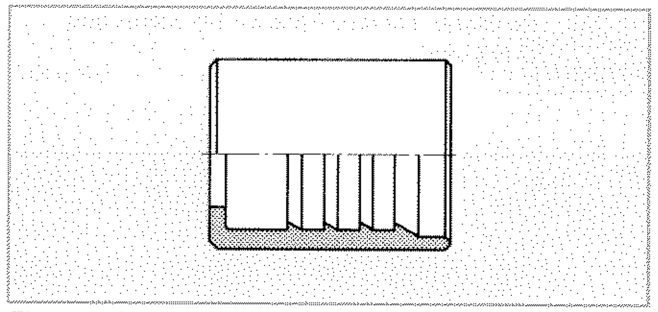


Fig. 203: *Crimping sleeve*

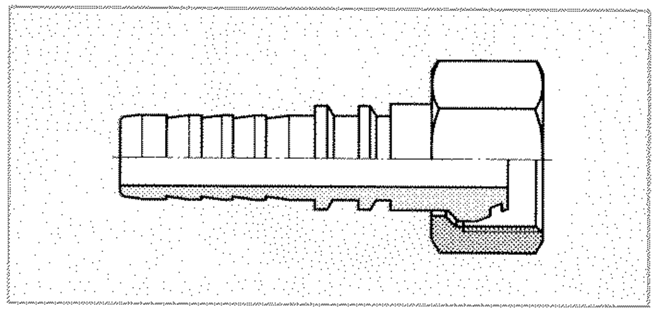


Fig. 204: *Straight hose end fitting for 24° connections with O-ring seal*

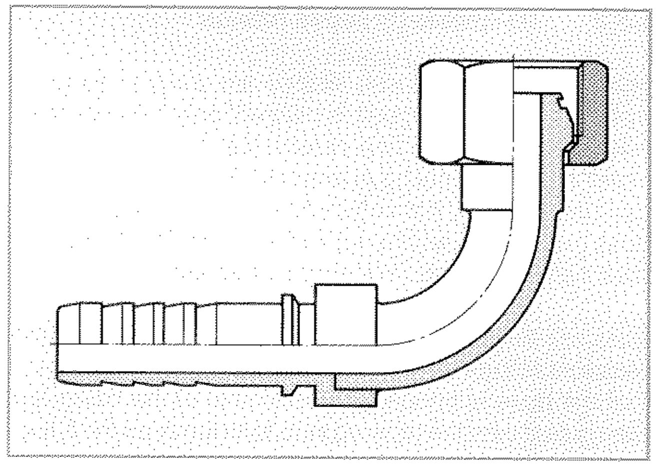


Fig. 205: *90° elbow with 24° connection and O-ring seal*

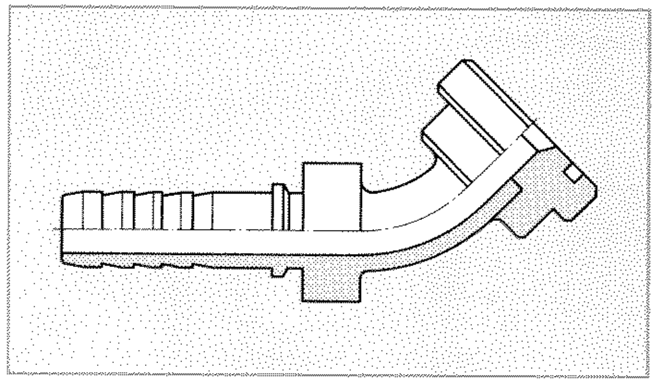


Fig. 206: *45° bend for SAE flange connection*

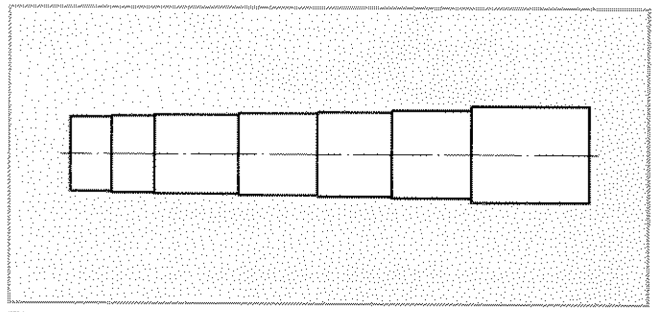


Fig. 207: *Hose, e.g. 4 SPA-SAE 100 R9R, 4 SH-DIN 22023/2*

5.3.2 Re-usable threaded fittings

Hose lines with re-usable fittings are used in hydraulic systems so that, if a repair is needed, only the hose itself has to be replaced. This means that the maintenance department does not have to stock a large number of pre-fabricated hoses; it is sufficient for them to have coils of hose that can be cut to length as required. The fittings, as their name suggests, are simply used again.

Basically the same type of connections are available in reusable fittings as in press-fit fittings. The difference is that the hose is gripped by the sleeve of the fitting and then pulled over the tapered stub to sit firmly on the fitting. The securing function between hose and fitting is provided by the specially shaped sleeve. The connection between hose and fitting limits the maximum operating pressure in this case. It is generally lower than with the press-fit fittings.

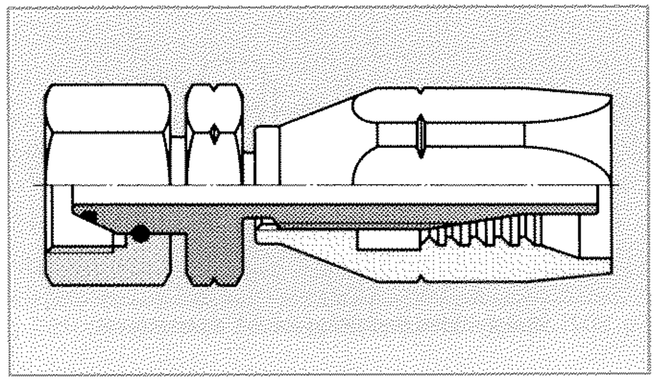
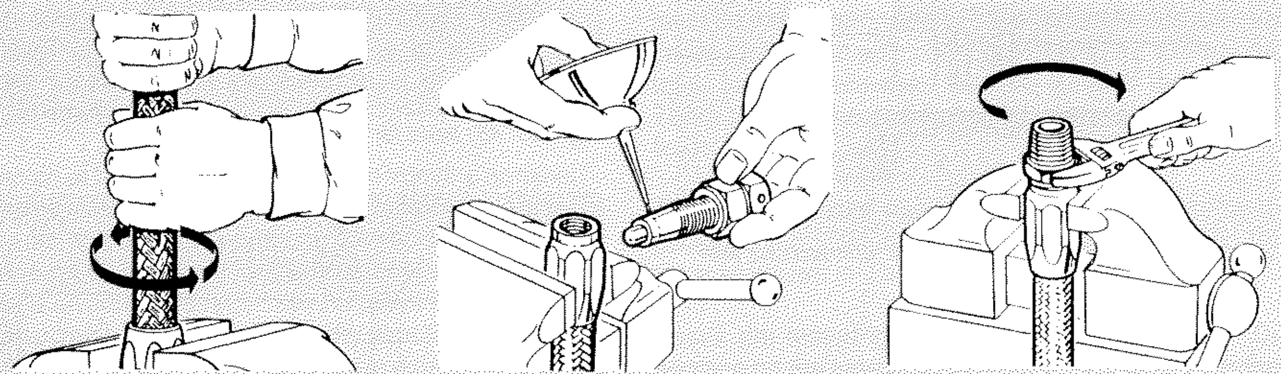


Fig. 208: *Re-usable threaded fitting for 24° connection, O-ring seal*



Cut the hose to length with a square cut using a fine-toothed hacksaw. Remove any pieces of rubber or wire. Grip the fitting in a vice and screw in the hose anti-clockwise until it will go no further, then unscrew a quarter turn.

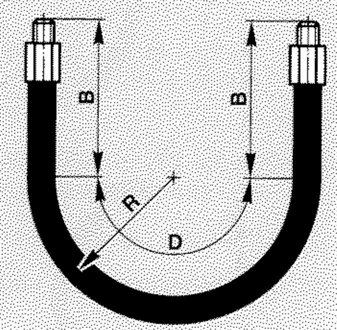
Lubricate the hose bore and nipple thread generously with oil or grease (hydraulic fluid is unsuitable).

Screw the nipple into the fitting and hose, turning clockwise from the hexagon until the hexagon is just against the fitting. Do not tighten.

Fig. 209: *Instructions for re-usable threaded fittings*

Hydraulic fluid	Hose material	Permitted temperature
HL and HLP to DIN 51 524, parts 1 and 2 (mineral oils)	NBR	100 °C
HFA and HFB to VDMA 24 317 (water and water-in-oil emulsions)	NBR	55 °C
HFC nach VDMA 24 317 (water glycol)	NBR	70 °C
HFD-R nach VDMA 24 317 (phosphate ester)	Polyamide/EPDM	80 °C
HFD-U nach VDMA 24 317 (polyol ester)	NBR	80 °C

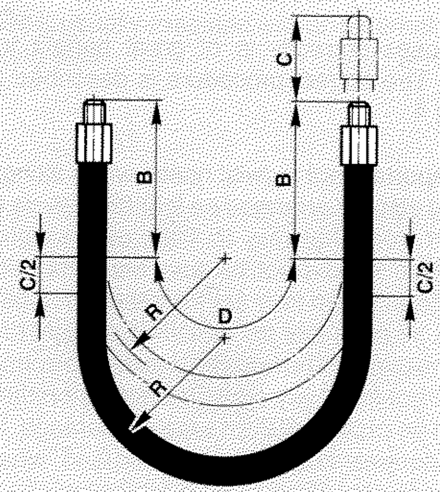
Table 59



Designing fixed hoses

Total length L= 2 B + 3.14 • R
(3.14 • R = D)

R ≥ minimum bending radius



Designing moving hoses

Total length L= 2 B + 3.14 • R + C
(3.14 • R = D)

The additional length C must be taken into account with reciprocating motions

Hose quality *)	Inside diameter DN / Additional length *B* in mm	6	8	10	12	16	20	25	32	40
	Additional length *B* in mm	90	100	110	120	130	140	160	180	200
1 ST and 1 SN	Permitted operating pressure *p* in bar	225	215	180	160	130	105	88	63	50
	Minimum bending radius *R* in mm	100	115	130	180	200	240	300	420	500
2 ST and 2 SN	Permitted operating pressure *p* in bar	400	350	330	275	250	215	165	125	90
	Minimum bending radius *R* in mm	100	115	130	180	200	240	300	420	500
4 SP and 4 SH	Permitted operating pressure *p* in bar	450	—	445	415	350	350	280	210	185
	Minimum bending radius *R* in mm	150	—	180	230	250	300	340	460	560

Table 60 *) Hose to DIN 20066

Length *L* in mm	up to DN 25	Tolerances DN 32 to DN 50	DN 60 to DN 100
to 630	+ 7 mm - 3 mm	+ 12 mm - 4 mm	
630 to 1250	+ 12 mm - 4 mm	+ 20 mm - 6 mm	+ 25 mm - 6 mm
1250 to 2500	+ 20 mm - 6 mm	+ 25 mm - 6 mm	
2500 to 8000		+ 1,5 % - 0,5 %	
over 8000		+ 3 % - 1 %	

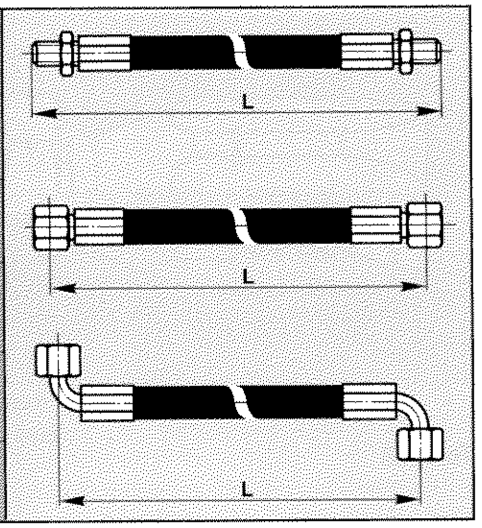


Table 61: *Length tolerances for hoses installed to DIN 20 066*

5.3.3 Installing hoses

Installing hoses correctly is of great benefit to their service life. The correct procedure is described in DIN 20066, Part 4.

It is most important not to twist hoses when fitting them. They should also be arranged so that there is no tensile stress apart from that due to their own weight. The bending radius should not be less than the minimum permitted value. When they are installed with a bend the length must be such that the specified amount of bend-free zone is retained. The hose fittings must be chosen so that they do not exert any additional stress on the hose. Hoses must be protected from external damage. Sharp-edged components should be guarded and, where necessary, hoses should be given a protective sheath.

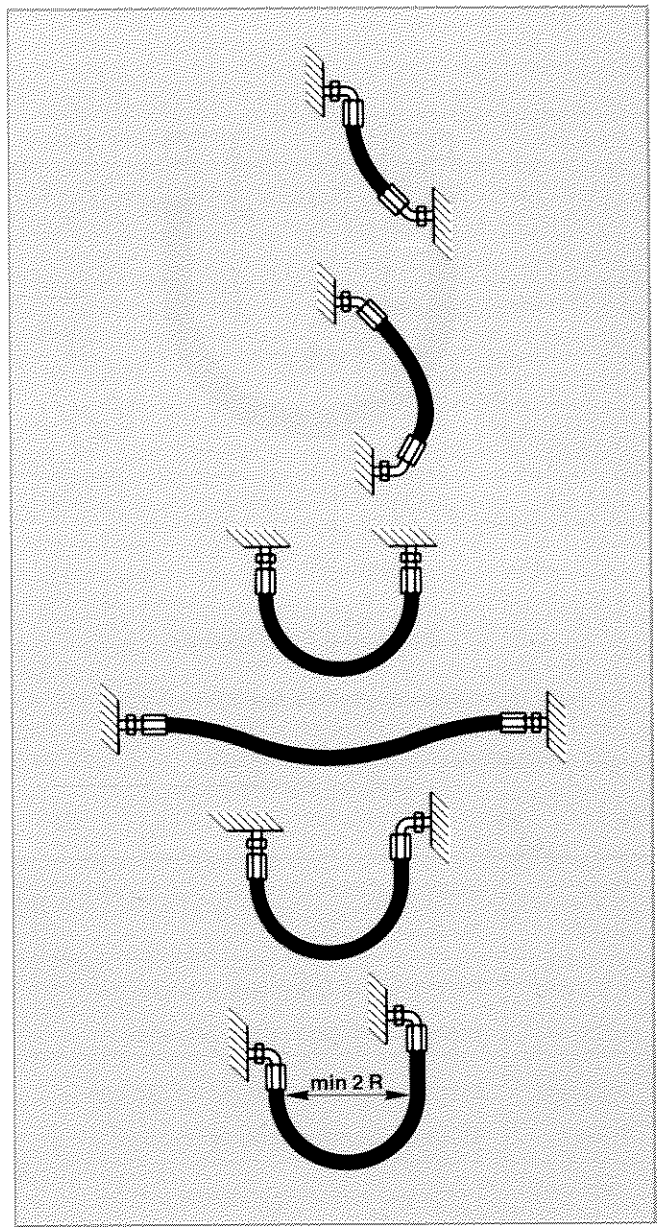


Fig. 210: *Examples of correctly installed hoses*

5.3.4 Prerequisites for safe hoses

Select hoses of adequate bore according to the operating pressure and working conditions involved.

- apply only according to the relevant standards
- proper fixing
- careful installation
- regular inspection for early detection of damage
- quick replacement of damaged hoses.

The following list of possible defects in hose lines has been taken from a brochure published by a German professional organization.

5.3.5 Typical defects in hose lines

1. Is the natural position or movement of the hose obstructed?
2. Is the hose subjected to tension, torsion or compression?
3. Is the bending radius of the hose less than the maker's minimum value during movement or when stationary?
4. Is the hose exposed to any external mechanical, thermal or chemical stress?
5. Have the hoses been painted?
6. Is the outer skin damaged (e.g. chafing, cuts or cracks)?
7. Is the outer skin brittle (e.g. cracked)?
8. Are there any pinch points?
9. Are there any kinks?
10. Are there any bulges?
11. Are there any leaks at the fittings?
12. Is the hose leaking fluid?
13. Is the hose coming out of the fitting?
14. Is the hose fitting damaged or deformed?
15. Are the hose fittings corroded?
16. Is there any discolouration of the exterior (e.g. due to solvents)?
17. Is the hose time-expired?

5.4 Quick-release couplings

Quick-release couplings are used for the quick coupling and release of sections of hydraulic systems. Their uses are many and varied but are usually related to the linking of certain items of equipment into the system for brief periods of time.

The coupling action automatically operates check valves to open up or isolate the flow of fluid. The check valves are designed to give a good seal when uncoupled; they can withstand operating pressure in that state.

The male part of the coupling is usually permanently attached to the pipework system and the female part to the hose. Any type of connection can be used between the female part and the pipework and the male part and the hose. The usual type of fittings are used for the connection to the pipework system. The connection to the hose is usually executed with an external taper having an O-ring seal.

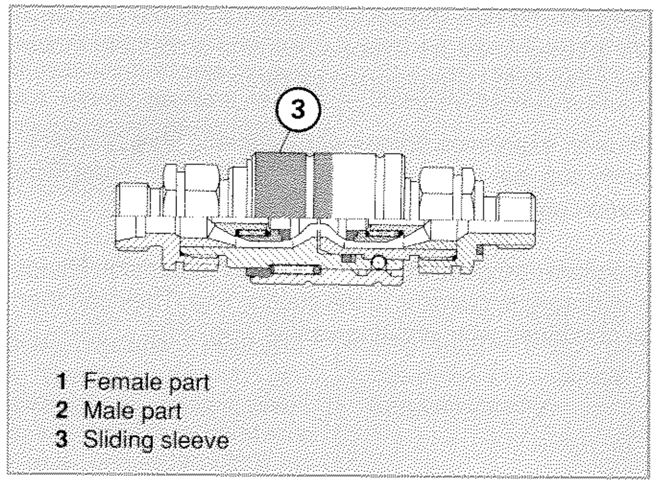


Fig. 211: *Quick-release coupling closed*

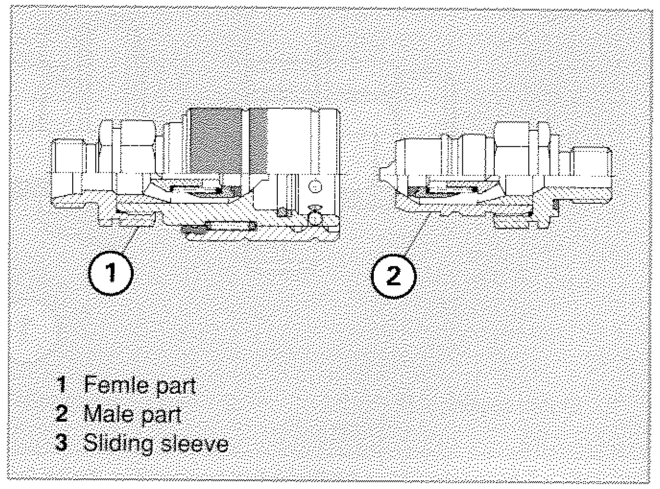


Fig. 212: *Quick-release coupling open*

5.5 Expansion fittings

Rubber expansion fittings are used in a pipework system to accommodate expansion, contraction, stress and movement caused by fluctuations in temperature, movement of foundations, fluctuating loads or vibration. They are also used for isolating structure-borne noise and shock and for accommodating inaccuracy in installation. The normal expansion fittings used in hydraulic systems are made of natural rubber or synthetic rubber in various qualities, e.g. EPDM chloroprene or NBR, which, depending on the pressure, is applied to a strong carcase of synthetic fibre or steel wire *(Table 62)*.

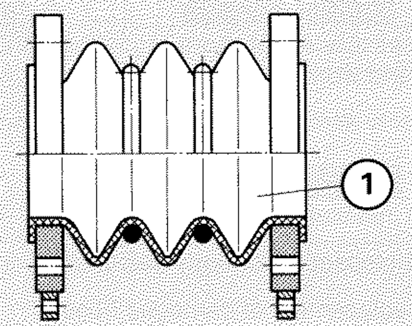
The connections to the pipework system use steel flanges of St 37.2 for sizes from DN 40 upwards or, if necessary, of other materials and normal pipe fittings for sizes up to DN 40 *(Figs. 213 to 215)*.

Standard rubber expansion fittings are widely available for sizes from DN 20 to DN 3600 and for nominal pressures of PN6, PN10 or PN16. Special connections such as SAE flanges are also available.

The pressure strength of rubber expansion fittings depends on the size and design of the fittings, the temperature to which they are exposed and the amount of movement involved *(see diagram in Table 63)*.

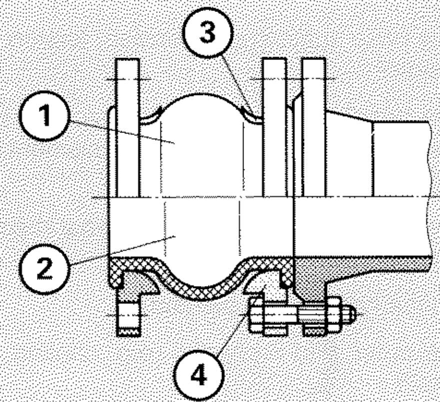
Usually, expansion fittings with synthetic fibre reinforcement can be used up to +10°C and with steel wire reinforcement up to +130°C.

The preferred point of fitting for rubber expansion fittings in hydraulic systems is in the suction line. They can then be combined with isolating valves *(Fig. 216)*.



