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Fluid Mechanisms

Hydraulic Mechanisms

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Contents

List of Symbols

Symbol	*Unit*	*Term*
A	[m^2]	area
A_1	[m^2]	piston area
A_2	[m^2]	annulus area on the piston rod side
D	[m], [mm]	diameter, piston diameter
E	[MPa]	Young’s modulus of elasticity
E_p	[J]	pressure energy
F	[N]	force
F_{CRe}	[N]	Euler’s critical force
F_r	[N]	radial force
F_s	[N]	spring force
F_t	[N]	theoretical force
G_{Gl}	[$N^{-1} \cdot m^5 \cdot s^{-1}$]	leakage permeability of the hydraulic pump
G_M	[$N^{-1} \cdot m^5 \cdot s^{-1}$]	leakage permeability of the hydraulic motor
J	[m^4]	moment of inertia of the piston rod cross-sectional area
K	[Pa]	bulk modulus of liquids
L	[mm]	guidance length
L_1	[mm]	ring length
M	[N · m]	torque
M_1	[N · m]	torque on the shaft of the hydraulic pump
M_2	[N · m]	torque on the shaft of the hydraulic motor
M_G	[N · m]	torque on the input shaft of the hydraulic pump
M_M	[N · m]	real output torque on the output shaft of the hydraulic motor
M_{Mt}	[N · m]	theoretical output torque on the output shaft of the hydraulic motor
P	[W]	power
P_1	[W]	input power of the hydraulic mechanism
P_2	[W]	output power of a hydraulic mechanism
P_h	[W]	hydraulic power
P_{ht}	[W]	theoretical hydraulic power at the output of the hydraulic pump
P_i	[W]	input power

P_l	[W]	power loss
P_{lRV}	[W]	power loss during liquid flow through the relief valve
P_{lrv}	[W]	power loss during liquid flow through the pressure reducing valve
P_m	[W]	mechanical power
P_{mt}	[W]	theoretical mechanical power on the output shaft of the hydraulic motor
P_T	[W]	transmitted power
Q_1	$[m^3 \cdot s^{-1}]$, $[dm^3 \cdot min^{-1}]$	volumetric flow at the motor input
Q_2	$[m^3 \cdot s^{-1}]$, $[dm^3 \cdot min^{-1}]$	volumetric output flow from the hydraulic cylinder
Q_G	$[m^3 \cdot s^{-1}]$, $[dm^3 \cdot min^{-1}]$	real volumetric flow of the hydraulic pump
Q_{Gl}	$[m^3 \cdot s^{-1}]$, $[dm^3 \cdot min^{-1}]$	loss volumetric flow of the hydraulic pump
Q_{Gt}	$[m^3 \cdot s^{-1}]$, $[dm^3 \cdot min^{-1}]$	theoretical volumetric flow of the hydraulic pump
Q_{HG}	$[m^3 \cdot s^{-1}]$, $[dm^3 \cdot min^{-1}]$	volumetric flow of the hydraulic pump
Q_l	$[m^3 \cdot s^{-1}]$, $[dm^3 \cdot min^{-1}]$	flow loss
Q_M	$[m^3 \cdot s^{-1}]$, $[dm^3 \cdot min^{-1}]$	input volumetric flow of the hydraulic motor
Q_r	$[m^3 \cdot s^{-1}]$, $[dm^3 \cdot min^{-1}]$	real volumetric flow
Q_{RV}	$[m^3 \cdot s^{-1}]$, $[dm^3 \cdot min^{-1}]$	volumetric flow through the relief valve
Q_{rv}	$[m^3 \cdot s^{-1}]$, $[dm^3 \cdot min^{-1}]$	volumetric flow through the pressure reducing valve
Q_t	$[m^3 \cdot s^{-1}]$, $[dm^3 \cdot min^{-1}]$	theoretical volumetric flow
Q_{TV}	$[m^3 \cdot s^{-1}]$, $[dm^3 \cdot min^{-1}]$	volumetric flow through the throttle valve
R	[m], [mm]	radius
Re	[-]	Reynolds number
Re_{crit}	[-]	critical value of the Reynold's number
S_e	[-]	safety factor
V	$[m^3]$, $[dm^3]$	volume
V_0	$[m^3]$, $[dm^3]$	nominal volume of the accumulator
V_A	$[m^3]$, $[dm^3]$	useful volume of the accumulator
V_g	$[m^3]$, $[cm^3]$	geometric stroke volume
V_{gG}	$[m^3]$, $[cm^3]$	geometric stroke volume of the hydraulic pump
V_{gM}	$[m^3]$, $[cm^3]$	geometric stroke volume of the hydraulic motor
a_s	[MPa]	material constant

b	[m], [mm]	width of the gears
b_s	[MPa]	material constant
c	$[J \cdot kg^{-1} \cdot K^{-1}]$	specific heat capacity
d	[m], [mm]	diameter, inner pipe diameter, piston rod diameter
d_s	[m], [mm]	spool diameter
e	[m], [mm]	eccentricity
e_l	$[J \cdot kg^{-1}]$	specific loss energy
g	$[m \cdot s^{-2}]$	acceleration due to gravity
h	[m]	height
h_l	[m]	loss height
i	[m]	radius of gyration of the piston rod cross-sectional area
l	[m], [mm]	length
l_{red}	[m], [mm]	reduced length of the piston rod
m	[kg]	weight, mass
m_g	[-]	gear module
n	$[s^{-1}]$, $[min^{-1}]$	speed
n_1	$[s^{-1}]$, $[min^{-1}]$	shaft speed of the hydraulic pump
n_2	$[s^{-1}]$, $[min^{-1}]$	shaft speed of the hydraulic motor
n_G	$[s^{-1}]$, $[min^{-1}]$	speed of the hydraulic pump
n_M	$[s^{-1}]$, $[min^{-1}]$	real speed of the hydraulic motor
n_{max}	$[s^{-1}]$, $[min^{-1}]$	maximum speed
n_{min}	$[s^{-1}]$, $[min^{-1}]$	minimum speed
n_{Mt}	$[s^{-1}]$, $[min^{-1}]$	theoretical speed of the output shaft of the hydraulic motor
p	[Pa], [MPa], [bar]	pressure
p_0	[Pa]	ambient pressure (usually the atmospheric pressure)
p_1	[Pa], [MPa], [bar]	input pressure
p_2	[Pa], [MPa], [bar]	output pressure
p_{abs}	[Pa], [MPa], [bar]	absolute pressure
p_l	[Pa], [MPa], [bar]	pressure loss
p_{max}	[Pa], [MPa], [bar]	maximum pressure
p_{min}	[Pa], [MPa], [bar]	minimum pressure
p_n	[Pa], [MPa], [bar]	nominal pressure
p_{rel}	[Pa], [MPa], [bar]	relative pressure

p_{RV}	[Pa], [MPa], [bar]	pressure on the relief valve
p_{sA}	$[N \cdot mm^{-2}]$	allowable specific pressure
r	[m], [mm]	radius
s	[m], [mm]	gap thickness
s_2	[mm]	cylinder wall thickness
t	[s]	time
v	$[m \cdot s^{-1}]$	velocity
v_1	$[m \cdot s^{-1}]$	velocity of the piston rod extension
v_2	$[m \cdot s^{-1}]$	velocity of the piston rod retraction
v_r	$[m \cdot s^{-1}]$	actual discharge velocity
v_t	$[m \cdot s^{-1}]$	theoretical discharge velocity
v_t	$[m \cdot s^{-1}]$	theoretical velocity of the piston rod extension
z	[-]	number of teeth of the drive wheel, number of pistons
α	[°]	angle of inclination of the plate, swivel angle
β	$[°C^{-1}]$	thermal coefficient of expansion of liquids
δ	$[Pa^{-1}]$	coefficient of volume compressibility
Δp	[Pa], [MPa], [bar]	pressure gradient, pressure drop
Δp_G	[Pa], [MPa], [bar]	real pressure gradient of the hydraulic pump
Δp_{Gt}	[Pa], [MPa], [bar]	theoretical pressure gradient of the hydraulic pump
Δp_l	[Pa], [MPa], [bar]	pressure loss
Δp_{lrv}	[Pa], [MPa], [bar]	pressure loss on the pressure reducing valve
Δp_M	[Pa], [MPa], [bar]	pressure gradient of the hydraulic motor
Δp_{PC}	[Pa], [MPa], [bar]	pressure gradient on the pressure compensator
Δp_{RV}	[Pa], [MPa], [bar]	pressure gradient on the relief valve
Δp_{TV}	[Pa], [MPa], [bar]	pressure gradient on the throttle valve
Δt	[°C]	change in temperature
ΔT	[K]	change in temperature
ΔV	$[m^3]$	volume change
ΔW	[J]	change in internal thermal energy
ζ	[-]	loss coefficient
η	$[Pa \cdot s]$	dynamic viscosity
η_{Gmh}	[-]	mechanical-hydraulic efficiency of the hydraulic pump

η_{GT}	[-]	total efficiency of the hydraulic pump
η_{Gvol}	[-]	volumetric efficiency of the hydraulic pump
η_{mh}	[-]	mechanical-hydraulic efficiency
η_{Mmh}	[-]	mechanical-hydraulic efficiency of the hydraulic motor
η_{MT}	[-]	total efficiency of the hydraulic motor
η_{Mvol}	[-]	volumetric efficiency of the hydraulic motor
η_{T}	[-]	total efficiency
η_{vol}	[-]	volumetric efficiency
λ	[-]	friction coefficient
λ_{m}	[-]	limiting piston rod slenderness
λ_{p}	[-]	piston rod slenderness
μ	[-]	discharge coefficient, flow coefficient
ν	[$m^2 \cdot s^{-1}$]	kinematic viscosity
π	[-]	mathematical constant
ρ	[$kg \cdot m^{-3}$]	density
σ_{CR}	[MPa]	critical stress
σ_{d}	[MPa]	allowable tensile stress
σ_{t}	[MPa]	tensile stress
σ_{u}	[MPa]	elasticity limit stress
τ	[Pa]	tangential stress
φ	[-]	throttle valve opening ratio
φ_{G}	[-]	control parameter of the hydraulic pump
φ_{M}	[-]	control parameter of the hydraulic motor
ω_{1}	[$rad \cdot s^{-1}$]	angular velocity on the shaft of the hydraulic pump
ω_{2}	[$rad \cdot s^{-1}$]	angular velocity on the shaft of the hydraulic motor
ω_{G}	[$rad \cdot s^{-1}$]	angular velocity of the input shaft of the hydraulic pump
ω_{Mt}	[$rad \cdot s^{-1}$]	theoretical angular velocity of the output shaft of the hydraulic motor

Abbreviations

AU	Polyester Urethane
CM	control mechanism
ČSN	Czech national standard
CV	check valve
DA	unloading valve
DV	directional valve
EPDM	Ethylen-Propylen-Dien
EU	Polyether Urethane
FPM	Fluor rubber
HM	hydraulic motor
HNBR	Hydrogenated acrylonitrile butadiene
HP	hydraulic pump
IIR	Butyl rubber
ISO	International Organization for Standardization
NAS	National Aerospace Standard
NBR	Acrylnitril-Butadien
PA	Polyamide
PC	pressure controller, pressure compensator
PTFE	Polytetrafluorethylene
PU	Polyurethan
RV	relief valve
SAE	Society of Automotive Engineers
SBR	Styren-Butadien
TPE	Thermoplastic Elastomer
TV	throttle valve
VG	viscosity class

1. Definition and division of fluid mechanisms

Fluid mechanisms can be characterized as devices that use fluid to transfer energy and information between the drive (input) member and the driven (output) member. The fluid of these mechanisms may be:

- liquid (e.g., oil or water) - then these are hydraulic mechanisms,
- gas (compressed air) - these are pneumatic mechanisms.

The liquid is the energy carrier in hydraulic mechanisms. The total energy is given by the sum of all the partial energies of the liquid, i.e., pressure, kinetic, potential, deformation, and thermal energies. The division of hydraulic mechanisms and the determination of basic parameters are based on laws defined in fluid mechanics.

When an ideal liquid flows in a hydraulic mechanism, the law of conservation of energy is applied, which can be expressed by the Bernoulli equation for an ideal liquid as follows:

$$\frac{p}{\rho} + \frac{v^2}{2} + g \cdot h = const. \ , \tag{1.1}$$

where p [Pa] is the pressure, ρ [$kg \cdot m^{-3}$] is the liquid density, v [$m \cdot s^{-1}$] is the liquid flow velocity, g [$m \cdot s^{-2}$] is the acceleration due to gravity, and h [m] is the height.

The first term in equation (1.1) represents the specific pressure energy; the second term is the specific kinetic energy, and the last term is the specific potential (positional) energy of the liquid. In each system, different types of energy are transferred simultaneously, and depending on the predominant type and magnitude of energy, fluid mechanisms can work on static or dynamic principles [1].

If the predominant type of energy is kinetic, these are hydrodynamic mechanisms (hydrodynamic torque converters and couplings, centrifugal pumps). If the pressure energy predominates, hydrostatic mechanisms (hydraulic pumps and hydraulic motors) are the subject of this study text. The potential energy is negligible in terms of hydrostatic mechanisms and is not considered.

1.1 Basic parameters hydraulic mechanisms

Pressure

If the liquid is in equilibrium, then due to forces proportional to the liquid's mass, the hydrostatic pressure is generated at every point inside the liquid. This pressure is equal to the ratio of the pressure force dF acting perpendicular to the area dA:

$$p = \frac{dF}{dA}, \tag{1.2}$$

where p [Pa] is the pressure, F [N] is the pressure force, and A [m^2] is the area.

The unit of the pressure is Pa (Pascal); it is a derived unit of the SI system as follows:

$$Pa = \frac{N}{m^2} = \frac{kg \cdot m}{m^2 \cdot s^2} = \frac{kg}{m \cdot s^2}. \tag{1.3}$$

Because the value of 1 Pa is relatively small for hydraulic mechanisms, multiple units such as kPa and especially MPa are used. In engineering practice, the older unit bar is the most used. Because the value of 1 Pa is relatively small for hydraulic mechanisms, multiple units such as kPa and especially MPa are used. In engineering practice, the older unit bar is the most used. The conversion of the units is as follows:

$$100\,000\ Pa = 100\ kPa = 0.1\ MPa = 1\ bar\,. \tag{1.4}$$

The pressure at a given point in the liquid is the same in all directions, but its magnitude varies depending on the height h of the liquid column. If only the earth's gravity acts on the liquid, then the hydrostatic pressure can be expressed by the equation:

$$p = \rho \cdot g \cdot h\,. \tag{1.5}$$

where p [Pa] is the hydrostatic pressure, ρ [kg · m^{-3}] is the liquid density, g [m · s^{-2}] is the acceleration due to gravity, and h [m] is the liquid height.

When the ambient pressure p_0 acts on the liquid level, the pressure can be determined for any point in the liquid at the depth h according to the formula:

$$p = p_0 + \rho \cdot g \cdot h\,. \tag{1.6}$$

where p_0 [Pa] is the ambient pressure, usually the atmospheric pressure.

Depending on the pressure expression, it is necessary to distinguish between absolute pressure values (relative to absolute zero) and relative pressure values (relative to the atmospheric pressure), as shown in Fig. 1.1. In the case of the relative pressure, due to its magnitude in relation to the atmospheric pressure, a further distinction is made between positive pressure (overpressure) and negative pressure (under pressure) [2].

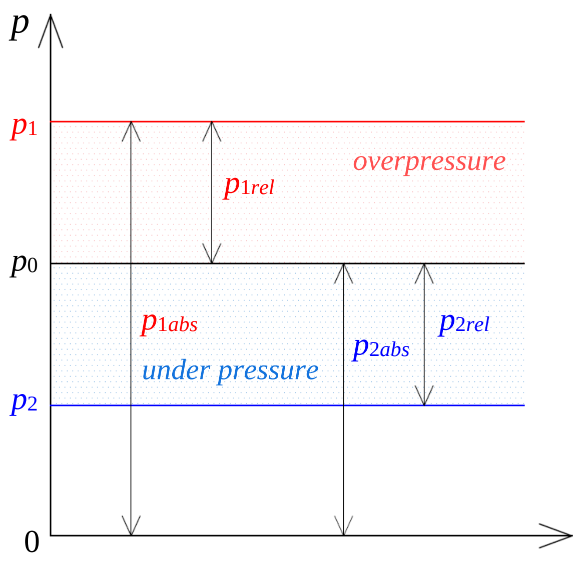


Fig. 1.1 Pressure expression

p_0 – atmospheric pressure, p_{1abs} – absolute pressure, p_{1rel} – relative pressure (overpressure), p_{2abs} – absolute pressure, p_{2rel} – relative pressure (under pressure)

If the mass forces acting on the liquid are neglected, Pascal's law can be used in hydrostatic mechanisms. If the pressure is increased at a certain liquid point, the pressure in the total volume of the liquid is increased since the pressure is propagated equally in all directions in the liquid. Then the pressure is expressed by the equation:

$$p = \frac{F}{A}. \tag{1.7}$$

where p [Pa] is the pressure, F [N] is the pressure force, and A [m^2] is the area.

Fig. 1.2 shows an example of a hydraulic jack. The action of the force F_1 on the surface of the smaller piston A_1 causes a pressure p in the liquid. The pressure action of the liquid is transmitted to the surface of the larger piston A_2, thereby achieving the force F_2 while:

$$p = \frac{F_1}{A_1} = \frac{F_2}{A_2} \rightarrow F_2 = F_1 \cdot \frac{A_2}{A_1}, \tag{1.8}$$

where F_1 and F_2 [N] are the forces acting on the jack pistons, and A_1 and A_2 [m^2] are the areas of the pistons.

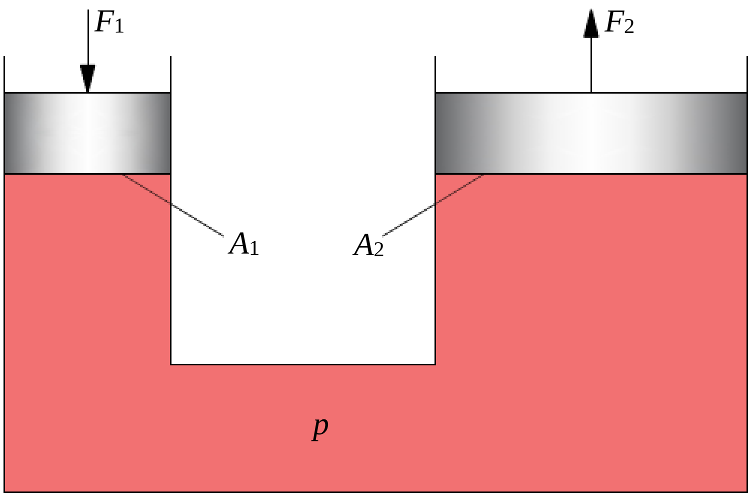


Fig. 1.2 Principle of Pascal´s law - hydraulic jack

In hydrostatic mechanisms, another expression of pressure is often observed. For example, the pressure gradient Δp represents the pressure difference in front of and behind the element (or conversely for hydraulic pumps). In addition, hydraulic mechanisms include terms such as nominal pressure, opening pressure, pressure drop during fluid flow and pressure pulsations or pressure peaks during system operation, etc. [10].

Volumetric flow

The liquid volumetric flow can be generally defined as the liquid quantity (volume), which flows through a given location per unit of time; it can be expressed by:

$$Q = \frac{V}{t} \ , \tag{1.9}$$

where Q [$m^3 \cdot s^{-1}$] is the liquid volumetric flow, V [m^3] is the liquid volume, and t [s] is the time.

The basic unit of the volumetric flow is $m^3 \cdot s^{-1}$. For practical reasons, the volumetric flow is usually given in $dm^3 \cdot min^{-1}$ (litres per minute). The conversion of units is as follows:

$$1 \ \frac{m^3}{s} = 1 \cdot 60\ 000 \ \frac{dm^3}{min} . \tag{1.10}$$

For the flow of an ideal liquid, the continuity equation, which expresses the law of conservation of mass, can be used. In the case of flow through two different pipe cross-sections (see Fig. 1.3) and considering an incompressible liquid, the volumetric flow is:

$$Q = A_1 \cdot v_1 = A_2 \cdot v_2 = const. \ , \tag{1.11}$$

where A_1 and A_2 [m^2] are the pipe cross-sectional areas, and v_1 and v_2 [$m \cdot s^{-1}$] are the liquid flow velocities.

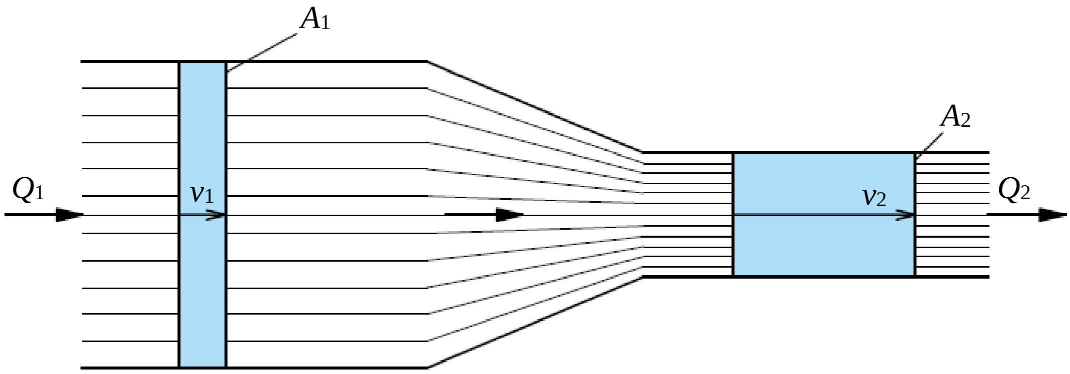


Fig. 1.3 Continuity equation

In the case of a real viscous liquid flow, the loss energy e_l must also be considered in equation (1.1). The loss of energy expresses the increase in thermal energy when overcoming resistance to liquid flow. This is the Bernoulli equation of an actual liquid, which can be written in the form:

$$\frac{p}{\rho} + \frac{v^2}{2} + g \cdot h + e_l = konst. \ , \tag{1.12}$$

where e_l [$J \cdot kg^{-1}$] is the specific loss energy.

The specific loss energy of a liquid can be expressed by means of partial energies by the equation:

$$e_l = \frac{p_l}{\rho} = g \cdot h_l = \zeta \cdot \frac{v^2}{2} , \tag{1.13}$$

where p_l [Pa] is the pressure loss, h_l [m] is the loss height, and ζ [-] is the loss coefficient including the effect of friction and local pressure losses in a system.

From the Bernoulli equation for the discharge of an ideal liquid from a vessel, considering the different pressures p_1 (above the level in the vessel) and p_2 (at the discharge) and neglecting the level drop in the vessel, the theoretical discharge velocity v_t can be determined as follows:

$$v_t = \sqrt{\frac{2 \cdot (p_1 - p_2)}{\rho}}, \tag{1.14}$$

where v_t [m · s^{-1}] is the theoretical discharge velocity.

The theoretical volumetric flow Q_t of a liquid is given by:

$$Q_t = S \cdot v_t = S \cdot \sqrt{\frac{2 \cdot (p_1 - p_2)}{\rho}}, \tag{1.15}$$

where Q_t [m^3 · s^{-1}] is the theoretical liquid volumetric flow.

The real discharge velocity is influenced by losses in the discharge port. These losses are characterized by the discharge coefficient μ. Then the actual discharge velocity is [3]:

$$v_r = \mu \cdot \sqrt{\frac{2 \cdot (p_1 - p_2)}{\rho}}, \tag{1.16}$$

where v_r [m · s^{-1}] is the actual discharge velocity, and μ [-] is the discharge coefficient.

The discharge coefficient is expressed by the equation:

$$\mu = \sqrt{\frac{1}{1 + \zeta}}. \tag{1.17}$$

The real liquid flow can subsequently be defined as:

$$Q_r = A \cdot v_r = A \cdot \mu \cdot \sqrt{\frac{2 \cdot (p_1 - p_2)}{\rho}} = \mu \cdot A \cdot \sqrt{\frac{2 \cdot \Delta p}{\rho}}, \tag{1.18}$$

where Q_r [m^3 · s^{-1}] is the real liquid volumetric flow.

The relation (1.18) can be used to define the liquid flow through elements (valves) of a hydraulic circuit, where area A characterizes the flow area of the element and Δp is the pressure gradient on the element.

The flow of real liquids can be in two modes. There is laminar flow (flow in layers) and turbulent flow (swirling flow with disordered particle motion), as is shown in Fig. 1.4.The flow regime is characterized by the dimensionless Reynolds number *Re*, according to the formula [4]:

$$Re = \frac{v \cdot d}{\nu}, \tag{1.19}$$

where Re [-] is the Reynolds number, v [m · s^{-1}] is the liquid flow velocity, d [m] is the inner pipe diameter (clearance), and ν [m^2 · s^{-1}] is the liquid kinematic viscosity.

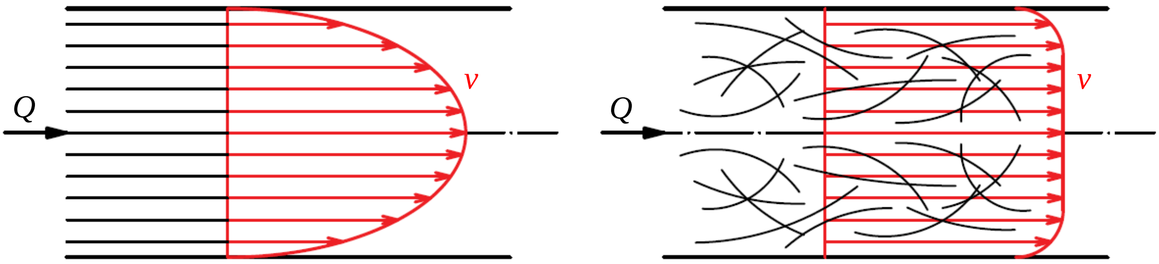


Fig. 1.4 Real liquid flow regimes, laminar flow (left), turbulent floe (right)

The transition between the two flow regimes occurs at a critical value of the Reynold's number Re_{crit}. For $Re \leq Re_{crit}$, the liquid flow is laminar. Similarly, for $Re > Re_{crit}$, the liquid flow is turbulent. The value of the critical Reynolds number is determined experimentally and varies for different types of hydraulic lines, as shown in Tab 1.1.

Tab 1.1 Values of the critical Reynolds number for different types of cross-sections [3]

Cross-section type	Re_{crit} [-]
smooth circular tubes	2000 ÷ 2320
flexible hoses	1600 ÷ 2000
smooth annulus slits	1100
distribution holes of cylindrical spools	260

As a liquid flows through a pipe, the friction causes a pressure loss. The friction loss equation (1.13) can be express the loss coefficient from equation:

$$\zeta = \frac{v^2}{2} \cdot \lambda \cdot \frac{l}{d}, \tag{1.20}$$

where λ [-] is the friction coefficient, l [m] is the pipe length, and d [m] is the inner pipe diameter.

The pressure loss Δp_l during the liquid flow through a pipe can be expressed from the equation (1.13) as follows:

$$\Delta p_l = \rho \cdot \frac{v^2}{2} \cdot \lambda \cdot \frac{l}{d}, \tag{1.21}$$

where Δp_l [Pa] is the pressure loss, and ρ [kg · m^{-3}] is the liquid density.

Values of the friction coefficient λ depend on the flow regime and the pipe character. Some calculation relations are presented in Tab 1.2.

Tab 1.2 Equations for calculation of the friction coefficient λ [5]

Laminar flow		Turbulent flow	
isothermal flow	$\lambda = \frac{64}{Re}$	for smooth pipe	$\lambda = 0.1\left(\frac{100}{Re} + \frac{k}{d}\right)^{0,25}$
non-isothermal flow (liquid temperature differs from ambient temperature)	$\lambda = \frac{75}{Re}$		
pipe with bend radius (lower value for higher bend radius)	$\lambda = \frac{82}{Re} \div \frac{155}{Re}$	for rough pipe	$\lambda = \frac{0,3164}{\sqrt[4]{Re}}$
hose with bend radius (lower value for higher bend radius)	$\lambda = \frac{80}{Re} \div \frac{108}{Re}$		

Power

The energy transfer in hydraulic (hydrostatic) mechanisms is carried out by means of a liquid. The energy transferred to the liquid in a hydraulic pump is given by the product of the liquid volume V and the liquid pressure p [8]. If the energy transferred is expressed in terms of power, it is possible to write:

$$P = \frac{V \cdot p}{t} = Q \cdot p \,, \tag{1.22}$$

where P [W] is the power, V [m^3] is the liquid volume, p [Pa] is the liquid pressure, t [s] is the time, and Q [m$^3 \cdot$ s^{-1}] is the liquid volumetric flow.

The power transfer in a hydraulic mechanism can be demonstrated using a block diagram (see Fig. 1.5).

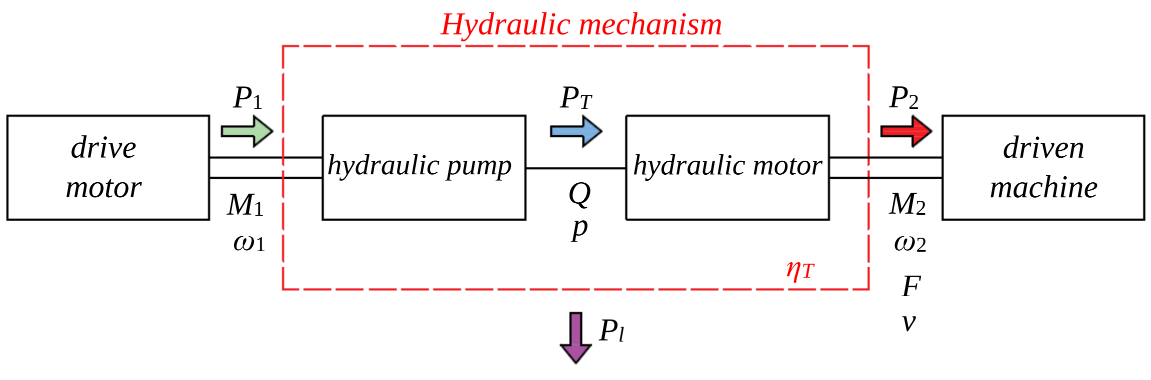


Fig. 1.5 Block diagram of power transmission in hydraulic mechanism

The drive motor of the hydraulic mechanism is usually an electric motor or an internal combustion engine whose power is defined by the rotary movement of the output shaft. The power input of the hydraulic mechanism (power of the drive motor) can be determined by the formula:

$$P_1 = M_1 \cdot \omega_1 = M_1 \cdot 2\pi \cdot n_1 \, , \tag{1.23}$$

where P_1 [W] is the input power of the hydraulic mechanism, M_1 [N · m] is the torque on the shaft of the hydraulic pump, ω_1 [rad · s^{-1}] is the angular velocity on the shaft of the hydraulic pump, and n_1 [s^{-1}] is the shaft speed of the hydraulic pump.

The transmitted power P_T can be described according to equation (1.22) as follows:

$$P_T = Q \cdot p \, , \tag{1.24}$$

where P_T [W] is the transmitted power, p [Pa] is the liquid pressure (or the pressure gradient), and Q [m^3 · s^{-1}] is the liquid volumetric flow.

Power losses occur during energy transfer in the hydraulic mechanism. The power loss P_l can be generally expressed by the equation:

$$P_l = Q_l \cdot p_l \, , \tag{1.25}$$

where P_l [W] is the power loss, Q_l [m^3 · s^{-1}] is the liquid flow loss, and p_l [Pa] is the pressure loss.

Therefore:

$$P_1 = P_T + P_l \, . \tag{1.26}$$

The output power of the hydraulic mechanism (power input of the driven machine) can be defined on the output component of the system, i.e., on the hydraulic motor. For the rotary motion of the output member, the output power is:

$$P_2 = M_2 \cdot \omega_2 = M_2 \cdot 2\pi \cdot n_2 \, , \tag{1.27}$$

where P_2 [W] is the output power of a hydraulic mechanism, M_2 [N · m] is the torque on the shaft of the hydraulic motor, ω_2 [rad · s^{-1}] is the angular velocity on the shaft of the hydraulic motor and n_2 [s^{-1}] is the shaft speed of the hydraulic motor.

For the linear movement of the output component, it is possible to write the formula:

$$P_2 = F \cdot v \, , \tag{1.28}$$

where F [N] the force on the piston rod of the hydraulic motor and v [m · s^{-1}] is the movement velocity of the hydraulic motor.

The total efficiency η_t of a hydraulic mechanism can be expressed as the ratio of the output power to the input power, according to the equation:

$$\eta_T = \frac{P_2}{P_1} \, , \tag{1.29}$$

where η_T [-] is the total efficiency of a hydraulic mechanism.

1.2 Applications and fields of use of the hydraulic mechanism

Hydraulic mechanisms are an integral part of machinery and equipment and are used in many different areas. The main areas of their application are:

- manufacturing machinery and equipment (machine tools and forming machines),
- machinery for earth and building works (loaders, excavators, backhoes, road rollers),
- machinery in metallurgy (blast furnaces, rolling mills, foundries),
- machinery for surface and mining operations (hydraulic reinforcements, hand tools, harvesters, mining locomotives),
- transport and manipulation equipment (forklifts, conveyors, rope winches, mobile cranes),
- agricultural and forestry machinery (tractors, mowers, harvesters, hydraulic grabs and arms),
- power engineering (control of hydropower and nuclear power plant dampers, tilting of wind power plant blades, tilting of solar panels),
- road transport (hydraulic brakes, power steering, shock absorbers, truck body tilt control),
- automotive industry - (progressive presses, high- and low-pressure presses, injection presses),
- rail transport (special railway machines - tampers, grinders, cleaners),
- aviation and rocketry (hydraulic control systems in aircraft, aircraft tractors, high-lift platforms, etc.),
- cultural and entertainment facilities (theatres, carousels),
- part of robots and manipulators,
- laboratories and testing rooms,

and many other applications.

The main advantages of hydraulic mechanisms are:

- to achieve high forces and torques,
- high specific power, compact dimensions, and low weight,
- the simplest conversion of rotary motion to linear motion,
- the ability to achieve high torques at operating speeds without gearing in low-speed hydraulic motors,
- easy energy distribution even to hard-to-reach areas of machines and equipment,
- simple and precise continuous control and regulation of output parameters,
- good dynamic properties of hydraulic drives, a simple reversal of movements and simple overload protection,
- possibility of connection to automated systems of working and production machines and equipment.

If the advantages of hydraulic mechanisms are mentioned, the basic disadvantages of these systems should also be mentioned:

- susceptibility to impurities in working liquid,

- sensitivity to temperature changes in the working liquid (significant liquid heating during system operation),
- relatively low efficiency,
- leakages in hydraulic mechanisms can lead to leakages of working liquid (possible negative environmental impacts).

1.3 Arrangement of hydraulic circuits

Hydraulic mechanisms are characterized by the circuit arrangement of individual elements. According to the location and arrangement of these elements, so-called open and closed circuits can be distinguished.

Open circuit

The open circuit is characterized by the serial integration of the tank between the hydraulic pump and the hydraulic motor, whereby all the liquid from the hydraulic motor is returned to the tank. The tank is typically dimensioned for several times the minute volumetric flow of the hydraulic pump. Open circuits are more commonly used in stationary hydraulics applications where larger tank sizes are not a problem. The tank has the function of a reservoir of working liquid, ensuring the possibility of level oscillation (differential hydraulic motors and accumulators), liquid stabilization, passive cooling, etc. Filtration and cooling of the working liquid are easier to solve with open circuits. An example of an open hydraulic circuit is shown in Fig. 1.6.

The source of pressure energy is the hydraulic pump (1), which sucks liquid from the tank (2) and supplies it to the hydraulic circuit. The liquid is carried through lines (pipes and hoses) to consumers, where the liquid's pressure energy is converted into mechanical energy, and the corresponding motion is generated. In this case, the consumer of the pressure energy is the linear hydraulic motor (3), which produces a linear motion. The inner part in the body of the hydraulic motor is separated by a piston. When the liquid is supplied to one or the other side of the motor (piston), the piston and piston rod of the hydraulic motor can be moved in the appropriate direction. The change in the direction of the liquid flow and, therefore, the movement of the hydraulic motor can be controlled by means of the distributor (4). In this case, it is a distributor of spool construction, which is manually operated by a lever. By changing the position of the lever, the slide valve moves in the distributor body, and different ways (input and output channels) are connected. The relief valve (5) is connected in parallel to the pressure source (hydraulic pump). The relief valve protects the circuit against an excessive increase of liquid pressure and overloading of the system; it is usually set to the so-called safety (maximum) pressure. When this pressure is exceeded, the relief valve will be open, and the liquid will flow back into the tank. However, the liquid flow through the relief valve relates to the conversion of the pressure energy into heat. The liquid is heated, and the energy is dissipated.

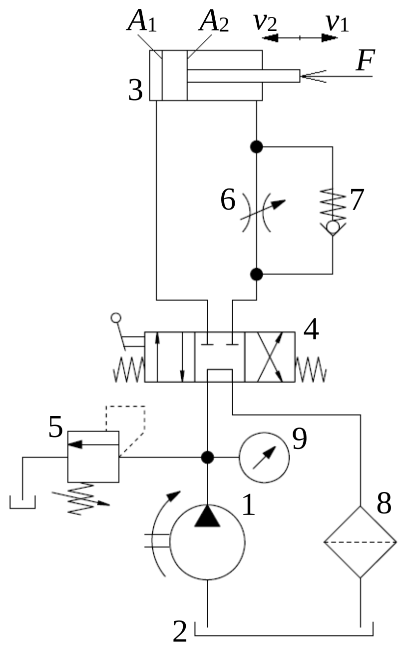


Fig. 1.6 Schematic of an open hydraulic circuit

1 – hydraulic pump, 2 – tank, 3 – hydraulic cylinder, 4 – directional valve, 5 – relief valve, 6 – throttle valve, 7 – check valve, 8 – return line filter, 9 - manometer

The movement velocity of the piston rod of the hydraulic motor depends on the volumetric flow of the liquid supplied to the hydraulic motor. By changing the size of the liquid volumetric flow, a change in the movement velocity of the piston rod of the hydraulic motor can be achieved. In the above circuit, the velocity change of the piston rod extension (v_1) is realized by means of the throttle valve (6). The throttle valve works on the principle of changing the flow area between the valve seat and the valve cone. The valve closing results in a reduction of the flow area and, thus, a decrease in the velocity of the extending piston rod of the hydraulic motor. Also, the liquid flow through the throttle valve is related to the conversion of part of the liquid's pressure energy into heat. In addition, the difference between the flow rate supplied by the hydraulic pump and the flow rate supplied to the hydraulic motor flows back into the tank through the relief valve, which is related to an additional energy loss. The retracting velocity of the piston rod of the hydraulic motor (in direction v_2) is not controlled in this case. When the piston rod of the hydraulic motor is moved to the left, the throttle valve is disabled, which is ensured by the check valve (7). The check valve allows the liquid to flow in one direction only. In the circuit shown, it is placed parallel in the branch to the throttle valve. When the piston rod of the hydraulic motor is moved in direction v_2, the liquid flows through the check valve. The low-pressure filter (8) is placed in the discharge branch to ensure continuous filtration of the working liquid when the system is in operation. The pressure in the hydraulic circuit is monitored using the manometer (9).

Closed circuit

A closed hydraulic circuit is characterized by connecting the output of the hydraulic motor to the input of the hydraulic pump. An auxiliary tank is connected to the circuit in parallel. Its volume is considerably lower compared to open circuits, usually only (20 ÷ 30) % of the minute flow of the hydraulic pump. Closed circuits are mainly used in mobile hydraulics applications, but they are also used in industrial hydraulics. In these systems, cooling and filtration of the working liquid are more difficult to solve. Leakages in hydrostatic converters are replenished in the main circuit by an auxiliary hydraulic pump via check valves. Rotary or linear hydraulic motors with a continuous piston rod can be used in a closed circuit. The motion reversal of the hydraulic motor is usually performed by a variable displacement hydraulic pump with liquid flow in two directions [6], [7].

An example of a closed circuit is shown in Fig. 1.7. The pressure energy source is the variable displacement hydraulic pump (1). The consumer is the rotary hydraulic motor (2). Both converters work in two directions. The motion reversal is realized by the hydraulic pump (e.g., by changing the angle of the hydraulic pump control plate). The circuit includes filling and safety blocks and a discharge block. In the filling and safety blocks, there is a pair of relief valves (4), by means of which the maximum pressure in the system is limited. When the circuit is operated, one branch is always high pressure, and the other branch is low pressure. The working liquid is always supplied from the tank to the low-pressure branch of the circuit by the hydraulic pump (3) through the check valve (11). The relief valve (6) is connected after the auxiliary hydraulic pump (3) to limit the maximum filling pressure (usually 20 ÷ 30 bar). The low-pressure filter (10) may also be part of the filling branch. The discharge block consists of a 3/3 hydraulically operated distributor (7) and the relief valve (5). The liquid is discharged into the tank from the low-pressure branch of the main circuit through the orifice (12). The liquid cooling is provided by the cooler (8), after which the low-pressure filter (9) is located.

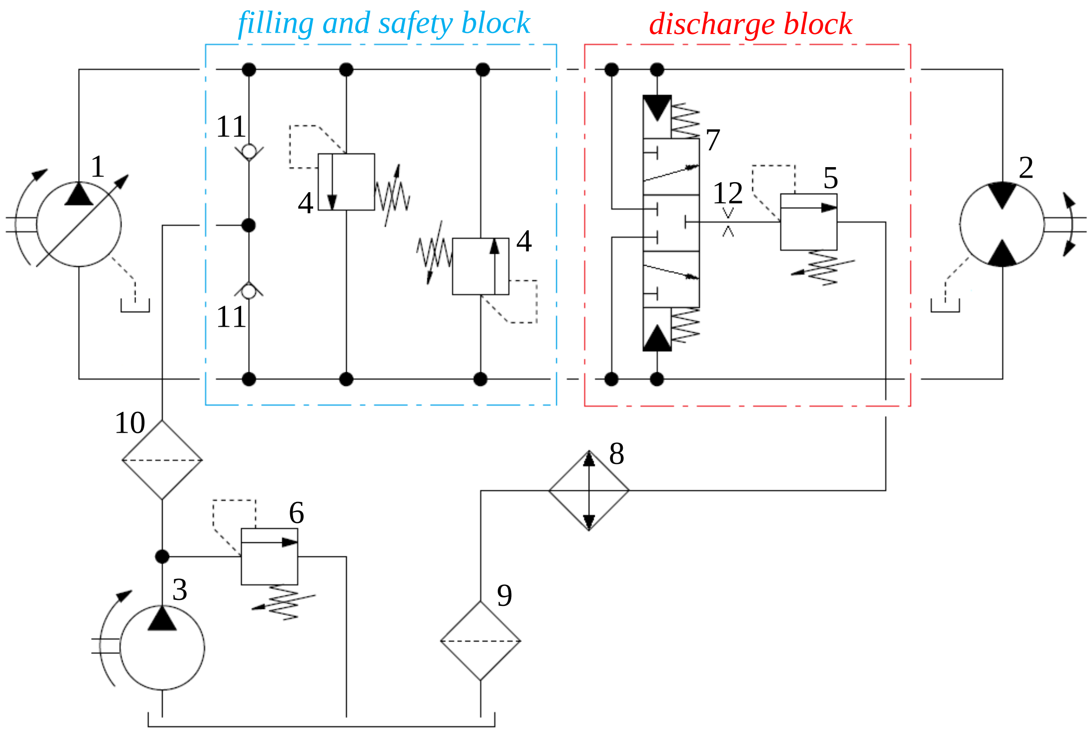


Fig. 1.7 Schematic of a closed hydraulic circuit

1 – variable displacement hydraulic pump, 2 – rotary hydraulic motor, 3 – auxiliary hydraulic pump, 4 – relief valve, 5 – by-pass valve, 6 – relief valve auxiliary hydraulic pump, 7 – directional valve, 8 – cooler, 9 – low-pressure filter, 10 – low-pressure filter, 11 – check valve, 12 - orifice

2. Properties of liquids

Liquid is an integral part of any hydraulic system. It performs a number of functions, and its condition significantly affects its service life. The primary function of a liquid is to transfer energy. In most applications, it is also used to transmit information or a signal that can be used to control the system. During the operation of hydraulic mechanisms, significant changes in pressure and liquid flow occur. When the liquid flows, friction is created, which increases its temperature. Most properties of liquids change with temperature and pressure. In addition, it is necessary to keep the liquid inside the system during operation. This is achieved by using a suitable type of seal at the joints of individual components and moving parts. Seals are usually made of flexible materials and must not be aggressively affected by the liquid in the whole range of operating pressures and temperatures. During the operation of a hydraulic system, there is also a movement of partial parts of individual elements (e.g., movement of distributor slide, piston, the piston rod of a hydraulic cylinder, pistons of a hydraulic pump, bearings, etc.). In these cases, the liquid must ensure lubrication of functional surfaces and prevent dry or boundary friction. All these aspects and many others need to be considered when designing a hydraulic system. The following chapters will describe the basic physical and chemical properties of fluids and present an overview of the most commonly used working fluids in hydraulics.

2.1 Density

Density, or specific mass ρ, is the basic quantity defining liquids. It is the weight of a liquid relative to its volume:

$$\rho = \frac{m}{V}, \tag{2.1}$$

where ρ [kg · m^{-3}] is the liquid density, m [kg] is the liquid mass, and V [m^3] is the liquid volume.

The density of liquids varies with pressure and temperature. Its change is due to thermal expansion or volume compressibility of liquids. The dependence of mineral oil density on pressure and temperature is shown in Fig. 2.1. The liquid density increases with increasing pressure. On the contrary, the liquid density decreases with increasing temperature. The liquid density is measured using densitometers.

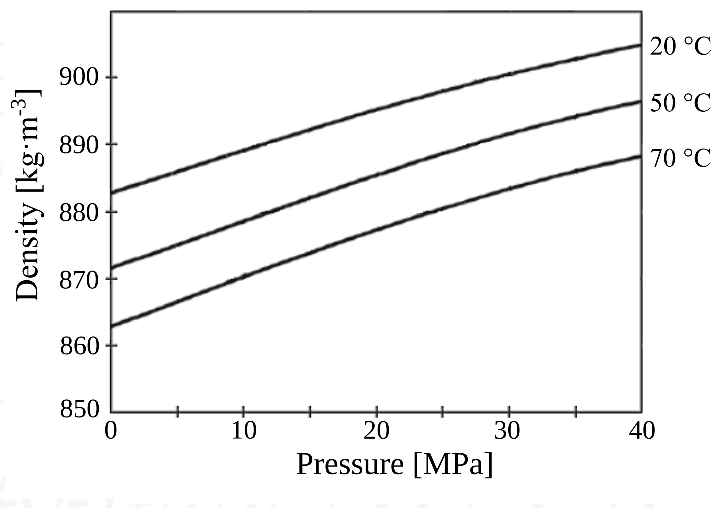


Fig. 2.1 Change in mineral oil density depending on pressure and temperature changes

The density occurs in a number of relations describing the flow of liquids. Its change with pressure and temperature is not so significant under normal conditions and is often considered constant in practical calculations.

The density of selected liquids is given in Tab 2.1. The values are considered at atmospheric pressure and a temperature of 20 °C (the exception is water, for which a value of 4 °C is used in the basic calculations).

Tab 2.1 Density of selected liquids

Liquid	Density ρ [kg · m^{-3}]
water	1 000
mineral oil	890 ÷ 960
synthetic liquids	950 ÷ 1 400
petrol	700 ÷ 750
mercury	13 547

2.2 Viscosity

Viscosity is a measure of internal friction that occurs when a fluid flows. Internal friction induces a tangential shear stress τ at the interface of the liquid flow layers, which is proportional to the change in velocity in the direction perpendicular to the liquid flow direction (see Fig. 2.2) [9], [13]. Based on Newton's law:

$$\tau = \eta \cdot \frac{dv}{dy}, \tag{2.2}$$

where τ [Pa] is the tangential stress, η [Pa · s] is the dynamic viscosity, and dv/dy [s^{-1}] is the velocity gradient in a direction perpendicular to the direction of motion.

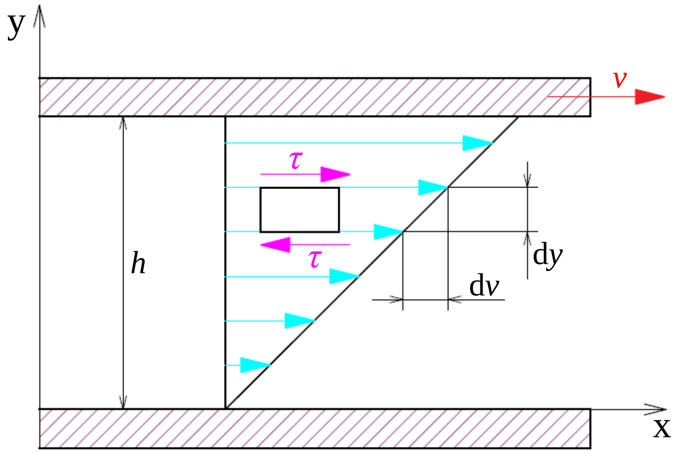


Fig. 2.2 Shear stress for laminar flow

In the equation (2.2), the internal friction of a liquid is represented by the dynamic viscosity η. In practice, however, the kinematic viscosity ν is more often used, which is given by the ratio of the dynamic viscosity to the liquid density according to the formula:

$$\nu = \frac{\eta}{\rho} \,, \tag{2.3}$$

where ν [$m^2 \cdot s^{-1}$] is the kinematic viscosity, η [Pa · s] is the dynamic viscosity and ρ [$kg \cdot m^{-3}$] is the liquid density.

The **viscosity** of liquids varies with **temperature** and **pressure.** The viscosity increases with increasing pressure. On the contrary, the liquid viscosity decreases with increasing temperature (but it is the opposite in the case of gasses; i.e., the viscosity increases with increasing temperature and decreases with increasing pressure). A liquid with a higher viscosity has a higher resistance to flow, leading to increased pressure losses in a system. Therefore, more energy must be expended for its movement. Conversely, at low viscosity values, internal friction decreases, and flow losses increase significantly. In addition, the magnitude of the viscosity also affects the lubrication of moving parts, and the Reynolds number depends on it. The dependence of the liquid viscosity on the temperature can be expressed using the **viscous curve** (see Fig. 2.3). The mentioned kinematic viscosity is such a fundamental and characteristic quantity that its value is often found in the designation of the working liquid itself. For mineral oils intended for hydraulic mechanisms (so-called industrial oils), viscosity grades are distinguished according to ISO 3448 with the designation VG (engine and gear oils are defined by SAE). The viscosity class number represents the kinematic viscosity [$mm^2 \cdot s^{-1}$] of oil at the temperature of 40 °C. There are a total of 18 viscosity classes according to the ISO standard; hydraulic oils fall into the VG10 to VG100 classes. The most commonly used are oils of viscosity classes VG22 (operation in arctic conditions), VG32 (winter operation), VG46 (summer operation), and VG68 (tropical conditions, circuits with high heat generation) [10].

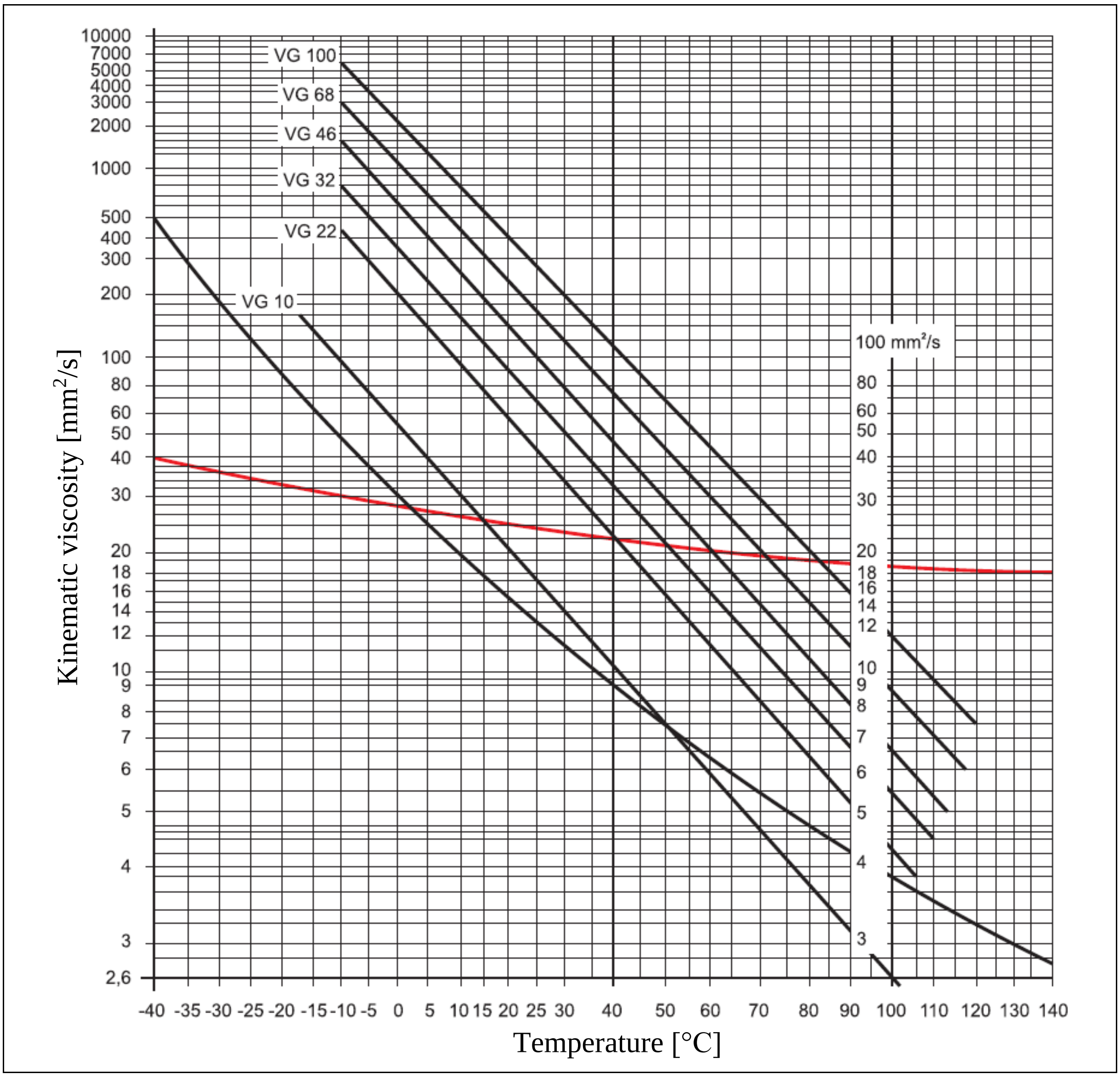


Fig. 2.3 Viscous curves of mineral oils [11]

The degree of the temperature dependence of the viscosity is given by the so-called viscosity index. The higher the viscosity index of a liquid, the flatter its viscosity curve and the less its dependence on temperature. Ideally, a liquid with as high a viscosity index as possible should be selected; for oils, a viscosity index of at least 90 is required (the viscosity index of the liquid can be increased by using additives).

When choosing a liquid, it is necessary to consider permissible values of viscosities of individual elements of a hydraulic system. These values are listed in manufacturers' catalogues. For example, up to four different values are commonly reported for hydraulic pumps. The **starting viscosity** is the maximum viscosity value at which the hydraulic pump can safely suck up liquid. It is obvious that it is mainly about starting the hydraulic pump at low ambient temperatures. This value varies for different design types of hydraulic pumps. It is suitable to heat the liquid at low temperatures before starting the hydraulic system, for example, by using heating elements. The **operating viscosity** indicates the range of viscosities suitable for the long-term operation of a system. The **optimum viscosity** range defines where a hydraulic pump operates with the highest efficiency. The **short-term minimum viscosity** represents a limiting value especially at high liquid temperatures. This is only a temporarily allowed minimum viscosity value. In this case, the lubricating ability of the liquid is significantly reduced, and there is a risk of seizing the hydraulic pump and, thus, its irreversible damage. The consequences of non-compliance with operating parameters (minimum viscosity) in an axial piston pump in swashplate design are shown in Fig. 2.4. The reduced lubricating ability of liquid caused an increase in friction between slipper pads and a swashplate of the hydraulic pump.

Fig. 2.4 Damage to the axial piston hydraulic pump due to a reduced lubricating liquid ability

Approximate values of the recommended viscosity of working liquids, according to the manufacturers of hydraulic pumps, are given in Tab 2.2 [12].

Tab 2.2 Approximate values of viscosity of working liquids

System operation	Construction design of hydraulic pump	Kinematic viscosity ν [$mm^2 \cdot s^{-1}$]
starting viscosity	vane hydraulic pumps	800
	piston hydraulic pumps	1 000
	gear and screw hydraulic pumps	2 500
operating viscosity	generally	16 ÷ 150
optimal range of viscosity	generally	16 ÷ 36
short-term minimum viscosity	generally	10
	vane hydraulic pumps	13

2.3 Volume compressibility

Volume compressibility is generally the ability of a liquid to **decrease** its **volume** when the external **pressure increases**. The volume decrease ΔV, which is caused by the increase in pressure Δp, is expressed using the coefficient of volume compressibility δ as follows:

$$\delta = \frac{\Delta V}{V \cdot \Delta p}, \tag{2.4}$$

where δ [Pa^{-1}] is the coefficient of volume compressibility, ΔV [m^3] is the volume change, V [m^3] is the original liquid volume at the pressure p, and Δp [Pa] is the pressure change.

The volume change ΔV is:

$$\Delta V = V_0 - V, \tag{2.5}$$

where V_0 [m^3] is the liquid volume after increasing the pressure to the value p_0.

And the pressure change Δp is:

$$\Delta p = p_0 - p, \tag{2.6}$$

where p [Pa] is the original pressure at the liquid volume V and p_0 [Pa] is the value of the increased pressure.

The coefficient of volume compressibility δ is equal to the inverse of the bulk modulus of liquids K (2.7). Both coefficients represent a measure of the stiffness of the liquid volume and can be characterized similarly to, for example, the Young’s modulus of elasticity of steels [2].

$$K = \frac{1}{\delta}, \tag{2.7}$$

where K [Pa] is the bulk modulus of liquids.

The bulk modulus of liquids varies with temperature and pressure, as shown in Fig. 2.5 [13]. However, this change is significant only with relatively large changes in temperature and pressure, and in standard calculations, the modulus of elasticity is considered constant. Some indicative values of the bulk modulus are given in Tab 2.3.