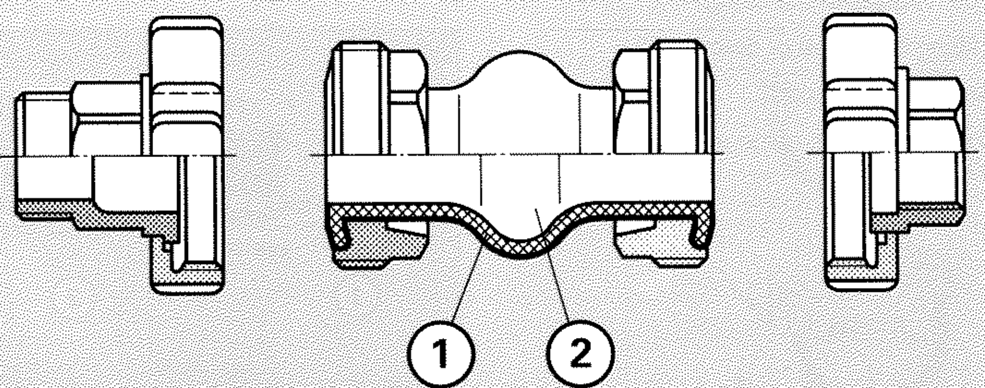
1 Inner and outer coating PTFE

Fig. 213: *Expansion fitting for flange connection to DIN 2632 and 2533, PN 10 and 16, DN 40 to 400, suitable for phosphate ester*



1 Outer coating CR
2 Inner coating NBR
3 from DN 250, flange with profile collar
4 up to DN 200, flange with solid profile

Fig. 214: *Expansion fitting for flange connection to DIN 2632 and 2633, PN 10 and 16, DN 40 to 400*

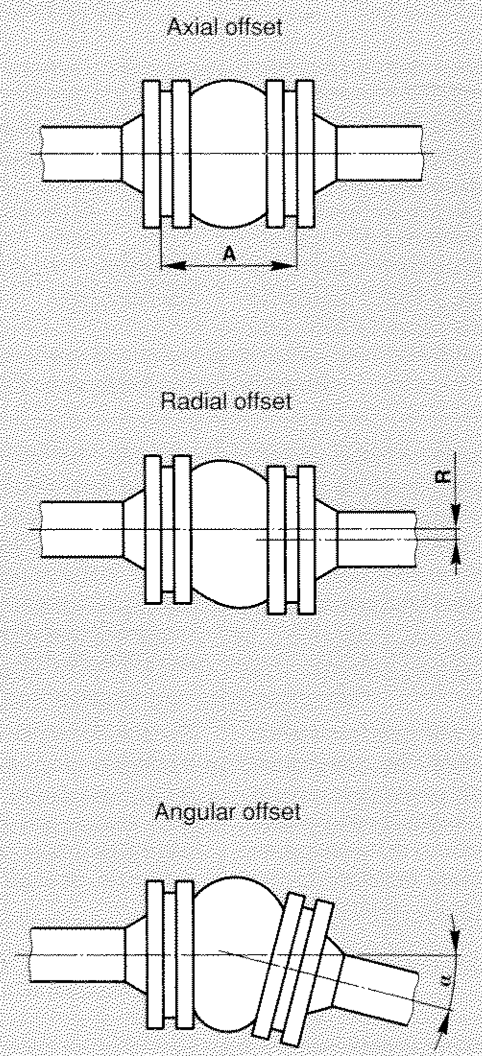


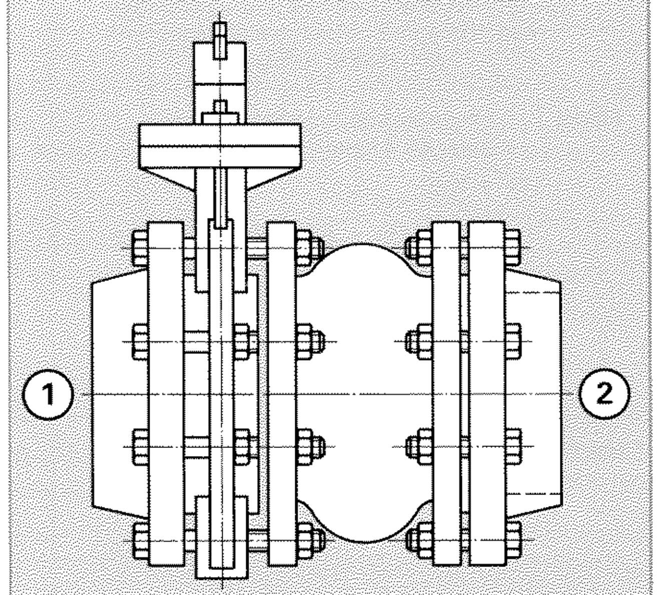
1 Outer coating CR   2 Inner coating NBR

Fig. 215: *Expansion fitting for threaded connection 3/4 to (a) 1 1/2" BSP, external or internal thread, suitable for mineral oil and water-glycol*

Type (abbreviation) according to international nomenclature/common trade names for types of rubber		Basic properties	Applications
EPDM	Ethylene propylene terpolymer/ Buna AP [3] Keltan [4] Vistalon [5]	Heat and weather-resistant quality with exceptional resistance to strongly oxidizing substances and many chemicals; practically impervious to gas (except hydrocarbon gases); temperature resistance in continuous use up to approx. +100°C, adequate flexibility down to approx. -25°C (not oil-resistant)	Suitable for water, hot water and exhaust steam, in chemical plants, especially for acids, alkalis, pickling solutions, hypochlorite solutions, etc.
NBR	Butadiene acrylic nitrile rubber Perbunan[2]	Excellent resistance to petrol and oil, exceptionally swell-proof, e.g. with petrol-benzol mixtures, practically impervious to hydrocarbon gases, temperature resistance in continuous use up to +90°C (not totally resistant to hot water)	Suitable for town gas, natural gas, heating oil, fuels, mineral oils, blast furnace gas, HL and HLP fluids (mineral oils) to DIN 51 524, parts 1 and 2 also HSC (water glycols) to VDMA 24 317
CR	Polychloroprene/ Neoprene [1] Baypren [2]	Universal quality with good resistance to oil, weather and fire, very good ageing resistance; resistant to various inorganic and organic chemicals; practically impervious to hydrocarbon gases; (not entirely resistant to hot water), temperature resistance in continuous use up to approx. +90°C; adequate flexibility down to approx. –20°C	Suitable for water supply systems, for cooling water, seawater, acids and alkalis, air, coking plant gas, paper making materials, sewage works
PTFE	Teflon	Resistant to acids in all concentrations, alkalis, chlorides, sulphates, solvents, bleaching agents, peroxides, phenols, oils, greases, water, steam, fuels; temperature resistance in continuous use from -70°C to +230°C and up to +280°C for short periods	Suitable for phosphoric acid esters HSD to VDMA 24 317 (multiple corrugations only)
Trade-marks: [1] Du Pont, [2] Bayer Ag, [3] Buna Huls, [4] DSM, [5] Esso			

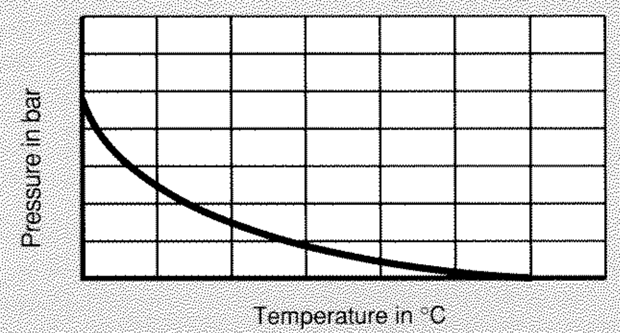
Table 62: *Expansion fittings - standard types, properties and applications*





1 Tank side
2 Pump side

Fig. 216: *Expansion fitting coupled to isolating valve*



DN	Offset in mm axial			radial			angular ±∢°	permitted number of stress cycles *)			Disturbing forces in N/mm ±25%	
	50 °C	100 °C	110 °C	50 °C	100 °C	110 °C	110 °C	50 °C	100 °C	130 °C	Up-setting	lateral
40	30	25	15	10	6	4	25	20 000	8 000	2 000	34	34
50	30	25	15	10	6	4	25	20 000	8 000	2 000	25	27
65	30	25	15	10	6	4	25	20 000	8 000	2 000	36	55
80	40	35	25	10	6	4	20	20 000	8 000	2 000	25	47
100	40	35	25	10	6	4	15	20 000	8 000	2 000	34	80
125	40	35	25	10	5	4	15	20 000	8 000	2 000	20	90
150	40	35	25	10	6	4	12	20 000	8 000	2 000	25	120
175	40	35	25	10	6	4	10	20 000	8 000	2 000	35	140
200	45	40	30	15	10	6	8	20 000	8 000	2 000	65	300
250	45	40	30	15	10	6	7	20 000	8 000	2 000	120	360
300	45	40	30	15	10	6	6	20 000	8 000	2 000	120	360
350	45	40	30	15	10	6	5	20 000	8 000	2 000	120	360

*) Permitted number of stress cycles refers to full deflection in axial or radial direction. Significantly higher figures are permitted for small amplitude vibration.

Table 63: *Movement accommodated by expansion fittings with steel wire reinforcement*

Flexible pipe fittings should always be installed where it is possible to keep an eye on them. Any rubber parts must never be coated with any substance such as paint. No lagging or other insulation should be applied.

It is better if flexible expansion fittings can be slightly compressed. Tension should be avoided in the operating state and torsion is not allowed at any time.

No more than one flexible pipe fitting may be installed between two fixed points.

Elastic pipe fittings must be protected against radiation and heat and, if necessary, must be covered or given a protective sheath.

If the hydraulic system pipework system is used for conducting electricity, e.g. for earthing welding transformers, there must be adequate bridging over expansion fittings. If such bridges are absent the pipe will act like an electrical resistance and could be damaged.

The unusual internal shape of an expansion fitting can cause turbulent flow which might cause a high noise level. A suitable guide tube must be fitted in such cases.

6 Production of hydraulic pipework

6.1 Introduction

Hydraulic pipework can be produced in two different ways - either by prefabrication in the workshop or on-site fabrication.

Technical and practical considerations often result in a combination of both methods being employed.

6.1.1 Prefabrication

Prefabrication means producing the pipework in a separate workshop away from the other activities on the site and then delivering it to the site for installation. The method requires a prefabrication department equipped with all the necessary means of production.

When the hydraulic power units have been installed, the site is surveyed for the hydraulic pipework and isometric drawings are produced. Production in the prefabrication department is then commenced.

Fully equipped workshop containers are used for the site installation work. They contain equipment for a limited amount of pipe-making and the execution of modifications when necessary.

Advantages of prefabrication

- fewer personnel, less material and less work on site without loss of flexibility
- more controllable methods for manufacturing the pipework with highly consistent quality through:
  - optimized production using permanently installed machines
  - the availability of modern, specialized equipment
  - the availability of skilled personnel
  - the use of automation
- short lines of communication between project management, design staff and production
- less transport of materials and equipment.

Another advantage of prefabrication is that the pipework can be flushed and pressure-tested in the workshop. The fitting of solderless fittings on site is not so good.

The use of prefabrication cuts production times quite substantially. It leads to a marked improvement in quality although heavy capital investment and the accumulation of practical experience are necessary at the start.

6.1.2 On-site fabrication

With on-site fabrication the production of the pipework advances with the general progress of the work of installation.

The pipes are produced in small quantities actually on the site itself or nearby after first taking measurements from the equipment already installed.

The production team and installation team work closely together and are interdependent. A well equipped workshop is an essential requirement.

Advantages of on-site fabrication

- quick response to modifications as work progresses
- early commencement of work
- short lines of communication between end user and installers.

6.2 Prefabrication of hydraulic pipework

The production of the pipework is based on the hydraulic circuit diagram. The items of hydraulic equipment shown in the diagram must first be set up and installed in accordance with the layout plan. The design of certain parts of the pipework can be established from the plan and prefabricated. Other parts are measured up on site after the other equipment has been installed.

6.2.1 Measuring the pipework

Here again the hydraulic circuit diagram is the most significant factor. It must contain information on the sizes and connections of all pipes.

Armed with this information the "global" setup of the pipework can be established on the site in discussion with the customer. Coordination with the installers of other systems, such as electricians, is essential. The other contractors must leave the chosen routes free for the runs of pipework. Each run is measured separately and accurately and entered into an isometric drawing together with any additional information. These drawings are subsequently handed over to the prefabrication department.

6.2.2 Producing the pipework

Prefabrication needs all the machines and tools for the work to be concentrated in a central workshop.

A range of permanently installed machines provides opportunities for using large, modern units such as CNC-controlled bending benches, automatic welding machines, flushing systems, X-ray equipment, pickling plant and paint spray equipment for external corrosion protection. Of course, the most economical passage for the material through the department is chosen.

Any good prefabrication workshop will have a separate room and separate tools for pipes made of stainless steel so that there is no cross contamination.

During production the pipes pass through the following stages:

- cutting to length
- bending
- fitting of means of connection
- pickling and passivating
- cleaning
- flushing and pressure testing
- application of corrosion protection
- preparing for shipment.

Other production activities are in progress at the same time:

- manufacture of supporting structures
- manufacture of any foundation structures needed
- gland and bulkhead units
- connection and distribution blocks
- other pipework components.

6.2.2.1 Cutting to length

It is preferable for the pipes to be cut to length on a machine. Pipe cutters cause the bore to be constricted.

6.2.2.2 Bending

A bent pipe is preferred to one fitted with an elbow fitting in order to keep the danger of leaks to an absolute minimum. Bending is quicker, cleaner and cheaper. Bends also cause less noise.

Good flow demands that bends should have the largest possible radius. The need to compromise with the amount of space required in the installation has led to the use of the following rule in hydraulics:

The radius should be three times the outside diameter $R = 3 \cdot d$

Small radii ($R = 1.5 \cdot d_o$) cause substantial pressure losses.

The following methods of bending are in common use:

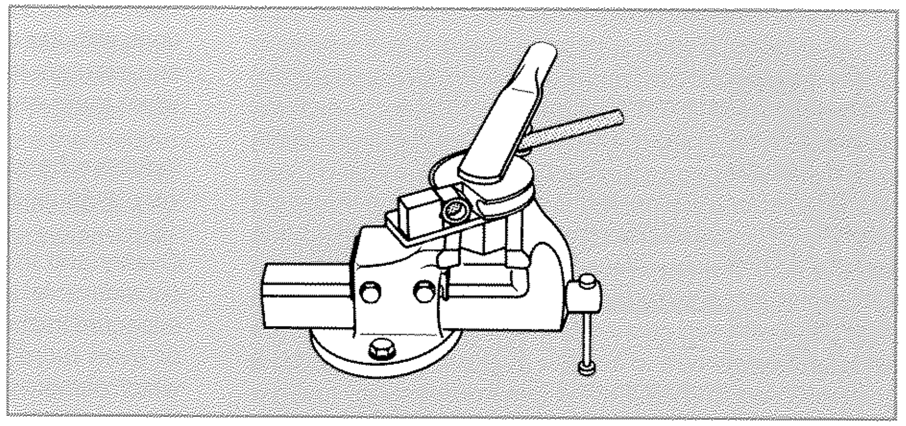


Fig. 217: *Bending by hand*

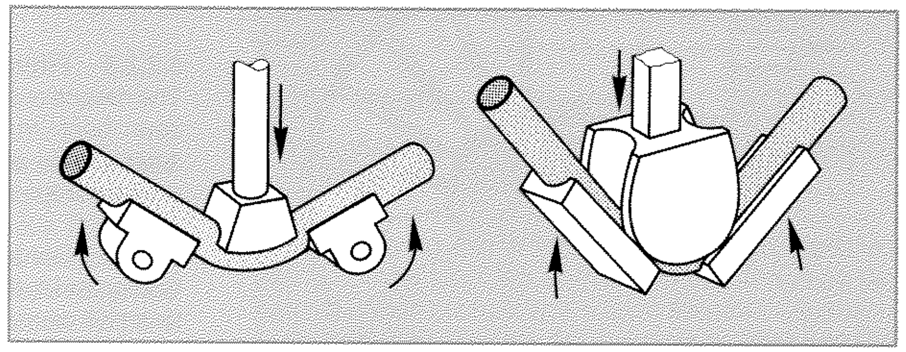


Fig. 218: *Press bending*

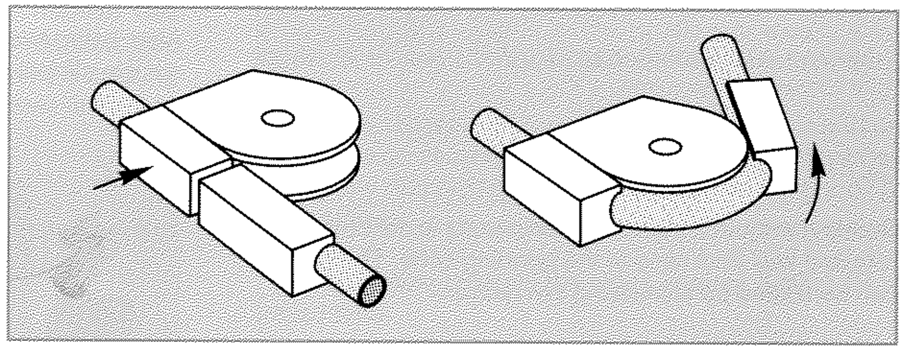


Fig. 219: *Fixture bending*

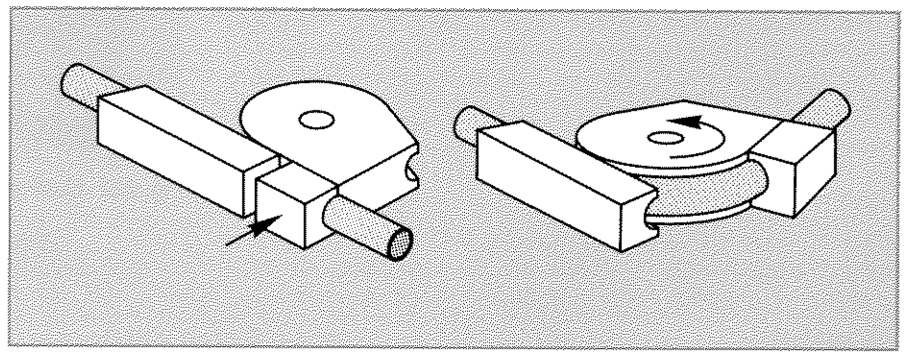


Fig. 220: *Stretch bending*

The stretch bending process is preferred because it is the most precise and its great advantage is that the pipe can be loaded internally during the bending.

Thin-walled pipes must be supported on the inside at the point of bending in order to prevent the bore going oval or deforming into a kink. A mandrel is used for this purpose.

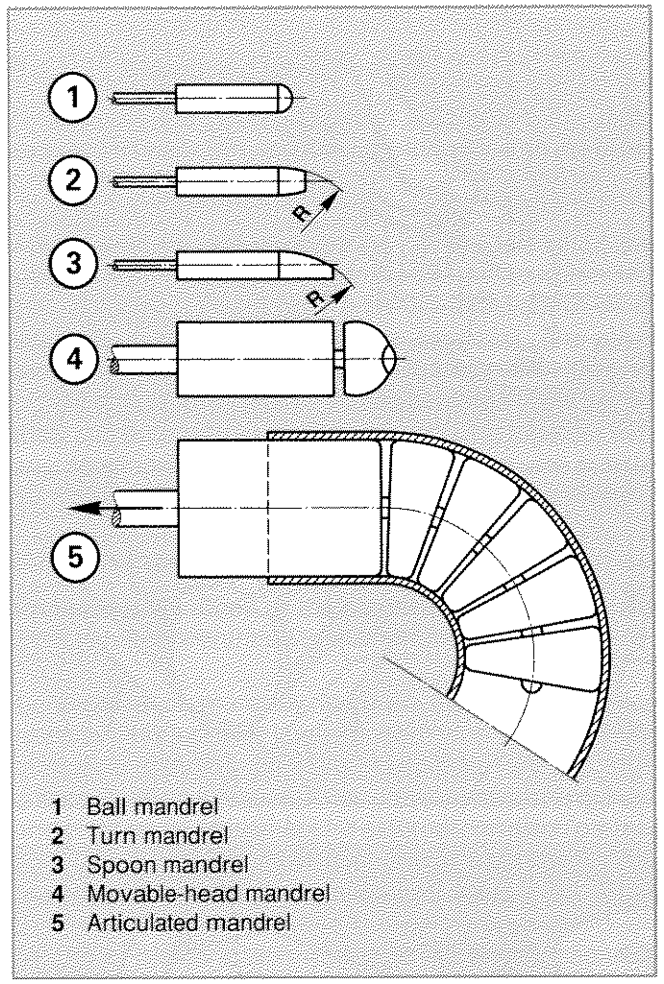


Fig. 221: *Different types of mandrel*

Bending without a mandrel often results in distortion and reduced wall thickness at the bending zone. The effects can be limited by increasing the radius.

Pipe bending equipment is manufactured in many different types and sizes. It ranges from the simple manual pipe bending kit to stand-mounted bending machines and single or multiple CNC-controlled bending machines. They are also powered in several different ways, e.g. mechanical, hydraulic and electric.

Linking a CNC-controlled bending machine to a CAD/CAM system allows one or more fully-automatic bending cycles to be initiated by simply entering the pipe number. The process starts with taking the material from store and ends with the delivery of the finished product. It is a fast and accurate means of production but is only economical when there are large quantities involved.

Inductive bending is a special process mainly used for very large pipes with thick walls.

Ovality and thinning of the wall thickness are normally considerable after inductive bending. If problems are expected the bending radii must be increased to $R = 5 \cdot d$ or even more. There are inductive bending machines which compress the pipe during the bending and the wall thickness in the bending zone can even be increased in this way.

6.2.2.3 Fitting of means of connection

The means of connection can be fitted after the pipe has been bent. In the case of welded joints it is always advisable to perform high-quality welding work at the prefabrication stage.

6.2.2.3.1 Solderless connections

Compression fittings and cutting ring fittings can be fitted to small diameter pipes by means of the appropriate connector body and a spanner.

Also, using hydraulic tools, the cutting ring and retaining nut are simple to fit on to the pipe.

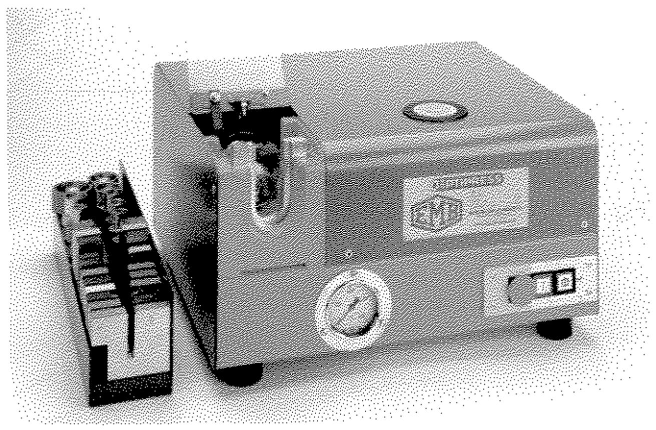


Fig. 222: *Cutting ring fitting machine*

6.2.2.3.2 Welded joints

Welded joints with an elastomer seal are better for preventing leaks and withstanding higher pressure (over 160 bar). Progress in welding techniques has allowed automatic control to be introduced with much better control of the process being obtained. Quality checks can be made by means of X-rays or ultra-sound.

The two basic methods of welding are manual and automatic.

The use of automatic welding equipment improves the overall quality of the pipework system. The latest generation of automatic welding machines regulate the speed of welding, the feed of filler metal and the electric current while welding is in progress according to various parameters that have been entered (e.g. material quality and wall thickness). The end result is a perfect weld which will pass any test.

The methods of welding used are:

- TIG (tungsten inert gas)
- MIG (metal inert gas)
- Arc welding
- Powder welding for thick pipe walls.

TIG welding

In this case the electric current is passed through a fixed tungsten electrode surrounded by argon gas. The filler metal is fed in either by hand or automatically. Due to its superior characteristics, TIG welding is often used for the root pass, whereby an anti-slag gas is passed through the pipe in order to prevent any internal combustion. With the two gases, TIG welding is very clean, i.e. no slag is produced. The wide popularity that the method has achieved is largely thanks to its suitability for automatic control. Computer-controlled machines with a rotating welding head and pulsating current guarantee first-class quality.

MIG welding

This method is especially suitable for large diameters and thick walls. The filler metal is also the electrode.

It melts very rapidly so automatic feeding is essential. The arc can be guided by hand or can be permanently set with a rotating workpiece.

The MIG method also has the welding zone enveloped in gas to prevent slag forming.

Arc welding

The arc evaporates the coating on the metal electrode to give protection to the pool of molten metal. The steel core of the electrode melts to deliver filler metal to the seam. The speed of this method lies between that of TIG and MIG.

Powder welding

Very thick walls, as encountered on cylinders, can be welded quickly and well by this method. The principle is the same as that for MIG welding and, in addition, the pool of molten metal is protected by a layer of powder flux. It is deposited around the electrode as welding progresses and care is taken that it does not mix with the molten pool. Pipes, welding ends and flanges made of St 52.4, widely used in hydraulics, need no prior or post heat treatment. Pipework made of St 52.4 should be heat treated before and/or after welding. If the system is exposed to extreme conditions it is essential for heat treatment to be carried out. Weld seams in which the root pass is laid by the TIG method with the addition of anti-slag gas need no post-finishing or hardly any at all. This is where the automatic welding equipment shows its true advantages. The subsequent boring out of small diameters, or grinding out in the case of larger diameters, can be significantly reduced.

6.2.3 Pickling and passivating

The purpose of pickling is to remove any contamination from the pipes, especially slag and weld spatter.

Pipes complying with DIN 2391 and DIN 2445, so-called precision steel tube, are pickled and passivated by the supplier. When such pipes are used the pickling process is only needed after the hot bending or after welding without anti-slag gas.

Pipes complying with DIN 2448 are supplied unpickled but they must always be subsequently pickled . It is essential for the rolled skin, coarse contamination and/or anti-corrosion coatings to be removed by sand-blasting, before any machining if necessary, i.e. before bending and welding.

The passivation protects the pipes against rusting for a limited time.

There are two methods of pickling:

- bath pickling and
- circulation pickling *(see Section 6.3.3.4)*.

Fig. 223: *Bath pickling shop*
Left to right
- *Degreasing bath (hot - 70°C)*
- *Acid bath (cold)*
- *Water bath*
- *Neutralizing bath, caustic soda with passivating additives (hot - 70°C)*

The pipes pass through the baths in sequence.

6.2.3.1 Bath pickling

Bath pickling is usually employed with prefabrication. The pickling shop should be sited close to the welding shop. Whether multi-stage or single-bath pickling is employed depends largely on the quantities of pipes needing treatment. Multi-stage pickling is usual when there are large quantities of pipes; single-bath pickling when there are small quantities of pipes.

Regardless of the type of pickling pipes must be degreased before they enter the pickling bath. In the case of multi-bath pickling there is a separate bath filled with a degreasing solution (P3). It is usually heated to 70°C.

Normally contaminated pipes will be degreased by a brief immersion. Very oily or greasy pipes will have to be degreased by hand with a cold degreasing agent. The next step of the multi-bath pickling process is for the pipe to be placed in acid. The acid bath contains about 20 to 30% pure hydrochloric acid. Depending on the amount of slag, the pipes remain in the bath for about 2 hours.

So that no acid is carried over into the neutralizing bath the pipes are dipped in a water bath after the actual pickling in order to remove as much acid as possible.

Neutralizing is carried out in the caustic soda bath. This is heated to about 70°C in order to accelerate the process. The bath also contains passivating additives to prevent film rust forming on the surface of the pickled pipes.

With single-bath pickling, pickling solution, neutralizing solution and passivating agents are all combined in a single pickling vat. In order to accelerate the process this too should be heated to about 70°C. The length of time the pipes spend in the heated bath depends on the amount and type of contamination and can be up to 8 hours. This long time means that the single-bath process is only suitable for small quantities of pipes. Nevertheless the quality of pickling is quite adequate. Naturally the pipes must be degreased before they are immersed.
Regardless of the type of pickling, pipes should be flushed through again after pickling with the type of hydraulic fluid that is to be used later. It cleans and preserves the interior of the pipes at the same time. Afterwards they should be carefully sealed for transport to site.

Depending on the circumstances the pipes may be treated with an external coating for protection against corrosion - refer to the chapter "External Corrosion Protection by Painting".

Bath pickling has the great advantage over circulation pickling that each individual pipe can be checked. There is no danger of any pickling solution remaining in the system.

Stainless steel pipes must also be pickled after welding. This is done by applying a pickling paste to the edge of the weld seam. The paste is washed off with water after allowing about one hour for it to work. Afterwards, the whole pipe is treated in the pickling bath once again so that any other contamination that might be present can be removed. Stainless steel pipes should also be flushed in the prefabrication shop before shipment to site.

6.2.4 Cleaning

When the pickled pipes have completely dried out they are flushed through with mineral oil in the same way as unpickled steel pipes. The best results are obtained by flushing the open pipes in a tank by means of a centrifugal pump. Filtration and regular renewal of the fluid are very important factors. The pipes can also be cleaned by pulling clean, non-linting, white cloths through them several times. Before they are sealed, they should be sprayed with oil inside by means of an air gun.

Depending on the hydraulic system for which the pipes are intended, extensive flushing can be performed with a separate flushing system.

In this case the pipes are coupled together in a ring and linked into the flushing system. The fluid is then circulated until the required cleanliness is achieved. Samples of fluid can be taken and tested in order to check the cleaning progress (see the chapter *"Filtration in Hydraulic Systems"*).

If the pipes are flushed before assembly, another flushing is advisable after the pipes have been assembled, using the system pumps or a separate flushing unit. On assembly, it is almost impossible to prevent some contamination entering the system despite even the most careful working practices.

6.2.5 Shipment

If a prefabrication department is to work efficiently it will have to be progressing orders for several different jobs at the same time. It is essential for the correct pipes to reach the correct destinations.

Under some circumstances it will be necessary to pack the pipes carefully in containers where they are protected against external damage, the ingress of water and dirt both during transportation and, perhaps most likely of all, when they arrive on site.

Large prefabrication departments make use of computerized order tracking.

6.2.6 Quality control

The monitoring and assurance of high quality during prefabrication makes the setting up of a quality assurance service essential. The specification for the quality control function is laid down in a quality manual. It must describe all guidelines and procedures in relation to the quality level. It must also contain an organization chart showing the hierarchy, management and responsibilities. Large projects will require supplementary instructions for special project-related quality specifications.

A well-equipped prefabrication workshop will have a range of quality control equipment including:

- Magnaflux test equipment
- X-ray equipment
- Metal check test equipment
- Ultrasonic test equipment
- Oil analysis laboratory.

6.3 Production of hydraulic pipework on site

The reasons for manufacturing pipework on site are many and varied:

- the absence of prefabrication facilities
- the customer's wishes (providing work for the local population)
- system requirements (e.g. all lines being welded together to form a whole).

The skilled work on the pipes such as cutting to length, bending and welding is almost the same whether performed in the prefabrication shop or on site. However, there are number of differences in the execution:

- Most activities take place on site
- More staff are required on site
- A production shop on or near the site must be provided
- The machines are simpler, have fewer capabilities and a lower capacity
- Work on large pipes or special procedures are very difficult
- Production conditions on site are less than ideal
- In consequence the customer often demands stricter checks, e.g. X-ray
- Apart from production facilities there must also be space for working and storage on site
- Internal inspection of endless welded lines without flanges or connectors is impossible.

6.3.1 Survey of pipework

Once again the circuit diagram is the most important aid. Often the work bench is immediately adjacent to the installation and the pipework can be produced by measuring and fitting according to the data given in the circuit diagram.

6.3.2 Bending

Pipes up to size DN 32 are usually bent by hand or by a small portable electro-hydraulic pipe bender.

Larger sizes must be heated before bending or be taken to the workshop. Sometimes it is necessary to weld in elbows or bends. These are of smaller radius than bent pipes and so are less efficient in flow. The greater number of welds also increases the cost.

6.3.3 Fitting the means of connection

6.3.3.1 Solderless connections

These can be fitted in the same way as for prefabrication and by hand, using spanners or the devices described in *Section 6.2.2.3.1.*

1 Saw pipe square and deburr. Do not use a pipe cutter - use a sawing machine if possible. Lightly deburr the inside and outside of each end. Clean.

2 Slide the retaining nut and cutting ring on to the pipe as shown.

3 Press the pipe against the pipe stop in the fitting and tighten the retaining nut by hand.

4 Firmly tighten the retaining nut until the cutting ring grips the pipe. This can be felt by the increase in tightening torque.

5 Finish tightening with one extra turn

Note: Hold the fitting with a spanner to prevent it turning.

6 Check
Examine the cut-in of the cutting ring. The pushed-up metal must fill the space in front of the cutting ring end. The cutting ring may rotate if tried but it must not move axially.

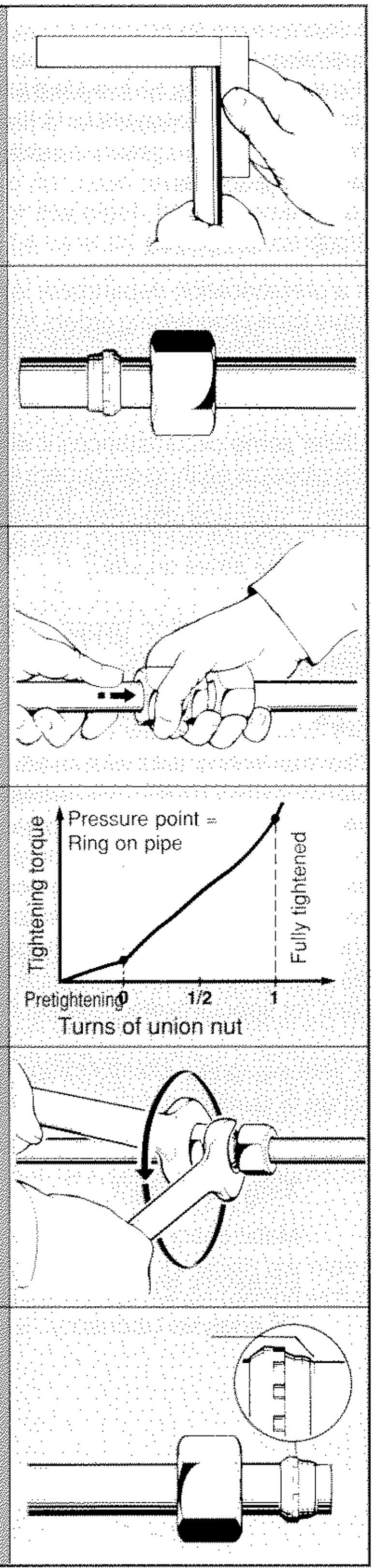


Fig. 224: *Instructions for assembling cutting ring fittings*

6.3.3.2 Welded joints

Production of the pipework on site uses the same welding methods employed for shop prefabrication. However, expensive automatic welding machines can only rarely be used on site (if at all). It is sometimes difficult to establish optimum welding conditions on site.

6.3.3.3 Cleaning

If pipes do not have to be permanently joined but employ connectors and flanges, the cleaning can be carried out as described in *Section 6.2.3.* If not, circulation pickling is the alternative.

6.3.3.4 Circulation pickling

The circulation pickling method is only ever used on site. The pickling of individual pipes is to be preferred if possible. Circulation pickling is only employed when the piping system is welded together as an "endless" system. Basically, the procedure for circulation pickling is the same as that described for the flushing of hydraulic systems in *Section 8*. However, it must be remembered that the pickling solution must not pass through hydraulic devices, manifold blocks, etc. Only the actual pipework itself may be pickled.

The pickling equipment comprises pumps, heaters, tanks and the necessary hose connections which must be suitable for the pickling solution.

The solution is pumped round the pipework relatively slowly. When acids are used in the solution the temperature is not of any great significance. A higher temperature of approximately 70°C is advisable when using single-bath pickling.

When acids are used for pickling, the pipework system must be flushed through with a neutralizing agent after the pickling solution.

Following pickling and neutralizing it is essential to flush out the pipework with a hydraulic fluid that is compatible with the actual fluid to be used later. The flushing of hydraulic systems is described in *Section 8.*

6.4 Summary

Both methods - prefabrication and on-site fabrication - have advantages and disadvantages. There are significant differences which have their effect on quality.

With prefabrication nearly all activities can be concentrated in a workshop specially equipped for the purpose. Here, under optimum conditions, there is a range of machines, ideally suited to automatic control, that can be used for prefabricating the pipework away from the site. The necessity for producing the pipework in sections between 6 and 7 m long at most offers many advantages in reference to cleaning, inspection and testing. Production quality is consistently high.

For the production of pipework on site, the means of production and the working staff must be concentrated on or near the site itself. This offers advantages in a fast response to any modifications requested. Short lines of communication to the end-user and other subcontractors are also good.

There are generally fewer machines than for workshop production and on-site working is made more difficult by the presence of other teams of workers. It is often difficult to achieve the required standards of cleanliness and quality.

It should be emphasized that permanently welded pipework systems should be avoided if possible.

7 Installation of hydraulic pipework

7.1 Preparations

The type of preparations required vary according to the scope of the project and the type of work involved.

7.1.1 Site equipment

The installation of hydraulic pipework will require:

- hand tools
- welding gear for fixing supports
- grinding machines
- lifting tackle
- storage facilities (e.g. containers)

Some of this equipment can be provided by the end-user if the pipework is manufactured on site. If the pipework is prefabricated the supplier will generally have to provide his own tools. The usual practice is to have a workshop container containing a workbench, welding equipment, machine saw, bending equipment and hand tools. The prefabricated pipework can be transported together with the container.

7.1.2 Other preparations

There must be co-ordinating talks between end-user, hydraulics suppliers and other sub-contractors. Cooperation, responsibilities, methods and safety regulations must be discussed and decided on.

7.2 The actual installation

The actual installation of the hydraulic pipework must be carried out very carefully so that the final quality matches that achieved in production. Special attention must be paid to:

- clean working
- weight of the pipework
- straight, direct and clear pipe runs
- stress-free mounting
- facilities for possible extensions.

7.2.1 Clean working

It is very important during the installation process to prevent any dirt getting into the hydraulic system. For one thing this means that the place of installation must be shielded against the wind which can cause dangerous contamination. Pipes, blocks and components must be carefully sealed for delivery to site.

7.2.2 Weight of the pipework

The pipework becomes heavier the higher the working pressure. This affects the installation procedures and the support structures. For practical reasons the larger pipes should be run in the floor or immediately above the floor. They must be properly secured.

7.2.3 Straight, direct and clear pipe runs

A straight run of pipe is economic. They are also clearer to see and therefore easier to maintain.
It is sensible to:

- work according to the main outline of the design
- bundle lines together
- avoid complex situations
- differentiate between different sub-systems.

7.2.4 Stress-free mounting

The pipework must be free from stress after it has been installed. Suitable countermeasures must be adopted if there are any problems in connection with stress due to tolerances, temperature or support design.

Some such measures are:

- the use of hoses
- the use of expansion fittings
- the use of packing pieces which must be measured and made after the installation has progressed to a certain point.

7.2.5 Facilities for extensions

It is necessary to know whether the pipework system will have to be extended at some later date. If so, suitable measures will have to be adopted during design and construction. These should include facilities for the draining of certain sections of the system or the immediate provision of connections with isolating valves at suitable points.

7.3 Assembling the pipework

The pipework should be assembled under clean conditions in order to prevent any early contamination of the hydraulic system. The workplace must be adequately protected against dust and other forms of dirt. Wind and draughts can carry many microscopic forms of contamination and should be kept at bay as much as possible.

The various connections of the pipes and items of equipment must remain plugged as assembly progresses. The plugs and caps should only be removed when the actual connecting-up is about to be performed. Connections and connection surfaces should be checked for cleanliness before connecting-up. It is not sufficient to rely solely on the flushing of the system that will probably take place after assembly.

Workshops, tools and other auxiliary equipment must be clean at the start and must remain so.

7.3.1 Mounting the pipes

Once the primary route for the pipework has been established, the first stage of site work is to install the support structures and pipe clips. This finally establishes the pipework routing which has only been preliminary up to this point.

It must be remembered that shock and vibration can cause extensive damage in a hydraulic system. Harmful vibration can be generated by the hydraulic system itself, e.g. due to cavitation, reciprocating motions, pressure pulsation, alignment errors, mechanical movements, etc. This means that the supporting structures or the building housing the equipment could be damaged. In the case of ships it is possible for shock and vibration from elsewhere to affect the hydraulic equipment.

The effects of the shock and vibration can be reduced by choosing the correct positions for the supports and clips for the pipework. A solid base or rigid structure in combination with strong pipe clips are the best means of preventing damage due to vibration. Clips with rubber inserts, hoses and expansion joints help to reduce the transmission of vibration. In extreme cases the support structures can be mounted on anti-vibration mountings of the spring or rubber type.

Flexible bulkhead adaptors employing rubber elements should be employed at the points where pipes pass through decks or bulkheads, i.e on board ship.

The spacing between pipe clips is specified in DIN 24 346.
Mechanical anchors can be used for fixing to concrete. Adhesive anchors are preferred for heavy lines.

It is very important for the pipes not to be subjected to any stress due to the mounting. Pipes which cannot be installed without straining them should be realigned or remade.

Pipes should be installed with a slight fall towards the tank.

Thermal expansion and contraction must be taken into account at the project design stage and either expansion loops, hoses or expansion joints must be provided. The movement of these devices must be allowed for in the fixings.

In order to facilitate assembly and disassembly when a repair is necessary, connectors and flanges should be offset from one another and be installed at the prescribed distance from the adjacent pipe.

When installing stainless pipes it is essential to avoid contact between the austenitic steel pipe and any ferritic steel structures through the use of suitable pipe clips. Corrosive attack can take place very quickly, especially in atmospheres containing salt and other corrosive substances.

7.4 Fitting hoses

Flexible hoses are often used:

- to isolate vibration
- to allow relative movement
- to connect two points of previously unknown distance apart.

The first two definitely involve some movement of the hoses.

Guidelines for the installation of hoses will be found in *Section 5.3.1.*

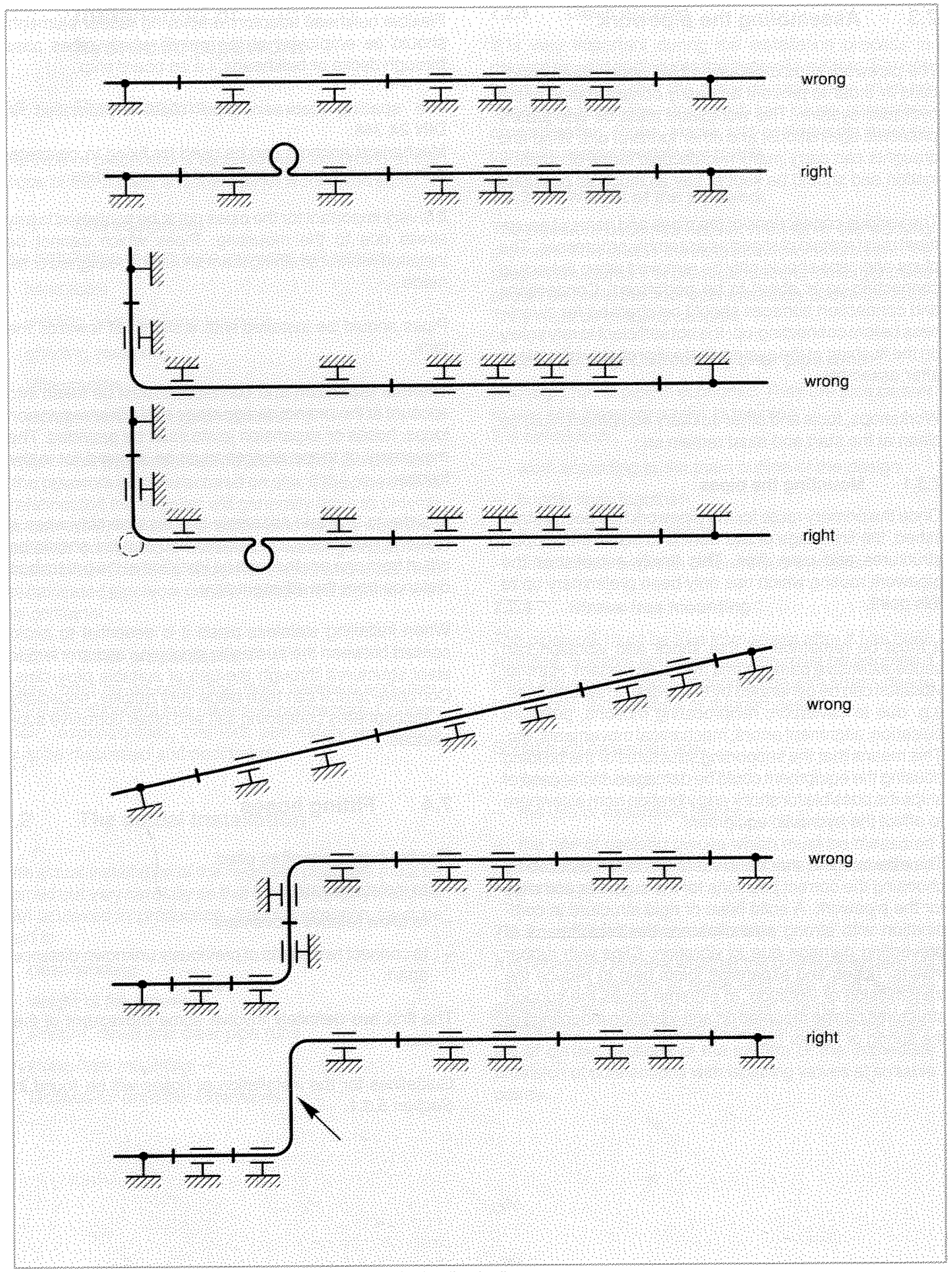


Fig. 225: *Examples of correctly and wrongly installed hydraulic pipelines*

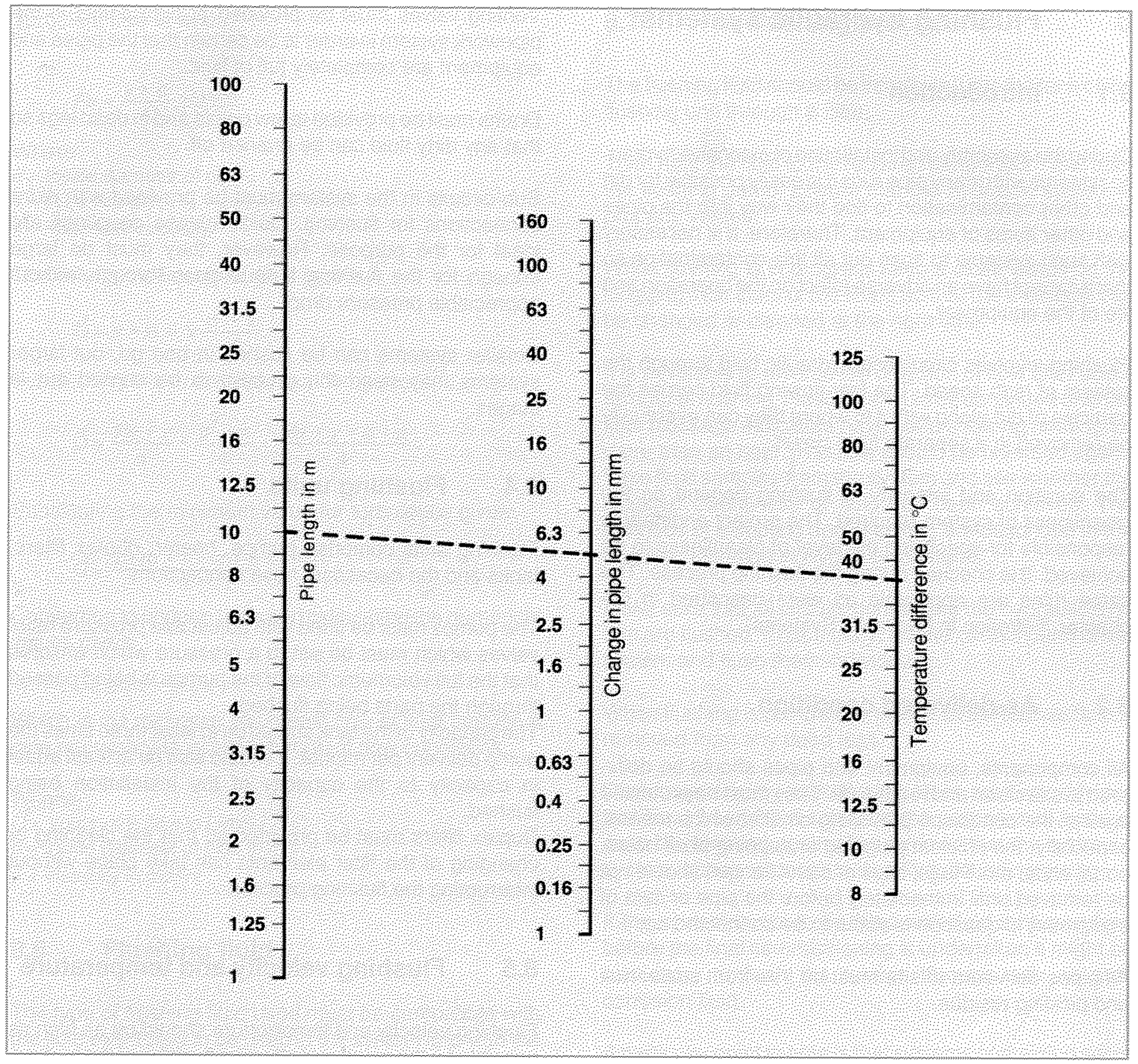


Diagram 61: *Pipe expansion and contraction with temperature*

7.5 Summary

The design, manufacture and installation of pipework systems are very closely related procedures. The process of installation really begins at the preliminary design stage as various aspects of installation must be allowed for on the drawing board. Suitable preparations can make for greater efficiency and economy. The reliability and safety of a hydraulic system depend to a large extent on the pipework so it is very important for all work on the pipework to be carried out carefully and to a high quality. Communication between contractor and end-user is of great significance in this respect.

8 Flushing hydraulic systems

8.1 Introduction

During the assembly and installation of extensive hydraulic systems with many pipe runs it is easily possible for dirt and other contamination to find their way into the pipes and other items of equipment. Therefore, it is necessary with such systems to flush out all this contamination so that it cannot have an adverse effect on the operation and life of the equipment.

Flushing involves circulating hydraulic fluid through the system at high speed. The fast-flowing fluid carries the particles of dirt along with it to where they can eventually be removed in a separate filter circuit.

The duration and intensity of flushing depend on the cleanliness of system required. The project engineer is responsible for specifying the level of cleanliness to be achieved. It is checked during the flushing process. The same rules are applicable as were described in the chapter "Filtration in Hydraulic Systems".

8.2 As-delivered condition

All components, equipment and pipes should be delivered to site clean and ready to fit. They must be protected against the conditions prevailing on site; which means, especially, the efficient plugging or capping of all holes, ports, ends, etc. Plugs, caps or blank flanges should not be removed until immediately before the pipe or item of equipment in question is about to be connected up.

All pipes delivered to site must be free from scale, rust and pickling residue.

8.3 Preparing the system for flushing

Items of equipment that could be damaged by the flushing must either be replaced by suitable devices or be by-passed using pipes or hoses.

For example, the high-pressure pumps of the system must be by-passed by suitable flushing lines and the servo valves in the system replaced with flushing plates. The elements should be removed from the system filters before flushing begins so that they are not made dirty by the flushing fluid. The filtering should be carried out by a separate flushing filter.

Venting valves must be provided at high points in the pipework system in order to be certain that the pipes and equipment are completely full of fluid.