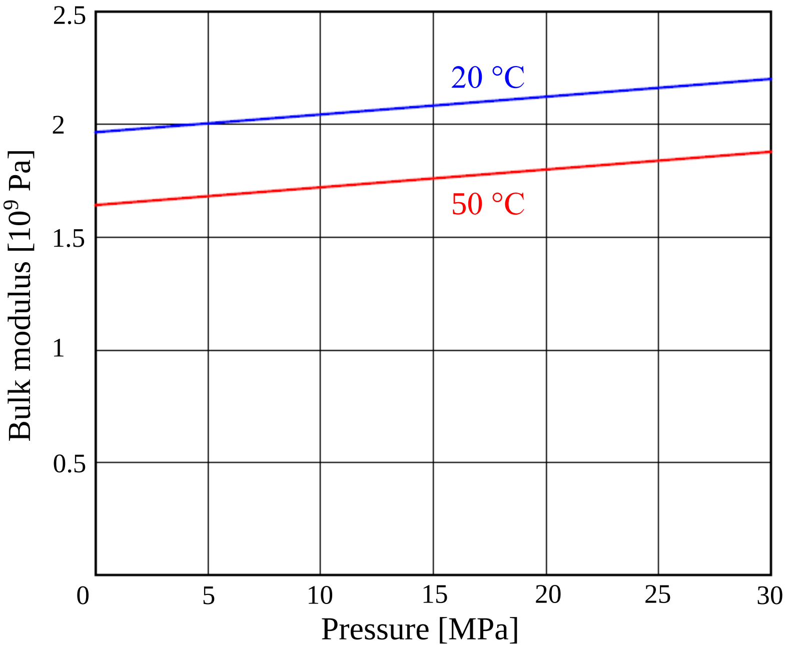


Fig. 2.5 The change in the bulk modulus of mineral oil depending on the change in temperature and pressure

Tab 2.3 Bulk modulus of selected substances [3]

Substance	Bulk modulus K [Pa]	Substance	Bulk modulus K [Pa]
water	$2 \cdot 10^9$	liquid HFD	$2 \cdot 10^9$
mineral oil	$1.4 \div 1.8 \cdot 10^9$	air	$1.4 \cdot 10^6$
liquid HFC	$3 \cdot 10^9$	steel	$2.1 \cdot 10^{11}$

The air content has a significant influence on the compressibility of liquids. Air occurs in liquids in two forms, namely dissolved and undissolved. The volume of dissolved air in oil at normal temperature and atmospheric pressure is approx. (8 ÷ 9) %, for water it is approx. 2 %. The air solubility in oil depends on pressure and temperature (the solubility increases with pressure and decreases with temperature) and is determined by the saturation state, which is proportional to the pressure at the liquid surface. The dissolved air does not affect the liquid compressibility. When the pressure in the system decreases below atmospheric pressure (approx. p_{abs} = 60 kPa),, e.g., at spool edges of distributor valves or in a suction line of hydraulic pumps, dissolved air is released in the form of bubbles (i.e., undissolved air). The higher the underpressure, the more intensively the dissolved air is released, which can lead to the formation of an air-oil mixture. The release of dissolved air from a liquid (oil) under negative pressure is called aeration (so-called “false” cavitation). The subsequent air dissolution is much slower compared to the release process, which results in a part of the air volume circulating in a system. In places with a large increase in pressure (e.g., on the discharge side of hydraulic pumps), these bubbles are compressed and subsequently implode, which can be manifested similarly to the Diesel effect and causes so-called cavitation erosion of surfaces of hydraulic elements [12].

Examples of cavitation damage to internal parts of hydraulic pumps are shown in Fig. 2.6. In the left part, the face plates of the gearing of gear hydraulic pumps are shown. Damage on the distribution plate surface of the axial piston hydraulic pump is shown on the top right. In the bottom right, material damage is visible in the discharge part of the hydraulic pump.

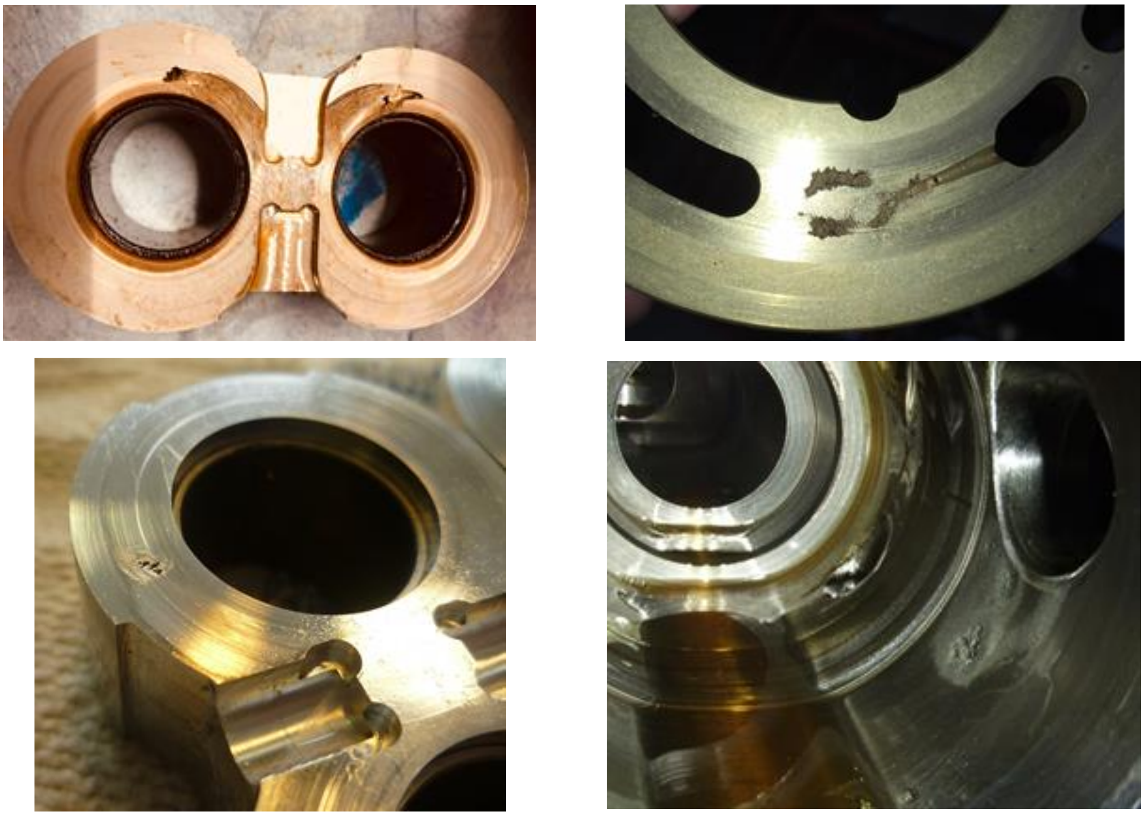


Fig. 2.6 Cavitation damage to internal surfaces of hydraulic pumps

Undissolved air also significantly affects liquid compressibility and can lead to premature deterioration of the liquid. In addition, heat transfer deteriorates, noise increases, and jerky movements and vibrations in the system can occur.

Air from mineral oil cannot be completely excluded. In order to prevent the above-mentioned phenomena, it is necessary to minimize the amount of underpressure in the suction pipe. This can be achieved by placing the hydraulic pump near the liquid tank, ideally as close to the liquid level as possible (or entirely below the level). It is also important to dimension the suction pipe correctly. Air must not be sucked in when the liquid level in the tank decreases.

2.4 Thermal expansion

Thermal expansion is the ability of a liquid to increase its volume when its temperature increases. This change is characterized by the thermal coefficient of expansion β:

$$\beta = \frac{\Delta V}{V \cdot \Delta t}, \tag{2.8}$$

where β [°C^{-1}] is the thermal coefficient of expansion of liquids, Δt [°C] is the change in liquid temperature, ΔV [m^3] is the change in liquid volume, and V [m^3] is the original liquid volume at the original temperature t.

When the liquid temperature increases by the value Δt, the liquid volume increases by the volume $\Delta V = V_0 - V$. The volume V is the original liquid volume, and V_0 is the volume after the temperature increase [2].

When liquid flows in a hydraulic system, the liquid is heated due to friction. The warming is even more pronounced in places where the flow area is narrowed and the flow is throttled. However, from the point of view of the whole system, this is not a sudden temperature change that would have a significant effect on its operation. Thermal expansion is of practical importance when sizing tanks, which must also accommodate an increased liquid volume during the equipment operation.

2.5 Specific heat capacity

It describes the ability of a liquid to receive and accumulate heat. Its value changes with temperature, and in hydraulic systems, it is important in calculations of heating and cooling of the circuit or in dimensioning of heat exchangers.

$$c = \frac{\Delta W}{m \cdot \Delta T}, \tag{2.9}$$

where c [J · kg^{-1} · K^{-1}] is the specific heat capacity of liquid, ΔW [J] is the change in internal thermal energy of the liquid, m [kg] is the weight of liquid, and ΔT [K] is the change in liquid temperature.

Indicative values of the specific heat capacity at 20 °C are given in Tab 2.4. These values simply show how much thermal energy is needed to heat 1 kg of a given substance by 1 °C.

Tab 2.4 Values of the specific heat capacity of selected substances

Substance	Specific heat capacity c [J · kg^{-1} · K^{-1}]
water	4 180
mineral oil	1 850
synthetic liquid HFC	3 000
steel	470
copper	390

2.6 Flash point

The point of liquids is determined in a test vessel. The liquid gradually heats up, and vapours are formed above the liquid level in the vessel. The flash point corresponds to the temperature at which vapours ignite after approaching a flame but do not burn permanently. In general, liquids are divided into four flammability classes. Hydraulic oils belong to the last flammability class IV (with a flash point between 100 and 250 °C). Nevertheless, there are strict safety regulations that limit the use of hydraulic systems with oils near open flames, hot metal, or in mines, for example. In these cases, it is necessary to use one of the fire-resistant working liquids. In addition to the flash point, the fire point is also determined for liquids. This corresponds to the liquid temperature, at which the vapours develop so intensively that they burn permanently after approaching the flame (the flash point has no practical meaning for hydraulic systems).

2.7 Freezing point

The freezing point of mineral oils is the temperature at which the viscosity increases so much that the oil stops flowing. For standard applications and when operating under normal conditions, the freezing point is not significant. It must be respected in hydraulic systems that are exposed to very low temperatures. However, the lowest operating temperature is usually determined by the starting viscosity of a given hydraulic pump. The freezing point can be modified by additives called depressants. For industrial oils, the pour point is sometimes also mentioned. It is approximately (4 ÷ 6) °C above the freezing point and represents the temperature at which the oil is still flowing [12].

2.8 Lubricating ability

There are a significant number of internal moving parts in hydraulic systems. These parts must be manufactured with great precision and very little clearance to ensure functionality while minimizing flow losses. During operation, there must be no metal-to-metal contact (so-called dry or mixed friction), significantly increasing wear and metal abrasion, inevitably leading to a system failure during long-term operation. Therefore, it is necessary to create a thin, continuous, and sufficiently strong layer of lubricating film between the friction surfaces. This is one of the many important tasks of liquid in a hydraulic circuit. The liquid forms a thin layer of lubricant on the sliding surface. This layer has high shear strength and ensures the separation of the friction surfaces even when they are in contact with each other. The resulting friction is called liquid friction or viscous friction. This property can be defined as the lubricating ability of the liquid. The natural lubricating ability of base mineral oils is only up to operating pressures of approx. 16 MPa. For higher pressures, lubricity or anti-wear additives are added to oils to improve lubricating ability. For extremely high pressures and shear stresses, high-pressure additives are added, which bind to the metal surface chemically (chemical adsorption). Ensuring a lubricating film is especially necessary for hydraulic pumps and hydraulic cylinders that are operated at high pressures [12], [14]. The clearances for the formation of a lubricating film in hydraulic elements are shown in Fig. 2.7.

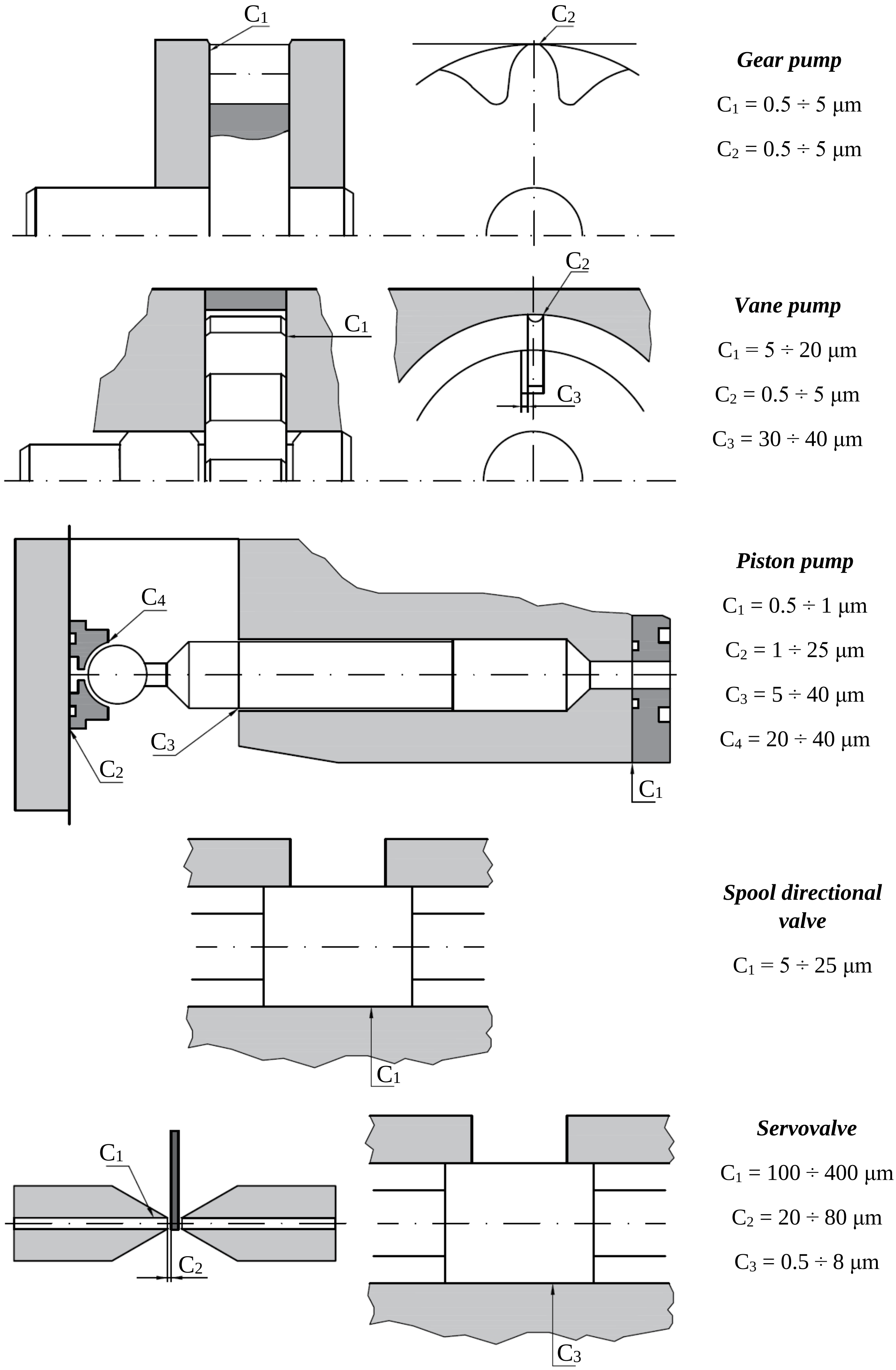


Fig. 2.7 Clearance sizes for lubricating film formation in hydraulic elements

2.9 Foaming of liquids

The foaming of liquids is an undesirable phenomenon. It is primarily created by the rise of air bubbles to a tank level. Content of water and other impurities in mineral oil can also increase the foaming intensity. Foam stability is related to the surface tension of a liquid. The air is separated from the liquid in the tank by sieves, and the foam is separated from the suction part by suitably placed baffles. The correct location of suction and return (waste) lines in the tank is also important. Air generally accelerates oil ageing and reduces the strength of the lubrication layer. To reduce the foaming of oils, additives called polysiloxanes are used [3], [12].

2.10 Water content

In the case of anhydrous liquids, the effect (and contamination) of the working liquid with water must also be considered. Water reduces fluid viscosity, and corrosion resistance and promotes oxidation and eventually rot. If water is contained in the oil, sediments can form in the tank (water has a higher density, and it sticks to the bottom of the tank). At low temperatures, ice crystals can form in the liquid, which can damage functional parts of hydraulic pumps or valves or clog control systems and filter inserts. The water content of oil cannot be completely avoided. It depends on the air temperature and humidity. A certain amount of water can be absorbed by the oil. The maximum amount is called the saturation level. If this level is exceeded, the oil will become cloudy.

When the pressure drops below the pressure of saturated water vapour, so-called cavitation bubbles of water vapour are formed. This process is accompanied by a significant increase in temperature and subsequent bubble implosion. When the bubble comes into contact with a device wall, cavitation erosion and surface damage to the hydraulic pump occur. This phenomenon, known as **cavitation**, has negatively affects oil hydraulics, especially on distributor plates of hydraulic pumps. In water hydraulics, the impeller blades of hydrodynamic pumps and turbines are particularly exposed to cavitation. Cavitation is accompanied by increased noise, vibration and shocks in systems and leads to damage to hydraulic pumps. Of course, the risk of cavitation is much higher for systems working with water emulsion and synthetic fluids containing water [12], [16].

2.11 Stability against oxidation

Hydraulic liquids, mineral oils and, in general, petroleum products have a natural chemical stability that depends on the type of liquid but also on the method of production. Under the influence of air, light, heat, radiation or chemical substances, oxidation processes occur in liquids. Under the term oxidation stability, it is possible to imagine the resistance of a liquid against oxidation and thus the so-called ageing of the liquid. The ageing of oils leads to their thickening, the formation of sludge and sticky deposits, and subsequently, it can lead to the clogging of pipes, control elements or filter inserts. The ageing is also negatively affected by water content, dirt, abrasion (especially from copper and its alloys) and contact with metal surfaces. Temperature is an essential factor that significantly accelerates oxidation processes. It is reported that at temperatures above 70 °C, oxidation reactions double with every 10 °C increase in temperature. Oxidation is also manifested by an increase in the acidity of the liquid. This can be used to diagnose the oil condition. The acid number is the amount (mass in milligrams) of potassium hydroxide (KOH) needed to neutralize one gram of oxidatively degraded oil. The acid number is data that, together with other factors (oil pollution, viscosity change, water content), indicates the ageing degree of oil and the moment of its replacement. Additives called antioxidants are used to increase the oxidation resistance of liquids [11], [12].

2.12 Corrosive action on metals

This is the chemical action of liquids on metal parts of a hydraulic system, especially on steel and copper. Water is corrosive to standard carbon and low alloy steels. In the case of using water, water emulsions, or synthetic liquids containing water, it is necessary to add so-called corrosion inhibitors to the working liquid. They create a protective layer on a metal surface and prevent corrosion from penetrating in depth. In addition to damaging metal surfaces themselves, rust can also act as a catalyst to accelerate oil oxidation reactions. Mineral oils are non-corrosive; on the contrary, they protect metal surfaces. However, some moisture is contained in every system. Water can enter the circuit by simple condensation of water vapour or, for example, by leakage from a cooling circuit. In such cases, the two liquids are mixed, and a water-in-oil emulsion is formed, which significantly reduces corrosion protection. In addition, at high operating temperatures, significant oxidation processes can occur in the liquid, leading to the formation of acidic waste products. These products are also corrosive to some metals. For these reasons, demulsifying (to prevent the formation of emulsions) and anti-corrosive additives are added to mineral oils of higher classes to increase protection [12], [14].

2.13 Compatibility with elastomers

Hydraulic hoses and sealing elements, which are an essential part of every circuit, are often made of elastomers. Different types of elastomers are used depending on the required properties and operating parameters of a given hydraulic system. The term compatibility of liquids with elastomers means the non-aggressive action of a liquid on these materials in such a way that their properties do not change in the full range of operating temperatures and pressures. Acrylonitrile butadiene rubber NBR is most commonly used for mineral oils and FPM (fluoroelastomers based on vinylidene difluoride) rubber (Viton) for synthetic liquids. When designing a hydraulic system, the mutual compatibility of the liquid and the used sealing material must always be verified, and the instructions of the manufacturers of hydraulic elements and seals must be followed.

2.14 Physiological and ecological properties of liquids

The physiological and ecological effects of working fluids on the environment and human health are currently an increasingly important topic. If water, which is ecological and harmless, is neglected, then all other liquids are harmful in some way.

From a physiological point of view, skin contact with the liquid can lead to skin diseases, and at higher working temperatures, there can be intense evaporation of harmful substances. Mineral oils contain small amounts of carcinogenic hydrocarbons, which limit their use in agriculture. Some liquids are so harmful that their use is completely prohibited [12], [14].

From ecological properties, the biological degradability and the water hazard class are mainly evaluated. In the case of biodegradability, the time taken for substances to decompose into water and carbon dioxide is determined. When water is contaminated, a thin film of oil forms on the level and prevents the dissolution of oxygen, which can lead to the death of aquatic animals. Petroleum products are classified as moderately to highly dangerous. For these reasons, there is an effort to replace mineral oils with less environmentally harmful liquids [12], [14], [15].

3. Liquids of hydraulic circuits

A number of factors must be considered when choosing a working liquid. First of all, the question is where and under what conditions the equipment will be operated. Requirements for systems operating in potentially explosive atmospheres will be different compared to equipment operating in open spaces. Furthermore, it is necessary to consider the function of a given hydraulic system. In general, the basic requirements for hydraulic liquids include good lubricity, oxidation stability, anti-corrosive action, suitable viscosity and high viscosity index. Depending on the situation, additional requirements may arise for the liquids, such as non-flammability (more precisely, higher resistance to combustion) or environmental safety. Last but not least, price is also an important deciding factor.

Most liquids cannot fully meet all these requirements, but some liquid properties can be partially modified. Ingredients (or additives) are used to modify and improve properties.

Water is a special chapter. **Water** is used in a number of applications where kinetic energy is primarily used to transfer power. Such mechanisms are referred to as hydrodynamic (centrifugal pumps, turbines). This is a separate scientific field and is not part of this study text. From the point of view of hydrostatic mechanisms, water itself is basically not used. This is mainly due to its low viscosity, poor lubricity, corrosive effect on metals and high freezing point (the change of state occurs at 0 °C). On the other hand, water is relatively cheap and available, non-flammable and ecological. For these reasons, various mixtures and emulsions of other liquids with water are formed, and some of them will be mentioned later [20].

Hydraulic liquids can be divided into three basic groups according to their use:

- **mineral oils** (hydrocarbons of petroleum origin) or oils of other bases are predominantly used in conventional systems without any special requirements for flammability and ecological safety,
- in applications with increased requirements for non-flammability, so-called **non-flammable liquids** are used,
- **ecological liquids** are used in applications where ecological safety is required.

3.1 Mineral oils

Mineral oils are extracted from petroleum and are the product of distillation, condensation and subsequent refining (or upgrading). The resulting properties of oils depend on the length of hydrocarbon molecules, and the degree of refining and are further modified by the addition of additives. Mineral oils for use in hydraulic systems are classified according to ISO 11158 [17] into five groups.

HH (previously marked H) – are mineral base oils without added additives. These oils are prone to oxidation; they were used in the past in simple hydraulic systems operating in areas of lower to medium pressures, without increased requirements for viscosity and lubricity. The range of permitted working temperatures is (-10 ÷ 90) °C. The seal was usually made of NBR (acrylonitrile butadiene) rubber. Currently, they are practically not used.

HL – mineral oils with anti-oxidation and anti-corrosion additives. They are characterized by improved corrosion protection and higher resistance to ageing. It is also possible to use them in circuits with greater heat generation (typically circuits with throttle valves) and higher circulation numbers. Lubrication ability is not modified; they are suitable for applications with standard lubrication requirements. Working pressure up to 20 MPa, operating temperatures in the range of (-10 ÷ 90) °C, seals used are usually made of NBR or FPM (fluoroelastomers based on vinylidene difluoride) rubber. They are aggressive to lead. Typical applications are steel mills or rolling mills.

HM (HLP) – are mineral oils of the HL group, supplemented with anti-wear additives. These additives significantly reduce the mechanical wear of internal parts. They are suitable for systems with higher operating pressures (20 ÷ 40) MPa, with an operating temperature range of (-20 ÷ 90) °C. Seals are usually made of NBR or FPM rubber. Like the oils of the previous group, they are aggressive towards lead. Typical applications are systems with highly loaded components, and machines with high mechanical and thermal stresses operating all year round,, e.g., presses and die-casting machines.

HV – oils identical to the HM group supplemented with viscosity modifiers. They have a high viscous index and, therefore, a flatter viscosity curve. They are used in applications where there are large changes in ambient temperatures, typically as year-round oil filling of mobile construction machinery or in shipping. They have a wider range of operating temperatures (-35 ÷ 120) °C. Other properties are similar to HM oils.

HG (HLPD) – are HM-type oils with additives against jerky movement (so-called stick-slip effect). Detergent and dispersant additives (which are also common in motor oils) are added to these oils to help loosen sediments, impurities, and water from the oil. This also improves the oil's resistance to ageing. Typical use is in hydraulic systems with sliding bearings, in applications where there are frequent changes of motion and smooth movement is required even at low speeds (e.g., in mobile working machines, in hydraulic systems with hydraulic cylinders for precise position control and in regulated circuits with hydraulic cylinders). The operating temperature range is (-35 ÷ 125) °C. An increased release of impurities places greater demands on filtration. These oils absorb more water and should not be used in very humid environments.

Note: For individual oils, ranges of operating temperatures given by manufacturers of these oils are given. It should be kept in mind that, especially for stationary hydraulic systems, the maximum oil temperature should not exceed 55 °C. The minimum temperature must respect the starting viscosity, according to the type of hydraulic pump used.

3.2 Fire-resistant liquids

They are used in applications with safety requirements where the non-flammability of a working fluid is required. These are hydraulic systems operating in environments with high ambient temperatures or where there is a risk of fire or explosion. Specific applications include the mining industry, systems operating in the vicinity of hot metal or open flames (smelting and rolling mill equipment, die casting, hydraulic forging presses), control equipment for steam and gas turbines, equipment in the chemical industry, etc. [12], [16], [21].

For all liquids that contain higher water content, the service life of the metal parts of the system is significantly decreased. For example, the service life of rolling bearings is only about

15% compared to the service life when using mineral oils. Fire-resistant liquids of hydraulic systems are defined by ISO 12922 [18].

HFAE (HFA) – it is an oil-in-water emulsion. The mass fraction of water is more than 80% (maximum oil content 20%). This liquid has a very low kinematic viscosity (v is less than $1.5\ mm^2 \cdot s^{-1}$), leading to significant flow losses and reducing the system efficiency. Poor lubricity causes high wear of components. It is necessary to add additives to increase anti-corrosion protection. The maximum operating temperature range is (5 ÷ 55) °C. Intensive evaporation occurs at higher temperatures. The working pressure is up to 30 MPa. NBR or FPM rubbers are most often used as sealing material. They are used in mines (hydraulic stands and reinforcements), in steel mills for some simple types of forging presses, or in injection presses for light metals. Additives, so-called emulsifiers, are added to promote the formation and increase the stability of the emulsion. The liquid acts aggressively on zinc and aluminium. High water content increases the risk of cavitation. It is a very cheap liquid, practically non-flammable [22].

HFAS – these are synthetic aqueous solutions of chemicals (e.g., containing glycol) but without petroleum oils. The basic properties and operating temperature range are similar to HFAE liquids. They can be used in hydrostatic drives for pressures of max. 16 MPa, especially in the food industry.

HFB – it is a water-in-oil emulsion, and the maximum oil content is 60%. In many respects, similar properties and requirements apply to HFAE emulsions. Compared to HFAE liquids, they have a higher viscosity, better lubricating ability, and operating temperature range between (5 ÷ 60) °C and can be used for working pressures up to 25 MPa. The emulsion stability must be continuously checked. They belong to the category of liquids with limited flammability. The flash point of these liquids is about 430 °C. However, they do not meet the strict requirements of the mining regulations for non-flammability, and their use is prohibited in mines. Currently, they are practically not used anymore.

HFC – solutions of polymers in water, the water content in the range (35 ÷ 60) %. These are usually higher polyglycol solutions, there are no solution stability problems, and component wear is lower compared to emulsions. They contain water, so it is necessary to add anti-corrosive additives and continuously check their condition. They have higher viscosity (22 ÷ 68) $mm^2 \cdot s^{-1}$ and density (1040 ÷ 1090) $kg \cdot m^{-3}$, which increases pressure losses in a system and complicates filtration. The risk of cavitation must also be taken into account. They are compatible with most common seals. NBR, SBR (styrene butadiene), EPDM (ethylene propylene diene) rubber, IIR (butyl) and natural rubber can be used. It is not recommended to use FPM and AU (polyester urethane) seals. They are more suitable for lower temperatures; the operating temperature range is usually (-20 ÷ 60) °C, and the maximum temperature is often limited to 50 °C. They are characterized by a high viscosity index (up to 150) and very good resistance to burning (ignition temperature approx. 650 °C). They act aggressively on zinc, aluminium, cadmium, leather, and common types of coatings. They cannot be mixed with other liquids. They do not tolerate the presence of mineral oil. They are not toxic and can be used as ecological liquids. Solutions with higher water content are mainly used in mines. Other applications - steel industry, foundries, forging machines, blast furnace hydraulics, die casting machines, etc.

HFDR (HFD) – anhydrous synthetic liquids containing phosphoric acid esters. They are characterized by high resistance to ageing and good protection against wear (they react with metal surfaces). The absence of water allows their use in a wide temperature range (-25 ÷ 150) °C. The kinematic viscosity is (15 ÷ 100) $mm^2 \cdot s^{-1}$, and the density depending on composition is in the range (1100 ÷ 1500) $kg \cdot m^{-3}$. They are poorly compatible with most commonly used seals and coatings. It has a particularly aggressive effect on NBR rubber; the most commonly used is FPM (Viton), EPDM, or IIR rubber. They are sensitive to the presence of water, with which they hydrolyze. They are mainly used in operations with an extensive range of temperatures and pressures,, e.g., in aircraft hydraulics, for lubrication and control of steam turbines, in hydrodynamic couplings, in welding machines and in mining equipment. Physiologically, they are not harmful and are easily biodegradable, which allows their use also as an ecological liquid. The greater expansion of these liquids is mainly limited by their higher price. It is possible to mix them with mineral oils (improvement of viscosity curve and price reduction).