Drains must be provided at low points and in dead legs so that any dirty fluid can be drained off.

Sub-circuits in the system must be provided with extra connections for flushing. Quick-release couplings are ideal for the purpose. However, they must be large enough for the flushing fluid to pass through without appreciable pressure drop.

Smaller systems can be flushed in one go, but larger systems may need the process to be carried out in stages.

8.4 Flushing units

Flushing units comprise pumps, heater, duplex filters, tanks and the necessary hose connections.

The pumps must be fitted with adjustable pressure relief valves which must be set to a pressure which ensures that the full volumetric flow of the pumps is being passed through the parts being flushed.
The capacity of the tank should be at least three times the pump delivery per minute. It should also be at least equal in capacity to the capacity of the installation being flushed.
Duplex filters must be provided so that the cleaning or changing of the filter elements can take place without interrupting the flushing process.

8.5 Flushing velocity and temperature

Flushing velocity and temperature should be as high as possible for the quick and effective flushing of the hydraulic system.

Practical experience has shown that the temperature should be higher than the operating temperature if possible. In the case of mineral oils 60°C is recommended and with water-in-oil emulsions and water-glycol 50°C is recommended.

The flow velocity should be at least twice that during normal operation of the installation and the flow should be in the turbulent range at all points.

Turbulent flow is when:

$$Re = \frac{v \cdot d}{\nu} \geq 2320$$

Where
d = Inside diameter of pipe
v = Fluid velocity
ν = Kinematic viscosity.

The volumetric flow through a pipe is:

$$\dot{Q} = v \cdot A = v \frac{d^2 \cdot \pi}{4}.$$

Substituting this in the first equation gives:

$$\dot{Q} = \frac{Re \cdot \nu}{d} \cdot \frac{d^2 \cdot \pi}{4} = \frac{Re \cdot \nu \cdot d \cdot \pi}{4}.$$

Substituting the minimum value for turbulence gives:

$$\dot{Q} = \frac{2320 \cdot \nu \cdot d \cdot \pi}{4}.$$

Converting to uniform units and calculating the numerical values gives:

$$\dot{Q} \geq 0.11 \cdot \nu \cdot d$$

Which is the minimum volumetric flow of fluid through a pipe necessary to achieve turbulent flow conditions.

Where
$\dot{Q}$ = Volumetric flow in L/min
d = Inside diameter of pipe in mm
ν = Viscosity in mm²/s

8.6 Flushing fluid

The type of flushing fluid used must be compatible with the fluid eventually to be used in the system and with the materials used in the system, especially the seals.

Although the same fluid to be used subsequently can be used for flushing, a lower viscosity flushing fluid with no expensive additives can flush a system more quickly and more cheaply.

8.7 The flushing procedure

The flushing fluid should be introduced into the tank of the flushing unit through a filter.

Then heat the fluid to the required temperature and start the flushing pump. Ensure that the hydraulic system is properly vented.

While flushing is in progress keep a close watch on the clogging indicators of the filters so that the elements can be changed or cleaned at the right time.

It is advisable to reverse the direction of flow after about an hour of flushing. Flushing should be continued until the filter blockage indicators have been showing nothing for more than an hour. The flow should be reversed again. Take fluid samples from the system and see whether the required cleanliness of fluid has been achieved. The methods of sampling and checking are described in the chapter "*Filtration in Hydraulic Systems*".

Make sure that all circuits comprising the hydraulic system are flushed. If necessary, isolate certain parts of the system and flush them individually.

When flushing is complete ensure that all residual fluid is removed from any dead legs.

Obviously, any by-passes and other auxiliary devices that have been fitted solely for the purposes of flushing must be removed again afterwards so that the hydraulic system is then restored to its fully functioning state.

If the system is not to be filled with fluid for some time before the final commissioning is undertaken it might be necessary to flush the system through with an anti-corrosion fluid.

A particular point to note is that systems containing servo valves will have to be flushed for up to 48 hours in order to obtain the required degree of cleanliness.

Internal Protection of Hydraulic Components and Power Units

Erhard Wiesmann

1 General

When hydraulic components and equipment are stored for long periods of time it is possible for them to suffer gradual deterioration which can have an adverse effect on the subsequent commissioning. The residual fluid left inside the equipment tends to go resinous after a certain time which makes spools, for example, stiff to move and rotating parts have difficulty in forming a lubricating film quickly.

Such problems can be delayed by testing or filling items with a preserving oil before they are stored. Such oils remaining in the equipment provide better protection against corrosion and resinification for longer periods.

Table 64 lists the protective measures to be adopted in relation to storage conditions and storage time.

Seaworthy packing is usually necessary for extended storage or for export shipment and storage. Internal protection and packing methods are interactive and so should be coordinated for the best overall effect.

Parts and equipment should not be stored for more than two years because damage to seals can start after this period of time.

The protective measures described in this chapter are applicable to hydraulic equipment operated with the following fluids:

Mineral oil	HL, HLP	to DIN 51 524, Parts 1 and 2
Oil-in-water emulsion	HFAE	to DIN 24 320
Water-in-oil emulsion	HFB	to VDMA 42 317
Water based solutions	HFC	to VDMA 24 317
Phosphate ester	HFD-R	to VDMA 24 317

2 Methods of protection

For equipment operated with mineral oil-based fluid, internal protection is provided with either mineral oil-based fluid (Protective fluid A) or anti-corrosion oil (Protective fluid B).

For equipment operated with aqueous fire-resistant fluids of Groups HFAE, HFB or HFC, internal protection is possible with either Protective fluid A (mineral oil) or Protective fluid B (anti-corrosion oil), provided the protective fluid is flushed out of the system and its equipment before commissioning is started. The amount of residual protector in the fluid must not exceed 0.1%.

For equipment operated with anhydrous fire-resistant fluids of the phosphate ester type HFD-R, internal protection is provided by Protective fluid C. It should be noted that the same fluid is used for commissioning and operation. If a different fluid is used for commissioning than has been used for internal protection, the protector must be removed by flushing until the residual concentration does not exceed 0.2%.

Note

If any items of equipment incorporate seals or other components such as accumulator bladders made of EPDM, it must be ensured that no residual mineral oil gets into the operating fluid. Mineral oil is harmful to EPDM elastomers.

Equipment wetted or filled with mineral oil must be flushed out with operating fluid before the power unit is installed.

3 Protective fluids A, B and C

These protective fluids are widely used for internal protection in the hydraulics industry, including Mannesmann Rexroth.

3.1 Protective fluid A

Mineral oil HLP 68, DIN 51 524, Parts 1 and 2
Density approx. 0.87 kg/dm^3
Viscosity approx. 40 mm^2/s at 50°C

3.2 Protective fluid B

Anti-corrosion oil

This is a mineral oil-based protector with excellent ageing and anti-corrosion properties.

Density approx. 0.89 kg/dm^3

Viscosity approx. 44 mm^2/s at 50°C

If anti-corrosion oil is used for testing and filling hydraulic equipment it must be drained before commissioning the equipment with mineral oil.

3.3 Protective fluid C

Phosphate ester HFD 46-R
to VDMA 24317
Density approx. 1.125 kg/dm^3
Viscosity approx. 32 mm^2/s at 50°C

4 Internal protection procedure

The internal protection listed in *Table 64* is achieved by appropriate testing or filling of the hydraulic equipment and systems.

Testing means a brief period of operation of the tank or item of equipment filled with the appropriate protector followed by draining. All pipe connections must be plugged afterwards.

If *Table 64* indicates that internal protection must be provided by filling, testing must be carried out with the appropriate protector first. The protector remains in the equipment and components. The pipe connections are sealed with plugs or flanges. Tanks must not remain filled. It is sufficient to fill any equipment mounted on or in the tanks, such as pumps, control gear and filters, with the appropriate protective fluid.

Tanks for HL or HLP fluid are protected against corrosion by internal painting (a single-component polyurethane zinc-rich paint).

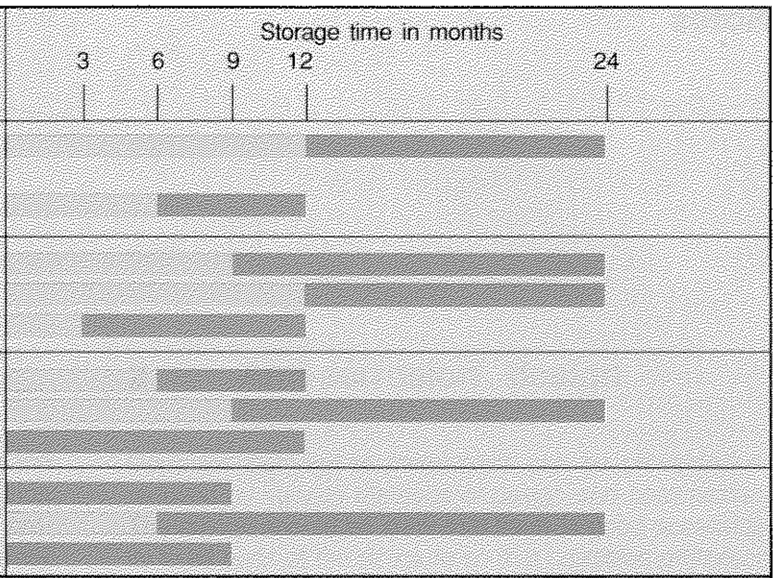
Tanks for HFAE, HFB, HFC or HFD-R fluids are best made of stainless steel. Tanks made of normal steel are protected by an internal coating resistant to the fluid (information on suitable resistant coatings will be provided by the fluid supplier).

In exceptional cases the interior of the tank can remain bare if commissioning is to be carried out soon. The surfaces can then be sprayed with anti-corrosion oil (Protective fluid B). The protective fluid must be removed subsequently with a cleaning fluid (e.g. cold degreasing agent) before the system is commissioned.

Note

The interior of the tank must be inspected visually before commissioning takes place and any contamination or condensation discovered must be removed.

Storage conditions	Packing	Protector	Storage time in months
Stored in dry places maintained at uniform temperature	Seaworthy	A B C	
	Non-seaworthy	A B C	
Stored outdoors	Seaworthy	A B C	
	Non-seaworthy*	A B C	



* Protected against damage and the ingress of water
A = Mineral oil
B = Anti-corrosion oil
C = HFD-R
Test with protective medium
Fill with protective medium

Table 64

If transport or storage times are longer than those shown in *Table 64*, the equipment manufacturer must be approached for advice on suitable protective measures to be employed.

5 External protection procedure

External protection against corrosion is provided by painting; full details are given in the chapter "*External Corrosion Protection by Painting*".

An epoxy resin undercoat is sufficient for external protection for periods of storage up to 6 months in dry places maintained at normal even temperatures.

If the storage time is more than 6 months, suitable finished paintwork must be applied. The type of coating system employed (single or two-component paints) depends on the type of attack to which the surfaces involved are exposed.

Care must be taken that the external paintwork does not suffer any damage during transport and storage.

Bare exposed parts must be sprayed with a waxy anti-corrosion substance.

Note

If any packed hydraulic equipment is unpacked for inspection purposes it is essential to replace the packaging carefully afterwards. New drying crystals must be put in if the packing is for ocean shipping.

External Corrosion Protection by Painting

Erhard Wiesmann

1 General

The reasons of painting items of equipment are many and varied. They include visual appearance, protection against corrosion, the provision of special surface properties such as good reflection of light as well as easy cleaning and resistance to chemical substances in the environment.

There is no single universal paint or other coating material. A particular paint must be chosen to suit the surface to which it is to be applied and its environment.

Paints may be of varying consistency ranging from liquid to paste-like. They may also be physically and/or chemically drying substances or blends which can be applied by brushing, spraying or other method.

An important constituent of a paint is the medium binding the particles of paint pigment together and to the surface. The pigment is responsible for the colour. The range of colours available will be found on the RAL shade card which is related to the colour register RAL 840 HR.

When applying paint, care must be taken to ensure that the whole structure receives good, uniform protection against corrosion suitable for the purpose. In the case of structures where the susceptibility to corrosion varies, extra corrosion protection must be provided in the more heavily stressed areas, e.g. by employing hot-galvanized components or using stainless materials.

2 Anti-corrosion design

Corrosion damage to metal parts can be prevented or reduced by various methods of design and construction. The design should be such that there is good accessibility to all parts of the structure so that corrosion protection can be applied, checked and kept in good order. If there are special reasons why this is impossible, the cleaning and painting of places which will no longer be accessible after assembly must be carried out beforehand.

Narrow gaps, voids and blind holes in which dirt can accumulate should be avoided.

Voids and the underside of metal plates on which condensation can form should be adequately ventilated.

When units are installed outdoors, with the attendant greater danger of corrosion, all weld seams should be continuous and there should be no sections open at the top.

Sharp cut edges and burrs should be avoided because the surface tension of liquid paints causes them to pull back from sharp edges and corners.

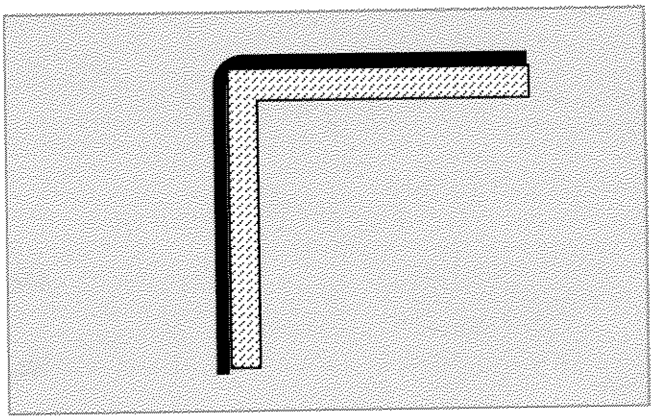


Fig. 226

3 Surface preparation

The effectiveness and life of anti-corrosion paints on metal surfaces depends to a large extent on good preparation of the surface to which the paint will be applied.

3.1 Preparation of steel parts

Steel parts are prepared by particle blasting to a standard cleanliness rating of SA 2 1/2. The standard ratings are laid down in DIN 55 925, Part 4.

SA 2 1/2 means that sufficient scale, rust and old paint have been removed to ensure that any residue remaining on the surface will simply be visible as slight shadows due to the toning of pores.

3.2 Surface preparation of hydraulic power units before applying the second undercoat

The surfaces must be carefully and completely cleaned with a suitable cleaning agent to remove all dirt, dust, grease and other substances which would have an adverse effect on the paint.

The cleaning agents must be handled in accordance with the relevant safety instructions.

4 Selecting paints according to corroding media and environmental factors

Both single-component paints and two-component paints are available. Which system is chosen depends on the corroding media to which the surface requiring protection will be exposed.

Note, for example, that a seawater or brackish water environment will require a different painting system to a dry or warm and moist environment.

Corroding media and environmental factors	Painting system Single-component system Alkyd resin top coat	Painting system Two-component system Polyurethane top coat
Dry temperate climate (DIN 50019)	●	
Warm humid climate (DIN 50019)	●	
Marine climate (DIN 50019)	●	
Fresh water	●	
Seawater or brackish water		●
Mineral oil (DIN 51524)	●	
Oil-in-water emulsion HFA (DIN 24320)		●
Water-in-oil emulsion HFB (VDMA 24317)		●
Aqueous polymer solutions HFC (VDMA 24317)		●
Phosphate ester HFD-R (VDMA 24317)		●
● resistant to		

Table 65

5 Single-component paint systems

In order to obtain the required properties of protection, more than one coat of paint must be applied to a surface, i.e. undercoat and top coat. Remember, of course, that the sequence of application - undercoat first, top coat second - cannot be reversed. And no more undercoat may be applied to parts which have already received their top coat.

Top coat Hydraulic power units	Alkyd resin paint (any desired RAL colour)
2nd undercoat Hydraulic power units	Epoxy undercoat RAL 7031 or RAL 6011
1st undercoat Hydraulic power units	Epoxy undercoat RAL 7031
1st undercoat Steelwork	Zinc-rich primer RAL 7000
Surface	Steel to normal grade of cleanliness SA 2 1/2

Table 66: *Painting procedures*

5.1 Paints

5.1.1 Zinc-rich primer

Zinc-rich primer is a high-quality single-component polyurethane based paint containing approximately 84% zinc in solid form. It is sprayed on to surfaces which have been previously sand or shot blasted (the necessary surface roughness for zinc-rich primer is 50 µm).

Where bruises, cracks or voids are present, zinc-rich primer clearly provides protection for a small, although limited, exposed surrounding area. It initiates a cathodic process which produces large quantities of a product of zinc corrosion which eventually covers the damaged area. In fact, the protection is due less to the cathodic process, which is only present initially, but more to the products of corrosion produced by the cathodic process preventing the zinc going into solution and thereby increasing the durability.

Zinc-rich primer has good adhesion, high abrasion resistance and good impact resistance. Subsequent welding is possible, although the welded area must be repainted after careful cleaning. Subsequent undercoats and top coats do not have to be applied immediately.

5.1.2 Epoxy undercoat

Epoxy undercoat comprises a medium containing a single-component epoxy resin ester with 54% solids.

The advantages of epoxy undercoating are:

- optimum corrosion resistance
- good resistance to chemicals, water and solvents
- good elasticity at high temperatures
- good adhesion.

5.1.3 Alkyd resin paint (top coat)

Alkyd resin paint comprises a medium of alkyd resin containing 56% solids. Its key features are good surface properties such as hardness, shine and scratch resistance.

Note:
The commonly used term "synthetic resin paint" does not describe the quality of a paint sufficiently well because there are a wide variety of different compositions and properties for such paints.

5.2 General features of single-component paints

Description		1st undercoat Steelwork	2nd undercoat Complete power unit with pipe work	Top coat Complete power unit (any desired RAL colour)
Product designation		Zinc-rich primer RAL 7000	Epoxy under coating RAL 7031 or RAL 6011	Alkyd resin paint RAL 6011
Chemical characteristics		Single-component polyurethane Zinc-rich primer	Epoxy resin ester (free from zinc chromate, lead and asbestos)	Alkyd resin base (lead-free)
Drying		Chemical under influence of air humidity	Air-drying	Air-drying
Paint	Solids % Specific gravity in kg/dm³	84 2,8	54 1,2	56 1,2
	Flashpoint in °C Danger class [1] Warning sign [1]	30 All	25 All Xn	24 All
Viscosity as-supplied mm²/s Working viscosity mm²/s		60 to 70 17 to 18	100 25	130 40
Thinning required % Pot life in h at 20 °C		10 6 to 8	15 —	15 —
Thinners	Flashpoint in °C Danger class [1] Warning sign [1]	Special thinners 43 All	Special thinners 24 All Xn	24 All Xn
Application		Brush or spray	Brush or spray	Brush or spray
Method		Spray gun, Airless	Spray gun, Airless, Esta	Spray gun, Airless
Theoretical consumption for 30 µm dry film in g/m² Coating thickness per application in µm Max. coat thickness in µm Spreading capacity in m²/kg with approx. 30% wastage allowance (coating thickness 40 µm)		200 bis 250 40 70 3	120 30 60 4	120 30 60 4
Dust-dry at 20 °C in h Transport-dry at 20 °C in h Re-coatable at 20 °C in h		0,25 8	0,5 1 8	4 16 6
Storage stability in months in original container at 5 to 40 °C		6	6	6
Max. temperature resistance of painting system in °C		-40 to 150	-40 to 150	-40 to 150
Surface preparation for painting		Sand-blasting SA 2,5 or free from rust, dust and grease Surface roughness ≥ 50 µm	Free from rust, dust and grease	Free from rust, dust and grease

Table 67: *Single-component paints*

[1] *Description of danger class and identification see Section 11*

6 Two-component paint systems

The advantages of two-component paints are their exceptional toughness, abrasion resistance, adhesion and resistance to chemical attack. More than one coat of paint must be applied to a surface, i.e. undercoat and top coat, in order to obtain the required properties of protection.

Remember, of course, that the sequence of application - undercoat first, top coat second - cannot be reversed. And no more undercoat may be applied to parts which have already received their top coat.

Top coating Hydraulic power units	Two-component polyurethane paint (any desired RAL colour)
2nd undercoat Hydraulic power units	Two-component epoxy undercoat RAL 7032
1st undercoat Hydraulic power units	Epoxy undercoat RAL 7031
1st undercoat Steelwork	Zinc-rich primer RAL 7000
Surface	Steel to normal grade of cleanliness SA 2 1/2

Table 68: *Painting procedures*

6.1 Paints

6.1.1 Two-component epoxy undercoat

Two-component epoxy undercoat is an epoxy resin paint with a total solids content of 67%.

The advantages of two-component epoxy undercoat are:

- universal application on almost any surface (including hot-galvanized parts)
- good filling power, good running qualities
- toughness, scratch resistance, excellent resistance to bubbling
- resistant to solvents
- exceptionally resistant to chemical attack.

6.1.2 Polyurethane paint

Polyurethane paint is of the reaction type on a polyurethane base with a solids content of 67%.

The drying process comprises a combination of solvent evaporation and chemical reaction.

Careful mixing before use will ensure the following properties:

- resistance to chemicals, water and solvents
- fire-resistance
- excellent hardness, abrasion resistance, filling and shine
- resistance to fire-resistant hydraulic fluids

Note:

Surfaces which have received a top coat of two-component paint can be painted again with two-component top coat within 14 days. After this time it will be necessary to roughen the surface first with a fine glasspaper.

Note:
The term "DD paint" stands for Desmodur and Desmophen (trade-marks of Bayer AG). The reaction between these two single components produces a polyurethane film.

6.2 General features of two-component paints

Description	**1st undercoat** Steelwork	**2nd undercoat** Complete power unit with pipework		**Top coat** Complete power unit (any desired RAL colour)	
Product designation	Zinc-rich primer RAL 7000	2 part Epoxy resin under coat RAL 7032		Polyurethane paint RAL 6011	
Chemical characteristics	Single-component polyurethane Zinc-rich primer	2 part epoxy resin with Polyamid hardener (free from zinc chromate, lead and asbestos)		2- part Polyurethane (lead-free)	
Drying	Chemical under influence of air humidity	Chemical reaction		Chemical reaction	
Paint		Undercoat	Hardener	Top coat	Hardener
Paint – Solids %	84	60	7	67	67
Paint – Specific gravity in kg/dm³	2,8	1,67	0,97	1,3	1,1
Paint – Flashpoint in °C	30	25	25	25	30
Paint – Danger class [1]	All	All	All	All	All
Paint – Warning sign [1]			✕ Xn		
Viscosity as-supplied mm²/s Working viscosity mm²/s	60 to 70 17 to 18	100 to 110 25		40 to 80 20 to 25	
Thinning required % Pot life in h at 20°C	10 6 to 8	5 to 10 12		aprox. 10 8	
Mixing ratio, base/hardener		87,5 : 12,5		100 : 40	
Thinners	Special thinners	Special thinners			
Thinners – Flashpoint in °C	43	25		24	
Thinners – Danger class [1]	All	All		All	
Thinners – Warning sign [1]		✕ Xn			
Application	Brush or spray	Brush or spray		Brush or spray	
Method	Spray gun, Airless	Spray gun, Airless, Esta		Spray gun, Airless, Esta	
Theoretical consumption for 30 µm dry film in g/m²	200 to 250	120		150	
Coating thickness per application in µm	40	40		35 to 40	
Max. coat thickness in µm	70	80		50	
Spreading capacity in m²/kg with aprox. 30 % wastage allowance (coating thickness 40 µm)	3	4		4	
Dust-dry at 20 °C in min		10		20	
Transport-dry at 20 °C in h	0,25	2		6 to 8	
Re-coatable at 20 °C in h	8	16		6	
Storage stability in months in original container at 5 to 40 °C	6	6		6	
Max. temperature resistance of painting system in °C	-40 to 150	-40 to 150		-40 to 150	
Surface preparation for painting	Sand-blasting SA 2,5 or free from rust, dust and grease Surface roughness ≥ 50 µm	Free from rust, dust and grease		Free from rust, dust and grease	

Table 69: *Two-component paints*

[1] *Description of danger class and identification see Section 11*

7 Applying paint

The manufacturer's instructions must be followed closely when applying paint.

In the case of steel parts where there are surfaces which will no longer be accessible after they have been joined by welding or bolting, the affected areas must be treated with zinc-rich primer first *(see Section 5.1.1)*.

The minimum coat thicknesses listed in *Tables 68 and 69* must be adhered to when applying the paints. However, the total thickness should not exceed 120 µm. Excessive thickness can cause surface tension which will have an adverse affect on the corrosion protection.

The following items must be left free of paint:

- nameplates and instruction plates
- plastics
- oil level sights and gauges
- piston rods and hoses.

All relevant regulations must be adhered to when using and applying paints.

8 Methods of painting

The application of paint by spray produces an excellent surface finish when the proper methods are used. However, there are many different methods of spraying, each having its own special properties.

8.1 Air spraying

Air spraying is excellent for large, flat surfaces. However, due to the high spray loss, it is less suitable for parts with a small specific surface area.

Paint wastage when spraying small suspended items is usually very high.

It is also difficult to spray inside voids and channels by this method *(Fig. 227)*. The large amount of compressed air in the paint spray causes an air cushion to form which makes the deposition of the paint difficult or impossible. There are, nevertheless, special nozzles and extensions which make it possible to paint inside voids by this method.

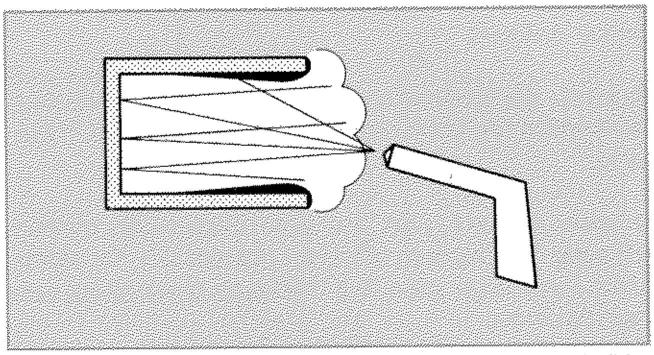


Fig. 227: *Air spraying inside voids causes a pressure build-up which, in extreme cases, can completely stop the paint being deposited*

8.2 Airless spraying

Airless spraying avoids some of the difficulties of air spraying. In this case the paint is expelled under high pressure and does not need compressed air to carry it. Its principal advantages are a high throughput (making it suitable for painting large surface areas) and practically no pressure build-up when painting inside voids and channels. Generally speaking, the paint is not atomized so finely as with the air spraying method, which makes control easier.

8.2.1 Hot airless spraying

The advantages of hot airless spraying are that it permits the use of high-viscosity, low-solvent paints. The paint is heated to about 55 to 70°C either directly in the can or in a heat exchanger. The elevated temperature reduces the viscosity of the paint.

The advantages of hot airless spraying are:

- economy due to the saving of thinners (cold spraying needs about 5 to 15% by weight of thinners
- faster drying
- thicker application if necessary
- less pollution.

8.3 Electrostatic spraying

The generation of an electric field between the spray gun and the item being painted causes the particles of paint to be attracted to the item so that the overspray loss is greatly reduced.

Naturally, electrostatic spraying is primarily intended for items which are good conductors of electricity, i.e. metal parts.

In terms of paint yield, the pure electrostatic method achieves the highest efficiency, particularly for items which have large gaps, e.g. hydraulic power units with their pipework.

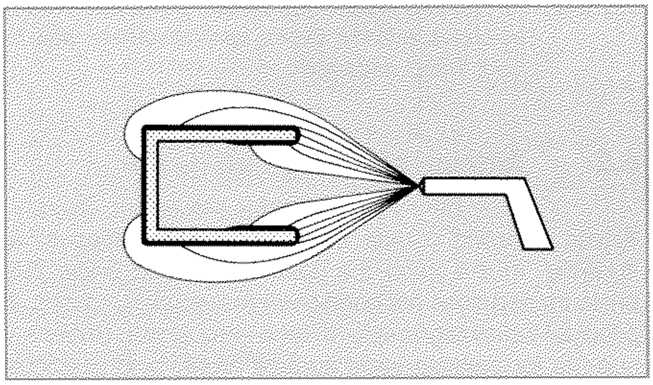


Fig. 228: *Electrostatic spraying gives good coverage, even on areas facing away from the spray gun. On the other hand, the Faraday cage effect makes painting the inside of voids difficult because the lines of force cannot penetrate inside the hollow section*

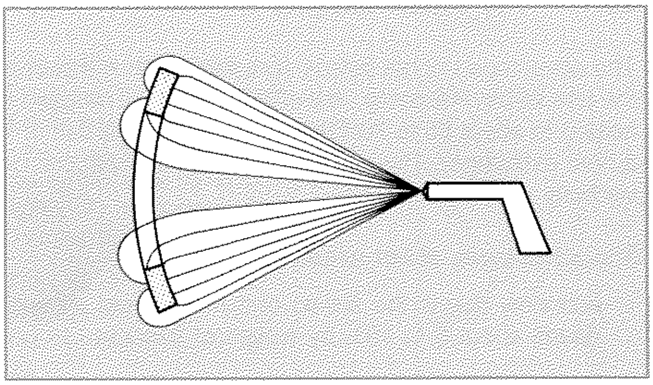


Fig. 229: *Electrostatic spraying produces a high paint yield and correspondingly little overspray*

9 Paintwork for hydraulic power units and equipment for hydraulic engineering

Customers in the hydraulic engineering field usually require corrosion protection for hydraulic power units and equipment to be in accordance with DIN 55 928, Part 5. DIN 55 928, Part 5 - Table 6 lists all the usual and well tried protection systems with different protection parameters. The different stresses due to atmospheric, chemical and mechanical effects are also listed.

Coat thicknesses are recommended to DIN 55 928, Part 5. Structural viscosity adjustments make it possible to apply paints thickly.

The customer specifies the corrosion protection according to the corrosion stresses involved.

a Light corrosion attack, e.g. installation in enclosed spaces, use protection parameters 6.11.1 to 6.11.5.

b Medium corrosion attack, e.g. installation outdoors, use protection parameters 6.30.1 to 6.30.3.

c Heavy corrosion attack, e.g. cylinders outdoors, use protection parameter 6.31.1.

Example of protection system designation to DIN 55 928, Part 5, Table 6, protection parameter 6.30.2 "Corrosion protection DIN 55 928 - T 05 - 6.30.2"

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24
		Type of painting							Applicable for														
		Undercoat[1] UC		Top coat[1] TC		Total system[1] UC + TC			Outdoor steelwork (conditions to DIN 55 928, Part 1)									Indoors e.g. workshops; accessible / inaccessible; Areas of components subject to similar internal stress					
Protection parameter	Binding element	No.	Total req. coat thickness µm	No.	Total req. coat thickness µm	Total req. coat thickness µm	Surface preparation to DIN 55 928 Part 4[2]	Usual medium for production painting Parameter	Agricultural atmosphere L	Town atmosphere S	Industrial atmosphere I	Marine atmosphere M	Chemical attack Ch	Road salt	Road sand and gravel	Mech. stress	Mech. stress, flying gravel	accessible LS	accessible I	accessible Ch	inaccessible LS	inaccessible Ch	Remarks
6-10.1	Oil, oil blends	1	40	1	40	80			—	—	—	—	—	—	—	—	—	x	—	—	0	—	
6-10.2		1	40	2	80	120			x	0	—	—	—	—	—	—	—	x	—	—	x	—	
6-10.3		2	80	2	80	160	Sa 2 1/2		x	x	0	—	—	—	—	—	—	•	—	—	•	—	
6-10.4		2	80	3	120	200			•	x	x	—	—	—	—	—	—		—	—		—	
6-11.1	Alkyd resin;	1	40	1	40	80	Fl[3]		—	—	—	—	—	—	—	—	—	x	—	—	0	—	Phthalic resin is an alkyd resin
6-11.2	Alkyd resin blends;	1	40	2	80	120		11, 24, 30	x	0	—	—	—	—	—	—	—	x	—	—	x	—	
6-11.3	epoxy resin ester	2	80	2	80	160	Sa 2[4]		x	x	0	—	—	—	—	—	—	•	—	—	•	—	
6-11.4		2	80	3	120	200			•	x	x	x[6]	—	x[6]	—	—	—			—		—	
6-11.5		2[5]	160	1	80	240			•	x	x	x	—	x[8]	—	—	—			—		—	DICK
6-12.1	Bitumen-oil blend	1	40	3	210	250	Fl[4]; Sa 2[8]		—	•	x	x	0	x	0	—	—		x	0		0	Bitumen-oil blends only for TC, UC with anti-corrosion pigment, medium: oil, alkyd resin or blends
6-20.1	Chlorinated rubber,	2	70	2	70	140	Sa 2 1/2	24, 30, 52, 53	x	0	0	—	—	—	—	—	—	x	x	—	x	—	DICK
6-20.2	vinyl chloride,	1	80	1	80	160			x	0	0	—	—	—	—	—	—	x	x	—	x	—	DICK
6-20.3	copolymer	2[5]	160	1	80	240	Sa 2 1/2, Fl[7]		•	x	x	x	x	x	—	—	—			x		x	
6-21.1	Chlorinated rubber	2	70	2	70	140			x	0	0	—	—	—	—	—	—	x	x	—	x	—	
6-21.2	blend, vinyl chloride copolymer blend	1	80	1	80	160	Sa 2 1/2	11, 24, 30	x	0	0	—	—	—	—	—	—	x	x	—	x	—	DICK
6-21.3		2[5]	160	1	80	240	Sa 2 1/2, Fl[7]		•	x	x	x	x[6]	x[6]	—	—	—			x[8]		x[8]	DICK
6-30.1		2	100	2	100	200				x	x	0	—	—	—	—	—		x	0	x	0	
6-30.2	Epoxy resin; polyurethane	1	80	2	160	240		30		•	x	x	x	x	0	—	—			x		x	DICK
6-30.3		1	80	3	240	320					•	x	x	x	x	0	—			x		x	DICK
6-30.4	Epoxy resin, abrasion-res.	2		1		2000[10]	Sa 2 1/2[9]						x	x	x	x	—						Painting method in 3 coats of 300 µm each, sprinkled with quartz sands between coats
6-30.5	Epoxid resin, abrasion-res.	2	600	1	4000	4600[10]										•	x						UC with quartz sand brushed, TC applied by spatula
6-31.1	Tar epoxy resin, tar pitch epoxy resin, tar/tar pitch polyurethane			3	360	360	Sa 2 1/2	30			•	x	x	x	x	0	—			x		x	DICK
6-31.2	Epoxy resin, polyurethane	1	60																				UC with zinc
	Tar/tar pitch epoxy resin, tar/tar pitch polyurethane			2	240	300	Sa 2 1/2[9]	30	—	—	•	x	x	x	x	0	—					x	DICK
6-40.1	Bitumen, filled	1	40	3	240	280	Fl[6]	11, 24, 30	x	x	x	x	0	0	—	—	—	x	x	0	x	0	Only for TC, UC with anti-corrosion pigment, medium: alkyd resin or blends
6-41.1	Coal tar pitch, filled	1	40	3	240	280	Sa 2[6]		•	•	x	x	x	x	—	—	—	•	x	x	•	x	
6-50.1	Silicone resin	1	30	2	50	80	Sa 3					—	—	—	—	—	—			—		—	For high temperatures (400°C), UC with zinc and TC with aluminium pigment and/or mica
6-52.1	Zinc alkali silicate	1	80		[11]	80			x	x	0	x	—	x	x		—	x	0	—	x	—	Without TC, for high temperatures (400°C)
6-53.1	Zinc ethyl silicate	1	80		[12]	80			x	x	0	x	—	x	x		—	x	0	—	x	—	

DICK — Thick coat system
x — Recommended
• — For exceptionally long protection only
0 — With comparatively low stress
— — Not recommended
No code — Indeterminate

1) Systems with the same system for UC and TC are generally used. However, compatible combinations of UC and TC with different media can be used, also combinations of thin-coat and thick-coat systems. The governing factor is the total coat thickness allowed. The number of coats indicated can be applied most uniformly by the airless spraying method. If this is not available the number of coats should be increased.

2) The information in Column 8 gives cleanliness standards for specific painting systems. They are not intended as a basis for comparing cleanliness standards.

3) If the object is suitable for flame cleaning and the work is performed by trained staff; but not for UC with zinc.

4) Only with red lead in the UC; for overhaul work also St 3 and, indoors, St 2.

5) 1 UC + 2 TC is also possible

6) For stresses M, road salt and Ch: Sa 2 1/2 or Fl with additional flame phosphating

7) Fl only with additional flame phosphating

8) With alkaline stress TC with highly saponifying softener

9) In special cases Sa 3; see DIN 55 928, Part 4, January 1977 edition, Section 4.1

10) With quartz sand

11) Soaping-resistant TC possible

12) Also usual with TC

Table 70: *Examples of proven corrosion protection systems for steel structures (except thin-walled, load-bearing components, hydraulic engineering, marine engineering)*

10 Special customized painting

The considerable expense of purchase, storage, application and disposal of special paints must be taken into account when considering special customized painting.

There are a number of points to watch:

- paints must be free from carcinogenic substances such as chromate and cadmium
- paints classified as poisonous, e.g. containing lead, must not be used
- paints, thinners and hardeners of danger class A1 (e.g. nitro-cellulose paints) must not be used.

11 Danger classes and warning signs

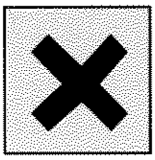
Dangerous substances must be marked in accordance with the appropriate regulations.

In the case of West Germany, typical warning signs are as follows:

poisonous

T

Slightly poisonous (injurious to health)



Xn

German danger classes are as follows:

Danger class A I: Liquids with a flashpoint below 21°C

Danger class A II: Liquids with a flashpoint between 21° and 55°C

Danger class A III: Liquids with a flashpoint between 55° and 100°C

12 The principal relevant standards

DIN 55 928

Corrosion protection for steel structures by painting and coating

Part 1: General

Part 2: Anti-corrosion design

Part 4: Surface preparation and testing

Part 5: Paints and protection systems

DIN 55 945

Paints, lacquers and similar coating materials
Terms

13 References

Jurgen Fichtner — Die bessere Lackierung kommt vom Reissbrett, Ingenieur Digest, March 1977

Karl-Albert van Oeteren — Zinkstaubanstrichstoffe und ihre Anwendung, Maschinenmarkt, Wurzburg 1970, No. 30

Packing and Transport

Erhard Wiesmann

1 General

The supplier is under an obligation to deliver his products to his customers in satisfactory condition in accordance with the customer's expectations and in compliance with the requirements of the contracts.

Hydraulic parts and equipment must be packed in such a way that they can arrive at their destination undamaged when despatched by normal means of transport. This means that they must be protected against moisture, rain, corrosion, shock, dust, dirt and any other damaging effects while they are in transport.

In order to ensure that the products reach the customer in the same condition as they leave the factory it is important for them to receive adequate protection against corrosion.

Equipment must be broken down into sections which can be packed so that their size and weight are appropriate for handling by the means available, such as fork-lift truck or crane.

The means of transportation and lifting must be appropriate to the weight of the packed goods, as must also the type of boxes, crates or pallets and any other devices associated with the lifting, transport and storage and the related stresses.

- Choose a method of packing which ensures that the goods reach their destination in good condition. The packing must be both economical and effective.
- Ensure that the customer's packing and shipping specifications are complied with.
- Finished equipment and components must be stored safely before shipment so that they cannot come to any harm or be confused with other equipment.
- Bright parts must be protected so that they do not suffer any corrosion before the system in which they are incorporated is put into service.

2 Corrosion protection for packed hydraulic equipment

During its journey from factory to site, especially overseas, hydraulic equipment is often exposed to adverse conditions for which the normal factory corrosion protection might not be adequate.

Some such conditions are:

- rain and/or seawater
- high humidity
- salt-laden air
- extreme high or low temperatures and temperature fluctuations
- chemical attack.

The nature and intensity of these conditions depend on the method of transport, its duration, any intermediate storage and the type and susceptibility of the equipment. The methods of corrosion protection described below provide the extra protection needed to nullify the adverse effects on the equipment which may be encountered.

2.1 Desiccant method

For complete effectiveness of corrosion protection and the desiccant it is necessary for the atmosphere inside the packing to be entirely isolated from the ambient atmosphere.

The equipment is sealed inside polyethylene film at least 0.2 mm thick or, if conditions are extremely bad, in a composite aluminium film (the composite film is 100% gas and water vapour-proof). Desiccant inside the sealed envelope keeps the air at a maximum relative humidity of 50%.

The quantity of dessicant to be used is laid down in DIN 55 474. The climatic conditions in the place of storage at the destination must also be taken into account.

Moisture indicators can be placed inside the envelope to monitor the amount of moisture absorbed by the desiccant.

In the case of envelopes made of the composite aluminium film there should be windows made of polyethylene or similar material welded in with the indicators placed behind them. If boxes are used for long-term storage a suitably large hole and cover should be provided in the side of the box through which the indicator inside the envelope can be examined.

2.2 VCI method

VCI stands for "volatile corrosion inhibitors" and is the name given to the method whereby bright metal parts are protected against corrosion by packaging them in an atmosphere saturated with anti-corrosion substances. The VCI method is preferred for protection during storage and shipment when the protective atmosphere can be effectively contained by this packaging.
Bright metal parts such as pistons, sleeves and the sealing surfaces of valves receive their VCI protection in the form of an anti-corrosion oil or by wrapping or packing in anti-corrosion paper.

The VCI anti-corrosion products continuously give off minute quantities of the inhibitor they contain or with which they have been treated. This means that anti-corrosion oil or paper does not actually have to be touching the metal in order to protect it; a small gap is perfectly satisfactory. Such protection is therefore ideal for irregular shapes such as tapped holes, pipes and machines.

The inhibitors prevent corrosion of metal surfaces by oxygen, water vapour and salt in the air and by wood acids, human sweat and industrial atmospheres. Also, any corrosion that has already started is stopped. The VCI method needs a sealed envelope to exclude the ambient atmosphere if it is to be totally effective.

VCI substances can be used for steel, iron, chromium, cast iron and aluminium, and there are also special substances for copper and its alloys.

However, the VCI method cannot be used for zinc, tin, cadmium, magnesium, lead or alloys of these metals.

3 Load dimensions, limits and regulations

The maximum size of loads for the means of transport to be used eventually for delivering the finished equipment should be taken into account at the initial design stage so that no insurmountable problems are encountered subsequently. Goods which do not exceed the dimensions and limits listed in *Tables 71 and 72* can be transported without special approval.

3.1 Load dimensions not requiring special approval

	Type	Load			Door		Floor	Max.
		Length	Width	Height	Width	Height	Height	Load
Rail	E: Open wagon, normal	12 500	2 760	2 000	1 800		1 235	21 t
Rail	G: Covered wagon, normal	9 000	2 700	2 100	2 000	2 000	1 245	21 t
Rail	KLM: Flat wagon, normal	12 500	2 700	2 000			1 250	23 t
Rail	RS: Flat bogie wagon	18 500	2 700	1 200			1 375	45 t
Road	Truck	6 500	2 400	2 650			1 350	8 t
Road	Trailer	8 000	2 400	2 650			1 350	14 t
Road	Articulated truck	12 500	2 400	2 350			1 650	25 t
Road	Semi-trailer	12 000	2 500	3 000			1 000	23 t
Road	Low loader	8 000	2 500	3 500			500	20 t

Table 71: *Load dimensions not requiring special approval - West Germany*

Note:
Low-loaders and articulated covered trucks are special-purpose vehicles.
The basic rule of the "undivided load" is applicable to the maximum load dimensions and payload limits. This means that the special-purpose vehicle may only carry one load. In the case of machines with accessories the weight of the accessories must not exceed 10% of the weight of the machine.

	Type	Length	Load Width	Height	Floor Height	Max. load
Road	Truck 6 500	2 400	2 650	1 350	*	
Road	Trailer	8 000	2 400	2 650	1 350	*
Road	Articulated truck	12 500	2 400	2 350	1 650	*
Road	Semi-trailer	12 000	2 500	3 000	1 000	*
Road	Low-loader	5 000	2 500	3 500	500	*
* = Max. loads vary according to country; they depend on permitted axle loads and wheelbases.						

Table 72: *Load dimensions exempt from special approval - Western Europe*

3.2 Load dimensions requiring special approval

Any road transport loads exceeding the widths listed in Tables 71 and 72 require the special approval of the authorities of the states and countries through which the load will pass. The route to be taken will be specified.

Major haulage companies operating at the heavy end of the market have usually been granted permanent special approval for carrying goods up to 3000 mm wide.

The height of road transport loads must not exceed the figures given in Tables 71 and 72. If necessary the equipment must be partially dismantled before loading so that, even including the packing, the height limit is not exceeded. Here too it is sensible to consult specialist haulage companies beforehand in order to investigate what is possible with the transportation of high loads.

In the case of rail transport the loading gauges of the different railway systems are the governing factor and the relevant authorities must be consulted if special sizes are involved.

3.3 Shipment of pressurized hydraulic accumulators

In the case of West Germany there are regulations governing the transport of dangerous substances and they specify that hydraulic accumulators charged with nitrogen can be transported either loose or installed in other equipment.

Shipment can be by road, rail, sea or air, although delivery by post is not currently allowed.

For all forms of transport a green danger sticker with the inscription "NON-FLAMMABLE COMPRESSED GAS" must be applied to the accumulators and the accompanying paperwork must also contain appropriate notes.

It is important for the transport arrangements to be such that the equipment is firmly anchored so that it cannot tip over or fall from the loading surface. Any hydraulic accumulators being delivered loose must be packed in cartons or crates.

3.4 Transport by sea

In order to remain within the maximum external dimensions of packing cases for transportation by sea (Fig. 230), the dimensions of the equipment packed must not exceed 2490 mm wide, 2225 mm high and 4000 mm long.

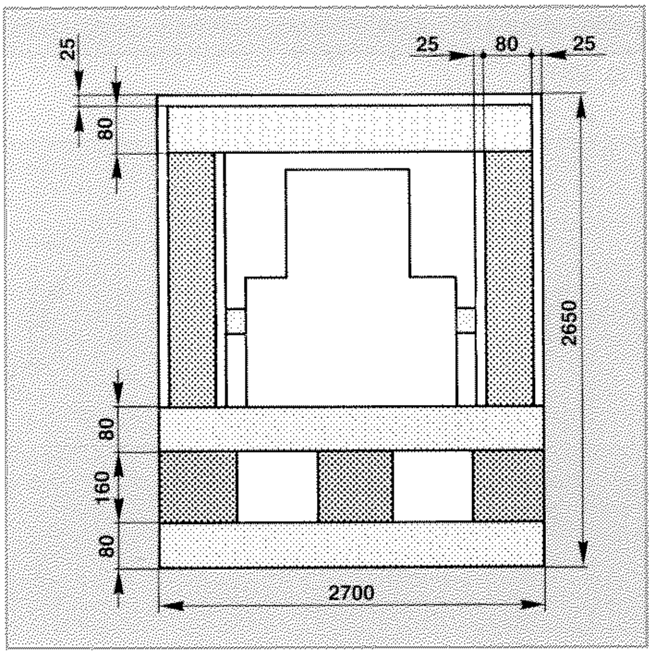
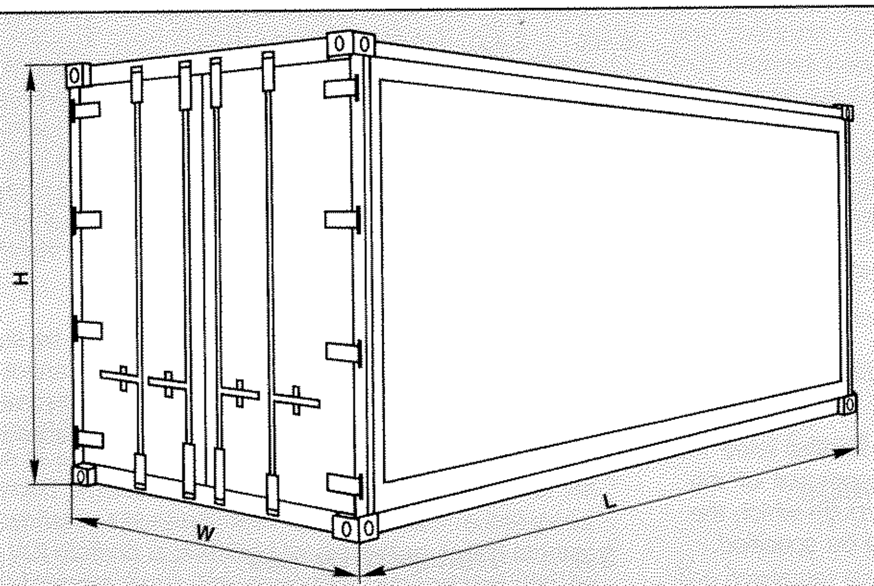


Fig. 230: *Heavy packing case for transport by sea*

3.5 Loading dimensions for sea containers



[1] Internal dimensions and weights can vary slightly between different makes and designs of container.

There are special containers available for unusual loads:
- 10'- and 30'-Container,
- Open Top Container,
- High Cube Container.