HFDU – anhydrous synthetic fluids of a different composition compared to HFDR liquid. Many synthetic liquids,, e.g., carbonates or silicone oils belong to this group. They are used sporadically. Chlorinated biphenyls (formerly HFDS liquids) have excellent properties from the point of view of hydraulic mechanisms. Their use is currently prohibited due to their carcinogenic effects and problematic environment impact.

When using all non-flammable liquids, it is necessary to take extra care in the design of hydraulic systems to consider the properties of the liquids and their compatibility with both the sealing material and the material of distribution systems and tanks.

3.3 Environmentally-friendly liquids

These are environmentally friendly liquids designed for applications with high ecological requirements. These liquids are mostly partially ecologically degradable. They are mainly used in mobile hydraulic equipment, working in protected water areas, in agriculture, forestry, or in stationary equipment in the food industry [12], [21]. They are defined by ISO 15380 [19].

HETG (HTG) – this is a group of vegetable oils, while rapeseed oil is most often used. They have a very good lubricating ability, and the viscosity index is also high (higher than 200). Compared to mineral oils, they are more expensive, have a shorter service life and are more susceptible to oxidation, especially at higher operating temperatures. The operating temperature range is usually between (-20 ÷ 70) °C. When the maximum temperature is exceeded, these oils degrade very quickly (much faster compared to mineral oils). Vegetable oils are non-toxic, non-hazardous to water, and their biodegradability is higher than 95%. They can be used without significant restrictions in standard manufactured hydraulic elements and devices. The seal material is ideally FPM rubber, but basically, the same materials as mineral oil can be used. Their typical use is for mobile working machines in agriculture and forestry.

It is also possible to improve the properties of vegetable oils by adding additives (e.g., to increase thermo-oxidation stability) or mixing them with mineral oils, but this significantly degrades their ecological properties.

HEPG (HPG) – these are polyglycols (identical to HFC liquids, but without water content). Water deteriorates the properties of polyglycols. If non-flammability is not required, its content is reduced to a minimum. They have excellent oxidation stability. A favourable course of viscosity depends on the temperature and can be used in the temperature range (-20 ÷ 90) °C. They are characterized by very good fluidity even at lower temperatures; their higher density causes an increase in pressure losses. They are biodegradable (up to 90%) and water-soluble. They are mainly used in water management, lock hydraulics, floating excavators, etc. Other properties are similar to mineral oils, with which they cannot be mixed.

HEES (HT) synthetic esters (the base is a synthetic ester), this group also includes chemically modified vegetable oils. They are the most biodegradable (up to 95%), do not have a corrosive effect, have an excellent lubricating ability, higher oxidation stability than HETG liquids, and have a favourable course viscosity depending on the temperature. Their properties at low temperatures are also very good, i.e., in the temperature range (-30 ÷ 90) °C. Other properties are similar to mineral oil-based liquids. They can be mixed with HETG oils but are not compatible with water. They are mainly used in agricultural and forestry machinery. A higher price of these liquids is a barrier to their wider use.

HEPR – polyalphaolefin and other synthetic hydrocarbons that are insoluble in water. They are used in hydrostatic drives of mobile and industrial hydraulic systems in the temperature range (-35 ÷ 80) °C.

4. Filters and filtration

In order to achieve the correct function of a hydraulic system, its high reliability and its lifetime, it is necessary to ensure optimal cleanliness of a working fluid. In general, impurities are substances that enter the hydraulic system and influence its function. According to their state, they can be divided into gaseous (air), liquid (water - it is considered an impurity only if anhydrous liquids are used) and solid particles. The content of air and water in working fluids affects their basic physical properties. As described in the previous chapter, it causes ageing of oils and the formation of sludge and deposits. This chapter will be devoted to the contamination of liquids in the form of solid particles, the consequences of such contamination and its elimination. Modern manufacturing processes make the production of machine parts more precise. Clearance sizes are reduced, and increasing demands are placed on the accuracy and speed of hydraulic mechanisms. It is reported that more than 70% of failures in hydraulic systems are due to insufficient fluid cleanliness [11].

4.1 Impurities and cleanliness classes of liquids

The most harmful impurities in hydraulic systems are particles characterized by high hardness. These are mainly scale, particles of steel, brass, bronze, aluminium or other metals and rust. Less harmful are soft particles, such as parts of hardened fabrics, abrasions of seals and rubber particles from hydraulic hoses. In addition to material properties, particle size and concentration are also important.

Impurities in a circuit cause:

- clogging of narrow gaps and crevices in hydraulic components - leading to function failure of these components and system operation,
- abrasive wear of functional surfaces of moving parts - these are sliding pairs (valves, hydraulic pumps, and hydraulic motors),
- erosive wear on sharp functional edges - especially of spools of directional valves and other valves, valve seats and other functional components,
- ageing of working fluid - chemical reactions and accelerated oxidation reactions can occur due to contamination.

Depending on how the impurities enter the hydraulic system, it is possible to consider external and internal contaminations. In the case of external contamination, these are particles from the system's surroundings. They can enter the circuit during equipment installation and repairs or through filling holes and unsealed connections, especially in the case of tanks. For this reason, the tank is always equipped with a cover and air filter. Another method of the external input of impurities is during the system filling with liquid or during the change of the working liquid. A new liquid does not usually meet the requirements for the operation of hydraulic systems in terms of cleanliness (this applies in particular to oils). Therefore, a sieve is usually placed in the filling hole of the tank, which is used as a filter to catch the biggest impurities. However, filling devices equipped with a filter unit are usually used to fill and change the liquid. The external contamination can also be impurities generated during the components' manufacture or during the installation of the hydraulic system. These are residues after machining hydraulic elements or impurities after welding, cutting and grinding during the equipment installation. Also, for these reasons, hydraulic circuits are put into operation at lower working parameters. In the so-called test operation, the circuit is flushed with liquid and impurities are captured by the internal filtration of the hydraulic system (it is usually recommended to replace the filter element material subsequently).

Internal contamination occurs during the operation of the hydraulic system. This is mainly abrasion from moving parts of hydraulic components. In addition, the level of internal contamination increases significantly if the required liquid cleanliness is not maintained.

It is possible to prevent external contamination of the hydraulic system, and it can be largely eliminated by following prescribed procedures. Internal contamination will always occur due to the wear and ageing of the system. In order to reduce the adverse effects of this contamination, it is necessary to select the correct filtration method when designing the system, to deal with system diagnostics and to ensure the timely replacement of filter elements and working liquid.

Size of impurities

Particles larger than 20 μm are referred to as coarse impurities. These impurities are generally the most dangerous. They cause abrasive and erosive wear, clogging of functional gaps and associated sudden failures of the system function. Fine impurities with a particle size of (5 ÷ 15) μm are also dangerous in larger amounts (concentration). They cause erosive wear of functional edges of components and negatively contribute to liquid ageing. The finest impurities are represented by particles with a size of 2 ÷ 5 μm. o give you an idea, the average thickness of a human hair is about 70 μm, and the particle sizes mentioned above are not visible to the human eye [23].

The relative particle size relative to the size of the functional gaps has a significant effect on the wear of the functional surfaces. An example where the particle is larger than the size of the functional gap is shown in Fig. 4.1 (left). It represents the clogging of the functional gap and can be caused by both hard and soft particles. An example where the particle size is the same as the functional gap size of the moving part of the hydraulic element is shown in Fig. 4.1 (in the middle). In this case (if a hard particle is considered), it is the critical size of the particle. The most abrasive wear of the functional surfaces occurs, the most abrasion occurs, and therefore also other impurities in the system. If the particle size is much smaller than the size of the functional clearances, as is shown in Fig. 4.1 (right), mainly erosive wear occurs (in the case of hard particles). This is particularly evident in places with high liquid flow velocities. Hard and soft particles of small sizes can also cause inaccurate valve operation [10].

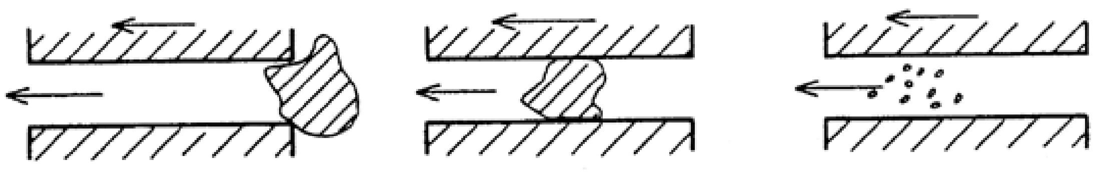


Fig. 4.1 Basic construction elements of valves for limiting and direction control of flow [24]

The minimum critical clearances vary depending on the design of hydraulic elements. The basic overview of the sizes of these clearances, including possible consequences due to liquid contamination, is given in Tab 4.1.

Tab 4.1 An overview of the sizes of these clearances, including possible consequences due to liquid contamination [25]

Hydraulic element	Clearance size range [μm]	Possible consequences due to liquid contamination
Gear converters	0.5 ÷ 5 (radial and axial clearances of gears)	increasing clearances, decreasing achievable pressure, decreasing efficiency, bearing seizing
Vane converters	0.5 ÷ 5 (radial clearance between vane and stator) 0.5 ÷ 20 (axial clearance between rotor and housing) 30 ÷ 40 (axial clearance between vane and rotor)	increasing clearances, decreasing achievable pressure, decreasing efficiency, bearing seizing
Axial and radial piston converters	0.5 ÷ 40 (clearance between pistons and block) 0.5 ÷ 1 (clearance between block and carrier plate) 20 ÷ 40 (clearance between slipper and piston) 1 ÷ 25 (clearance between slipper and swashplate)	increasing clearances, decreasing achievable pressure, decreasing efficiency, seizing of lapped sliding surfaces, seizing of pistons (or control elements), bearing seizing
Directional valves and valves	0.5 ÷ 5 (clearance between spool and housing)	increasing clearances, erosive wear of functional edges and surfaces, increased flow losses, loss of tightness (it can lead to loss of functionality), seizing of spools, burning of solenoids after seizing
Servo valves	0.5 ÷ 8 (clearance between spool and housing) 100 ÷ 400 (nozzle diameter) 20 ÷ 30 (clearance between nozzle and flap)	erosive wear of functional edges and surfaces, loss of control stage tightness, wear of functional edges or nozzle clogging - loss of control stage function

An example of damage to a proportional distributor due to impurities is shown in Fig. 4.2. The negative effect is usually caused by damage to the functional edges in the body of the distributor, which is usually made of softer material (grey cast iron). This is a combination of erosive and abrasive damage. Even slight damage to the functional surfaces of these valves results in functional failures of the hydraulic system.

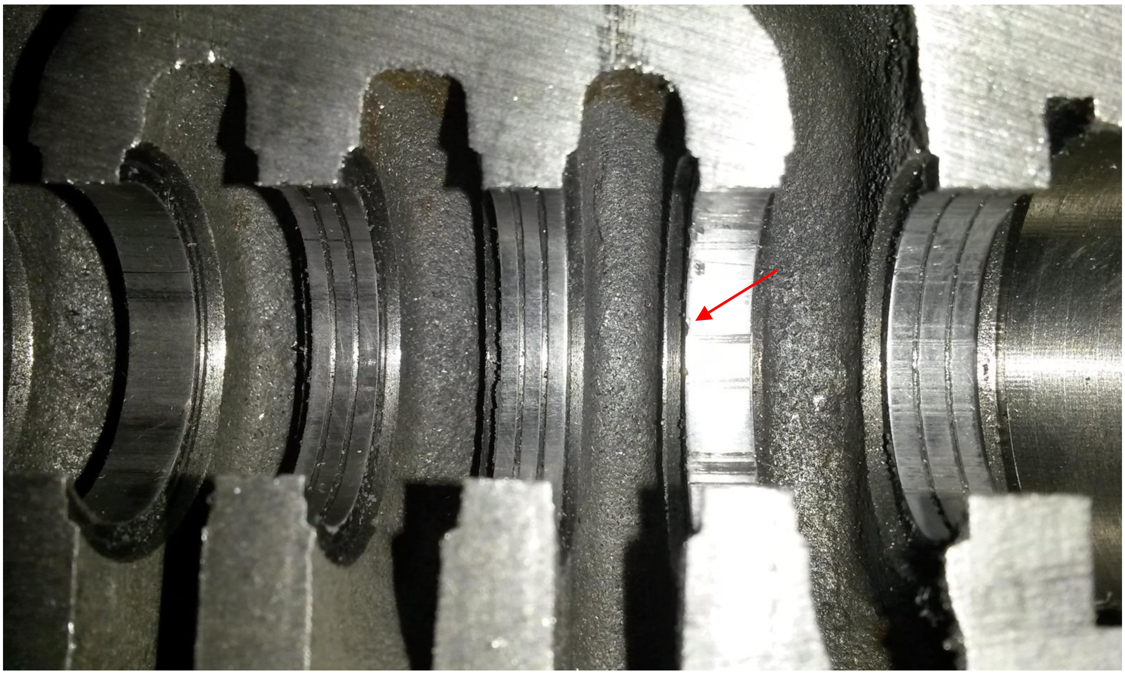


Fig. 4.2 An example of damage due to impurities in a hydraulic system with a proportional distributor

Similar damage as in the previous case is shown in Fig. 4.3. The damage due to impurities is visible in the body of the servo valve.

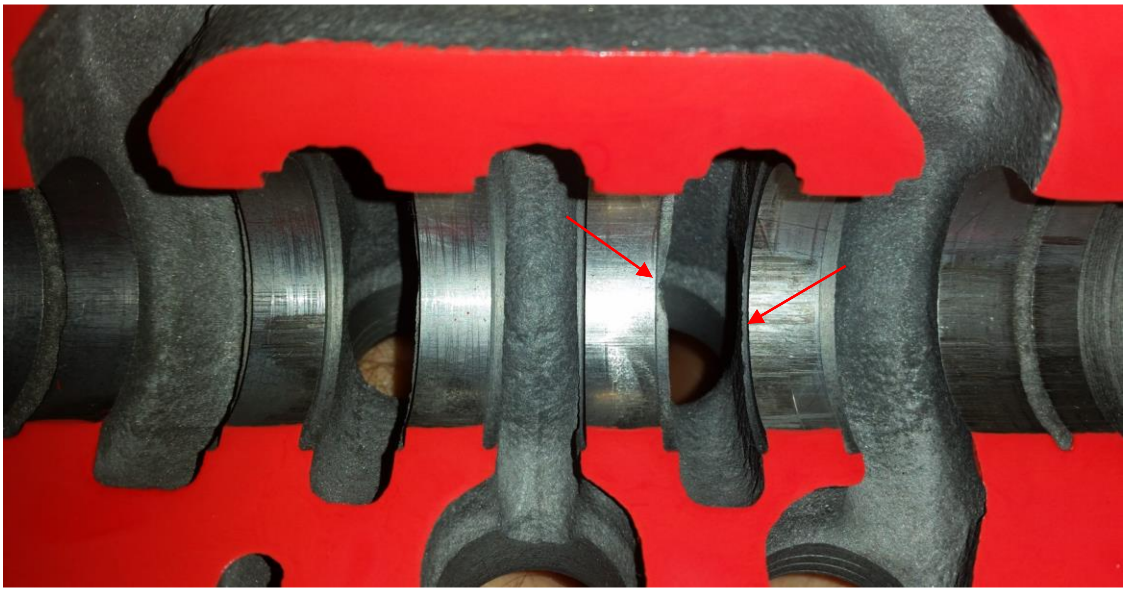


Fig. 4.3 An example of damage due to impurities in a hydraulic system with servo valve

The abrasive damage in an axial piston hydraulic pump in bent axis design is shown in Fig. 4.4. Specifically, this is damage to the front surface of the piston block, which is connected to the carrier plate of the hydraulic pump.

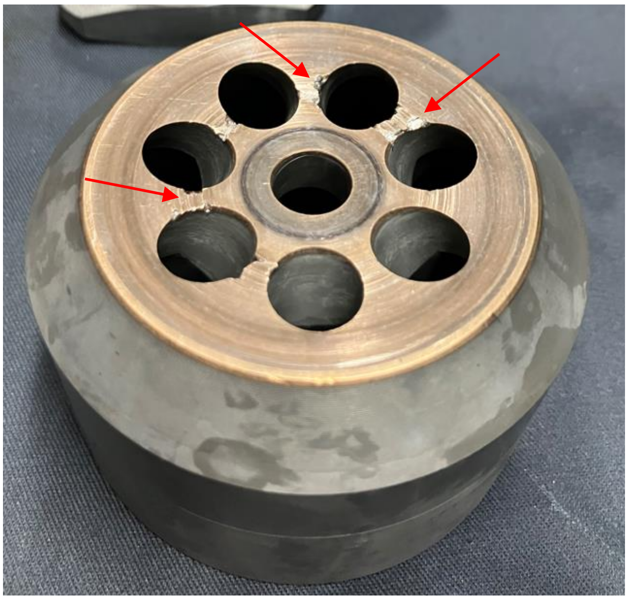


Fig. 4.4 An example of damage due to impurities in a hydraulic system with hydraulic pump

The damage to the external gear hydraulic pump is shown in Fig. 4.5. The damage is caused by the entry of impurities into the inter-tooth space of the hydraulic pump.

Fig. 4.5 An example of damage due to impurities in a hydraulic system with external gear hydraulic pump

Cleanliness class

The liquid's cleanliness and degree of contamination are determined based on the amount of impurities (number of particles in liquid) and the size of those impurities. The classification of the number and size of solid particles in hydraulic liquids is given by relevant standards. The most commonly used standards are the international standard ISO 4406 [26] and the American standard NAS 1638 [27]. Both standards specify individual classes of liquid cleanliness. The number and size of particles are determined from a liquid sample of a prescribed volume.

Tab 4.2 Liquid cleanliness classes according to the standard ISO 4406

Number of particles in 1 ml of liquid		**Code number**
more than	**maximum**	
1 300 000	2 500 000	28
640 000	1 300 000	27
320 000	640 000	26
160 000	320 000	25
80 000	160 000	24
40 000	80 000	23
20 000	40 000	22
10 000	20 000	21
5 000	10 000	20
2 500	5 000	19
1 300	2 500	18
640	1 300	17
320	640	16
160	320	15
80	160	14
40	80	13
20	40	12
10	20	11
5	10	10
2.5	5	9
1.3	2.5	8
0.64	1.3	7
0.32	0.64	6
0.16	0.32	5
0.08	0.16	4
0.04	0.08	3
0.02	0.04	2
0.01	0.02	1
0.005	0.01	0

The standard ISO 4406 specifies 28 classes of liquid cleanliness; see Tab 4.2. Based on the number and size of impurities in a 1 ml liquid sample, the code number of the liquid cleanliness is determined. The liquid cleanliness is subsequently defined by the cleanliness code (e.g., 17/15/12), which consists of three code numbers. The first number is given by the number of particles with a size larger than 4 µm. The second number represents the number of particles with a size larger than 6 µm. The third number is the number of particles with a size larger than 14 µm.

An example of determining the liquid cleanliness code according to ISO 4406 is presented in Tab 4.3. The test liquid volume contained 1215 particles larger than 4 µm, 268 particles larger than 6 µm and 31 particles larger than 14 µm. By subtracting the values for each cleanliness class, it is possible to determine the liquid cleanliness code according to the ISO standard as 17/15/12.

Tab 4.3 An example of determining the liquid cleanliness code according to ISO 4406 using Tab 4.2

Number of particles	Size of particles	Code number
1215	> 4 µm	17
268	> 6 µm	15
31	> 14 µm	12
Code of liquid cleanliness: 17/15/12		

The classification according to NAS 1638, specifies 16 individual classes of liquid cleanliness, see Tab 4.4. The cleanliness classes are defined by the maximum ranges of the number of particles of a given size in a 100 ml liquid sample. The resulting code designation of the liquid cleanliness class is determined by the highest class obtained in the liquid test sample. Only five particle size ranges are usually used to evaluate the liquid cleanliness class (the 2 - 5µm range is not commonly specified).

An example of determining the liquid cleanliness class according to the NAS 1638 standard is shown in

Tab 4.5. The test volume of the liquid contained 5443 particles with sizes ranging from 5 to 15 µm, 50 particles with sizes ranging from 15 to 25 µm and 21 particles with sizes ranging from 25 to 50 µm. Particles larger than 50 µm were not observed in the liquid. The o highest value obtained according to the NAS standard determines the liquid cleanliness class of 5.

Tab 4.4 Liquid cleanliness class according to the standard NAS 1638

Class	**Number of particles in 100 ml of liquid**					
	2 – 5µm	5 – 15µm	15 – 25µm	25 – 50µm	50–100µm	> 100µm
00	625	125	22	4	1	-
0	1 250	250	44	8	2	-
1	2 500	500	88	16	3	1
2	5 000	1 000	178	32	6	1
3	10 000	2 000	356	63	11	2
4	20 000	4 000	712	126	22	4
5	40 000	8 000	1 425	253	45	8
6	80 000	16 000	2 850	506	90	16
7	160 000	32 000	5 700	1 012	180	32
8	320 000	64 000	11 400	2 025	360	64
9	640 000	128 000	22 800	4 050	720	128
10	1 280 000	256 000	45 600	8 100	1 440	256
11	2 560 000	512 000	91 200	16 200	2 880	512
12	5 120 000	1 024 000	182 400	32 400	5 760	1 024
13	-	2 048 000	364 800	64 800	11 520	2 048
14	-	4 096 000	729 000	129 600	23 040	4 096

Tab 4.5 An example of determining the liquid cleanliness class according to the NAS 1638 standard using Tab 4.4

Number of particles	**Size of particles**	**Cleanliness class**
5443	5 – 15 µm	5
50	15 – 25 µm	1
21	25 – 50 µm	2
0	50–100 µm	00
0	> 100 µm	00
Liquid cleanliness class: 5		

Methods for determining liquid contamination

In the past, the gravimetric method was used to determine the concentration of particles in a liquid. This method was used to measure the mass of particles contained in a given liquid volume. However, it was impossible to determine the particle size, and it is not practically relevant for use in modern hydraulic systems at present.

One possibility for counting impurity particles is to use optical microscopy. A simple operational determination of liquid contamination is by using a portable oil laboratory. The measuring equipment contains a fine membrane filter through which a specified liquid volume flow. Subsequently, the microscope is used to compare the amount and size of impurities captured on the membrane filter with standards used to determine the liquid contamination. The evaluation of the number and size of particles can be manual or automatic. The accuracy of this method depends on the type of microscope used, but an accurate determination of the number of particles of the smallest sizes is difficult. This method usually evaluates only the number of particles larger than 5 μm and the number of particles larger than 15 μm. According to ISO 4406, the liquid cleanliness code contains only the last two numbers,, e.g. -/18/14. The advantage of this method is the elimination of the effects of air and water in the liquid on the measurement results [16], [23].

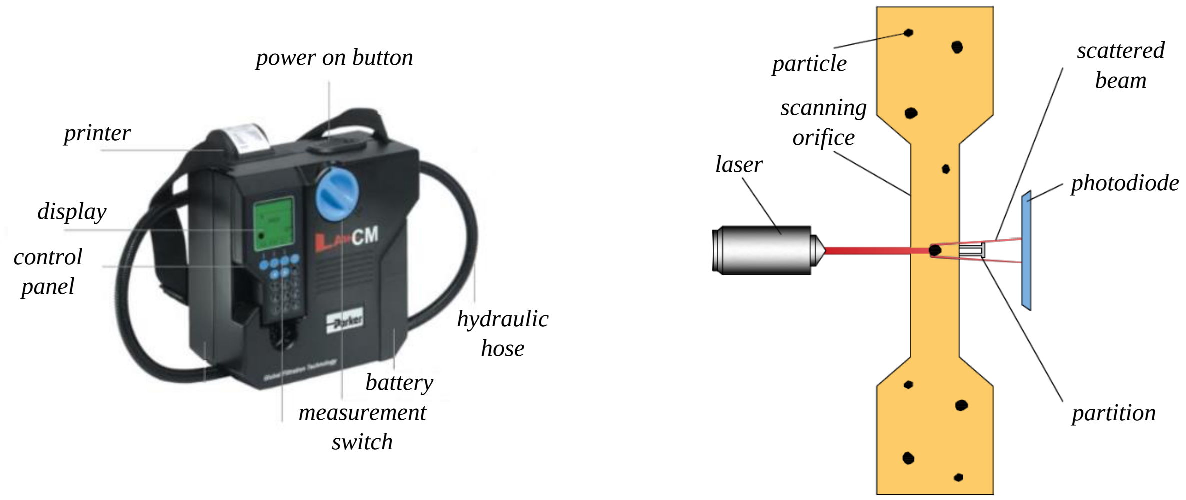
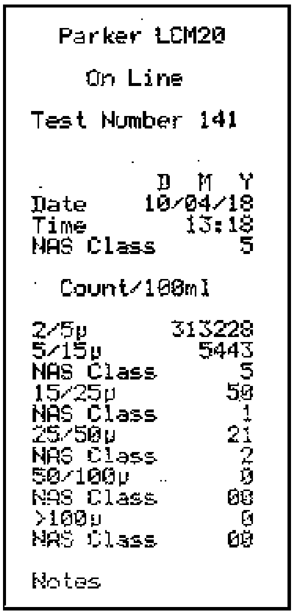


Fig. 4.6 Portable particle counter Laser CM20.2021 from Parker Hannifin company (left), functional principle (right)

A more accurate determination of impurities in the liquid, especially in terms of detecting the smallest particle sizes, is possible with automatic particle counters. In these counters, a specified volume of liquid flows through a sensing orifice and, in terms of particle detection principles, light and laser counters are used. The light counter consists of a light source emitted into the sensing orifice area and a photodiode. As the particle is transmitted through the scanning orifice, the white beam of light is shielded depending on the particle size. The particle size is evaluated as the difference between the emitted light amount and the light amount captured by the photodiode. In the case of the laser counter, a laser beam is emitted into the scanning orifice area. As the particle passes through the scanning orifice, the laser beam is scattered, which is captured by the photodiode. Laser counters are more sensitive and accurate. The main advantage of automatic particle counters is the simplicity of use and the quick evaluation of the size of the liquid contamination. However, the measurement results may be distorted as the particles may overlap each other as they pass through the scanning orifice. The resulting accuracy of these counters can also be negatively affected by air bubbles and water in the liquid. An example of an automatic laser particle counter Laser CM20.2021 from Parker Hannifin company is shown in Fig. 4.6 (left) and its functional principle is shown in Fig. 4.6 (right) [28].

A demonstration of the evaluation of the liquid cleanliness class according to the NAS 1638 standard using the Laser CM20.2021 particle counter is shown in Fig. 4.7. The measurement result corresponds to the example given in

Tab 4.5.



Parker LCM20

On Line

Test Number 141

	D M Y
Date	10/04/18
Time	13:18
NAS Class	5

Count/100ml

2/5µ	313228
5/15µ	5443
NAS Class	5
15/25µ	50
NAS Class	1
25/50µ	21
NAS Class	2
50/100µ	0
NAS Class	00
>100µ	0
NAS Class	00

Notes

Fig. 4.7 An example of evaluation of the liquid cleanliness class according to NAS 1638 standard using the Laser CM20.2021 particle counter

In addition to the above methods for determining the liquid contamination, devices that use electrical, magnetic or ultrasonic measurement principles can also be used.

Level of liquid cleanliness in hydraulic equipment

The required liquid cleanliness level is defined by the required liquid cleanliness class according to ISO 4406 or NAS 1638 standards. The recommended cleanliness class of the individual elements of hydraulic systems is usually indicated in product catalogues of manufacturers of these elements. Filters and filtration systems are used to achieve the required liquid cleanliness in hydraulic equipment. The filtration fineness depends on the type of hydraulic equipment and is also based on the construction design of hydraulic components. In general, it is necessary to ensure the filtration level of a hydraulic system that is required by its most sensitive element. The recommended cleanliness classes of hydraulic systems are given for guidance in Tab 4.6.

Tab 4.6 Recommended cleanliness classes of hydraulic systems [23], [29]

Hydraulic systems	Recommended cleanliness class		Filtration fineness	Filtration in circuit	Typical application
	ISO 4406	NAS 1638			
Systems with high sensitivity to impurities and high reliability requirements, highly loaded circuits with servo valves, filling of servo drives	16/12/9	3	1÷2 μm	Return and pressure	aerospace and rocket industry, laboratory equipment
Systems with high sensitivity to impurities, hydraulic circuits with servo valves, control and regulating systems	18/13/10	5	3 μm	Return and pressure	aviation, servo systems
Systems with proportional valves	19/14/11	6	5÷6 μm	Return or pressure	industrial robots, machine tools
Modern industrial hydraulic systems, medium pressure hydraulic systems	20/16/13	8	5÷10 μm	Return	general engineering hydraulics, mobile hydraulics
Industrial hydraulics, systems with larger clearances operating at lower pressures, lower sensitivity to impurities	21/17/14	10	do 25 μm	Return	heavy engineering and metallurgy hydraulics, mining and other systems

From the point of view of the individual system elements, it is possible to determine the maximum allowable operating contamination of a working liquid, as is shown in Tab 4.7. These values are only indicative. In the case of systems operating at higher operating pressures or higher operating liquid temperatures, it is recommended to increase the filtration class.

Tab 4.7 Maximum allowable operating liquid contamination [30]

Hydraulic elements	Maximum allowable liquid contamination (designation according to ISO 4406 standard)
Hydraulic pumps and hydraulic motors	20/18/15 (under high load 18/16/13)
Directional valves and other valves	20/18/15
Proportional valves and servo valves	18/16/13 (for some types 17/15/12)

4.2 Filter element materials

Filters are used in hydraulic systems to capture impurities and filter the liquid. The functional element of a filter is the filter insert. Filter inserts are manufactured in various construction designs and materials. The choice of filter insert depends on many factors,, e.g., the type of filtration, the required fineness of filtration (depending on the type of hydraulic equipment), the working liquid used, the magnitude of operating pressures and temperatures, etc. In terms of the filtration effect, filter element materials can be divided into:

- surface filters,
- depth filters.