Standard container	External dimensions			Internal dimensions, approx. [1]			Door aperture		Weights, approx. [1]			Volume m^3
	L mm	*W* mm	*H* mm	Length mm	Width mm	Height mm	Width mm	Height mm	Max. cross weight kg	Tare weight kg	Max. pay load kg	
20' x 8' x 8'	6058	2438	2438	5900	2335	2258	2335	2145	20320	2000	18320	31.8
20' x 8' x 8.6'	6058	2438	2591	5900	2335	2395	2335	2292	20320	2200	18120	33.1
40' x 8' x 8.6'	12192	2438	2591	12011	2342	2407	2335	2292	30480	3800	26680	67.7

Fig. 231

4 Lifting points for hydraulic equipment

Hydraulic equipment must be designed so that it can be carried and lifted safely by industrial trucks such as fork-lift trucks and lifting gear such as cranes.

The lifting points for cranes are needed for handling within the factory, for loading and unloading for transport and for installation on site.

4.1 Positioning of lifting points

Lifting lugs must be attached in the direction of pull so that they cannot be bent or otherwise deformed by a pull at an angle.

Equipment to be transported horizontally, e.g. accumulator racks, must be fitted with lifting lugs which allow the equipment to be turned from the vertical to the horizontal.

4.2 Load limits for lifting lugs

In connection with the load limits for lifting lugs it must be remembered that the permitted limits decrease according to the angle of pull of the chain or strop.

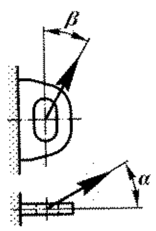
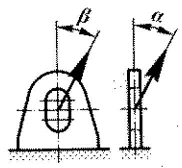
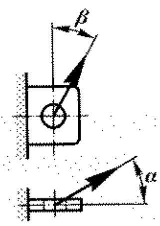
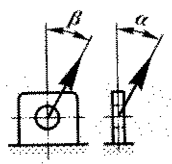
					
α in °	β in °	Max. load in t	Max. load in t	Max. load in t	Max. load in t
0	0	3.4	2.6	1.00	1.20
0	30 to 45	2.3	2.0	0.80	1.40
0	45 to 60	2.1	1.6	0.90	0.90
0	60 to 90	1.7	1.3	0.50	0.50
30 to 45	30 to 45	1.0	0.8	0.20	0.20
30 to 45	45 to 60	0.8	0.6	0.20	0.20
30 to 45	60 to 90	0.6	0.5	0.16	0.16
45 to 60	30 to 45	0.3	0.3	0.08	0.08
45 to 60	45 to 60	0.3	0.2	0.07	0.07
45 to 60	60 to 90	0.2	0.2	0.06	0.06
Type of lifting lug		Type: A	Type: B	Type: C	Type: C
Welded seam		4	4	3	3

Table 73: *Load limits for lifting lugs*

4.3 Dimensions of lifting lugs

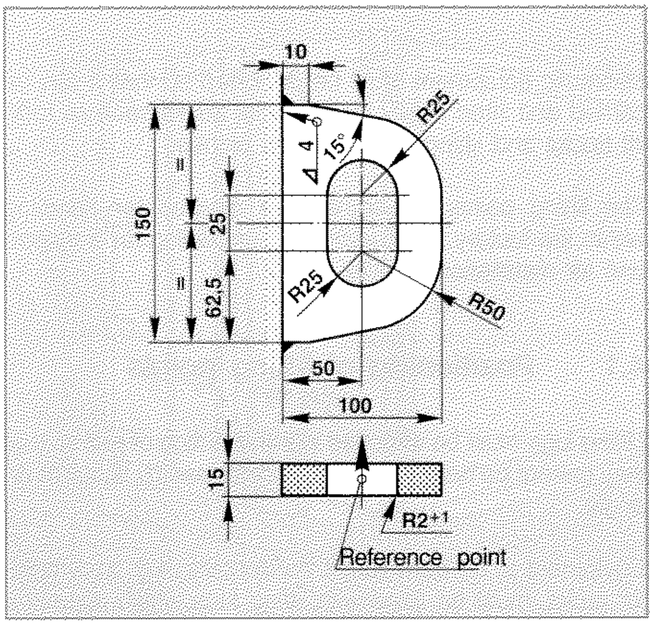


Fig. 232: *Lifting lug Type A for attaching container hooks to vertical surfaces*

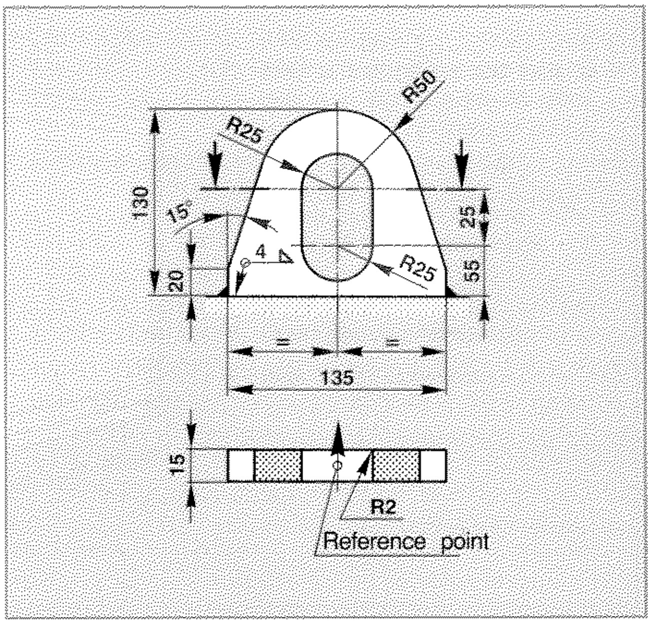


Fig. 234: *Lifting lug Type B for attaching container hooks to horizontal surfaces*

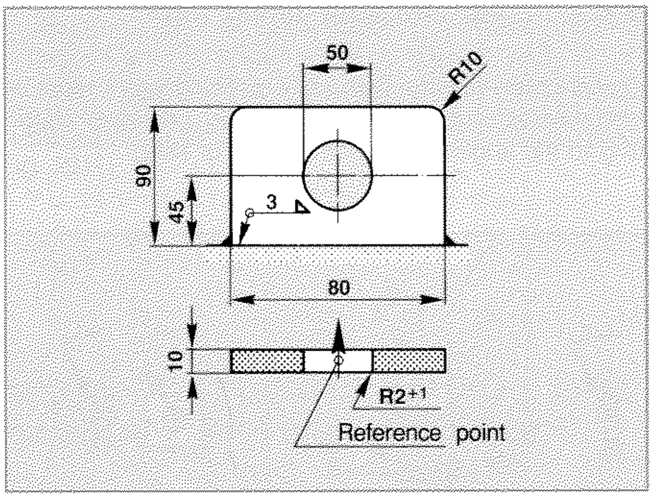


Fig. 233: *Lifting lug Type C for attaching hooks to vertical and horizontal surfaces*

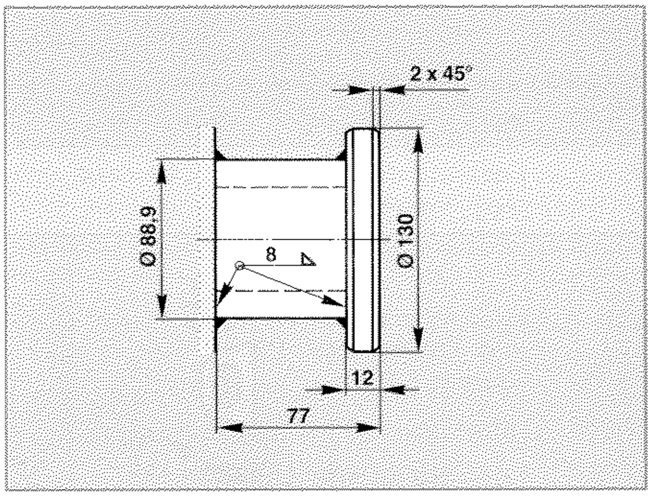


Fig. 235: *Post-type lifting lug for attaching strops*

5 Packing hydraulic equipment

The methods of packing hydraulic equipment described below vary according to the destination.

5.1 Delivery inside West Germany

Smaller items of equipment (up to 50 kg) are delivered in cardboard containers or cartons. The goods must be packaged in suitable materials inside the cartons so that they do not suffer any damage during transport. Any empty spaces must be filled with a suitable filling material. For gross weights over 50 kg skids must be attached to the bottom of the packing, or the packing must be mounted on a pallet, so that it can be lifted with a fork-lift truck. Corrosion protection of bright metal parts is by the VCI method *(see Section 2.2)*.

5.2 Delivery within the EEC

The same packing as described in Section 5.1 is appropriate.

Corrosion protection is by the VCI method.

5.3 Delivery to Eastern Block countries and overseas

Seaworthy packing, i.e. packing cases, is employed in the majority of cases. Corrosion protection is by the dessicant and VCI methods. Other details are as described in *Section 5.1.*

5.4 Delivery to the USSR

Seaworthy packing is employed in the majority of cases with corrosion protection by the dessicant and VCI methods using composite aluminium film.

Other details are as described in *Section 5.1.*

6 Packing hydraulic power units

The methods of packing hydraulic power units described below vary according to the destination.

Export packing must be designed for a total transport and storage period of from 6 to 24 months.

Exposed bright metal parts must be sprayed with a corrosion inhibitor.

6.1 Delivery within West Germany

Items of hydraulic power units such as tanks, valve tables, valve stands and pump sets are bolted to a wooden base and covered with polyethylene film for protection against water and dust.

Awkward items such as cylinders and accumulators are packed on pallets or skids depending on their size.

6.2 Delivery within the EEC and to Eastern Block countries

Hydraulic power units are shipped in wooden crates with the equipment sealed inside polyethylene film. Corrosion protection by the dessicant method can be provided if the customer wishes.

6.3 Delivery overseas in seaworthy packing

For transport by sea, hydraulic power units are loaded in wooden packing cases lined with waterproof bituminized paper (although the bottom of the case is not lined so that any water which gets in can drain away). Corrosion protection for periods up to 6 months is provided by the desiccant method.

6.4 Delivery to the USSR or transport periods over 6 months and storage times up to 24 months

Hydraulic power units are transported in wooden packing cases lined with waterproof bituminized paper (the bottom of the case is not lined).

Corrosion protection for the equipment is provided by the dessicant method using composite aluminium film.

Indicators for monitoring the condition of the desiccant can be placed inside the envelope if the customer wishes *(see Section 2.1)*.

Protection against internal corrosion with a suitable oil is necessary to maintain the equipment in good condition during long periods of transport and storage (see *"Internal Protection of Hydraulic Components and Power Units"*).

7 Identification and marking

7.1 Identification of individual components and spare parts

All individual components, including loose items in a packed unit, must be identified separately by the manufacturer/supplier. The information must also be given on the packing note/delivery note.

7.2 Marking of packed goods

Each item of packed goods must be marked either with a stencil using seawater-resistant, non-fading paint or with labels provided by the customer. In the case of stencilling the size of the lettering should be appropriate to the size of the unit. With unpacked items of equipment and those mounted on skids the marking should be directly on the equipment itself.

Basically, all units should be marked on both long sides.

7.3 Handling symbols/Danger symbols

The following international symbols should be applied to identify any goods requiring special handling.

Danger symbols should be taken from the IMDG Code (Fig. 236). The symbols should be marked on the sides containing the identification marks. Centre-of-gravity symbols should be marked on both sides and ends.

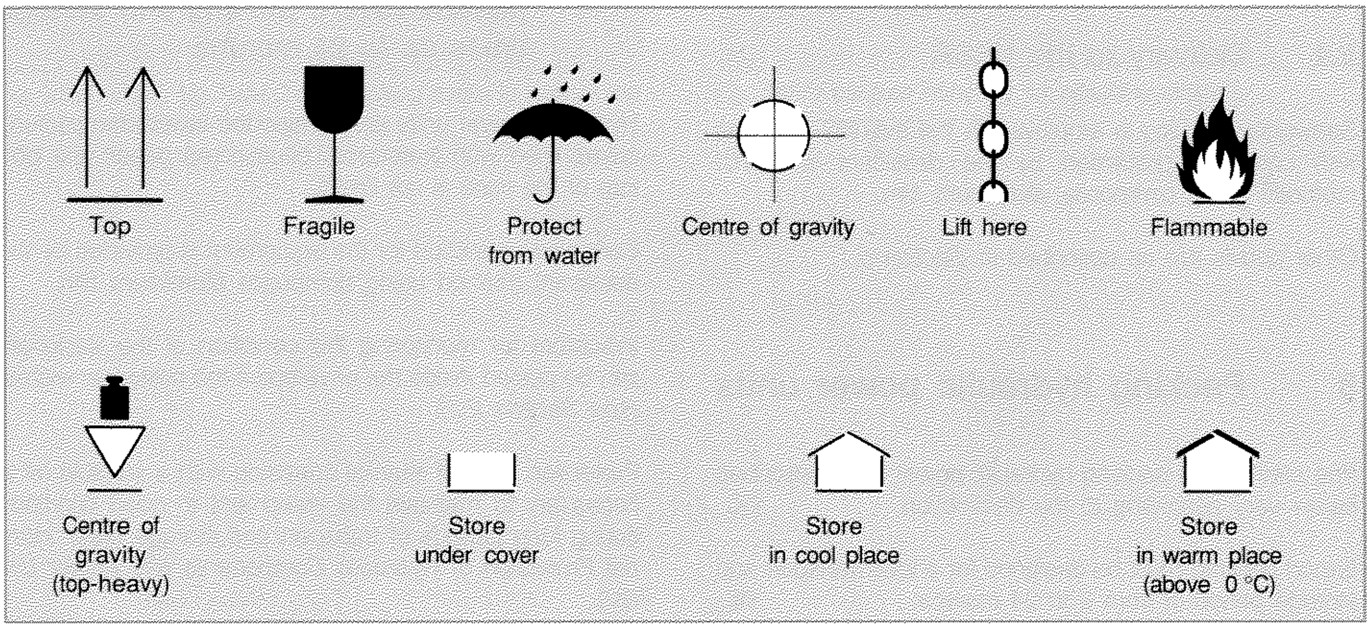


Fig. 236: *Handling and danger symbols*

8 References

Company Publication: Verpackungs- und Versandvorschriften für Hydac-Blasen- und Membranspeicher, die mit Vorfülldruck (Stickstoff) geliefert werden.
HYDAC GmbH, Sulzbach

Company Publication: Allgemeine Verpackungsbedingungen für Maschinen- und Anlagenbauteile
Mannesmann Anlagenbau AG, Theodorstraße 90, D-4000 Düsseldorf 30

Commissioning Hydraulic Installations

Franz X. Feicht

1 Preparations for commissioning

1.1 Checking the fluid tank

Oil tanks in hydraulic installations are checked for cleanliness before they leave the factory and all apertures are tightly sealed.

Due to unfavourable circumstances during transportation from the factory to the place of installation and any intermediate storage that might become necessary, it is possible for oil tanks to become contaminated with water, dust, etc. before it is time to commission the installation. If extended intermediate storage (i.e. for more than six months) is expected from the outset, proper corrosion protection may make it necessary to seal the return lines and suction lines inside the oil tank in order to prevent drainage.

During the process of installation on site it often happens that the connecting points for return lines remain open for long periods of time.

For all these reasons, therefore, it is essential for the fluid tank of an installation to be inspected and cleaned if necessary before it is filled with fluid (do not use linting cleaning cloths).

1.2 Checking the pipework between hydraulic installation and actuators

If the commissioning is not carried out by the same people who install the pipework between hydraulic installation, control gear and actuators, it is essential for the pipework to be inspected first, either a full or random inspection. Satisfactory operation of the system and long service life for the components are very closely related to the internal cleanliness of the pipework.

The correct connection of the individual actuators to the hydraulic installation according to the circuit diagram should also be checked because any mistakes mean wasted fluid and unnecessary "messing about" afterwards.

1.3 Aligning the pump and electric motor

The alignment between the pump and electric motor can sometimes become disturbed during transport and the fitting of other components, etc.

Some types of pump canot tolerate either radial or axial loads and, in addition, flexible elements in couplings will only permit a small amount of parallel or angular offset.

This point should never be forgotten, even when everyone is urging the commissioning to be started immediately.

1.4 Gas charging of any accumulators

Only nitrogen should ever be used for charging hydraulic accumulators.

The accumulators should be charged to the value of charging pressure indicated on the circuit diagram (the fluid side of the system must be depressurized when this is done). So that the nitrogen volume of the gas bottles can be properly utilized, there are so-called "charging units" which enable the nitrogen bottle pressure to be suitably "pumped up".

In the case of piston-type accumulators connected to gas bottles it is sometimes possible to use the hydraulic system to "pump up" the nitrogen pressure.

Remember, however, that this procedure is very dangerous with bladder-type accumulators because of the risk of over-straining the bladder.

1.5 Filling with fluid

Regardless of the type of container in which the fluid is delivered it will not be clean enough for immediate use so must be passed through a filter before entering the system. The absolute filtration rating of the filling filter must be at least identical to that of the filters in the system.

1.6 The commissioning team

For reasons of safety, only persons directly involved with the commissioning should be allowed on site while the commissioning is taking place.

It is easy to make this statement but in reality there are probably several other companies seeking to complete their work urgently at the same time.

Remember, nevertheless, that after a serious accident it will be no good saying "They should have known better".

2 Commissioning

2.1 Pressure settings

Release the pressure settings of all pressure relief valves, pressure reducing valves and pump pressure regulators except those valves which have a fixed TÜV (safety approval society) setting.

2.2 Servo systems

In the case of servo systems remove the servo valves and replace them with flushing plates or, even better, by directional control valves of the same size. Actuators must be short-circuited. Fluid temperatures attained throughout the hydraulic system during flushing should be at least equal to the normal service temperature attained during subsequent operation. Filter elements must be changed as necessary.

The flushing time can be calculated from the following formula

$$t = \frac{V}{Q} \times 5 \quad [h]$$

V = Tank capacity in L
Q = Delivery rate of high pressure pump in L/min

2.3 Commissioning of pumps

Some types of pump need to be filled with fluid before being started up for the first time in order to prevent bearings and other mechanisms from running dry.

Start and stop the pump drive motor briefly in order to check the direction of rotation. If it is correct, the pump can be run continuously and the system checked for leaks and correct flow.

Provided no defects are discovered, the setting of the pressure relief valve can be slowly increased to the value given on the circuit diagram and locked.

Filling pumps and boost pumps can be comissioned in the same way. At the same time, the system must also be bled.

The next stop is to commission any pilot fluid pumps. If no defects are discovered in this circuit, the control fluid pressure can be set to the prescribed value and locked. The pilot circuit must also be bled.

The next step is to commission the main pumps one by one.

Once again:

Bleed both the inlet side and the system side and check the direction of rotation on the first brief start. While the various pumps are being set to work a close watch must be kept on the level of fluid in the tank. It must be topped up if necessary until the whole system is full.

Each pump should be kept under constant watch until it is pumping smoothly and continuously. A pump which is sucking in the occasional air bubble or fluid containing large numbers of bubbles will give off loud knocking noises or a very loud continuous noise. Since very high rates of pressure rise are achieved in positive displacement pumps, the compression of the air bubbles causes severe localized overheating of the fluid and therefore damage.

Unless a pump starts drawing bubble-free fluid after a few minutes of running, it should be shut down quickly in order to ascertain the cause.

2.4 Commissioning control gear and actuators

Control gear and actuators should be put into service at low rates of flow and low pressure in order to avoid any damage due to incorrect circuitry, either electrical or hydraulic.

Once it has been established that the circuitry is correct, the actuators correctly controlled and the limit switches correctly positioned, the pressures and flow rates can be increased to the prescribed values.

Throughout the whole time a close watch must be kept on:

- tank fluid level
- tank fluid temperature
- all components for external leaks
- sources of noise
- pump and motor body temperatures
- clogging of filters.

2.5 Other settings

Set pressure relief valves to the prescribed value and lock.

Set pump pressure regulators and lock.

Set pressure reducing valves, pressure sequence valves and pressure unloading valves.

Adjust tank level monitor.

Set pressure switches and differential pressure switches.

Set temperature controllers.

Set switching points of temperature monitoring devices.

2.6 Other checks

Is the pipework properly secured to withstand fluctuations in pressure?

Are the fixing points positioned correctly?

Are hoses fitted so that they do not chafe when pressurized?

2.7 Commissioning of systems with proportional valves

This type of system should be commissioned as described previously.

The functions can be exercised by "emergency manual control". In order to avoid damage, proportional valves should first be operated using a portable electronic service unit. This type of auxiliary electronic equipment is available from equipment manufacturers as a service pack. Its use is not confined to commissioning, but is useful later as a service aid for checking over the proportional valves and their amplifier boards.

2.8 Commissioning systems with servo valves

The servo valves must be removed before starting commissioning. They should be replaced either by flushing plates or, if the system allows, by directional control valves of the same size. When flushing is complete, the servo valves can be replaced but the greatest cleanliness must be exercised. Once again the whole hydraulic control system should be operated manually first.

Battery-powered or mains-powered portable control units are also available for use with servo valves. Again, these are extremely convenient both for commissioning and for subsequent fault-finding.

2.9 Commissioning high-speed systems

Typical high-speed systems are those found on presses, plastics injection moulding machines, special-purpose machine tools, rolling mills, crane control systems, etc. and it is often impossible to commission them and optimize their operation with conventional tools and instruments such as pressure gauges, thermometers, etc.

For commissioning and optimizing these more complex systems it is necessary to have an appropriate amount of sophisticated instrumentation suitable for the particular application that is capable of multi-channel, simultaneous recording of a range of parameters, such as pressures, electrical signals, strokes, speeds, flow rates, etc., in order to determine such factors as dead times, peak pressures, pressure dips, motion overlaps, etc. so that the safety aspect can be clearly analyzed and documented for the eventual end-user.

3 The most common errors in commissioning

Apart from maintenance, commissioning is possibly the single most important process affecting the service life and correct function of a hydraulic installation.

Obviously, therefore, it is essential to eliminate as many errors as possible from the commissioning process.

The most common errors are:

- forgetting to check the tank fluid level
- forgetting to filter the fluid put into the system
- forgetting to check over the installation before starting commissioning (involving subsequent modifications and waste of fluid)
- forgetting to bleed parts of the system
- setting pressure relief valves too close to the working pressure
- setting pump pressure regulators higher than or the same as the pressure relief valve
- not flushing servo systems for the correct length of time
- not taking any notice of abnormal pump noises (cavitation, suction line leaks, excessive air in the fluid)
- neglecting lateral strain on cylinder rods (i.e. installation errors)
- not bleeding cylinders (leading to seal damage)
- setting limit switches too finely
- setting pressure switches without allowance for switching hysteresis
- not filling hydraulic pump and motor bodies with fluid before the first start
- failing to keep a record of settings
- not locking or sealing setting devices
- allowing too many people around the installation during commissioning.

4 Summary

Depending on the size and complexity of an installation, the commissioning can be carried out either by those who will be responsible for its subsequent operation (provided they have sufficient hydraulic knowledge or have been properly trained) or by manufacturers' personnel using the appropriate instrumentation and tools.

Past experience has shown quite clearly that customers often save nothing in the long run when attempting to cut costs by not employing a commissioning specialist from the manufacturer.

Maintaining Hydraulic Installations

Franz X. Feicht

1 Introduction

In accordance with DIN 31051, the generic term "maintenance" can be broken down into the following three activities:

1.1 Servicing

Action taken to maintain the status quo, i.e. ensuring that wear margins and reserves are used up at as slow a rate as possible during the useful service life of the equipment.

1.2 Inspection

Action to ascertain the actual status quo, i.e. determining how and why wear margins and reserves are being used up.

1.3 Repair

Action taken to restore the status quo, i.e. to re-establish wear margins and reserves and restore loss of performance.

With hydraulic installations the term "loss of wear margins and reserves" means:

- increased clearance between spools and bores
- worn dynamic sealing elements
- erosion of control lands
- fatigue of rolling bearing materials
- excessive clearance between plain bearings and shafts
- cavitation damage to pumps and valves
- chemical changes to fluids.

Each one of these "forms of wear" slowly uses up the reserves provided at the design stage until the point is reached where either a component fails abruptly or prescribed reference values can no longer be attained. The latter is not always identical to the failure of a piece of equipment.

2 Maintaining hydraulic installations

The enormous variety of applications for hydraulics means that the type of hydraulic equipment encountered ranges widely from the simplest unit incorporating perhaps one pump supplying one actuator to major complex installations incorporating several pumps and highly sophisticated control gear.

The maintenance needed by these installations must be planned and executed according to such factors as the required availability, the value of the installation, its usage (intermittent or multi-shift continuous) and the consequences of a breakdown (an individual unit performing some subsidiary task or an important sub-system which would result in the shutdown of a whole line should it fail).

2.1 Inspection

The individual points of a particular installation requiring inspection should be put down in "inspection lists" so that it is possible for them to be attended to with adequate thoroughness by persons of different levels of skill.

In the case of large installations there will be different inspection points according to different periods of service, e.g. daily inspection points, monthly inspection points and inspections shortly before an extended shutdown, e.g. holidays.

The principal inspection points are as follows:

2.1.1 Check tank fluid level

A low fluid level is usually indicative of a loss of fluid through external leaks. After major repair work the fluid level can sometimes continue falling slowly for some time as the system automatically bleeds itself of air.

A high fluid level can be indicative of parts of the system at a higher level draining while shut down due to the ingress of air. Water can gain access to the hydraulic system through leaks in water/fluid heat exchangers.

2.1.2 Check operation of heat exchangers

Air/oil type

In places heavily contaminated with dust it is very easy for the heat-dissipating surface of the heat exchanger to deteriorate very quickly due to a build-up of dust. Should the flow of cooling air be carrying fine droplets of fluid from a possible external leak the dissipation of heat will be stopped almost entirely very quickly indeed.

Water/oil type

The water circulating through the heat exchanger must be clean so that no insulating deposits of sludge can interfere with the transfer of heat. This is particularly important if the flow of water is regulated in order to maintain a constant fluid temperature and so there is no high water velocity to flush away any sludge that might build up. It is important for neither the water side nor the fluid side of water-type heat exchangers to be allowed to drain otherwise only part of the surface area will be utilized for heat transfer.

2.1.3 Check for external leaks (visual examination)

Examine all rigid pipes, flexible hoses (particularly the connectors), pumps, control gear, hydraulic motors and cylinders.

2.1.4 Check fluid service temperature

A rise in the fluid temperature can be due to a number of causes:

An inefficient heat exchanger (dirty surface, broken fan, insufficient circulating water, high circulating water temperature, sludge build-up in heat exchanger, etc.).

Heat being generated in hydraulic pumps or motors due to damage to either rolling-bearings or plain-bearings.

Heat radiation from the fluid tank, the piping and the hydraulic components reduced by a build-up of dirt.

Increased internal leakage in individual components, pressure relief valves lifting at too low a pressure, failure of load sensing control, system being operated outside the permitted limits, etc.

2.1.5 Check pressure settings

Check the operating pressures of primary pressure relief valves, secondary pressure relief valves and control pressure relief valves; check gas pressure of hydraulic accumulators and pressure setting of pressure reducing valves, pressure sequence valves and unloading valves.

2.1.6 Check leakage rates

With hydraulic motors and some hydraulic pumps it is possible to assess the internal wear by measuring the leakage fluid rate. This also applies to a number of control valves and check valves. The slow extension or retraction of a cylinder under external load with its isolating devices closed can be indicative of defective piston seals.

2.1.7 Check the fluid purity

A visual examination will only provide a rough estimate of the condition of the fluid, i.e. cloudiness, darker appearance than when new, sediment in the fluid tank.

There are three methods of measuring the purity accurately:

- Gravimetric determination of solids by the filtering of a certain quantity of fluid (e.g. 100 ml) and weighing of the filter paper before and after filtration. This will give a figure of solids content in mg/l. However, the actual solids content will be somewhat higher than the figure recorded because even very fine filter papers have an average pore size of about 0.8 µm.

  Although very fine contamination of this size will not actually cause any wear at normal mechanical fits, it can contribute to erosion in places where the fluid velocity is very high (e.g. in pressure relief valves, pressure reducing valves, grooves in control valves, etc.).

  The gravimetric method also says nothing about the composition of the solids nor anything about the size distribution. Gravimetric analysis can only be performed in the laboratory or in a laboratory van specially equipped for fluid analysis.

- Particle counting by electronic counting and sorting devices.

  Such devices can function completely automatically. However, fluids containing water, mixtures of fluids and fluids with an excessive solids content cannot be analyzed by these devices. As they are very costly, they are normally only found in the laboratories of large companies, filter manufacturers, fluid suppliers and, sometimes, hydraulic component manufacturers. With suitable calibration, the results can be compared directly with the contamination classes of MAS 1638 or SAE standards.
- Microscopic analysis

  In this case a few precisely-measured drops of fluid are placed on a highly absorbent surface such as a filter paper and the solids remaining on the surface after the fluid has been absorbed are counted under the microscope. This method also allows an assessment of the size distribution of the solids to be made and a rough estimate of the composition, e.g. metal particles, seal residue, silicate, fibres, etc. It is a method which allows a quick analysis to be made on site, where the other methods described can only be performed in the laboratory or in specially equipped vans.

A very important factor with all of the methods described above for determining solids content is "how" and "where" the fluid samples are taken from the system. The best results are obtained from samples taken from a system that is operating normally.

Sampling from the pressure line as close as possible to the pump has proved to be very effective. The pressure line usually contains pressure measuring points which can also be used for taking samples. The closer the sampling point to the pump delivery, the greater is the pulsation which obscures the difference between lamina flow and turbulent flow (ISO 4021 defines sampling precisely).

If the sample is taken through a tube or hose, a large quantity of fluid (at least 2 litres) must be passed through it first at as high a velocity as possible in order to make certain that there are no solids left in the hose or fittings which could falsify the sample. Taking a static sample from the fluid tank - after the system has been shut down for as long as possible - will give a result that bears no relation to the solids content of the fluid actually flowing through the system. However, since this method is very simple it is still very popular. The procedure should at least comply with the instructions in Cetop RP 95 H, Section 3.

2.1.8 Check for filter clogging

A visual check of the type of filter elements in widespread use today will reveal nothing.

The amount of clogging can only be determined by measuring the pressure drop across the filter element (or by measuring the pressure head before the element - with return line filters if there is no fluid resistance left on the clean side). So that such checks can be performed without the need for instruments, the filters used in the system should always incorporate a visual clogging indicator or be equipped for continuous electric monitoring of clogging. Solids in the fluid are the main cause of wear in hydraulic components. Between 75 and 80% of all equipment received from customers for general overhaul has failed due to wear and erosion caused by an excessive solids content in the fluid. About 10% of equipment shows some cavitation damage. When using HFA, HFC and HFD fluids the proportion with cavitation damage rises to about 15 to 20%.

The amount of solids smaller than the pore size of the filters used in the system is also a significant factor because they promote erosion and, through a kind of lapping, increase spool clearances, wear away control surfaces and ruin dynamic sealing elements. Sudden equipment failure is a rare occurrence but the actual service life of the component is reduced to a fraction of the design value.

2.1.9 Check the chemical properties of the fluid

It is inevitable that the chemical stability of the hydraulic fluid will be affected to some extent in time by the severe stress placed upon it by such things as pressure fluctuations, high flow velocities, high shear loads, localized overheating, absorption of oxygen from the air, contact with various metals, elastomers and plastics and the absorption of condensation and solid particles.

Consequently, regular tests should be made on the neutralization number, saponification number (soaping number), oxidation products content, viscosity and viscosity index.

The tests are more important for HFA, HFC and HFD fluids than for mineral-oil fluids and so should be performed more frequently because it is constantly being proved that major damage can be caused by chemical changes which take place very rapidly although outwardly everything appears satisfactory and in order.

If the operator of the system does not have his own laboratory, an agreement can usually be made with the fluid supplier to perform the tests on a regular basis.

When there is only a relatively small amount of fluid involved it is probably cheaper and simpler to change the fluid periodically in order to avoid the on-going expense of chemical tests.

2.1.10 Check bearing temperatures

When the first pitting appears in the races of rolling-contact bearings, the extra power loss causes a small rise in temperature in the vicinity of the bearing. However, in order to detect such a change it is essential to have a reference value which must have been taken at precisely the same point during a specific work cycle and after the bearing has been run in.

2.1.11 Check for noises

In this case too the object is to detect changes from the status quo when the equipment was new.

Pressure relief valves normally make a hissing noise when they lift. Therefore, if there is any chattering or whistling it is a safe bet that there are some damaged pressure relief valves or pressure reducing valves in the system.

Cylinders may make chattering or groaning noises when they extend or retract which can be an indication of worn guides, excessive strain (e.g. rusted pivots), unsuitable fluid, etc.

An increase in hydraulic pump or motor noise with increasing pressure can be indicative of erosion or cavitation damage to control surfaces, excessive clearance in displacement elements or incipient bearing damage. Unpleasantly loud noises from pumps, regardless of the pressure but gaining greatly in intensity as the speed rises, are a signal for insufficient boost pressure or excessive negative pressure in the suction line.

2.1.12 Check power and speed

It is possible to assess the overall condition of an installation by checking the extension and retraction times of cylinders against the guaranteed values and by measuring the output speeds of motors and power consumption of pumps as appropriate.

2.1.13 Check pipes and hoses

Examine all pipes, connectors and fittings for leaks and ensure that the pipes are firmly supported everywhere. Loose pipes can eventually chafe through and fittings can be overstrained.

Kinks and crush points increase the flow resistance causing loss of power and overheating. Flexible hoses should be examined for chafing and bulges and their correct installation should be double-checked. All the hose fittings should be inspected for leaks.

2.1.14 Check the hydraulic accumulators

The gas charging pressure should be checked at regular intervals with the hydraulic system depressurized and always at roughly the same ambient temperature. This applies to bladder-type, piston-type and diaphragm-type accumulators. Weighted accumulators and spring accumulators, encountered only rarely nowadays, need only be checked for leaks on the fluid side.

2.2 Servicing

In actual practice the divisions between inspection, servicing and repair are not so strict as the definitions would suggest. It is common, for example, for servicing to be carried out at the same time as inspection.

The main servicing work involves:

2.2.1 Topping up the fluid

The fluid with which the system is topped up should always be the same as that already in the system. This is a particularly important point to note in the case of mineral-oil fluids from different manufacturers because, although they may comply with DIN 51 524, Part I or Part II, they may differ in base oil and additives.

In the case of HFA, HFC and HFD fluids it is very important not to mix different makes of fluid. Even when fluids of the same type, but from different suppliers, are mixed there is a possibility of liability for any damage being refused.

Numerous additives for hydraulic fluids are now on the market which are supposed to reduce mechanical friction, eliminate the stick-slip effect almost entirely, extend the life of the fluid and perform many other useful functions.

Before contemplating the use of such additives it is absolutely essential to obtain the approval of the fluid supplier in order to be certain that the two are compatible. However, it must be said that hydraulic fluid suppliers very often refuse all liability if additives not of their own make are mixed with their hydraulic fluid. Hydraulic equipment manufacturers adopt a similar attitude because it is enormously costly and time-consuming to ascertain the long-term effects and true compatibility when there are fluctuations in the mix ratios and applications vary widely.

2.2.2 Changing the fluid

Hydraulic fluid should be replaced as soon as any changes in its chemical properties start to take place, e.g. the presence of oxidation products, increase in neutralization number or saponification number, disappearance of essential additives, change in viscosity, etc.

The fluid should also be changed when the fine contamination (i.e. the content of solids smaller than the pore size of the system filters) has increased to a point where greater long-term wear must be anticipated (e.g. over 250,000 particles between 5 and 15 μm per 100 ml). Expensive purification processes such as centrifuging (if the type of fluid will allow) or filter presses are only economical if the quantity of fluid is large. Since the amount of fluid in the actual system, i.e. pumps, pipes, control gear and cylinders, is sometimes several times more than the capacity of the tank it is not enough to change just the amount of fluid in the tank. This is especially true if the old fluid is already showing signs of chemical instability.

The oil tank must be cleaned out every time the fluid is changed.

An important point to remember, both when topping up and changing the whole quantity of fluid, is that new fluid as-supplied is not normally immediately suitable for use in a hydraulic system due to the quantity of solids which it contains. The chain of supply from manufacturer to end-user is often very long, passing through storage tanks, sea and road -going tankers, tank cars and various types of container, and the required purity cannot be guaranteed.

Therefore, for topping-up and for complete renewal the new fluid must be passed through a suitable filter first. Its pore size must be at least equal to that of the filters fitted in the system but it is better for it to be smaller.

This is a very important point requiring close attention otherwise there is a possibility of malfunctions immediately after the fluid has been changed.

2.2.3 Cleaning the filters

Filter elements should also be cleaned or changed every time the fluid is changed.

With the types of filter fabrics in widespread use today cleaning is not a practical proposition so the elements are changed. The amount of clogging of a filter element can only be discovered from the pressure drop across it since nothing can be seen with the naked eye. Without artificial aid, the human eye is only able to see particles larger than about 45 to 50 μm in size. This means that all filters used now should be the type incorporating a visual or electric clogging indicator showing when maximum clogging has been reached. Filters without such a monitoring device must be changed at regular intervals which should be short enough to ensure that the by-pass valve does not lift or the clogged element burst.

2.2.4 Readjusting pressures

Readjust all pressure settings - pressure relief valves in the working circuit, control circuit and low-pressure circuit, also readjust pressure reducing valves, pressure sequence valves and unloading valves.

2.2.5 Dealing with leaks in the pipework

Resealing work on the pipework should only be performed with the system depressurized. Leaks from fittings incorporating soft seals such as O-rings cannot be eliminated by re-tightening because the seals will either have broken or gone hard. A new seal will have to be fitted in order to effect an improvement.

2.2.6 Cleaning the installation

The exterior of a hydraulic installation should be cleaned occasionally so that it is easier to find leaks, so that no dirt is carried in when topping up the fluid or changing filter elements, to protect cylinder rods from being scored and so as not to interfere with the amount of heat dissipation that the design allows for.

When cleaning an installation, however, it is important not to allow any cleaning liquids to gain access to the hydraulic system.

If high-pressure steam cleaning equipment is used, care must be taken that tank covers, pipe glands, shaft seals, electrical equipment, etc. are able to withstand it.

2.2.7 Servicing of pressure vessels

Each state or country has its own regulations governing pressure vessels. In the case of West Germany the main set of regulations are called *"Verordnung ueber Druckbehaelter"*.

The prime object of the rules laid down in these regulations and others is operating safety and the elimination of as much danger to persons and property as possible.

In the case of hydraulic installations, oil tanks are "pressure vessels" if they are operated at a pressure above atmospheric (e.g. on forging presses). Hydraulic accumulators of the bladder type, piston type and diaphragm type with their associated gas bottles are also pressure vessels within the meaning of the regulations.

With a few exceptions, hydraulic accumulators are always charged with nitrogen.

When carrying out any servicing or maintenance on hydraulic installations incorporating accumulators it is absolutely essential for the fluid side of the system to be depressurized first. In order to ensure that this instruction is observed, a clear sign (in several languages if necessary) should be placed in a prominent position on the installation:

DANGER PRESSURIZED ACCUMULATORS
Depressurize the fluid side of the system before commencing any repairs

2.2.8 Replacing worn components

The circumstances of hydraulic component failure are very difficult to predict. The conditions of the particular application are a very important factor, e.g. dynamic pressure stresses, flow velocities, type of fluid, thermal stresses, environmental factors, etc.

For known applications it is possible to predict approximate average service life figures for rolling-contact bearings and dynamically loaded seals. Precautionary renewal of rolling bearings, cylinder seals, dirt wipers, hoses, etc. can be sensible if the total failure of such items can have serious consequences.

For example, this applies if:

1) damage to a rolling bearing can cause the total destruction of a whole drive system
2) uncontrollable cylinders can result in collisions with other parts of machines
3) burst hoses can cause major environmental pollution or danger to life and limb.

The precautionary renewal of wearing parts during servicing is also a good idea if the system is employed on multi-shift working and is integrated into a production line which would suffer a total shutdown should the system suffer a malfunction.

Some of the other components of systems which must be regarded as wearing parts are:

- all elastomer and polyurethane based sealing elements exposed to static and dynamic loads
- inserts of pressure relief valves, pressure reducing valves, pressure sequence valves and unloading valves
- solenoids, when very high switching frequencies are involved
- filter elements, if there is no monitoring of clogging
- inserts in flexible shaft couplings
- high-pressure hoses, if conditions cause the manufacturer to guarantee only a limited number of stress reversals.

2.3 Repair

2.3.1 Fault-finding

Any repair to a system must be preceded by successful fault-finding.

It requires a systematic approach, one of the basic necessities for which is the easy availability of all the relevant documentation.

It is desirable for a circuit diagram containing lists of components and a function sequence chart to be kept constantly available close to the installation.

Past experience has shown that it is an excellent idea to paste the diagrams on to boards and then either varnish them or cover them with Perspex to prevent them becoming dirty or damaged. Good illumination is also beneficial.

Important measuring instruments should also be available close by, especially on the larger installations. The most important fault-finding tools for a troubleshooter are pressure gauges of the necessary measuring ranges. In order to maintain accuracy and an acceptable service life, pressure gauges should normally be chosen so that only 2/3 to 3/4 of the maximum scale reading is utilized. Pressure gauges can be connected quickly and without loss of fluid through Minimess quick-release connectors and hoses of 1.8 to 2.4 mm bore. It is important to re-seal the connecting points afterwards with the screw caps to keep out dirt and to protect the sealing surface.

Thermometers
(electronic thermometers with surface sensors are best)

- for quickly finding points of high temperature due to leaks, power loss, etc.

Multimeter

- for checking the resistance of solenoids
- for measuring the voltage at solenoids (the voltage is not measured at the coil terminals in the switchgear cubicle)
- for measuring the current at proportional valves and servo valves

Naturally, there are also many other specialized measuring instruments available on the market for measuring voltage and current which have been specifically designed to be very easy to use.

Industrial stethoscope

- for pin-pointing noise sources through structure-borne sound

Stopwatch

- for measuring extension times and retraction times of cylinders
- for measuring leakage rates from hydraulic pumps and motors

Tachometer

- for measuring the speed of hydraulic motors.

Measuring flow rates

The electrician has an easy task measuring current compared with the hydraulic engineer trying to measure flow rate.

The system has to be opened up in order to fit a flow-meter, with the attendant danger of fluid contamination and fluid loss.

Only few instruments are able to withstand the pressures involved and the measuring range is often relatively small. Consequently, the hydraulics engineer usually has to measure flow rate indirectly through the operating times of cylinders and the speeds of hydraulic motors.

Standard tools

2.3.2 Fault rectification

Generally speaking, faulty equipment should not be repaired on site because neither the necessary tools nor the necessary cleanliness are normally available there. Whenever possible the procedure should be to change complete items or sub-assemblies locally

- so that the system is only opened up for as long as is absolutely necessary
- so that fluid loss is kept to a minimum
- so that the down time is kept to a minimum by using overhauled and pre-tested equipment.

Another very important factor after the failed item of equipment has been pin-pointed is to ascertain whether the failure has caused any contamination of the remainder of the system due to broken pieces of metal or large quantities of metal residue.

If inspection of the filters or tank reveals such contamination, it is absolutely essential for the whole installation to be cleaned, e.g. by flushing, before it is returned to service. The filter elements should also be changed and, depending on the particular type of installation, it might be necessary to change the fluid.

In the case of larger installations the fluid can be cleaned either by filter or centrifuge.

This is the only way of preventing further malfunctions and the possible failure of other components due to secondary damage.

As an extra safety factor, particularly with sensitive systems which have to perform very precise control functions, it is sensible to fit the filters, for a limited period of time (e.g. until they reach their clogging limit), with elements of the next smaller pore size. This is nearly always possible with modern filters. When a system has been returned to service after repair it should be kept under close observation for some time in order to be certain that the repair has actually eliminated all the problems. Sometimes there is some irregular operation for a period if part of the system has not been properly bled of air, e.g. cylinders "spring" or overrun positions, pumps give off loud knocking noises, etc. If systems are self-bleeding, this type of irregularity might last for several hours.

2.3.3 Repair of hydraulic components

Before the actual repair of any hydraulic components is started, a decision must be taken on which components can be overhauled by the operator himself and which can only sensibly be overhauled by the equipment manufacturer.

The repair of hydraulic components needs a suitably equipped workshop in which the standards of cleanliness are above average compared with a normal mechanical engineering workshop. Obviously, this is only an economic proposition if the company involved possesses large amounts of equipment containing a high proportion of hydraulic systems.

The second need is for properly trained staff and all the relevant documentation.

However, both of these preconditions are useless and inefficient if there is not also a comprehensively stocked store of spares available.

What components can be repaired or overhauled by the plant operator for an economic outlay?

2.3.3.1 Hydraulic cylinders

The principal types of damage suffered by cylinders are:

- Worn seals - these can be changed. In some cases special tools are needed for fitting them.
- Scored pistons, scored cylinder walls, scored rod guides and rod surface damage due to impact or weld spatter. If the damage is not too bad, cylinder bores can be reclaimed by honing and piston rods by dechroming, regrinding and rechroming. Some subsequent work on the cylinder ends may be necessary in order to restore the specified clearances for seals.

2.3.3.2 Check valves

Nearly all types of check valve have metallic seats. Often the cone is harder than the seat in the body.

The main problems with this type of valve (non return valves, prefill valves, logic elements etc.) are:

- Hardened seats. They can be easily changed. Great care must be taken when opening up the body because the springs inside are sometimes very strong.
- Internal leakage due to damaged valve seats. Such damage is nearly always caused by foreign objects or erosion.

  A repair can be effected either by re-machining the seat or by changing both seat and cone.

  The re-machining of body seats needs absolute precision since practically no misalignment between seat and guide is allowed.

  A static leakage test is easy to perform.

2.3.3.3 Directional control valves

Some directional control valves, especially the smaller ones, are impossible to repair economically apart from fitting new seals or new solenoids.

Readjustment is necessary with some types when a new solenoid is fitted. This can only be performed with suitable equipment.

Directional spool valves

Apart from the fitting of new seals and new operating devices, such as solenoids, lamps or parts of manual, mechanical, hydraulic or pneumatic mechanisms, it is practically impossible for the plant operator to repair this type of valve because it requires intervention in the geometry of the design and involves honing or lapping of bores and the making of suitable precision-ground oversize spools. The very small clearances that are commonplace nowadays require very high precision machine tools in order to restore the components to their new state. The subsequent functional testing also requires a test stand.

2.3.3.4 Pressure valves

Both the spool type of pressure valve and the poppet type are equally popular in hydraulic installations.

In the case of the spool type the basic approach to repairs described in *Section 2.3.3.3* is also applicable.

Poppet valves, on the other hand, can be repaired by fitting a new cone and seat on the pilot side and a new bush and cone on the main side.
Nowadays, however, it is becoming ever more common for pressure valves to have a screw-in cartridge in the body so that a repair can be made by simply fitting a complete new cartridge.

The cost of a functional test to assess the condition of a pressure valve is quite high and therefore often uneconomical for the plant operator.

2.3.3.5 Flow control valves

Throttle valves and throttle check valves are available in line-mounting versions, subplate-mounting versions and cartridge versions for fitting in control plates.

Repairs to the smaller sizes of such valves are confined to the fitting of new seals. Any other repairs such as the replacement of worn parts are usually uneconomical.

In the case of the more costly types, such as fine throttle valves, large sizes of throttle and throttle check valves, twin throttle check valves and deceleration valves with or without check valves, repairs other than the straightforward renewal of seals can be economic provided the necessary hand tools and machine tools are available.

Of course, in order to restore such valves to their original (new) quality, it would be necessary to be in possession of the manufacturer's production drawings and, in order to set the valves for a particular characteristic, a sophisticated test stand would be needed.

Flow control valves make the highest demands on tools, qualified personnel and test stand facilities. Repairs other than the renewal of seals can, therefore, only be recommended to plant operators in the very rarest of cases.

2.3.3.6 Proportional valves

The three types are proportional directional control valves, proportional pressure valves and proportional flow valves. Such types are so widespread nowadays that they can be regarded as popular standards. The range of possible applications for them is also expanding very rapidly. It is not advisable for plant operators to attempt repairs other than the renewal of seals. However, this does not mean that the valves are so sensitive, fragile or complex that a fitter who is able to repair all other types of equipment would not have the skill to be successful with the repair of proportional valves. The problem is that the cost of test stand equipment including electronic gear and recorders for measuring and setting the required characteristics is so high that even equipment manufacturers who handle repairs cannot make the necessary investment until they can be assured of at least 5 to 7 hours usage per day. Until this level of business is attained, overhauled valves are tested and documented on the production test stands.