Surface filters capture contaminants only on their outer surface. They are usually made from coiled steel or plastic fibres, or non-woven fabric, and have a similar function to sieves. The filtration fineness is determined by the mesh size (pores) of the filter insert. Impurities bigger than the pore size are captured on the insert surface, but smaller particles are propagated further into the system; see Fig. 4.8 (left). Surface filters are only used for coarser filtration, usually larger than 20 μm. The advantage of these inserts is their small thickness, easy cleaning by flushing and, therefore, the possibility of their reusing. However, they are characterized by a relatively small absorption capacity and quickly become clogged.

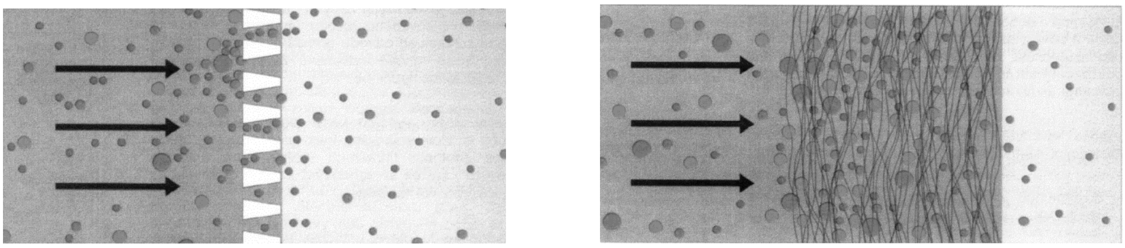


Fig. 4.8 Surface filter (left), depth filter (right) [5]

In the case of depth filters, the filter inserts are constructed of multi-layered fibrous materials (paper, glass and artificial fibres, textiles, metal microfibres, etc.). The liquid flowing through the individual layers of the insert is gradually removed from impurities, as is shown in Fig. 4.8 (right). The inserts have a higher thickness with more pores, which also ensures a higher filter capacity. Depth filters are used in most common hydraulic systems. In some types of inserts, an absolute filtration capacity of up to 1 μm can be achieved. These filter inserts are usually not cleanable and are designed for single use. When the insert becomes clogged with impurities, the pressure gradient on the filter rises rapidly, indicating the time to replace it. An example of a depth filter is shown in Fig. 4.9 (left).

The overview of selected materials of filtration inserts is given in Tab 4.8.

Tab 4.8 Filter element materials [11], [31]

Filter element material	Filtration fineness [μm]	Filtration effect	Application
Fabric made of steel fibres (cleanable)	10÷800	Surface filtration	For coarser filtration in the return line, pressure, or suction filters of conventional hydraulic systems. Possible to use for lubricating liquids and, in the case of stainless steel, also for water.
Non-woven textile (One-time use)	25÷60	Surface filtration	These are high-strength plastic fibres. They are designed for the coarser filtration of emulsions, coolants, lubricants, and water.
Paper fibres (One-time use)	10÷25	Depth filtration	The insert consists of layers of impregnated paper coiled on a supporting sieve. It is the cheapest type of filter inserts used in the return line and pressure filters of conventional hydraulic systems.
Glass fibres (One-time use)	1÷20	Depth filtration	For very fine filtration in the return line and pressure filters. For general use, also suitable for systems with high liquid cleanliness requirements (circuits with proportional valves and servo valves).
Non-woven steel fibres (One-time use)	5÷15	Depth filtration	They are characterized by a very good filtration capacity. Use in the return line and pressure filters for filtration of aggressive liquids at high temperatures up to 160 °C. Applications in the chemical industry.

When the liquid flows through the filter, the filter insert acts as a flow resistance, where its size depends on the selected filtration fineness (the finer the filtration, the higher the resistance). This resistance causes a pressure loss which increases with the level of clogging of the filter insert. When designing the filtration, it is necessary to select the correct filter size to avoid rapid clogging of the filter elements or their rupture. The lifetime of the filter inserts is given by the size of the allowable pressure gradient. Its maximum value varies for different types of inserts and is specified in the manufacturers' product catalogues. The dependence of the pressure gradient Δp across the filter element on the amount of captured impurities in grams is shown in Fig. 4.9 (right).

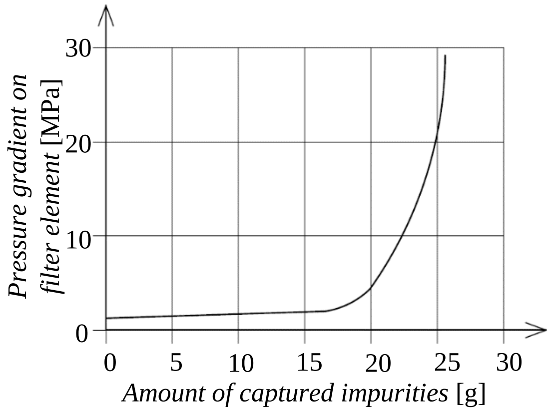


Fig. 4.9 Depth filter from Bosch Rexroth company [32] (left), the dependence of the pressure gradient Δp across the filter element on the amount of captured impurities in grams [5] (right)

4.3 Filters

The filtration method in hydraulic systems depends on the required liquid cleanliness (filtration fineness), operating parameters and system design. The filters used in these systems can be divided according to their location in the hydraulic circuit into:

- low-pressure filters (return line filters and bypass filtration),
- pressure filters,
- suction filters,
- fillers and breathers.

An example of filter arrangement in an open hydraulic system is shown in Fig. 4.10.

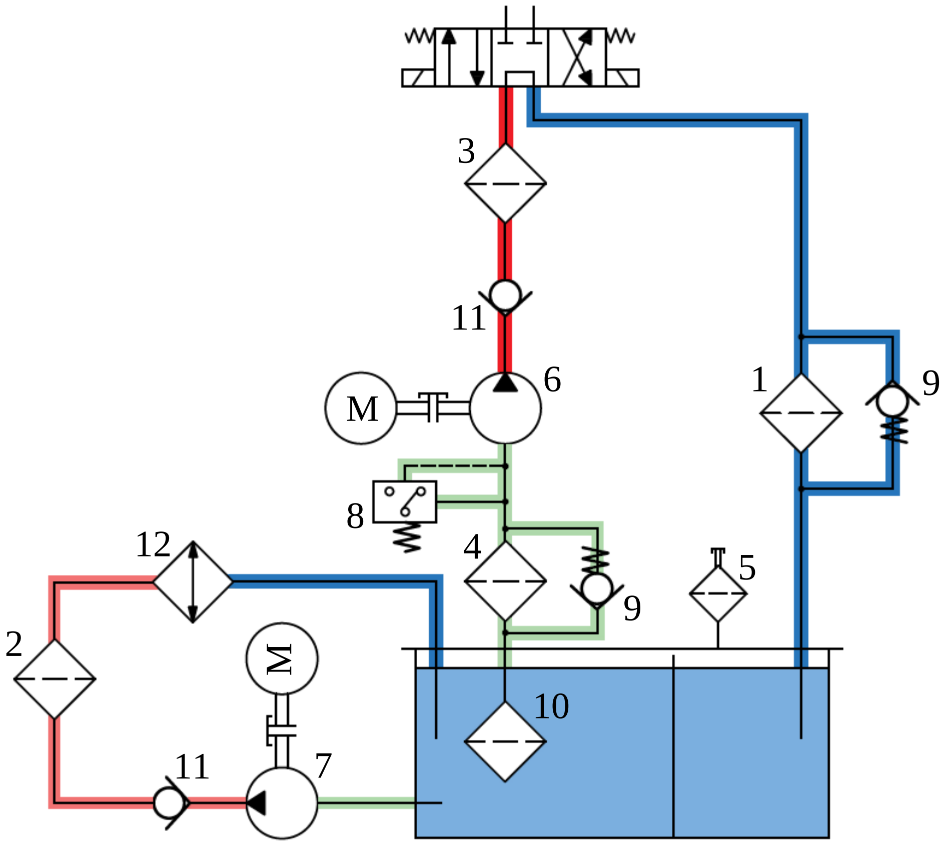


Fig. 4.10 Examples of filter arrangement in open hydraulic system

1 – mounted return line filter, 2 – low-pressure filter in secondary filtration circuit (bypass filtration), 3 – high-pressure filter, 4 suction line filter, 5 – filler and breather, 6 – hydraulic pump of the main circuit, 7 – hydraulic pump of the secondary filtration circuit, 8 – under pressure switch, 9 – bypass check valve with spring, 10 – suction basket, 11 – check valve, 12 - cooler

Low-pressure filters

Low-pressure filters are most often located at the end of the return line and can be mounted directly on a tank cover. In this case, these are so-called waste filters or return line filters. The liquid is filtered during the flow back into the tank. An example of a low-pressure filter design is shown in Fig. 4.11. The main parts of the filter are the cover (1) and the housing (2), in which the filter insert (3) is placed. The catch basket (4) prevents the entrance of impurities into the tank when changing the filter insert. The filter insert clogging indicator (5) is a standard part of these filters. Most filter design solutions work on the principle and direction of liquid flow shown in Fig. 4.11. The contaminated liquid enters the filter through input A, flows through the outer surface of the filter insert (liquid filtration occurs) and leaves the filter through the inner part of the filter insert through output B. This solution is advantageous because it uses a larger area around the perimeter of the filter insert to filtrate the liquid. The inner part of the insert is usually reinforced with a cylindrical metal sieve, which prevents its collapse and rupture when the pressure gradient increases (clogging of the filter insert).

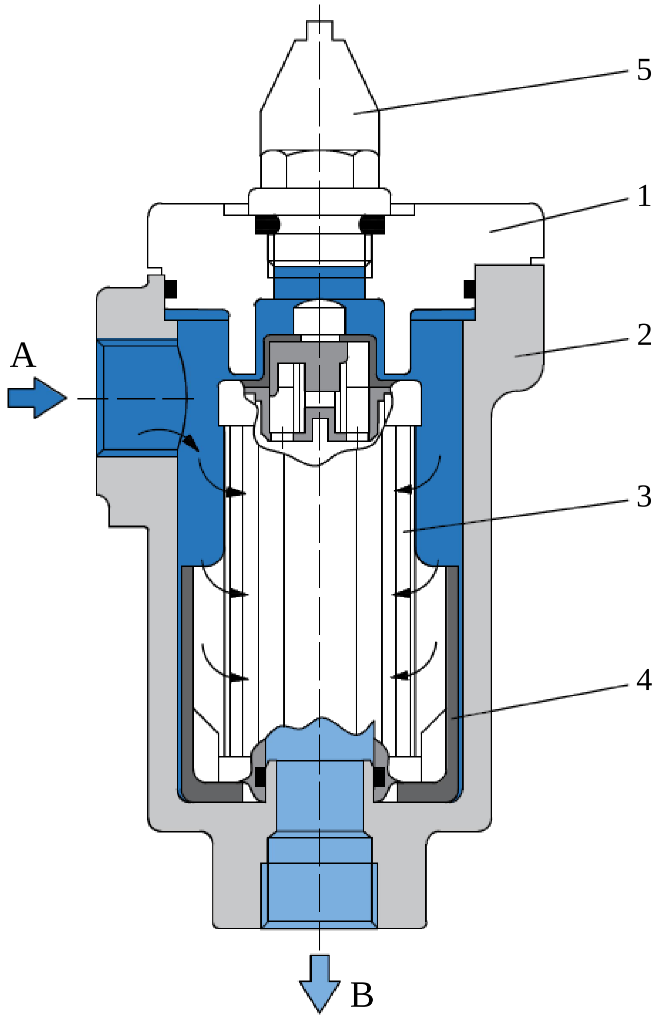


Fig. 4.11 Low pressure filter

1 – fixing flange, 2 – filter housing, 3 – cover, 4 – catch basket, 5 – filter element 6 – clogging indicator

The filters have a prescribed liquid flow direction that must be followed during installation. Liquid flow through the filter in the opposite direction is not so advantageous. The filter area is significantly reduced, the filter insert becomes clogged more quickly and the pressure gradient increases, which can lead to the insert being pulled out of the cover. In this case, the captured impurities are flushed into the tank, and the filter is no longer in operation. Example of filter element damage caused by incorrect filter assembly is shown in Fig. 4.13 (right).

Return line filters are practically always used with a bypass check valve with a spring connected in parallel, which in some cases is directly a constructional part of the filter. This valve has a safety function. When the filter insert is clogged, the pressure gradient on the filter increases, the spring of the check valve is compressed, and the liquid flows through the check valve into the tank. The check valve protects the filter insert from its rupture, but the filter loses its filtering ability during the flow through the check valve.

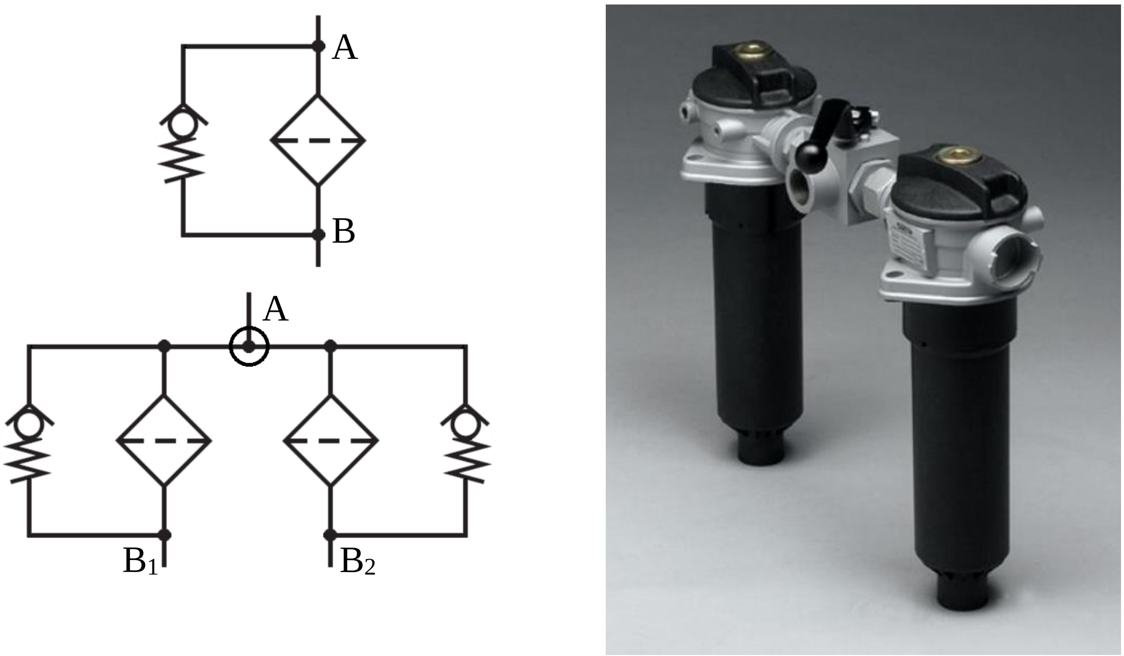


Fig. 4.12 Graphic symbol of return line filter with bypass check valve (top left)), double return line filter with the possibility to switch flow direction from Hydac company [34] (right), graphical symbol (bottom left)

The advantages of waste filters are low acquisition costs and, simple maintenance - the filter is usually placed in an easily accessible location, and the filter insert contamination indicator is clearly visible. This is a relatively efficient method of filtration, with the possibility of achieving high filtration fineness. The use of the return line filter allows the filtration of the whole volumetric flow of the hydraulic pump.

Fig. 4.13 Low-pressure filter with optical-electrical signalization of filter element contamination indicator (left), damage to the filter element when the filter was incorrectly connected to the hydraulic system (right)

The disadvantage is that filtration is interrupted depending on the operation of the hydraulic system. The filter must be dimensioned for the maximum system flow rate (in the case of circuits with accumulators, the possible sum of the flow rate of the hydraulic pump and the flow rates from the accumulators). During cold starting (higher liquid viscosity) or pressure peaks, the check valve may open shortly, and contaminated liquid may flow into the tank. When replacing the filter insert, the system operation must be stopped. This disadvantage can be eliminated by using a double waste filter, as is shown in Fig. 4.12 (right). The element consists of two identical filters and a switching shut-off valve. When the hydraulic system is in operation, the liquid always flows through only one of the two filters and the filter insert of the second filter can be changed during the operation.

Another solution for low-pressure filtration is to use a low-pressure filter in a separate filtration circuit (circulating filtration). The liquid is delivered to the filtration circuit by a secondary hydraulic pump (which increases the acquisition cost). The advantage is that the filtration process does not depend on the operation of the main hydraulic circuit. The filter can be dimensioned for a smaller fluid flow. The secondary filtration circuit may also include a cooler to ensure the cooling of the working liquid.

Pressure filters

Pressure filters are located in the pressure branch of the hydraulic circuit behind the hydraulic pump. They are used in justified cases, especially in circuits with elements particularly sensitive to liquid cleanliness (e.g., in circuits with servo valves). The filter is usually placed close to the sensitive element. Pressure filters are very efficient, and fine filtration is chosen. However, they are significantly more expensive compared to the return line filters. They must be of robust construction to be resistant to high operating pressures in the hydraulic system, where the filter inserts must also be adapted to these pressures. Replacement of the filter insert during the operation of the hydraulic system is only possible in the case of using a pair of pressure filters with the possibility of switching the flow direction. An example of a pressure filter is shown in Fig. 4.14. The function principle is similar to low-pressure filters.

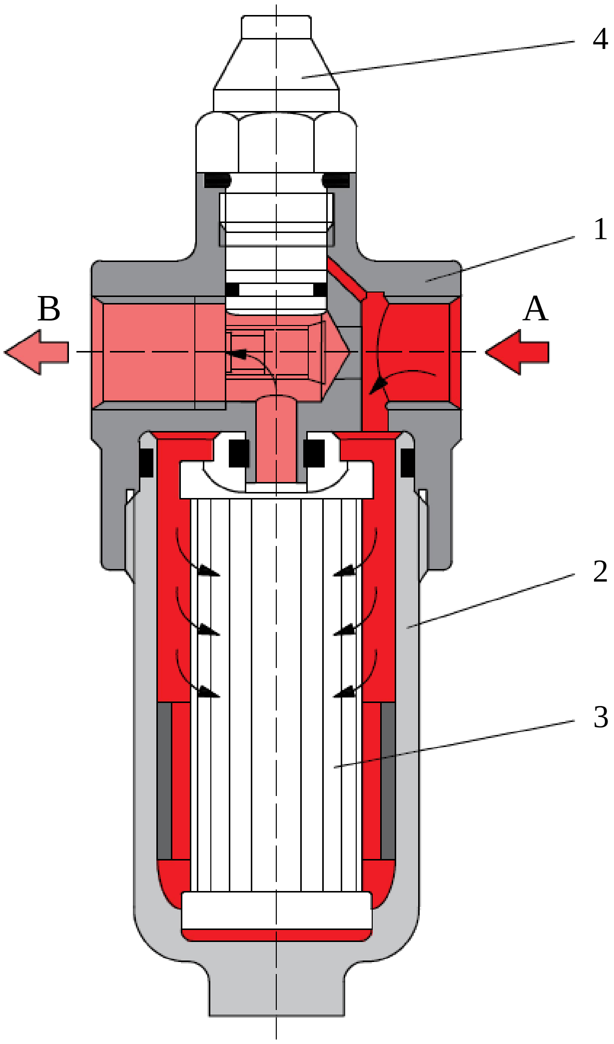


Fig. 4.14 Pressure filter

1 – filter head, 2 – filter housing, 3 – filter element, 4 – clogging indicator

Suction filters

The suction filter can be located between the hydraulic pump and the tank in the suction pipe. It serves to protect the hydraulic pump from coarse impurities. The operation of these filters is problematic, and they are only used in exceptional cases. When the filter insert is clogged, the resistance in the suction line of the hydraulic pump increases significantly. This leads to an increase in the required underpressure in the suction line of the hydraulic pump, the risk of cavitation increases, and the suction ability of the hydraulic pump may be limited. If the suction filter is used, the filter should be equipped with a bypass and an underpressure sensor that stops the hydraulic pump if the allowable underpressure value is exceeded. Suction filter inserts should only be chosen with very coarse filtration (70 ÷ 100 μm).

More often, suction baskets are used in hydraulic systems, which are installed under the liquid level in the tank. This is only to protect the pump from coarse impurities. Also, if the suction basket is used, the underpressure size in the suction line of the hydraulic pump should be monitored.

Fillers and breathers

These filters are placed on the tank cover or into the filling hole of the tank. When the liquid level in the tank changes, the air filler Fig. 4.15 and breather are connected to the tank surrounding. In this case, the air filters prevent the entrance of impurities into the tank from its surrounding.

Fig. 4.15 Air filter from Bosch Rexroth company [33]

Filling filters Fig. 4.16 (left) are used for filtration when filling the hydraulic system with liquid. They usually include a filter insert and a sieve to capture impurities. Some fill and air filters may also contain materials that serve to absorb moisture see in Fig. 4.16 (right).

Fig. 4.16 Filing filter from MP Filtri company [35] (left), breather with moisture absorption [33]

Clogging indicators of filter element materials

The pressure or pressure gradient, which represents the resistance of the filter insert, is monitored using the filter insert contamination indicators. Reaching a critical value (when the filter insert is clogged) is indicated visually or by an electrical signal; see Fig. 4.17.

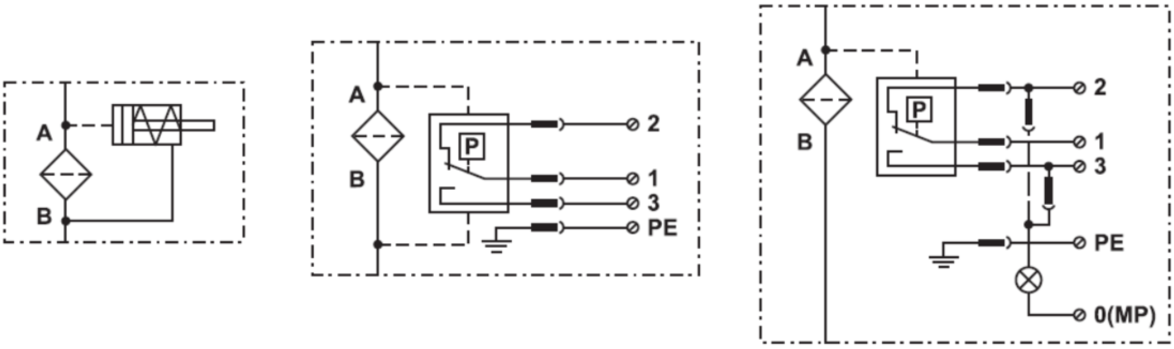


Fig. 4.17 Examples of connection of filter element contamination indicator [11], optical signalization (left), electrical signalization (in the middle), optical-electrical signalization (right)

There are a large number of filters, filter inserts and equipment. Selected filter products from Bosch Rexroth company are shown in Fig. 4.18.

Fig. 4.18 Filter products of Bosch Rexroth company [32]

5. Hydrostatic pressure converters

Hydrostatic pressure converters are elements in which mechanical energy is converted into liquid pressure energy, or conversely. They work on the volumetric principle, which means that energy conversion occurs by transporting a particular liquid volume from the input of the converter to its output. The pressure energy E_p of flowing liquid is directly proportional to the pressure p and the volume V of the liquid:

$$E_p = p \cdot V \, . \tag{5.1}$$

The change in pressure energy when the liquid flows through a hydrostatic pressure converter is given by the formula:

$$\Delta E_p = \Delta E_{p2} - \Delta E_{p1} = p_2 \cdot V - p_1 \cdot V = \Delta p \cdot V \, . \tag{5.2}$$

The hydraulic power P_h corresponds to the amount of pressure energy transferred (work done) of the liquid per time [8]:

$$P_h = \frac{d\Delta E_p}{dt} = \frac{d(\Delta p \cdot V)}{dt} = \frac{d\Delta p}{dt} \cdot V + \Delta p \cdot \frac{dV}{dt} \, . \tag{5.3}$$

If the steady state is considered, then Δp = const. and the change in pressure with time is zero, i.e., $d\Delta p/dt = 0$. The change in the liquid volume V in time t can be expressed by the volume flow rate Q of the liquid, and the equation is modified as follows:

$$P_h = \Delta p \cdot \frac{dV}{dt} = \Delta p \cdot Q \, , \tag{5.4}$$

where P_h [W] is the hydraulic power, Δp [Pa] is the pressure gradient (pressure difference), and Q [$m^3 \cdot s^{-1}$] is the liquid volumetric flow.

Thus, for an ideal (lossless) hydrostatic pressure converter, the equality of powers during the conversion of pressure energy into mechanical energy (and conversely) can be considered:

$$P_h = P_m \, , \tag{5.5}$$

where P_m [W] is the mechanical power.

In practice, there will always be losses in the operation of converters. A part of the pressure and mechanical energy is converted into heat. This is the so-called energy dissipation, and it is an irreversible process. In general, there are three types of losses in hydraulic systems.

Volumetric losses

These are leakage losses. They are manifested by liquid flow through functional gaps of moving parts. Their size depends on pressure gradient, dimensions, and geometry of functional gaps, or on physical properties of liquids (especially on viscosity). For hydrostatic pressure converters, the magnitude of volumetric losses can be expressed by the **volumetric efficiency** η_{vol}.

Pressure losses

They are caused by resistance to the flowing liquid. This includes hydraulic friction and local losses. Frictional losses are caused by internal friction of the liquid and friction of the liquid in contact with a body surface. Local losses are structural resistances to flow that result in a change in the direction or velocity of liquid flow.

Mechanical losses

Mechanical losses represent the friction of moving mechanical parts and are sometimes also referred to as passive resistances.

For hydrostatic pressure converters, mechanical and pressure losses are presented together, and their magnitude is defined by the **mechanical-hydraulic efficiency** η_{mh} (it is the product of mechanical and pressure efficiencies).

Values of flow and mechanical-hydraulic efficiencies for hydrostatic pressure converters are based on steady-state measurements and are a function of many parameters.

The total efficiency η_T of a hydrostatic pressure converter is given by the product of the individual partial efficiencies:

$$\eta_T = \eta_{vol} \cdot \eta_{mh} \ . \tag{5.6}$$

The power P of a real hydrostatic pressure converter will therefore be equal to its input power P_i, reduced by the total efficiency of the hydrostatic pressure converter η_t, according to the equation:

$$P = P_i \cdot \eta_t \ . \tag{5.7}$$

Hydrostatic pressure converters can be classified according to a number of different aspects. If the basic parameter of their division is the way of movement of the input or output elements, then the converters can be divided into:

with rotary motion – **hydraulic pumps** and **rotary hydraulic motors**,

with swivel motion – **rotary actuators**,

with translational (linear) motion – **hydraulic cylinders**.

The main manufacturers of rotary converters are the companies Bosch Rexroth, Parker, Yuken, Danfoss, Atos, Linde Hydraulics, Kawasaki Hydraulics, and Poclain Hydraulics, whose technical data and mentioned parameters were used in the creation of this chapter.

5.1 Converters with rotary motion

Hydrostatic pressure converters with the rotary motion of the input component are hydraulic pumps. In the case of rotary motion of the output element, these are rotary hydraulic motors. These converters are available in many different construction designs. Depending on the type of working element by which the transmission and transfer of energy are achieved, the rotary hydrostatic pressure converters are divided into gear, screw, vane and piston. They are further divided according to design, arrangement, etc. Usually, the differences between hydraulic pumps and hydraulic motors of the same design are very small. Tab. *5.1* shows the indicative parameters and properties of some rotary hydrostatic pressure converters. Their properties are graded from 1 (best) to 4 [12], [15].

Tab. 5.1 Overview of basic parameters of hydrostatic pressure rotary converters

Converter	**Pressure [MPa]**	**Volumetric flow [$dm^3 \cdot min^{-1}$]**	**Efficiency [-]**	**Noise**	**Service life**	**Sensitivity to viscosity**	**Price**
External gear	25	300	0.85 ÷ 0.9	4	3	1	1
Internal gear	30	300	0.91	1 ÷ 2	2	2	2
Vane	17.5	200	0.8	1 ÷ 2	1	3	2
Radial piston	70	200	0.92	3	2	1	3
Axial piston	45	3 500	0.94	3	2	1	3
Screw	16	2 600	0.9	1	1 ÷ 2	1	3

5.1.1 Hydraulic pumps

Hydraulic pumps are elements with the rotary motion of the input component. They are an essential part of all hydraulic systems. They are used to convert the input mechanical energy (from an electric motor or combustion engine) into the output liquid pressure energy. In order for the liquid from the tank to get into the hydraulic pump, it is necessary to create a vacuum in the suction part. Most hydraulic pumps are characterized by a good suction ability, which allows them to be placed above the liquid level in the tank (often on the tank cover). If the hydraulic pump has a worse suction ability, it is necessary to place it below the liquid level in the tank, or it can be filled with overpressure created by another so-called filling hydraulic pump. The underpressure in the suction part needs to be monitored because cavitation can occur at high values of negative pressure. The purity of the used liquid is also an important factor for the correct operation of the hydraulic pump. Impurities cause wear on friction surfaces, which leads to a subsequent increase in functional clearances and reduces the total efficiency of the hydraulic pump. The recommended filtration may vary for individual construction designs of hydraulic pumps and is specified in the manufacturers' catalogues.

The defining parameter of hydraulic pumps (and all rotary converters) is the **geometric stroke volume** V_g. This is the liquid volume that is transported through the hydraulic pump in one revolution. It defines the size of the transducer and is usually given in cm^3 in the manufacturers' catalogues. Basically, it is the volume of internal functional parts, and its determination varies for different construction designs of hydraulic pumps. Depending on the possibility of changing the geometric stroke volume of the converter, fixed displacement hydraulic pumps (with non-adjustable geometric stroke volume) and variable displacement hydraulic pumps (with adjustable geometric stroke volume) are divided. The possibility of changing the geometric stroke volume is given by the converter design. The hydraulic symbols are shown in Fig. 5.1. Although hydraulic pumps have different designs, this is not specified in the hydraulic drawings. The graphical symbol of the fixed displacement hydraulic pump (with constant geometric volume) and with one direction of rotation of the input shaft is shown in Fig. 5.1 (top left). The graphical symbol of the variable displacement hydraulic pump with two directions of flow (shown with the sloping arrow through the converter) and two directions of rotation (i.e., with the possibility of rotation of the input shaft in both directions) is shown in Fig. 5.1 (bottom left). The graphical symbol also shows leakages with a dashed line.

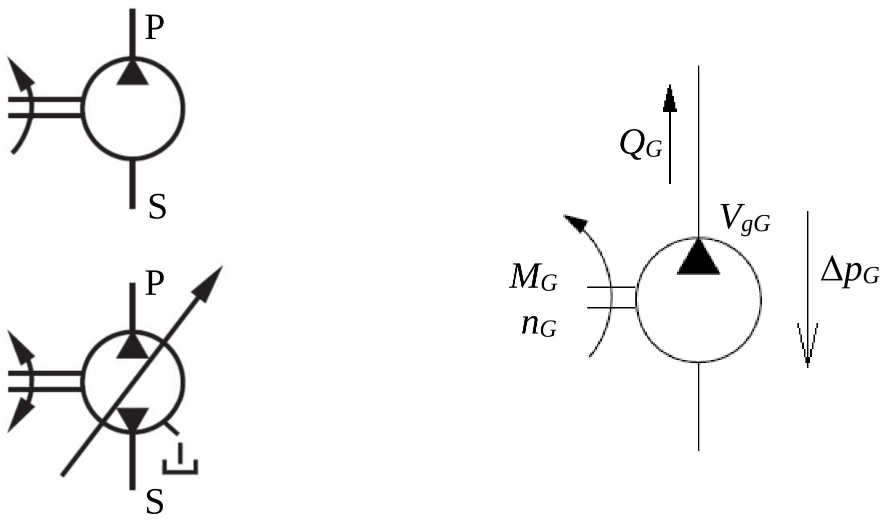


Fig. 5.1 Graphical symbols of hydraulic pumps

Based on the geometric stroke volume and input speed, it is possible to determine the theoretical volumetric flow of the hydraulic pump Q_t from the equation:

$$Q_{Gt} = V_{gG} \cdot n_G \ , \tag{5.8}$$

where Q_{Gt} [m^3·s^{-1}] is the theoretical volumetric flow of the hydraulic pump, V_{gG} [m^3] is the geometric stroke volume of the hydraulic pump, and n_G [s^{-1}] is the speed of the hydraulic pump.

The real volumetric flow Q_G of the hydraulic pump will be smaller due to leakages (flow losses) and can be expressed by the volumetric efficiency of the hydraulic pump η_{Gvol}, according to the formula:

$$Q_G = V_{gG} \cdot n_G \cdot \eta_{Gvol} \ . \tag{5.9}$$

To determine the theoretical pressure gradient Δp_{Gt} of the hydraulic pump, power equality (5.5) is assumed. The mechanical power P_m is on the pump input. It is the power of the drive motor (it represents the rotational motion of the output shaft of the drive motor). From the point of view of the hydraulic pump, this is the input power of the hydraulic pump:

$$P_m = M_G \cdot \omega_G = M_G \cdot 2\pi \cdot n_G \ , \tag{5.10}$$

where P_m [W] is mechanical power on the input to the hydraulic pump (power of the drive motor, input power of the hydraulic pump), M_G [N · m] is the drive torque moment on the input shaft of the hydraulic pump, ω_G [rad·s^{-1}] is the angular velocity of the input shaft of the hydraulic pump, and n_G [s^{-1}] is the speed of the hydraulic pump.

At the output of the hydraulic pump, the theoretical hydraulic power P_{ht} is given:

$$P_m = P_{ht} = \Delta p_{Gt} \cdot Q_{Gt} = \Delta p_{Gt} \cdot V_{gG} \cdot n_G \ , \tag{5.11}$$

where P_{ht} [W] is the theoretical hydraulic power at the output of the hydraulic pump and Δp_{Gt} [Pa] is the theoretical output pressure (pressure gradient) from the hydraulic pump.

By substituting equations (5.10) and (5.11) into equation (5.5), it is possible to determine the theoretical pressure at the outlet from the hydraulic pump:

$$\begin{gathered} P_m = P_{ht} \ , \\ M_G \cdot \omega_G = \Delta p_{Gt} \cdot Q_{Gt} \ , \\ M_G \cdot 2\pi \cdot n_G = \Delta p_{Gt} \cdot V_{gG} \cdot n_G \ , \\ \Delta p_{Gt} = \frac{M_G \cdot 2\pi}{V_{gG}} \ . \end{gathered} \tag{5.12}$$

To determine the real pressure gradient of the hydraulic pump, it is necessary to include the mechanical and pressure losses η_{Gmh} of the hydraulic pump:

$$\Delta p_G = \frac{M_G \cdot 2\pi}{V_{gG}} \cdot \eta_{Gmh} \ . \tag{5.13}$$

The hydraulic power of the hydraulic pump in accordance with equation (5.7) is given by the formula:

$$P_h = P_m \cdot \eta_{GT} \ . \tag{5.14}$$

where η_{GT} [-] is the total efficiency of the hydraulic pump.

Hydraulic pumps are produced in many different designs. The main output parameters of hydraulic pumps and the operating parameters of hydraulic systems are the volumetric flow and pressure. When choosing the optimal hydraulic pump type suitable for a given application, a number of other factors must be considered, such as lifetime, environmental influences (temperature, humidity), noise requirements, choice and possibilities of using a drive motor, range of working speeds and, of course, efficiency and price. In addition, some construction designs of the hydrostatic converters allow a continuous change of the geometric stroke volume.

Selected types of hydraulic pumps and their function principles will be described in the following chapters.

5.1.2 Static characteristics of hydraulic pumps

In the previous chapter, the output parameters of the hydraulic pumps were derived from the basic equations. Their real values are influenced by the efficiencies. The problem, however, is that efficiency is not constant but varies with a number of other factors (with pressure, speed, liquid viscosity, etc.). For a better understanding of the above relationships and the function of the hydraulic pumps, their basic static characteristics will be presented. The static characteristic is usually given as a function of one variable, while the other variables are considered constant. It is measured at a steady state, and some other parameters can be determined from its course [36].

Flow characteristic $Q = f(\Delta p_G)$

The flow characteristic represents the dependence of the flow volumetric flow Q_G of the hydraulic pump on its pressure gradient Δp_G at constant speed n_G (see Fig. 5.2 (left)) and in the case of the variable displacement hydraulic pump also at constant geometric stroke volume (control parameter φ_G). The theoretical flow Q_{Gt} of the hydraulic pump is constant, independent of the pressure gradient. It can be calculated according to equation (5.8). In fact, as the pressure gradient increases, the flow losses increase. This is expressed by the measured static characteristic, which is inclined by the angle α. From the slope of this characteristic, the loss flow Q_{Gl}, or the so-called leakage permeability G_{Gl} of the hydraulic pump, can be determined:

$$tg\alpha = \frac{Q_{Gl}}{\Delta p_G} = G_{Gl}\,. \tag{5.15}$$

The course of the characteristic can be approximated, for example, by the relation:

$$Q_G = Q_{Gt} - Q_{Gl} = V_{gG} \cdot n_G - G_{Gl} \cdot \Delta p_G\,. \tag{5.16}$$

The volumetric efficiency η_{Gvol} can be calculated at any point of the characteristic according to the equation:

$$\eta_{Gvol} = \frac{Q_G}{Q_{Gt}}\,. \tag{5.17}$$

Flow characteristic $Q_G = f(n_G)$

This characteristic represents the course of the flow Q_G of the hydraulic pump depending on its speed n_G at a constant pressure gradient Δp_G. In the case of the variable displacement hydraulic pump, also at constant geometric stroke volume (control parameter φ_G). Characteristic 1 for an ideal (lossless hydraulic pump) and characteristic 2 for an actual real hydraulic pump are shown in Fig. 5.2 (right). It is evident that tg $\delta = V_{gG}$. The course of the characteristic can be approximated by the formula:

$$Q_G = Q_{Gt} - Q_{Gl} = V_{gG} \cdot n_G - Q_{Gl} \tag{5.18}$$

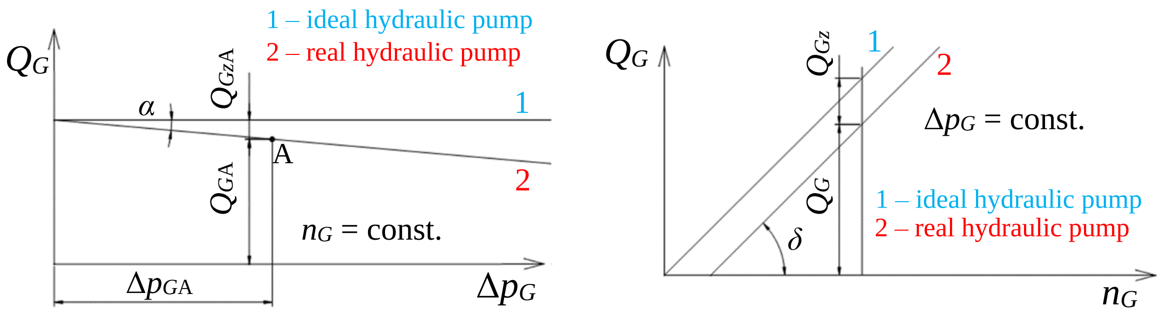


Fig. 5.2 Flow characteristic of hydraulic pump $Q_G = f(\Delta p_G)$ at constant speed n_G (left), flow characteristic of hydraulic pump $Q_G = f(n_G)$ at constant pressure gradient Δp_G (right)

Flow characteristic $Q_G = f(\varphi_G)$

In the case of a variable displacement hydraulic pump, the flow rate Q_G can be defined depending on the control parameter φ_G, while the other parameters are considered constants shown in Fig. 5.3. The control parameter φ_G of the hydraulic pump is given by the ratio of the current geometric stroke volume V_{gG} of the hydraulic pump to its maximum geometric stroke volume V_{gGmax}:

$$\varphi = \frac{V_{gG}}{V_{gGmax}} \,. \tag{5.19}$$

The course of the characteristic can be expressed by the formula:

$$Q_G = \varphi_G \cdot V_{gGmax} \cdot n_G \,. \tag{5.20}$$

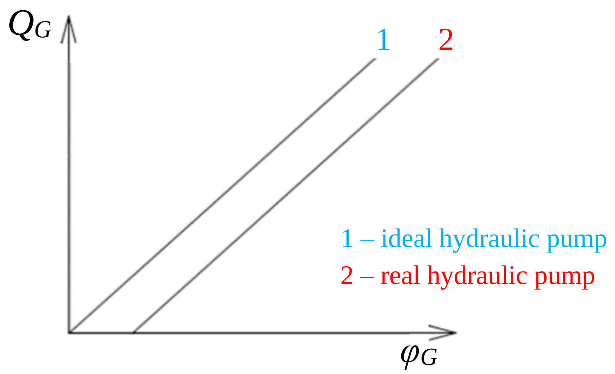


Fig. 5.3 Flow characteristic of hydraulic pump $Q_G = f(\varphi_G)$

Complex characteristics of hydraulic pump

The complex characteristic allows observing the dependence of several quantities. The relative values of the flow rate Q_G/Q_{Gmax} of the hydraulic pump, the values of the pressure gradient Δp_G on the hydraulic pump, the relative speed n_G/n_{Gmax} of the hydraulic pump and the relative values of the power P_G/P_{Gmax} of the hydraulic pump are shown in Fig. 5.4. The ranges of the total efficiency η_{Gt} of the hydraulic pump are also shown in the complex characteristic.

Complex characteristics can be found in the catalogues of some manufacturers of hydraulic pumps and are used to determine ideal operating conditions.

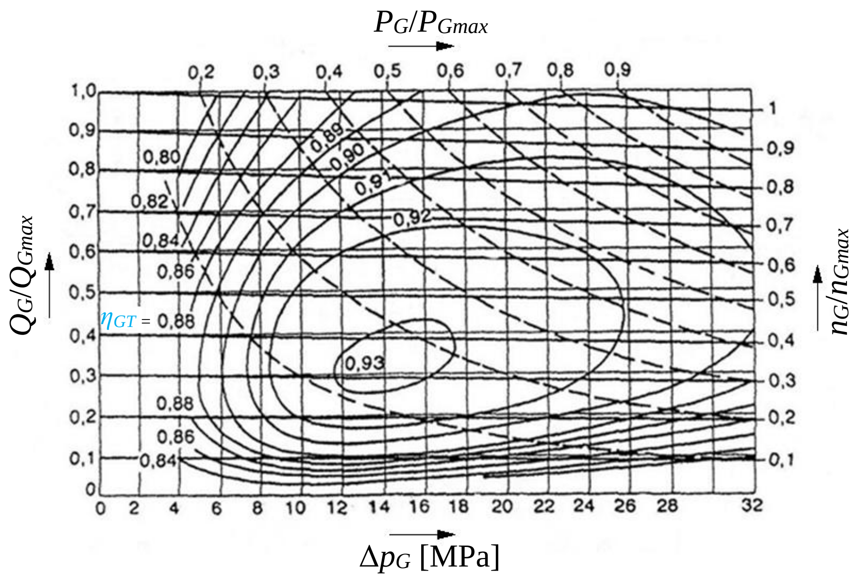


Fig. 5.4 Complex characteristic of hydraulic pump

The dependencies of the volumetric η_{Gvol} and mechanical-hydraulic η_{Gmh} efficiencies as a function of the pressure gradient Δp_G and the speed n_G of the hydraulic pump are shown in Fig. 5.5. Both efficiencies can also be expressed using the previously mentioned equations (5.9) and (5.13).

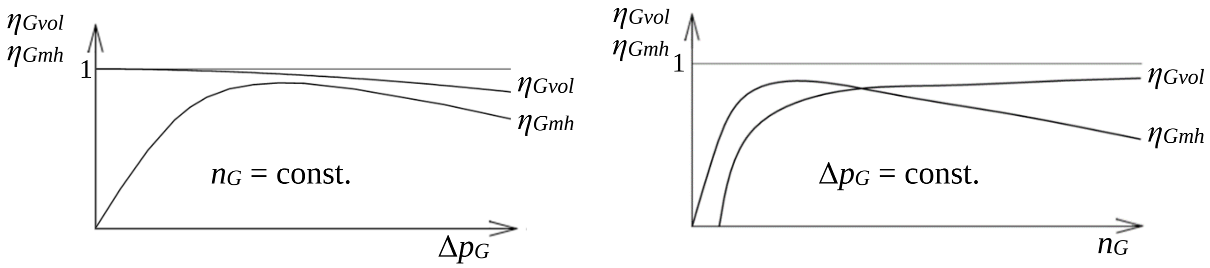


Fig. 5.5 Efficiency dependence of the hydraulic pump on pressure $\eta_G = f(\Delta p_G)$ (left), efficiency dependence of the hydraulic pump on speed $\eta_G = f(n_G)$ (right)

5.1.3 Rotary hydraulic motors

In general, hydraulic motors are the output elements of hydraulic systems. Rotary hydraulic motors convert the input liquid pressure energy into the output mechanical energy of a rotating shaft. Therefore, their function is inverse to that of hydraulic pumps. They are very close in design to hydraulic pumps and can also be designed as fixed displacement hydraulic motors (with non-adjustable geometric stroke volume) and variable displacement hydraulic motors (with adjustable geometric stroke volume). As is shown in Fig. 5.6, their graphical symbols are also similar; only the inner triangle is drawn in the opposite direction.

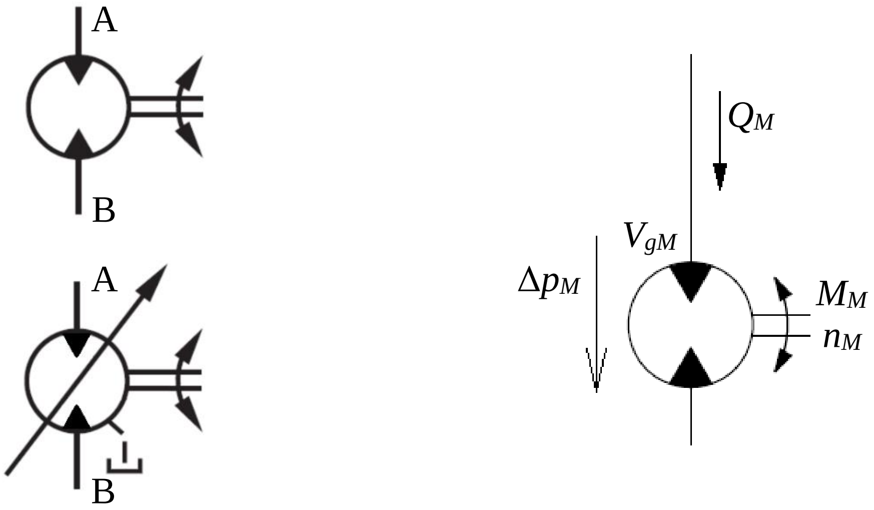


Fig. 5.6 Graphical symbols of rotary hydraulic motors

The output parameters of rotary hydraulic motors are torque and speed (angular velocity). The theoretical speed n_{Mt} of the output shaft of the hydraulic motor can be determined based on the knowledge of the geometric stroke volume V_g of the motor and the liquid input volumetric flow Q_M as follows:

$$n_{Mt} = \frac{Q_M}{V_{gM}}, \tag{5.21}$$

where n_{Mt} [s^{-1}] is the theoretical speed of the output shaft of the hydraulic motor, Q_M [$m^3 \cdot s^{-1}$] is the input volumetric flow of liquid, and V_{gM} [m^3] is the geometric stroke volume.

The real speed of the hydraulic motor n_M will be lower due to leakages (leakage flows) and can be expressed using the flow efficiency η_{Mvol} of the hydraulic motor, according to the equation:

$$n_M = \frac{Q_M}{V_{gM}} \cdot \eta_{Mvol} \,. \tag{5.22}$$

In order to determine the theoretical torque M_t of the hydraulic motor, it is again possible to proceed from the power equation (*5.5*). The hydraulic power P_h, which is at the input of the hydraulic motor, is represented by the pressure gradient of the fluid Δp and the volumetric flow rate Q:

$$P_h = \Delta p_M \cdot Q_M = \Delta p_M \cdot V_{gM} \cdot n_{Mt} \,, \tag{5.23}$$

where P_h [W] is the hydraulic power at the input of the hydraulic motor (input power of the hydraulic motor), Δp_M [Pa] is the liquid pressure gradient of the hydraulic motor and V_{gM} [m^3] is the geometric stroke volume.

The theoretical mechanical power P_{mt} of the rotary motion on the output shaft of the hydraulic motor is given by the product of the theoretical output torque M_{Mt} and the angular velocity ω_{Mt}:

$$P_{mt} = M_{Mt} \cdot \omega_{Mt} = M_{Mt} \cdot 2\pi \cdot n_{Mt} \,, \tag{5.24}$$

where P_{mt} [W] is the theoretical mechanical power on the output shaft of the hydraulic motor, M_{Mt} [N · m] is the theoretical output torque on the output shaft of the hydraulic motor, ω_{Mt} [rad·s^{-1}] is the theoretical angular velocity of the output shaft of the hydraulic motor, and n_{Mt} [s^{-1}] is the theoretical speed of the output shaft of the hydraulic motor.

By substituting equations (5.23) and (5.24) into equation (5.5), it is possible to derive the theoretical torque M_{Mt} on the output shaft of the hydraulic motor:

$$P_h = P_{mt} \,,$$

$$\Delta p_M \cdot Q_M = M_{Mt} \cdot \omega_{Mt} \,,$$

$$\Delta p_M \cdot V_{gM} \cdot n_{Mt} = M_{Mt} \cdot 2\pi \cdot n_{Mt} \,,$$

$$M_{Mt} = \frac{\Delta p_M \cdot V_{gM}}{2\pi} \,. \tag{5.25}$$

In order to obtain the real torque M_M, it is necessary to include the mechanical and pressure losses of the hydraulic motor:

$$M_M = \frac{\Delta p_M \cdot V_{gM}}{2\pi} \cdot \eta_{Mmh} \,. \tag{5.26}$$

As with hydraulic pumps, to determine the mechanical power of the hydraulic motor, it must be in accordance with equation (5.7):

$$P_m = P_h \cdot \eta_{MT} \,, \tag{5.27}$$

where η_{MT} [-] is the total efficiency of the hydraulic motor.

The choice of rotary hydraulic motors is usually based on the operating conditions and required parameters. In terms of the output speed, rotary hydraulic motors can be divided into low-speed motors with speeds up to 500 min^{-1} and high-speed motors (500 ÷ 10 000) min^{-1}. Output parameters can be further modified by using gearboxes.

5.1.4 Static characteristics of rotary hydraulic motors

As in the case of hydraulic pumps, the basic static characteristics of rotary hydraulic motors can be defined [36].

Flow characteristic $Q_M = f\,(\Delta p_M)$

The flow characteristic represents the dependence of the flow volumetric flow Q_M to the hydraulic motor on its pressure gradient Δp_M at constant speed n_M (see Fig. 5.7 (left)) and in the case of the variable displacement hydraulic motor also at constant control parameter φ_M.

The control parameter φ_M of the hydraulic motor is given by the ratio of the current geometric stroke volume V_{gM} of the hydraulic motor to its maximum geometric stroke volume V_{gMmax}:

$$\varphi_M = \frac{V_{gM}}{V_{gMmax}}. \tag{5.28}$$

The leakage permeability G_{MB} of the hydraulic motor at point B of the characteristics is given by the formula:

$$G_{MB} = \tan\beta = \frac{Q_{MB} - V_{gM} \cdot n_M}{\Delta p_{MB}}. \tag{5.29}$$

Torque characteristic $M_M = f\,(n_M)$

The torque characteristic represents the dependence of torque M_M on the output shaft of the hydraulic motor on its speed n_M at a constant pressure gradient Δp_M on the hydraulic motor, as shown in Fig. 5.7 (right). Starting torque M_{Ms} and the limiting speed of the hydraulic motor n_{Mlim} are drawn in this characteristic. At lower speeds than the limit speed, the hydraulic motor cannot be operated due to uneven operation (jerky motion and stopping). The course of characteristic 1 is given for the ideal hydraulic motor; characteristic 2 corresponds to the real hydraulic motor, including the consideration of losses.

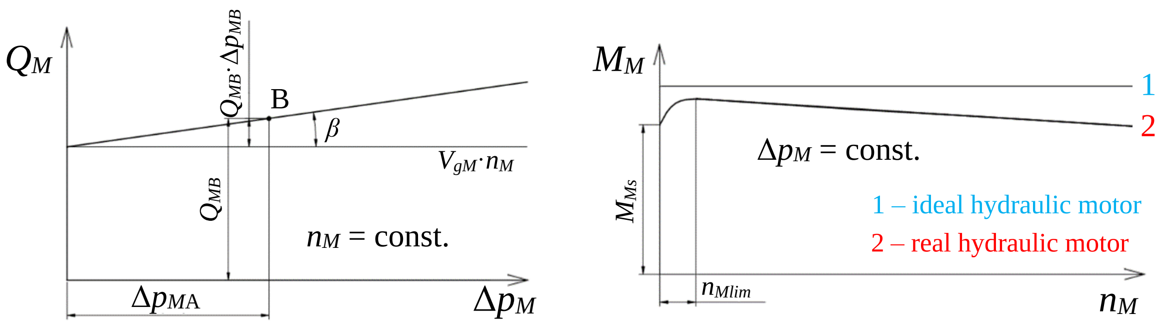


Fig. 5.7 Flow characteristic $Q_M = f\,(\Delta p_M)$ of hydraulic motor (left), torque characteristic of hydraulic motor $M_M = f\,(n_M)$ (right)

Torque characteristic $M_M = f\,(\Delta p_M)$

The torque characteristic represents the dependence of the torque M_M on the output shaft of the hydraulic motor on its pressure gradient Δp_M at a constant pressure speed n_M. The course of characteristic 1 is given for the ideal hydraulic motor; characteristic 2 corresponds to the real hydraulic motor including the consideration of losses, as is shown in Fig. 5.8 (left).

Speed characteristic $n_M = f\,(M_M)$

The speed characteristic (see Fig. 5.8 (right)) represents the dependence of the speed n_M of the hydraulic motor on its output torque M_M at a constant volumetric flow Q_M on the input of the hydraulic motor. An important parameter of the speed characteristic is the stiffness k_M, which can be expressed by the formula:

$$k_M = tan\,\gamma = \frac{\Delta M_M}{\Delta n_M}. \tag{5.30}$$

The course of characteristic 1 is given for the ideal hydraulic motor; characteristic 2 corresponds to the real hydraulic motor.

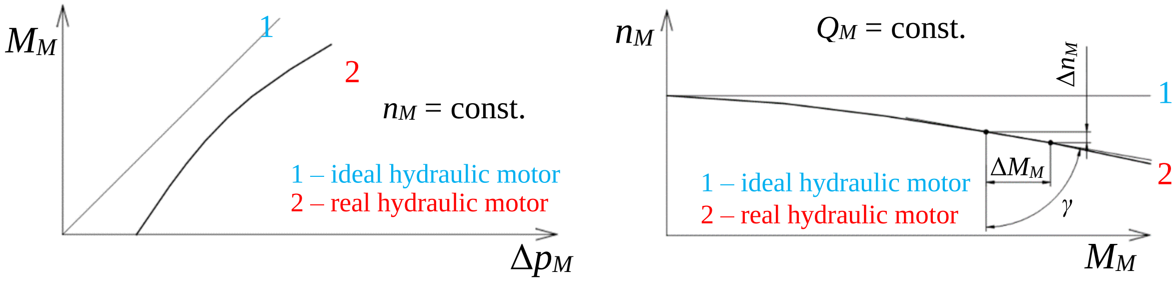


Fig. 5.8 Torque characteristic $M_M = f(\Delta p_M)$ of hydraulic motor at constant motor speed n_M (left), speed characteristic $n_M = f(M_M)$ at constant volumetric flow Q_M on motor input (right)

Complex characteristic of a hydraulic motor

Also, in the case of hydraulic motors, the manufacturer's catalogue may contain a complex characteristic where several quantities are shown in one graph. The values of the volumetric flow rate Q_M through the hydraulic motor, the values of the pressure gradient Δp_M on the hydraulic motor, the speed n_M of the hydraulic motor, the values of the power P_M of the hydraulic motor and the values of torque M_M on the output shaft of the hydraulic motor are shown in Fig. 5.9. Basically, it is a combination of the above dependencies. The characteristic also shows the areas that indicate the total efficiency η_{MT} of the hydraulic motor at the given operating parameters.

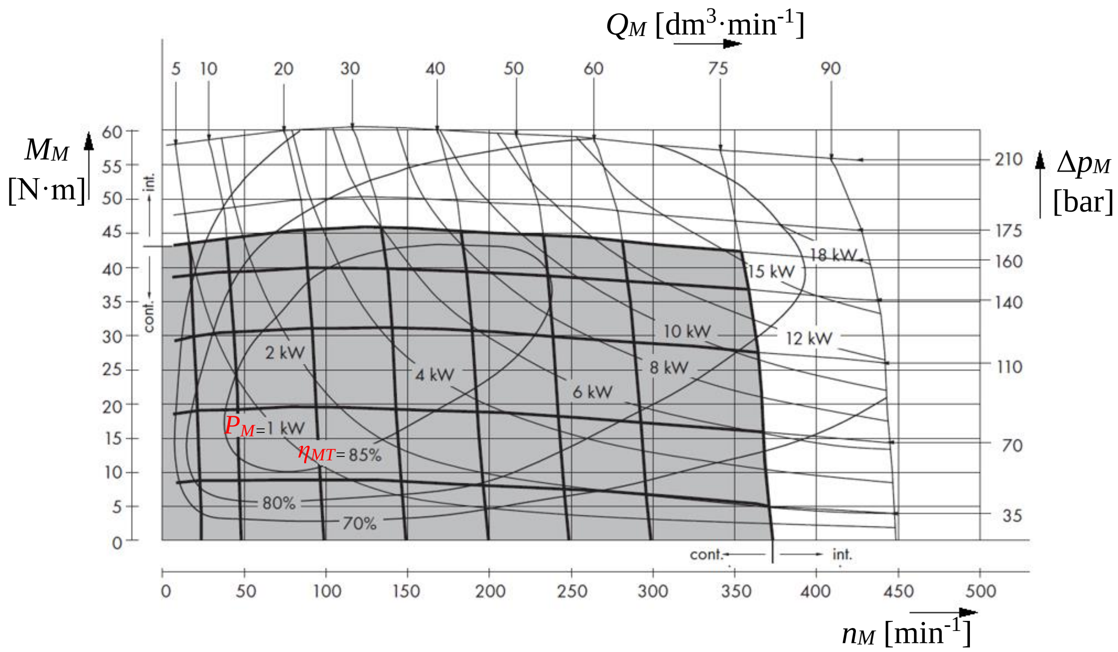


Fig. 5.9 Complex characteristic of hydraulic motor

5.1.5 Gear hydraulic pumps and motors

They are the basic and most widely used types of hydraulic pumps and are also used as rotary hydraulic motors in simple applications. They are characterized by their simple construction, good suction ability and operation reliability. Depending on their construction, gear converters are divided into internal gear, external gear and epicyclic gear converters. All gear converters are produced only with a constant geometric stroke volume.

External gear pumps

The function principle of the external gear hydraulic pump is shown in Fig. 5.10. The hydraulic pump consists of the drive (1) and driven (2) gears. The drive gear is connected to the drive shaft (3), which passes through the shaft seal (4) located in the front cover (5). The liquid is transported from the input area (suction) in tooth gaps around the circumference of the wheels to the output area of the hydraulic pump (discharge). The gears are supported on bearings (6), in which a specially shaped seal is placed. The bearings are usually sliding and must be able to withstand high pressures. There are axial clearances between the surfaces of the gear and the bearing housing. These clearances reduce the volumetric efficiency of the hydraulic pump. In the basic design, the hydraulic pump has fixed axial clearances. This type can be used for lower pressures and has significantly lower efficiency compared to the more commonly used hydraulic pump with the compensation of axial clearances. The liquid is guided into the sealed groove at the discharge side, where the pressure of the liquid acting on the outer surfaces of the bearing housings presses them against the gears [11], [23].