2.3.3.7 Servo valves

What has been said in the previous section is even more true of servo valves. As the grinding of the spool into the sleeve requires special machines (the hydraulic zero overlap is not the same as the mechanical zero overlap, amongst other things), adjustment and test stand expenses are among the highest needed by any hydraulic equipment. The necessary capital investment for servo valve overhaul therefore cannot be recommended to a hydraulic equipment user unless he has several hundred servo valves in constant use.

The servicing of servo valves by plant operators presents no problems. It is easy to clean or change the protective filter for the first stage. Also, for trained personnel, it is a relatively easy matter to adjust the zeros without needing much in the way of instruments.

However, it must be emphasized once more that maintenance personnel working on proportional valves and servo valves should be trained by the manufacturer before attempting any work on actual installations.

2.3.3.8 Hydraulic accumulators

Hydraulic accumulators are subject to the regulations governing pressure vessels which are so comprehensive as to exceed the bounds of a detailed description in this chapter.

Weight loaded accumulators and spring accumulators are practically never encountered nowadays except for a few specialized applications. With both types the fluid side is sealed from the surrounding atmosphere by pistons incorporating soft seals.

In piston-type accumulators the fluid side is also separated from the gas side by a piston with seals made of elastomer, fabric or polyurethane. The plant operator can change such seals very easily.

In the case of the much more popular bladder-type and diaphragm-type accumulators the fluid side is separated from the gas side by a bladder or diaphragm made of an elastomer material. The type of elastomer chosen depends on the type of fluid. When bladders or diaphragms are changed (both are wearing parts) it is important for the new ones to be made of a material compatible with the fluid.

Bladder accumulators should only be repaired by properly trained personnel because any errors are nearly always very dangerous, both for plant and personnel. Hydraulic accumulators are normally charged exclusively with nitrogen in order to prevent fire and explosion. Some small accumulators are welded together and so must be regarded as disposable items, i.e. they cannot be repaired.

2.3.3.9 Hydraulic pumps, hydraulic motors

Gear pumps, gear motors

Seals can be renewed on all types of gear pump. A general overhaul, on the other hand, is usually uneconomic, especially if the body forms one side of the sealing gap to the displacement elements. Bearing damage (they are usually plain bearings) or damage due to foreign bodies usually results in total destruction.

Vane-type pumps (fixed and variable displacement)

Only properly trained personnel are qualified to carry out repairs and overhauls on these units. In addition to the replacement of seals, a general overhaul through the changing of sub-assemblies is an economic proposition in most cases.

In Europe at least it is of prime importance to keep losses to a minimum so components are often "matched pairs" in order to achieve absolute minimum clearances and therefore absolute minimum volumetric losses.

Whether or not the overhaul of a pump has been successful can only be demonstrated by a run on a test stand. If the pump characteristics are to be adjusted or transfer functions tested, the instrumentation of the test stand will need to be very comprehensive.

If necessary, of course, the installation itself can serve as the test stand; although this should not be taken as normal practice with important installations because its availability as a test stand can never be guaranteed.

Axial piston pumps, axial piston motors

Once again the golden rule is that only specially trained personnel should be entrusted with repair work.

General overhauls are economic in most cases with the controllable versions of these pumps and motors as well as minor repairs such as the renewal of seals. A general overhaul for the non-controllable version is not usually economic depending on the extent of the damage.

The capital investment for test stands and instrumentation capable of handling this type of pump and motor of different outputs and with different methods of control ranges upwards from 0.25 to 0.75 million DM.

The pay-back period can only become reasonable if very high usage is achieved.

2.3.3.10 Slow-speed hydraulic motors (fixed displacement)

General overhauls and minor repairs are also economic with this type of motor.

Properly trained personnel are essential once again, as is a test stand to prove the repair.

Capital investment for the test stand is once more very high because of the low speed and high torque of the units.

2.3.3.11 Hydraulic accessories

Hydraulic accessories take the form of:

filters, pressure switches, pressure gauge selectors, pressure gauge isolators, air/oil heat exchangers, water/oil heat exchangers, heaters, etc. Some of these items are repairable whereas for others it is not an economic course of action.

2.3.4 Repair and general overhaul of hydraulic equipment

The bottom line as far as the repair and general overhaul of hydraulic equipment is concerned is that the equipment manufacturer can do it more economically and more safely.
This is because:

- the repair departments of equipment manufacturers work to the same quality standards on general overhauls as for the production of new equipment
- repair personnel are trained in exactly the same way as personnel who work on new equipment
- spare parts are made on the same machines that produce new parts
- full test facilities are available
- overhauled equipment is usually subjected to stiffer tests than new equipment
- equipment manufacturers guarantee the same performance from overhauled equipment as from new equipment
- some equipment manufacturers offer the same guarantee period for overhauled equipment as for new equipment

3 Summary

The scope and scheduling of inspection and maintenance should be planned and documented according to the size and significance of the particular installation.

Any inspections or maintenance demanded by law must be carried out independently of similar measures aimed at maintaining the function of the installation. For reasons of safety for the operating personnel and the avoidance of possible material damage, they should also be properly documented.

For fast and successful fault-finding or repair work on installations it is a sensible idea to have all the necessary documentation and principal instruments needed available locally.

If the maintenance engineer is not qualified in both hydraulics and electrics (very desirable but unfortunately rare), a hydraulics specialist and electrical specialist must be called in together in order to pin-point and rectify the fault as quickly as possible and without shunting the problem back and forth.

Repairs to equipment should be restricted to the replacement of entire components. Except for minor repairs which can be handled safely, equipment should be returned to the manufacturer for general overhaul in order to be certain that the replacement equipment eventually held in store will be totally reliable.

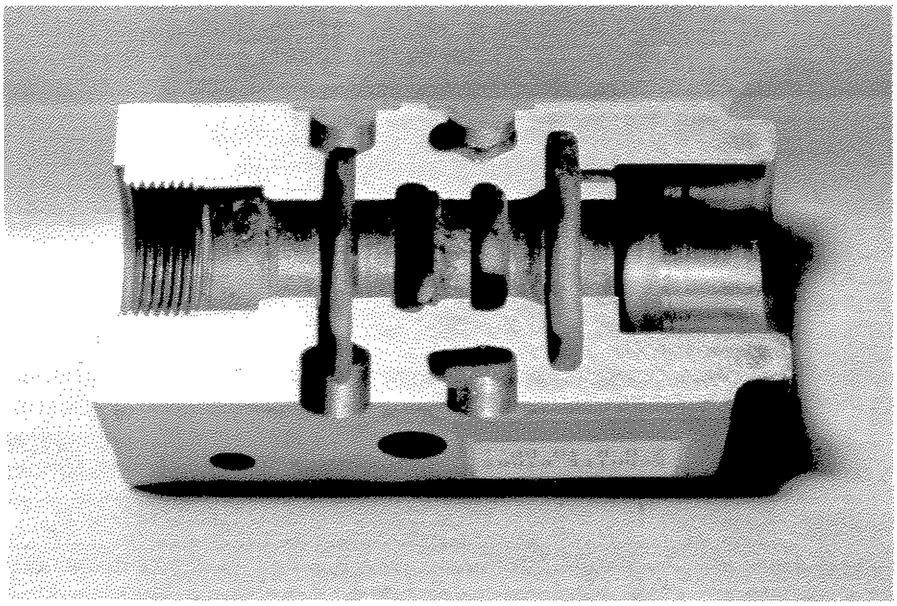


Fig. 237:
Erosion of the control land of a pressure reducing valve after approx. 12,000 hours. The pressure drop across the land is 50 to 65 bar

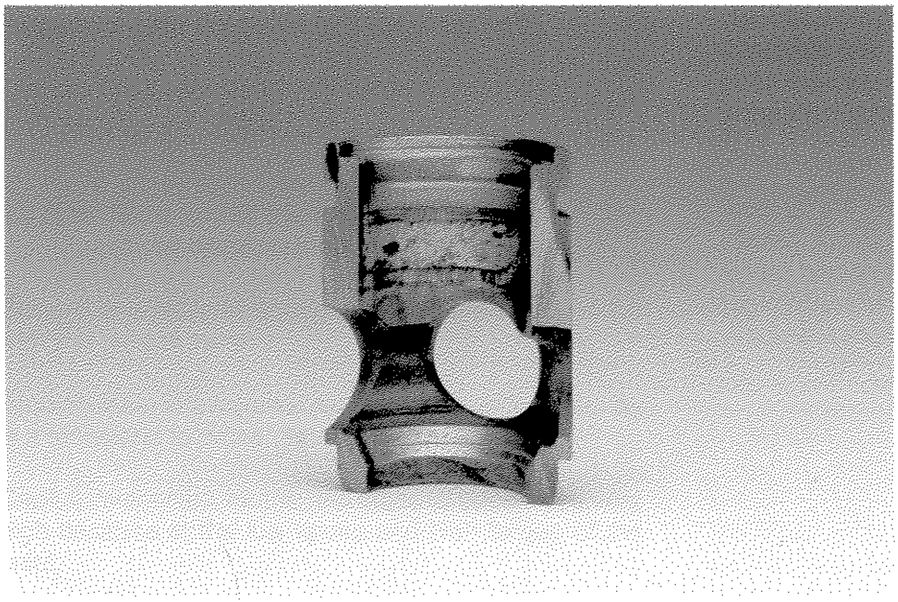


Fig. 238
Traces of seizure between spool and bush of a pilot-operated pressure relief valve.
Cause: High-frequency, low-amplitude vibration

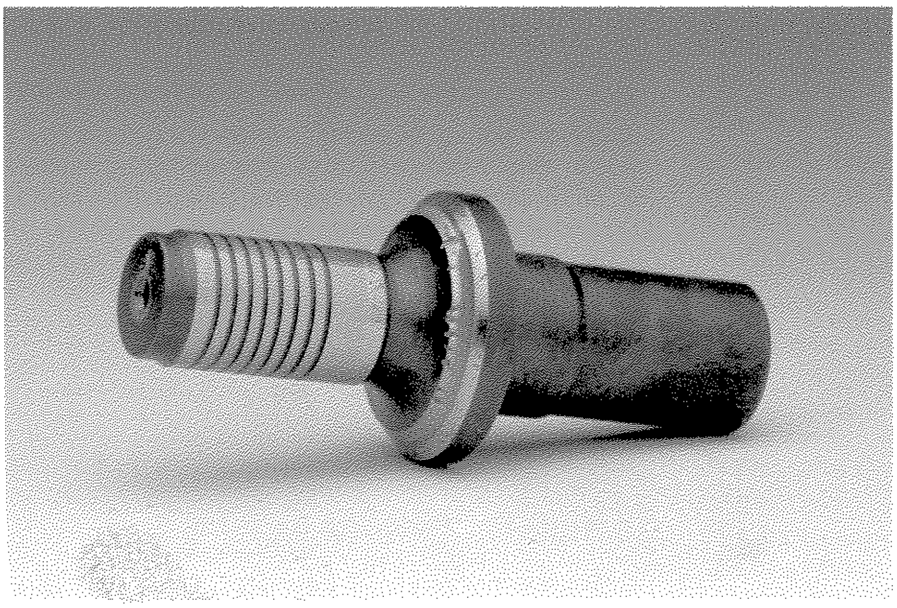


Fig. 239:
Erosion of the seat of the pilot cone of a pre-fill valve caused by a very high pressure drop and excessive solids in the fluid

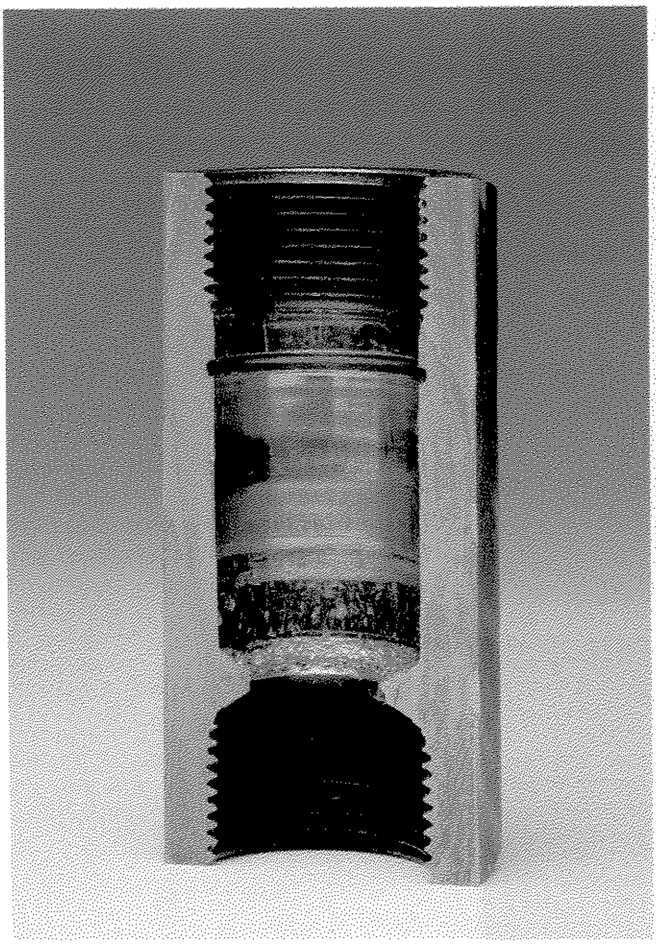


Fig. 240:
Seat of a check valve ruined by foreign bodies

Fig. 241:
Normal wear of the cone of a direct-operated pressure relief valve operating very frequently and after a long service life

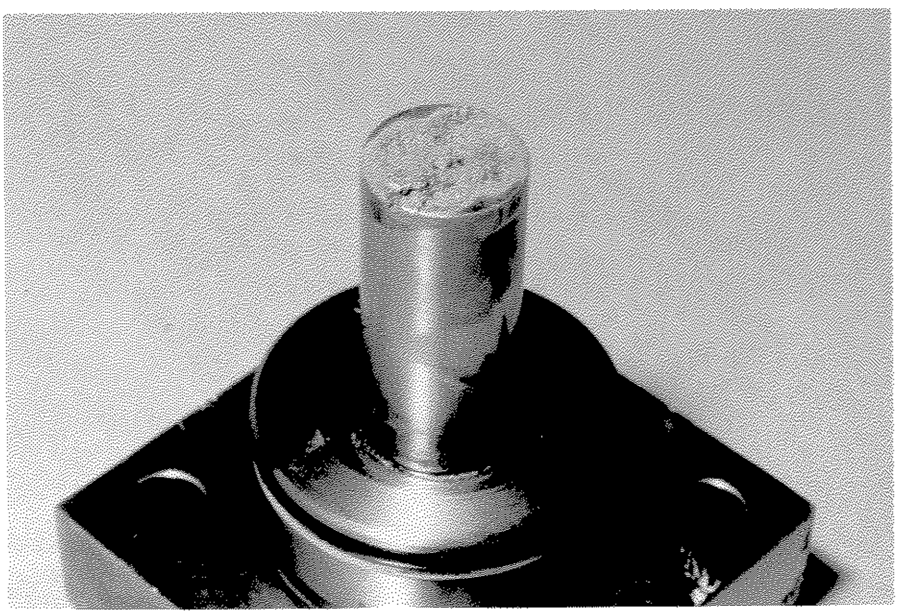


Fig. 242:
Cavitation damage to a throttle screw. Remedy: Increased downstream pressure

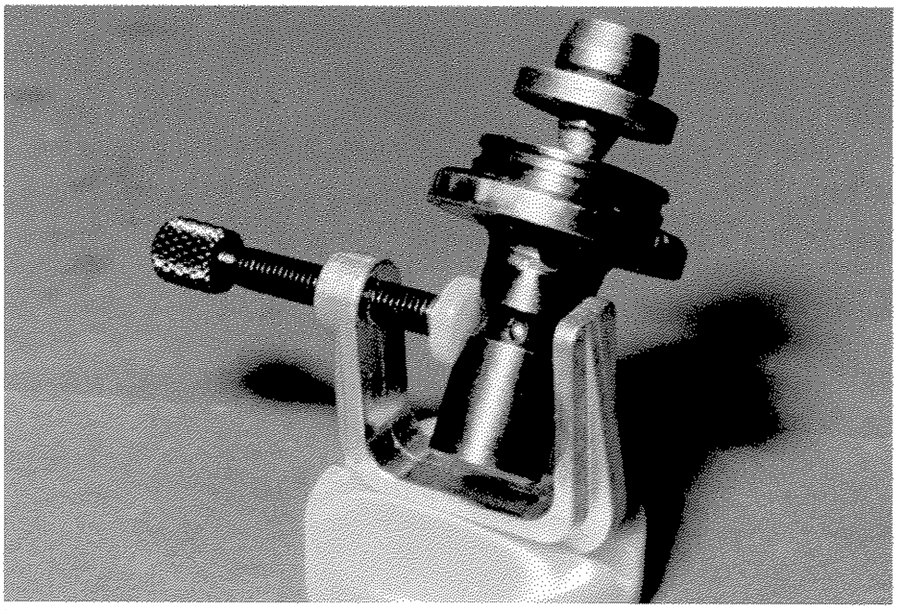


Fig. 243:
Erosion of the cone of a direct-operated pressure relief valve.
Cause: High pressure drop and high solids content in the fluid combined with frequent operation

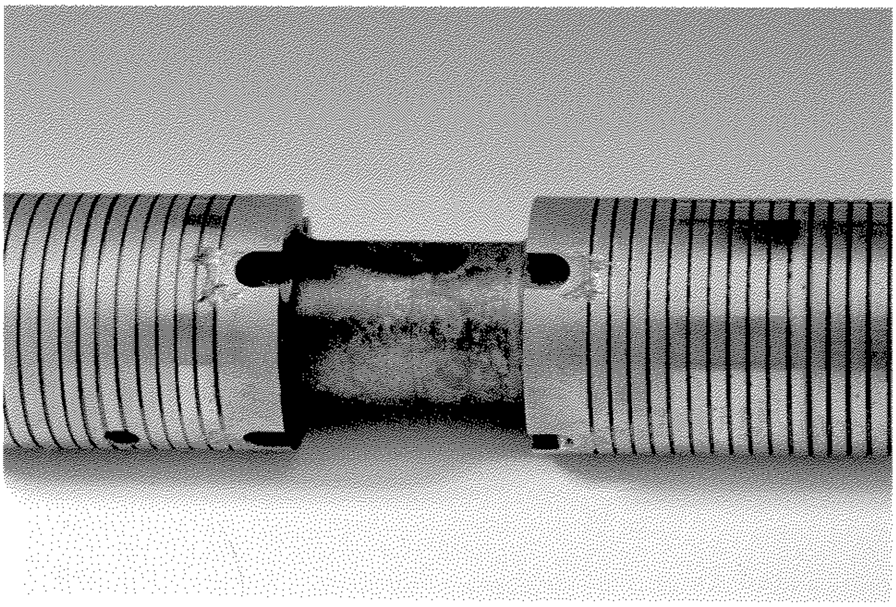


Fig. 244:
Erosion around the fine control grooves of a Size 25 directional control valve.
Cause: The valve was used to unload a large volume of fluid under high pressure. The solids content of the fluid was over 12 to NAS 1638

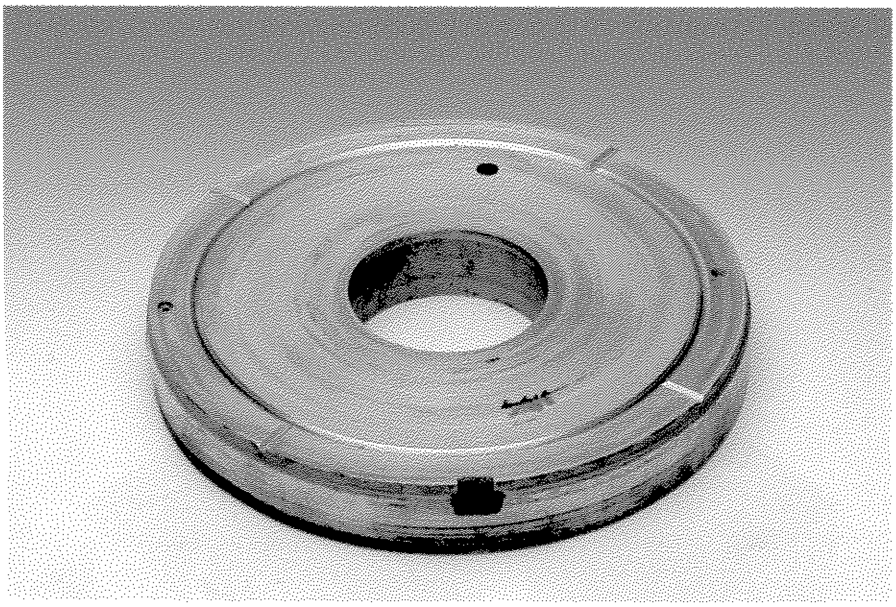


Fig. 245:
Cavitation damage to the end cover of a vane pump caused by insufficient pressure at the pump inlet

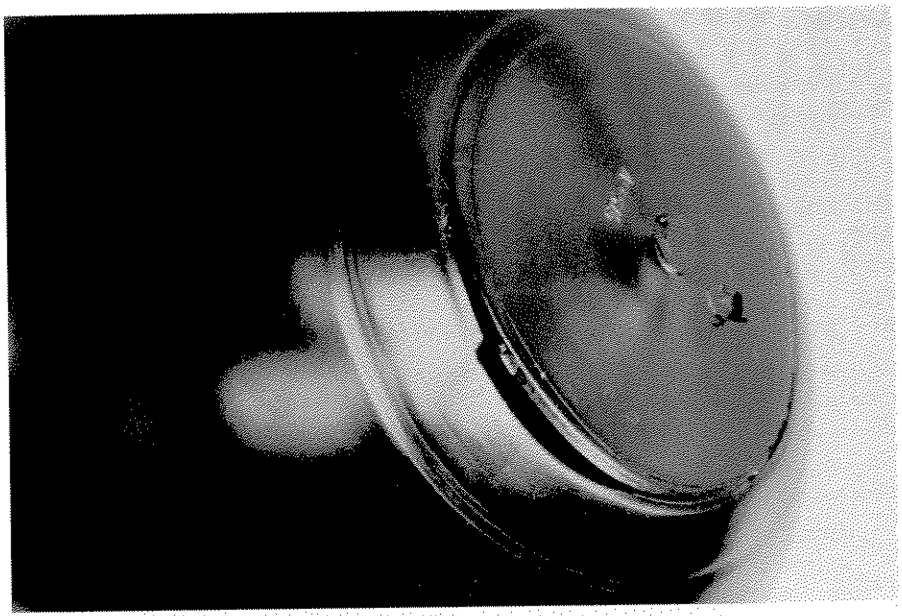


Fig. 246:
Torn accumulator bladder.
Cause: An excessive fluid flow rate pulled the bladder under the valve plate

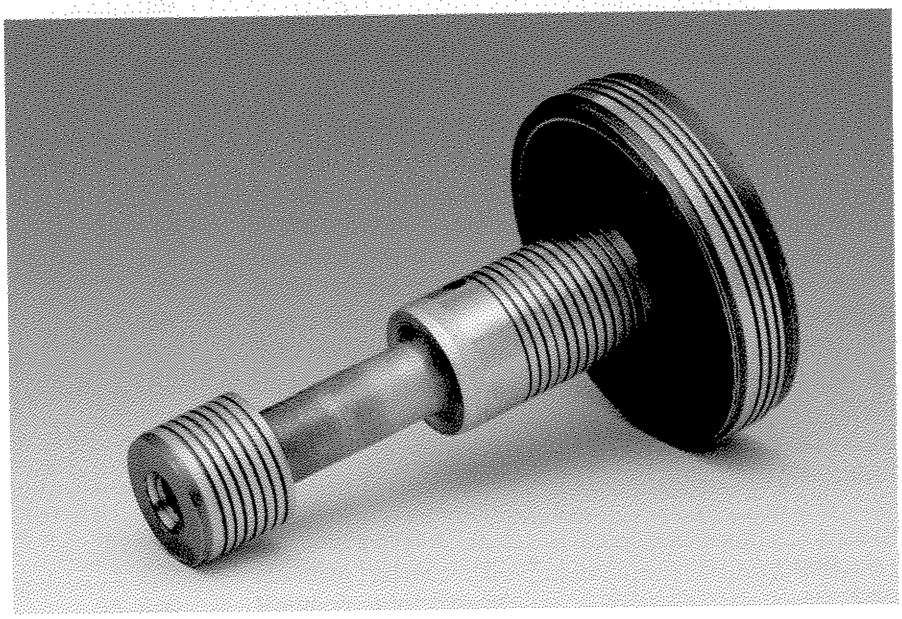


Fig. 247:
Worn control land on the piston of a pilot-operated pressure relief valve caused by excessive solids in the fluid