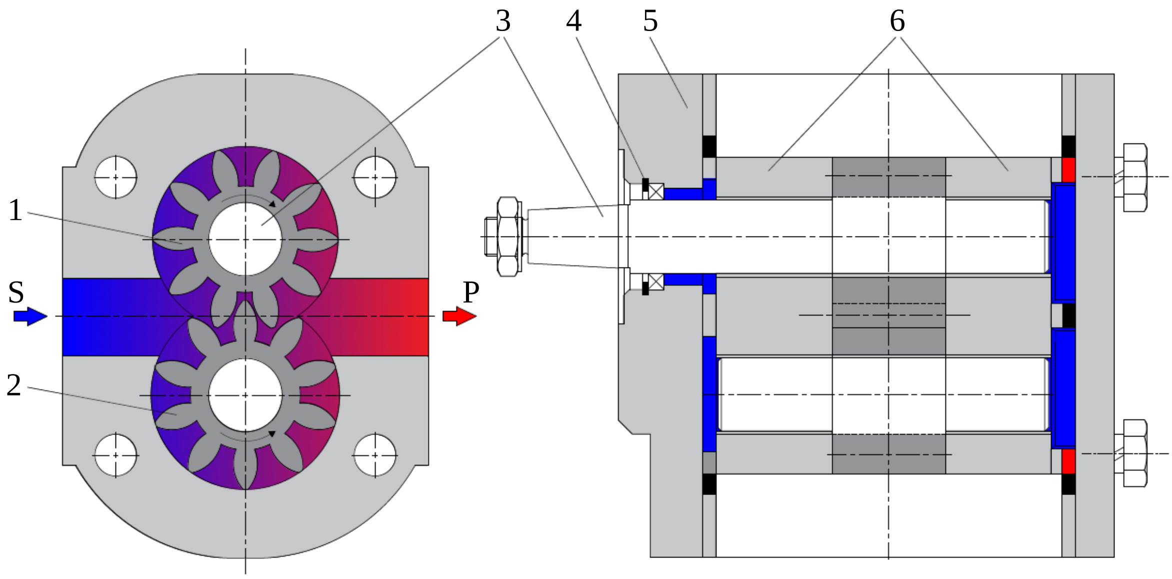


Fig. 5.10 External gear hydraulic pump

1 – drive gear, 2 – driven gear, 3 – drive shaft, 4 – shaft seal, 5 – front cover, 6 – bearing housings

In addition to the axial clearances, the hydraulic pump also has radial clearances around the circumference of the head circle of the gears. During the operation of the hydraulic pump, the gears are pressed by the higher pressure from the discharge against the suction side, resulting in radial clearances on the discharge side. It is possible to define these clearances by hydrostatic balancing of face plates.

The pressure distribution and pressure forces acting on the gears during operation are not even. This leads to increased requirements for the dimensioning of the pins and bearings. In addition, a small liquid volume is closed in the space between the head and the root of engaged teeth. This liquid is subsequently compressed and causes an additional load on the gear axles and bearings, resulting in a pressure increase. In order to prevent this from happening, there are relief grooves in the bearing housings to drain the liquid from these spaces.

The geometric stroke volume of the hydraulic pump can be approximately determined from the equation [12]:

$$V_g = 2\pi \cdot b \cdot m_g^2 \cdot z \,, \tag{5.31}$$

where V_g [m^3] is the theoretical geometric stroke volume of hydraulic pump, b [m] is the width of the gears, m_g [m] is the gear module, and z [-] is the number of teeth of the drive wheel.

An increase in the geometric stroke volume (while maintaining the wheel diameter) can be achieved by increasing the wheel width. Increasing the tooth module is not suitable because it leads to a reduction in the number of teeth, which has a negative effect on leakages and increases noise and flow pulsations. Flow pulsations are determined by the tooth gap size that expels the liquid at the discharge side. Reduction of pulsations can be ensured by the integration of a pair of gears which are rotated relative to each other by half the tooth pitch. Basically, these are two hydraulic pumps in one body with standard suction and discharge. In this way, the pulsations can be reduced by up to 10%. External gear hydraulic pumps are usually produced with straight teeth and are very noisy in operation. A noise reduction can be achieved by using helical or herring-bone gearings, but this significantly increases their purchase price. Flow control is only possible by changing the speed,, e.g. with an electric motor using a frequency converter. The basic parameters and properties of external gear hydraulic pumps including their applications, are shown in Tab. 5.2.

Tab. 5.2 Basic parameters and properties of external gear hydraulic pumps

<table>
<tr><th>Speed
n [min^{-1}]</th><th>Geometric stroke volume
V_g [cm^3]</th><th>Flow
(at n = 1500 min^{-1})
Q [dm^3 · min^{-1}]</th><th>Nominal pressure
p_n [MPa]</th></tr>
<tr><td>500 ÷ 6 000</td><td>0.2 ÷ 200</td><td>6 ÷ 250</td><td>21 ÷ 25</td></tr>
<tr><td>Operating viscosity
ν [mm^2 · s^{-1}]</td><td>Starting viscosity
ν [mm^2 · s^{-1}]</td><td>Minimum viscosity
ν [mm^2 · s^{-1}]</td><td>Total efficiency
η_T [-]</td></tr>
<tr><td>20 ÷ 100</td><td>2 000</td><td>12</td><td>0.75 ÷ 0.90</td></tr>
<tr><td colspan="2">Advantages</td><td colspan="2">Disadvantages</td></tr>
<tr><td colspan="2">Low cost, simple construction, good suction ability, high specific power and compact design, low sensitivity to viscosity, the possibility of tandem connection of more hydraulic pumps.</td><td colspan="2">Noise, flow pulsations, only with a constant geometric stroke volume, and axial clearance compensation are required for higher efficiency (it increases their price).</td></tr>
<tr><td colspan="4">Applications</td></tr>
<tr><td colspan="4">Simple applications in mobile and stationary hydraulics, low pressure hydraulic pumps of machine tools, lubrication technique, auxiliary hydraulic pumps of axial piston hydraulic pumps, loaders, dumpers, and servo control of trucks and tractors.</td></tr>
</table>

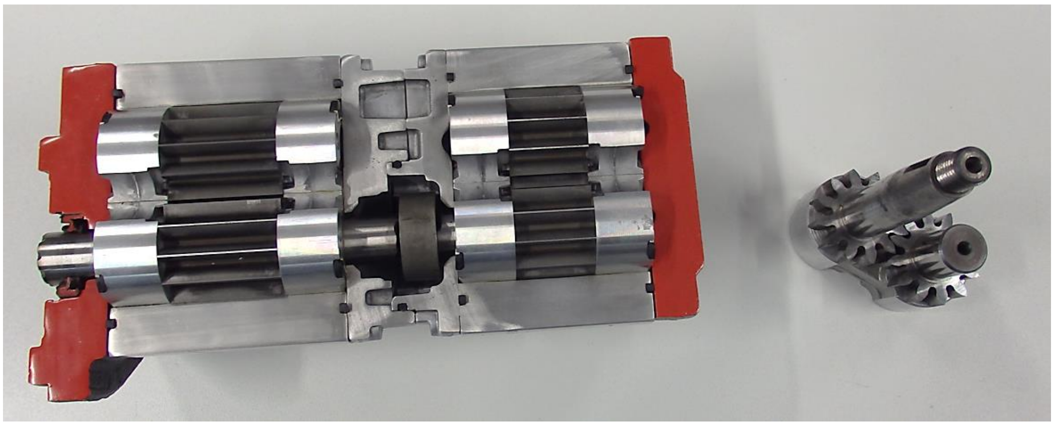


Fig. 5.11 An example of two series-connected external gear pumps

External gear motors

They are practically identical in their construction to hydraulic pumps. The motor's function is reversed, i.e. the liquid is fed into the space between the pair of gears. One wheel is connected to the output shaft. When the input space is filled with the liquid, a pressure gradient is created on the motor (between the input and output). After overcoming the resistance, it will set the wheels in motion.

They are used only in the version with axial clearance compensation. Compared to hydraulic pumps, hydraulic motors are usually required to operate in both directions of rotation, for which the clearance compensation must be adapted to their construction. Rolling bearings are used to ensure the reliable starting of these hydraulic motors. They are not suitable for low speeds where the motor efficiency decreases considerably and the stick-slip effect becomes apparent. The basic parameters and properties of these hydraulic motors are given in Tab. 5.3 [12], [6].

Tab. 5.3 Basic parameters and properties of external gear hydraulic motors

Minimum speed n_{min} [min^{-1}]	Maximum speed n_{max} [min^{-1}]	Torque M [N · m]	Nominal pressure p_n [MPa]
200 ÷ 500	2 000 ÷ 4 000	2 ÷ 130	9 ÷ 30
Geometric stroke volume V_g [cm^3]	Operating viscosity ν [mm^2 · s^{-1}]	Total efficiency η_T [-]	
1.4 ÷ 80	12 ÷ 800	0.8	
Advantages		Disadvantages	
Low price, simple construction, high specific power and compact design, low sensitivity to viscosity.		Noise, low efficiency, only with a constant geometric stroke volume, axial clearance compensation required (it increases their price), not suitable for low speed, difficult starting.	
Applications			
Simple applications, agricultural machinery, mobile technology, screw conveyors.			

Internal gear pumps

This design of the hydraulic pump (see Fig. 5.12) consists of the gear wheel with internal gearing (1), which rotates freely in the body of the hydraulic pump and the pinion (2), which is connected to the drive shaft (3). The space between the wheel and the pinion is limited by the crescent (4), which also separates the suction and discharge parts of the hydraulic pump. Thus, both wheels rotate in the same direction and the liquid is transported in their tooth gaps from the input to the output of the hydraulic pump [10], [30].

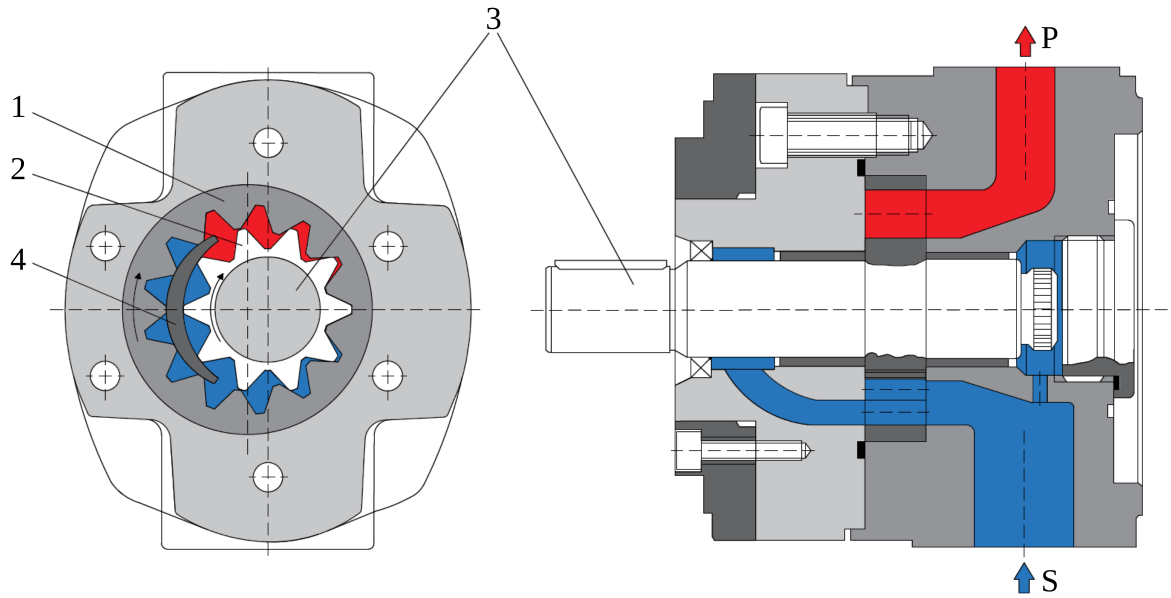


Fig. 5.12 Internal gear hydraulic pump

1 – gear wheel with internal gearing, 2 – pinion, 3 – drive shaft, 4 – crescent

The drive shaft is housed in plain bearings, to which the forces from the pinion are transmitted. The space of the outer surfaces of the gears is limited by a plate embedded in the bearing covers. During the operation of the hydraulic pump, the working pressure of the liquid acts on the distance plate and creates the required pressure force. Radial clearances are solved by a pressure segment, which ensures the necessary pressure on the wheel with internal gearing. This construction is characterized by very low volumetric flow losses, which increases the total efficiency of the hydraulic pump [12], [6].

The geometric stroke volume of the internal gear pump can be calculated in the same way as for the previous type using equation (5.31), where z is the number of teeth of the pinion. When the internal and external gearing engage, where the pinion has a convex tooth face and the gear is concave, the contact pressure is reduced, and the filling and emptying of the tooth gaps is slower. In the case of hydraulic pumps, this leads to a reduction in noise, vibrations and flow pulsations. The basic parameters and operating properties are presented in Tab. 5.4.

Tab. 5.4 Basic parameters and properties of internal gear hydraulic pumps.

Speed n [min^{-1}]	Geometric stroke volume V_g [cm^3]	Flow (at n = 1 500 min^{-1}) Q [dm^3 · min^{-1}]	Nominal pressure p_n [MPa]
500 ÷ 4 500	3 ÷ 250	6 ÷ 350	17 ÷ 35
Operating viscosity ν [mm^2 · s^{-1}]	Starting viscosity ν [mm^2 · s^{-1}]	Minimum viscosity ν [mm^2 · s^{-1}]	Total efficiency η_T [-]
15 ÷ 100	2 000	10	0.85 ÷ 0.90
Advantages		Disadvantages	
Very quiet, low pulsations, high specific power and compact design, higher working pressures than external gear pumps, the possibility of tandem connection of multiple hydraulic pumps.		Only with a constant geometric stroke volume, higher price compared to external gear pumps.	
Applications			
Presses, injection presses, quiet high-pressure aggregates, machine tools.			

Hydraulic motors of this construction are seldom used.

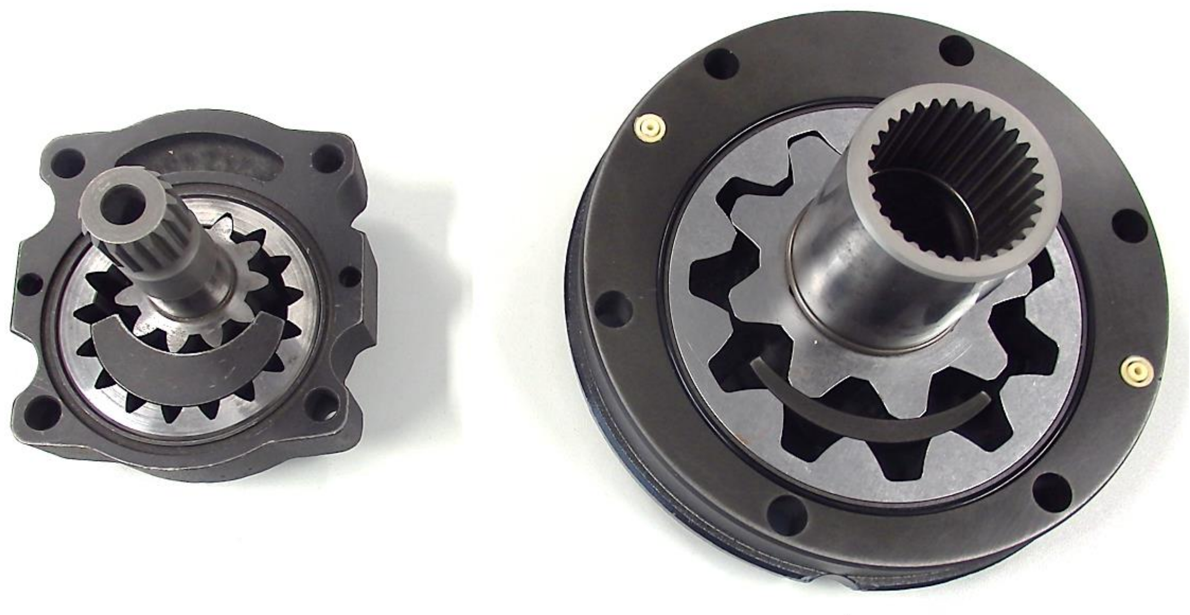


Fig. 5.13 An example of internal gear pumps

Ring gear pumps (Gerotor pumps)

In practice, this type of hydraulic pump is often referred to as a so-called Gerotor. The main parts of the hydraulic pump (see Fig. 5.14) are the external gear (1) with internal gearing and the pinion (2) connected to the drive shaft (3). The pinion is eccentrically mounted to the outer wheel and has one less tooth. When the pinion rotates, the outer gear rotates through the gearing. The difference in the number of teeth creates a gradually increasing and decreasing internal volume during movement, thereby achieving a suction and discharge effect. The suction and discharge are realized by means of kidney-shaped grooves in the side distributor plate and are divided by approximately half of the revolution of the pinion [23].

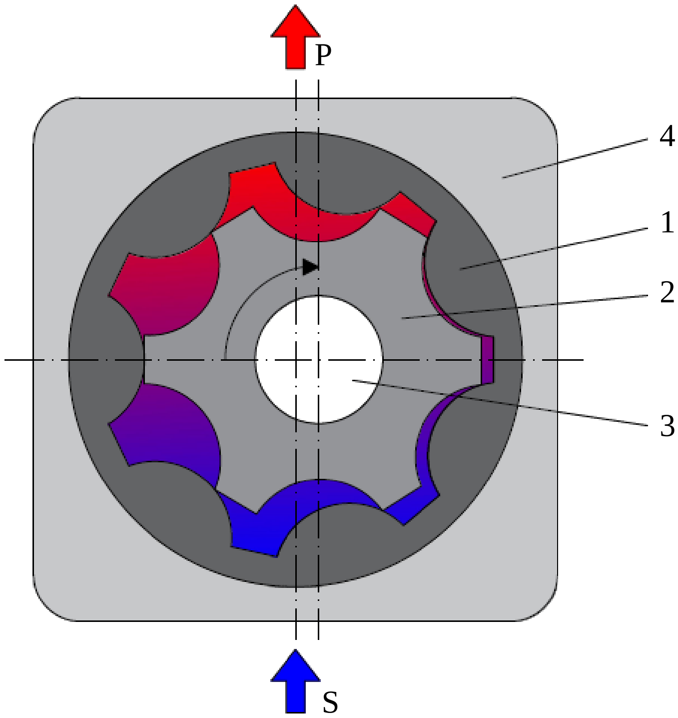


Fig. 5.14 Ring gear pump

1 – gear wheel with internal gearing, 2 – pinion, 3 – drive shaft, 4 – hydraulic pump body

The geometric stroke volume can be approximately determined by the equation [12]:

$$V_g = \frac{\pi}{z_1} \cdot (R_1^2 - R_2^2) \cdot b \cdot z_2\,, \tag{5.32}$$

where V_g [m^3] is the geometric stroke volume of the hydraulic pump, z_1 [-] is the number of teeth of the pinion, z_2 [-]is the number of teeth of the outer wheel, b [m] is the width of gears, R_1 [m] is the radius, $R_1 = R - r + e$, R [m] is the stator radius, r [m] is the shaft radius, e [m] is the eccentricity, and R_2 [m] is the radius, $R_2 = R - r - e$.

Hydraulic pumps of this type are used for lower pressures, usually up to a maximum of 16 MPa. This is due to their construction and function. Internal volumes are limited by the contact of relatively small tooth surfaces. This also reduces the efficiency of these hydraulic pumps, and both gears must be manufactured with high precision, which increases production costs. The basic parameters and operating properties of ring gear hydraulic pumps are presented in Tab. 5.5.

Tab. 5.5 Basic parameters and properties of ring gear hydraulic pumps

Speed n [min^{-1}]	Geometric stroke volume V_g [cm^3]	Flow (at n = 1 500 min^{-1}) Q [dm^3 · min^{-1}]	Nominal pressure p_n [MPa]
max. 4 000	3.2 ÷ 40	4 ÷ 400	12 ÷ 16
Operating viscosity ν [mm^2 · s^{-1}]	Starting viscosity ν [mm^2 · s^{-1}]	Minimum viscosity ν [mm^2 · s^{-1}]	Total efficiency η_T [-]
16 ÷ 150	2 000	10	0.70 ÷ 0.85
Advantages		Disadvantages	
Very small storage space, quiet operation, good suction ability, high specific power and compact design.		Low efficiency, lower working pressures, only with a constant geometric stroke volume.	
Applications			
Simple applications, hydraulic servo controls, filling pumps of axial piston hydraulic pumps, filtration and cooling circuits.			

Epicyclic gear motors

Hydraulic motors of this construction are used in various designs. The function, in this case, is different from ring gear hydraulic pumps. They are most often manufactured in the Geroler version, as shown in Fig. 5.15, with a so-called movable connecting axis. The outer wheel (1) is fitted with pulleys and must be immovable. The internal pinion (2) has one less tooth than the number of pulleys. When the liquid is fed into the hydraulic motor, the internal pinion is set in motion by the pressure and starts to rotate around its own axis. Due to its placement in the housing, this causes simultaneous rolling of the pinion against the pulleys and eccentric rotation in the opposite direction. For this reason, these motors are sometimes referred to as orbital motion motors. This motion is transmitted to the output shaft of the motor by means of the drive (cardan) shaft (3) [11].

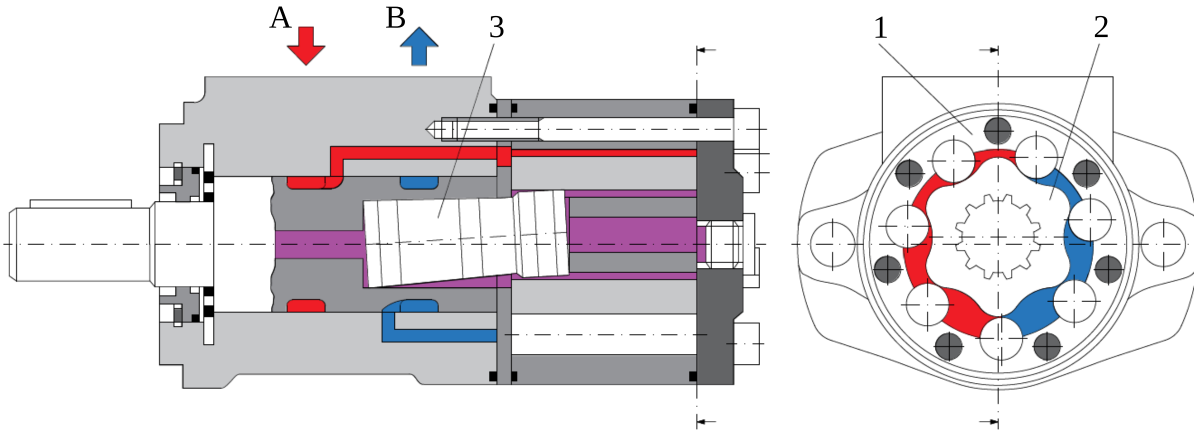


Fig. 5.15 Epicyclic gear hydraulic motor with drive (cardan) shaft

1 – outer wheel with pulleys, 2 – internal pinion, 3 – drive (cardan) shaft

In order to ensure the pinion rotation, the resulting internal volumes must always be alternately connected to the input (pressure) and output, for which the radial distribution plate must be adapted. Then, the ratio of the number of teeth of the outer and inner gear determines the gear ratio in relation to the output shaft rotation. This fact allows the use of eccentric hydraulic motors also as low-speed motors (for low output speed). The more complicated construction and the necessity to convert the pinion movement increase the production cost.

For an idea, it is also possible to mention the version of the motor with a fixed connecting axis. Epicyclic gear hydraulic motor type MZD from Bosch Rexroth is shown in Fig. 5.16. The design of the distributor plate of this motor is shown in Fig. 5.17 [11].

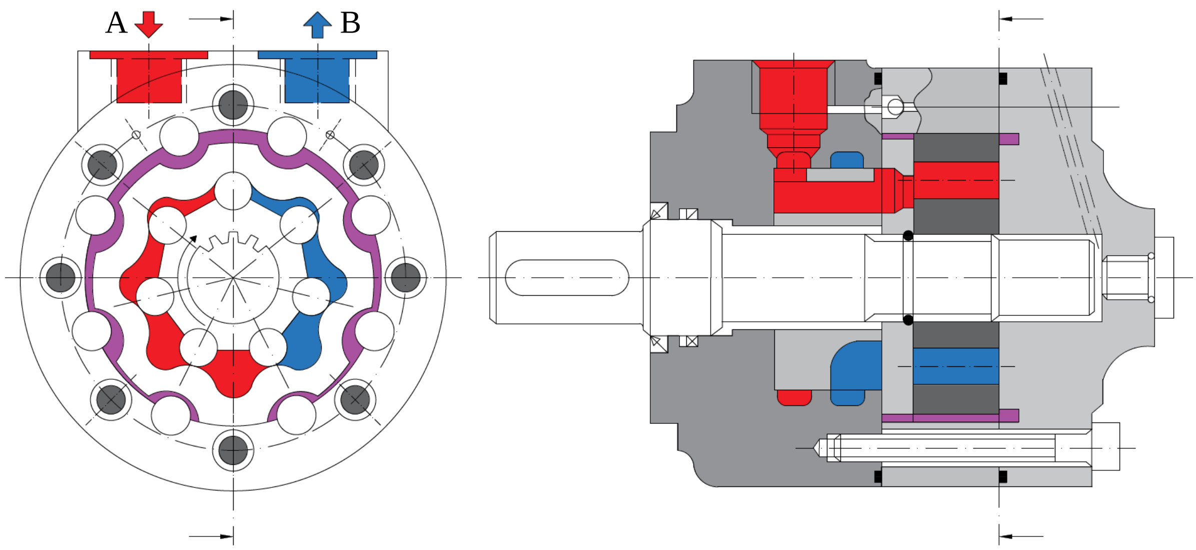


Fig. 5.16 Epicyclic gear motor, type MZD (Bosch Rexroth)

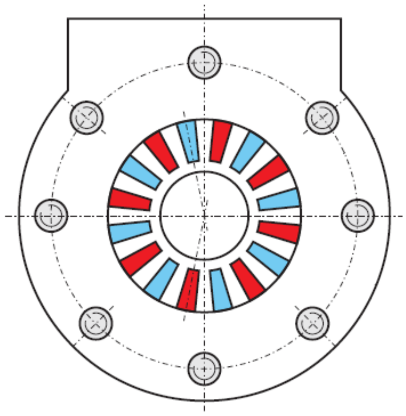


Fig. 5.17 The design of the distributor plate of epicyclic gear motor, type MZD (Bosch Rexroth) [11]

The basic properties and parameters of these hydraulic motors are shown in Tab 5.6. Due to the different constructions, the values correspond to the drive shaft design, and the parameters of the epicyclic hydraulic motors are in brackets.

Tab 5.6 Basic parameters and properties of epicyclic gear hydraulic motors

Minimum speed n_{min} [min^{-1}]	Maximum speed n_{max} [min^{-1}]	Torque M [N · m]	Nominal pressure p_n [MPa]
10 ÷ 50 (500)	80 ÷ 1 500 (4 000)	25 ÷ 1 000 (0.7 ÷ 25)	14 ÷ 17.5
Geometric stroke volume V_g [cm^3]	Operating viscosity ν [mm^2 · s^{-1}]	Total efficiency η_T [-]	
8 ÷ 800	60 ÷ 120	0.82 ÷ 0.85	
Advantages		Disadvantages	
High specific power, also for use as a low-speed hydraulic motor, suitable for difficult working conditions.		Higher price, lower efficiency, only with a constant geometric stroke volume.	
Applications			
Usually as a low-speed hydraulic motor in mobile hydraulics.			

5.1.6 Screw pumps

The screw-type converters are used only as hydraulic pumps. They are manufactured with either two or three warm gears, as shown in Fig. 5.18. One warm gear is driving (1), fixedly connected to the input shaft and the remaining warm gears (2) are driven. In the case of this hydraulic pump type, the liquid in body 5 is only transported by the warm gears from the input (suction) area (3) to the output (discharge) area (4). The helix is continuously opened at the output, eliminating flow pulsations. They are characterized by very good suction ability and quiet operation. In the case of the three warm gear design, they are pressure-balanced, which allows them to be operated even at high speeds and thus achieve a high volumetric flow with a relatively small geometric stroke volume [23], [29].

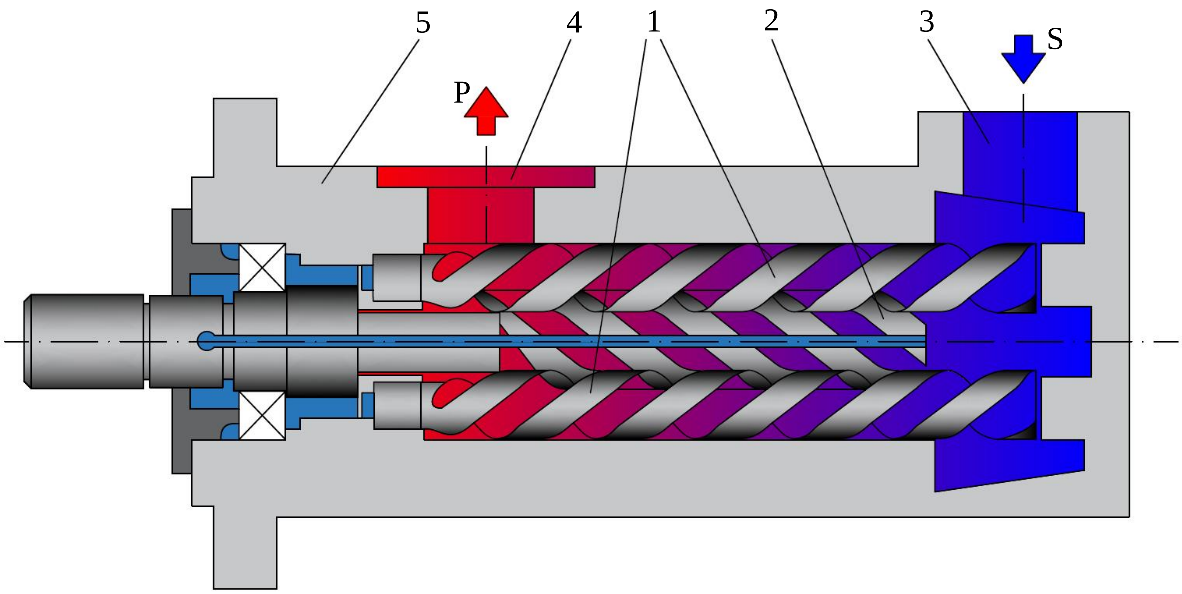


Fig. 5.18 Screw hydraulic pump

1 – drive warm gear, 2 – driven warm gears, 3 – suction part of hydraulic pump, 4 – discharge of the hydraulic pump, 5 – housing of the hydraulic pump

They are mainly used in applications where quiet operation is required. Their low viscosity requirements also allow their use in the lubrication technique. The production of warm gear is relatively expensive and significantly increases the purchase costs. The basic properties and parameters of screw pumps are shown in Tab 5.7 [38].

Tab 5.7 Basic parameters and properties of screw hydraulic pumps

Speed n [min^{-1}]	Geometric stroke volume V_g [cm^3]	Flow (at n = 1500 min^{-1}) Q [dm^3 · min^{-1}]	Nominal pressure p_n [MPa]
1 000 ÷ 4 500	9 ÷ 1 750	13 ÷ 2 500	8 ÷ 16
Operating viscosity ν [mm^2 · s^{-1}]	Starting viscosity ν [mm^2 · s^{-1}]	Minimum viscosity ν [mm^2 · s^{-1}]	Total efficiency η_T [-]
8 ÷ 100	2 500	2	0.9
Advantages		Disadvantages	
Very quiet, pulsation-free flow, reliable and with high lifetime, also used for high-viscosity liquids.		High price, lower working pressures, only with a constant geometric stroke volume.	
Applications			
Hydraulic pumps for cooling and filtration circuits, liquid transport in lubrication technique.			

5.1.7 Vane pumps and motors

Vane converters are mainly used as hydraulic pumps in hydraulic systems, but they can also work as hydraulic motors. They are characterized by a relatively simple construction and, compared to gear converters, they have smaller dimensions at the same volumetric flow rate, and work with a quieter operation and with small pulsations. In some designs, they also allow a continuous change of the geometric stroke volume and, therefore, the possibility of controlling the output parameters. In general, vane converters are very sensitive to maintaining the prescribed liquid viscosity. Due to the generated centrifugal forces acting on vanes, they are used for limited speeds.

Vane pumps

Vane hydraulic pumps can be divided according to different designs:

- by a number of strokes per revolution:
  single chamber (with a so-called unbalanced rotor),
  double chamber (with two strokes per revolution),

- according to the sense of rotation:
  unidirectional (for one sense of rotation),
  bidirectional (for both senses of rotation),

- according to the possibility of control:
  with constant geometric stroke volume (fixed),
  with adjustable geometric volume (variable).

As shown in Fig. 5.19 (left), the main parts of each vane hydraulic pump are the stator (1) and the eccentrically mounted cylindrical rotor (2), which has radial grooves in which vanes are placed. The rotor is connected to the drive shaft; during its rotation, the vanes are pushed against the inner stator surface due to centrifugal force. Due to the eccentric placement and extension of the vanes, a gradually expanding wedge-shaped space is created in the first half of the revolution, which is connected to the suction side of the hydraulic pump. In the second half of the rotor speed, this space is decreased and connected to the discharge. The suction and discharge sides are provided by an axially located distribution plate with kidney grooves. The distance between the discharge and suction grooves must be longer than the vanes' spacing. Otherwise, the input and output of the hydraulic pump would be connected. In the case of the single chamber vane pump, the rotor is not pressure balanced and is subject to significant forces that must correspond to the bearings dimensioning [11], [12].

The vanes can be single (usually used for hydraulic pumps with rotation in one direction only) or double (for bidirectional hydraulic pumps), which have the better sealing ability, as shown in Fig. 5.19 (right). It is the sealing ability of the vanes that significantly affects the volumetric efficiency of the hydraulic pump. The centrifugal force acting on the vanes may not be sufficient at start-up, at low operating speeds, and at higher pressures. For this reason, in some cases, springs are placed in the space under the vanes, or a groove is made into the space under the vanes for the liquid supply from the pressure part, where the liquid pressure ensures the pressure of the vanes against the stator. The liquid also ensures the lubrication of friction surfaces.

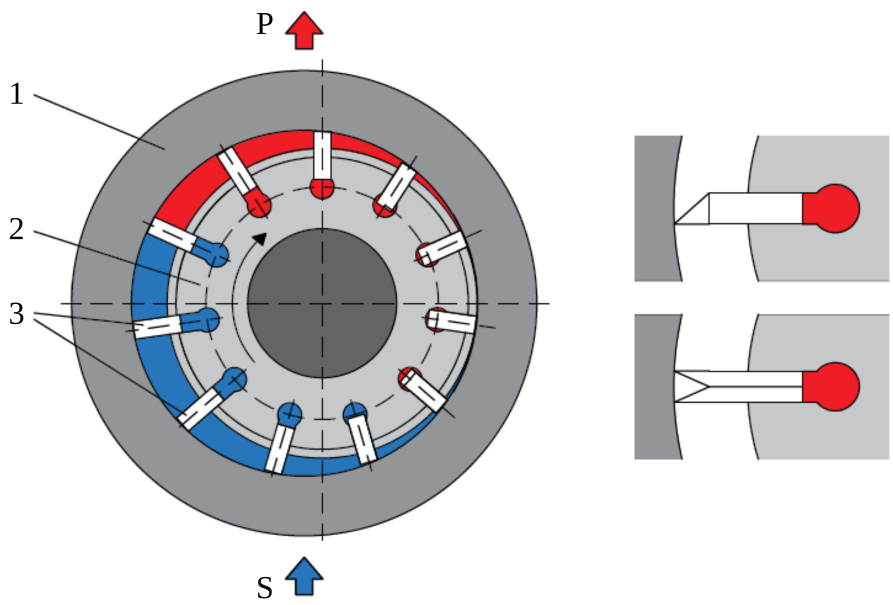


Fig. 5.19 Single chamber vane pump (left), vanes (right)

1 – stator, 2 – rotor, 3 - vanes

The converter of the above construction can be in a fixed design or with the possibility of changing the geometric stroke volume. A directly controlled variable displacement vane pump is shown in Fig. 5.20. The change in geometric volume or reverse movement of the hydraulic pump is caused by a change in eccentricity. This is achieved by the position change of the stator ring (1). In the simplest design of the hydraulic pump, the maximum eccentricity is given by means of the set screw (3) which acts against the spring (4). In the initial setting, the stator is in an eccentric position due to the spring force. The required system pressure is adjusted by spring preload using a set screw (5) [30].

This is the so-called **constant pressure control**. As the liquid pressure increases, the pressure force acting on the inner stator surface increases. If the horizontal component of this force exceeds the value set on the spring, the stator ring is moved to the right, reducing eccentricity. This reduces the geometric stroke volume and also the volumetric flow of the hydraulic pump (at constant speed). The volumetric flow at a given time corresponds to the current consumption of the system. If this consumption is zero, the pressure rises to the set value, and the stator ring moves to the zero-eccentricity position. Thus, the hydraulic pump maintains a constant output pressure and supplies only the minimum amount of liquid (needed to cover the flow losses) to the system. In the case of an increase in the liquid consumption by the system and a consequent decrease in the output pressure, there will be an imbalance of the forces acting on the stator and an almost immediate increase in eccentricity and, thus, an increase in volumetric flow. This control method of the hydraulic pump is energy efficient, and reduces power loss and liquid heating. If a fixed displacement hydraulic pump was used in the system, the required pressure would be limited by a pressure valve through which the liquid would flow at a given pressure gradient, resulting in power dissipation. Mechanically controlled hydraulic pumps are used only for lower pressures up to about 10 MPa and smaller volumetric flows up to about 50 $dm^3 \cdot min^{-1}$ [38].

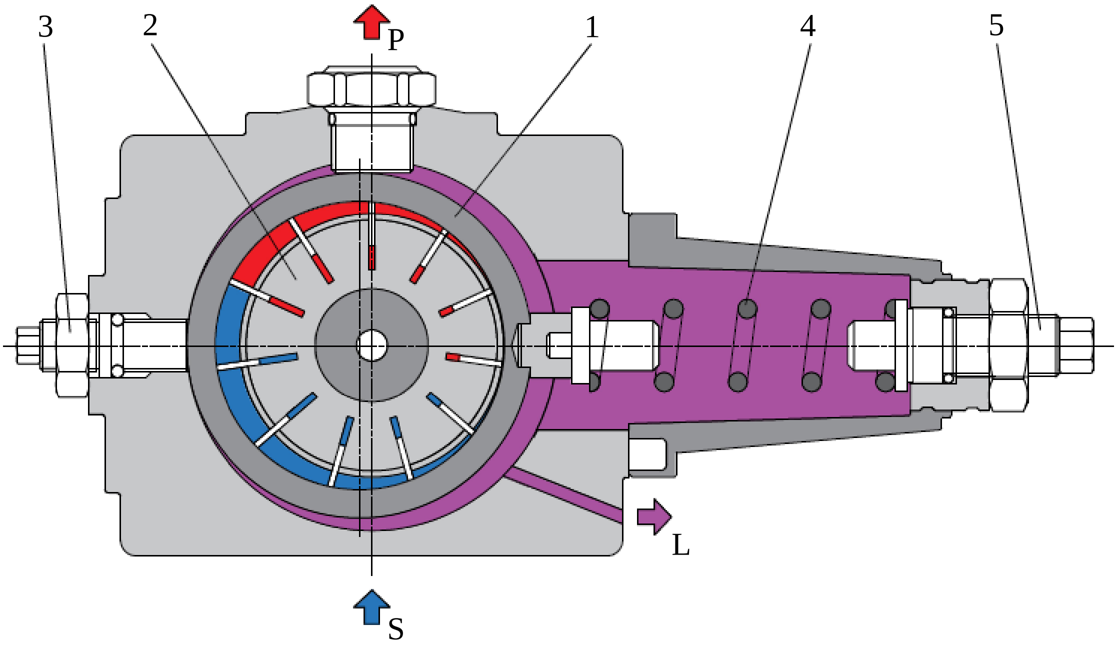


Fig. 5.20 Single chamber variable displacement vane pump with mechanical control

1 – stator ring, 2 – rotor, 3 – set screw for adjustment of maximum eccentricity, 4 – spring, 5 – set screw

Hydraulic pumps with hydraulic control are used for higher pressures and powers. The hydraulic pump is equipped with the pressure controller (3), and the change in eccentricity (position of the stator (1)) is achieved using the liquid pressure. The initial position of the hydraulic pump is depicted in Fig. 5.21. The maximum pressure in the system is adjusted by means of a set screw (6), which is used to obtain the required spring (7) preload. After starting the hydraulic pump, the pressure in the system is less compared to the required pressure. The liquid of the current pressure is acting on the surface of the smaller piston (4) and, at the same time, is fed to the larger piston (5) by drilling in the spool (8). A larger piston area will ensure a higher force effect and move the stator to the left. This results in an increase in the geometric stroke volume of the pump, the output flow, and the system pressure [11].

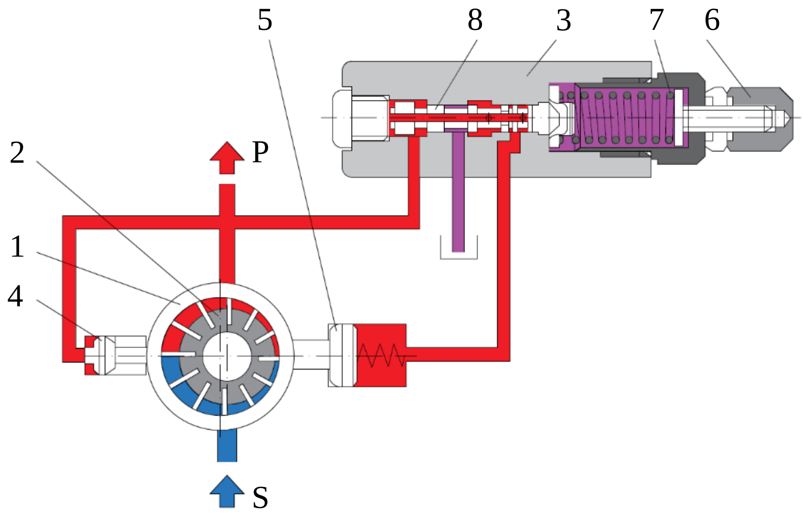


Fig. 5.21 Vane hydraulic pump with mechanical-hydraulic control ($p < p_{max}$)

1 – stator, 2 – rotor, 3 – pressure controller, 4 – left (smaller) piston, 5 – right (larger) piston, 6 – set screw, 7 – spring, 8 - spool

After reaching the adjusted pressure in the system Fig. 5.22, the liquid pressure acting on the front area of the controller spool overcomes the resistance adjusted on the spring (upper spring) (7). The spool is moved to the right and connects the channel of the large piston to the tank. This will relieve the force acting on the stator from the right. The left piston 4, which still remains under pressure, moves the stator to the right so that minimum eccentricity can be achieved. Only a minimum volume of flow is supplied to the system. As with the mechanically controlled hydraulic pump (see Fig. 5.20), this is the constant pressure control. It is an economical control method with lower power losses and low liquid heating [11].

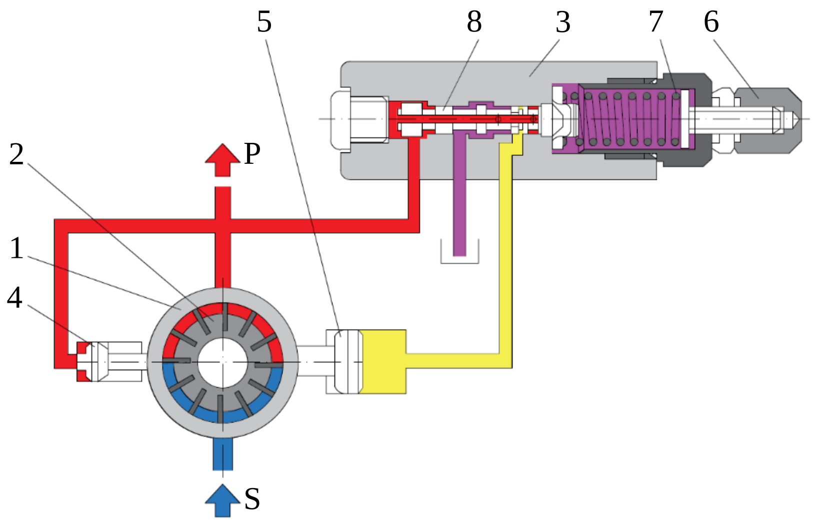


Fig. 5.22 Vane hydraulic pump with mechanical-hydraulic control (p_{max})

1 – stator, 2 – rotor, 3 – pressure controller, 4 – left (smaller) piston, 5 – right (larger) piston, 6 – set screw, 7 – spring, 8 - spool

The basic parameters and properties of single chamber vane pumps are shown in Tab 5.8.

Tab 5.8 Basic parameters and properties of single chamber vane pumps

Speed n [min^{-1}]	Geometric stroke volume V_g [cm^3]	Flow (at n = 1 500 min^{-1}) Q [dm$^3 \cdot$ min^{-1}]	Nominal pressure p_n [MPa]
max 2 000	8.5 ÷ 125	13 ÷ 200	7 ÷ 17,5
Operating viscosity ν [mm$^2 \cdot$ s^{-1}]	Starting viscosity ν [mm$^2 \cdot$ s^{-1}]	Minimum viscosity ν [mm$^2 \cdot$ s^{-1}]	Total efficiency η_T [-]
20 ÷ 100	500 ÷ 800	16	0.8
Advantages		Disadvantages	
Small storage space, low flow pulsation and noise, the possibility of changing geometric stroke volume.		Sensitive to pressure peaks and viscosity changes, for relatively low pressures, poor efficiency at higher pressures.	
Applications			
As fixed/variable displacement hydraulic pumps in mechanical engineering, they can replace axial piston hydraulic pumps in some less demanding applications.			

In addition to the above-described design of vane hydraulic pumps, double-chamber vane hydraulic pumps are also used. They are only in the fixed displacement design without the possibility of changing the geometric stroke volume. In this case, the rotor is pressure balanced, and the forces acting on bearings are significantly lower. The hydraulic pump shown in Fig. 5.23 has the stator (1) in an elliptical shape, and the rotor (2) is not eccentric in relation to the stator. The stator shape makes it possible to divide the internal space into two parts, resulting in two strokes per revolution of the rotor [11], [23].

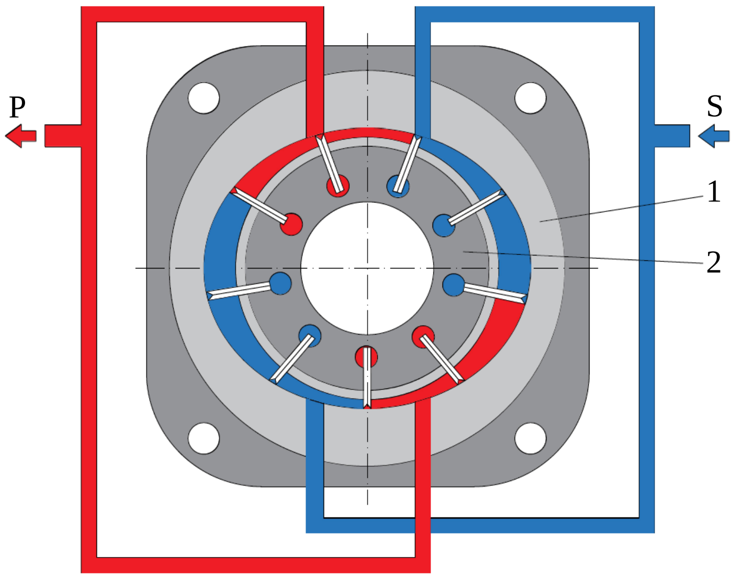


Fig. 5.23 Double chamber vane hydraulic pumps

1 – stator, 2 – rotor

Hydraulic pumps of this design are characterized by better efficiency and the possibility of their use for higher pressures compared to the single chamber vane pumps.

Vane motors

Vane hydraulic motors work on the same principle as hydraulic pumps. They are only in the fixed displacement design, usually with the ellipse-shaped stator. Compared to hydraulic pumps, the vanes of this hydraulic motor are always pushed by means of springs against the stator ring. They are only seldom used in hydraulic systems.

5.1.8 Axial piston hydraulic pumps and motors

In these converters, the liquid flow is realized by means of pistons which perform a linear reciprocating motion in cylinders. The pistons are axially placed relative to the axis of rotation of the converter, hence the name of this construction group. The number of pistons is always odd due to the reduction of flow pulsations. They are manufactured in both the fixed geometric stroke volume and variable geometric stroke volume versions. Depending on the internal arrangement of the pistons, axial piston converters are further divided into converters in a swash plate or bent axis designs.

Axial piston pumps in swash plate design

These are the most commonly used variable displacement hydraulic pumps in modern hydraulic systems. They are manufactured in a large number of partial construction designs for a wide range of many different applications. The drive shaft (1) of the hydraulic pump (see Fig. 5.24) is fixed to the cylinder block (2), to which the rotary motion is transmitted. The cylinders contain pistons (3) with ball joints which connect the piston to the slipper (4). The slippers are supported by the swashplate (5). This plate is inclined to the vertical axis by the angle α. No rotational motion is transmitted to the swashplate, but in some constructions, it is possible to change the plate inclination (angle α). The angle of inclination of the plate limits the size of the stroke of the pistons, which perform a linear reciprocating motion in the cylinder block. In the first half of the converter revolution, the piston performs the linear motion towards the plate (this achieves the suction effect). In the second half of the revolution, the piston is moved away from the plate (thus achieving the liquid discharge). The liquid suction and discharge mode is realized by means of channels in the cylinder block and the fixed distribution plate 6, in which the suction and discharge grooves are kidney-shaped [30].

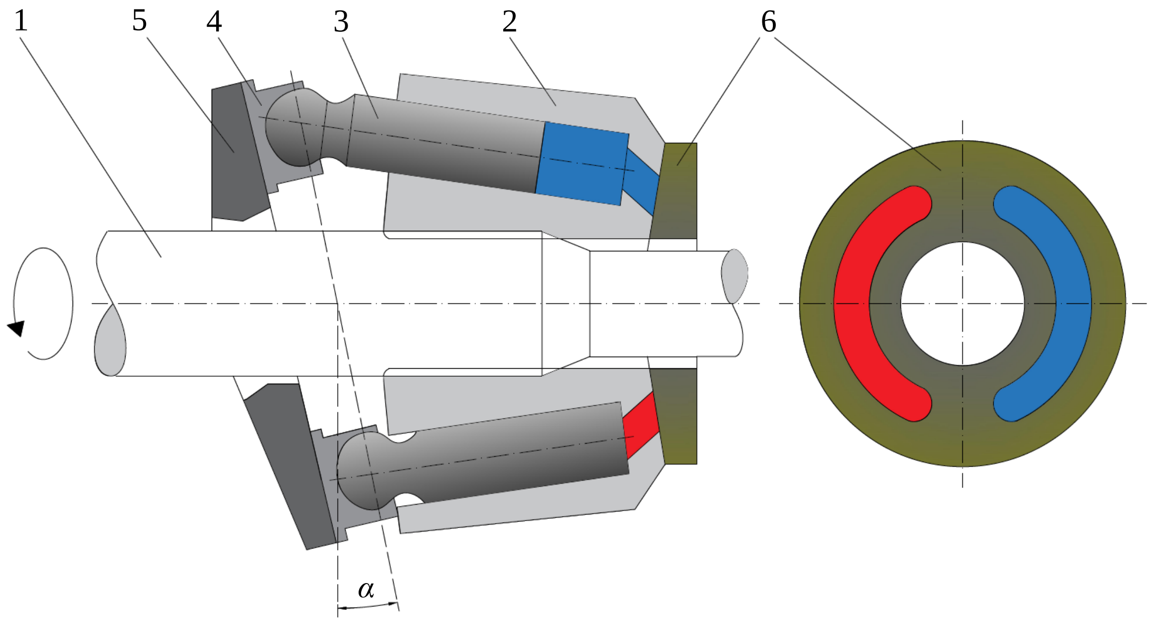


Fig. 5.24 Axial piston hydraulic pump in swashplate design

1 – drive shaft, 2 – cylinder block, 3 – pistons with ball pins, 4 – slipper, 5 - swashplate, 6 – control plate

It is possible to determine the piston stroke *h* as follows:

$$h = D \cdot tg\alpha \,, \tag{5.33}$$

where *h* [m] is the piston stroke, *D* [m] is the diameter of the pitch circle of pistons, and *α* [°] is the angle of inclination of the plate in relation to the vertical plane.

The theoretical geometric stroke volume V_g of the hydraulic pump is given by the formula [12]:

$$V_g = \frac{\pi \cdot d^2}{4} \cdot h \cdot z \,, \tag{5.34}$$

where V_g [m^3] is the geometric stroke volume, *d* [m] is the piston diameter, and *z* [-] is the number of pistons.

The slippers of these converters are hydrostatically balanced by the liquid pressure, which is supplied by drilling in the pistons and slippers into the grooves on the underside of the slippers. This also ensures lubrication and reduces friction between the swashplate and the moving slipper. The thickness of the lubricating film is usually in the range of (1 ÷ 25) µm, and it is necessary to ensure a suitable filtration of the liquid to prevent damage to the converter due to impurities.

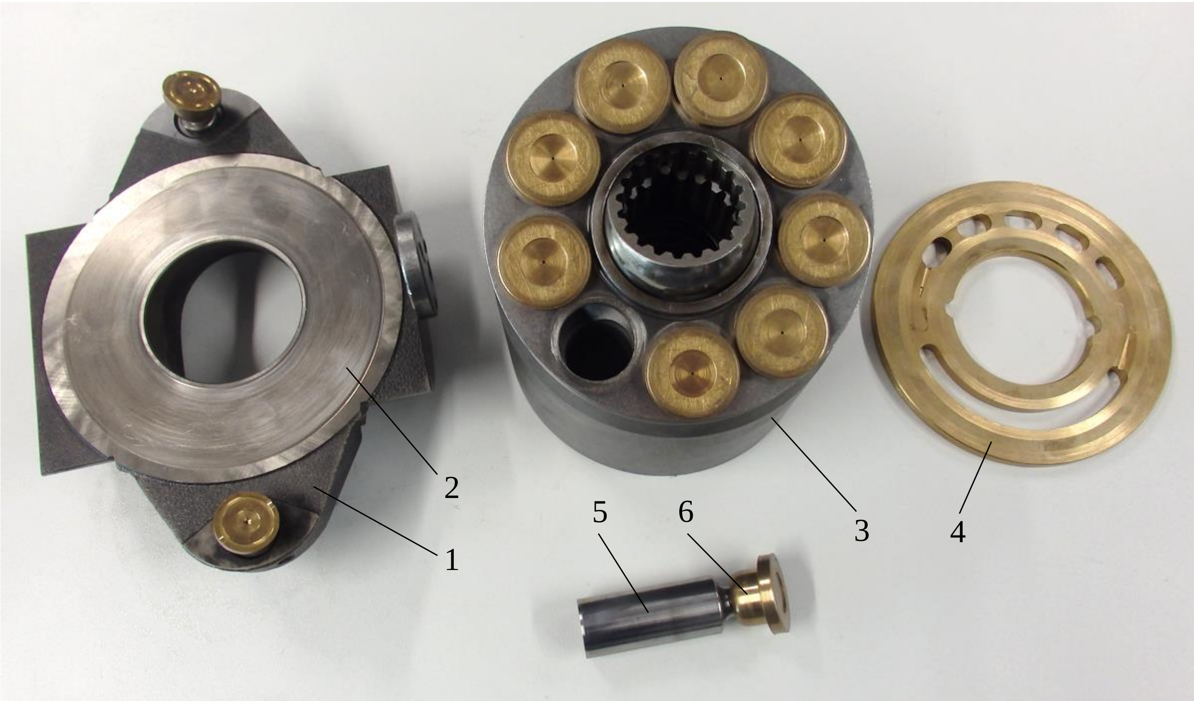


Fig. 5.25 Main parts of axial piston pump in swash plate design

1 – swashplate, 2 – support surface, 3 – cylinder block, 4 – control plate, 5 – piston, 6 - slipper

If the angle of the plate inclination is constant, then these are fixed displacement hydraulic pumps with liquid flow in one direction. Their use is possible in open circuits in some stationary hydraulic applications. If the plate angle of inclination can be changed, then these are variable displacement hydraulic pumps (with the possibility to change the geometric stroke volume). When the plate is inclined in one direction only (the angle of inclination $\alpha = 0° \div \alpha_{max}$) it is the hydraulic pump with variable geometric stroke volume and one direction of liquid flow. In the basic position, the maximum angle of the plate inclination is adjusted (by the piston with spring), and the change of the inclination angle is realized,, e.g., by a servo cylinder acting against the spring force. These hydraulic pumps are used in open circuits, especially in modern stationary hydraulic applications. For a design with the possibility of the plate inclination in both directions (the inclination angle $-\alpha_{max} \div 0 \div +\alpha_{max}$), it is the variable displacement hydraulic pump with two directions of flow. The basic position of the plate is at the angle $\alpha = 0°$; in this case, the flow of the hydraulic pump is zero. The plate flipping over the zero angle allows changing the flow direction without changing the rotation of the input shaft, while the plate inclination is usually possible by the angle α up to ± 15°. In this case, these are the most commonly used hydraulic pumps for closed circuits of mobile hydraulic systems. The inclination of the support plate is most often adjusted by means of a servo valve, which is controlled by control valves that are usually integrated into the body of the hydraulic pump. In most cases, such hydraulic pumps are supplemented with mechanical-hydraulic or electro-hydraulic controllers, which enable continuous control of output parameters.

The construction of the axial piston hydraulic pump in swashplate design allows to production of these converters in a version with a continuous shaft. This is very advantageous in many applications because an additional hydraulic pump can be connected to this shaft. It can provide the function of a filling hydraulic pump (in a closed hydraulic circuit), or it can be used to realize an external cooling and filtration circuit or other auxiliary functions of the system. The basic parameters and properties of axial piston pumps in the swashplate design are shown in Tab 5.9.

Tab 5.9 Basic parameters and properties of axial piston pumps in swashplate design

Speed n [min^{-1}]	Geometric stroke volume V_g [cm^3]	Flow (at n = 1 500 min^{-1}) Q [dm^3 · min^{-1}]	Nominal pressure p_n [MPa]
max 3 700	10 ÷ 1 000	15 ÷ 1 500	32 ÷ 40
Operating viscosity ν [mm^2 · s^{-1}]	Starting viscosity ν [mm^2 · s^{-1}]	Minimum viscosity ν [mm^2 · s^{-1}]	Total efficiency η_T [-]
10 ÷ 100	1 000 ÷ 1 600	5 ÷ 10	0.90 ÷0.92
Advantages		Disadvantages	
High pressures, the possibility of changing geometric stroke volume (suitable for control and regulation), very good efficiency, the possibility of design with the continuous shaft.		Higher price, more demand on fluid cleanliness, smaller angle of inclination (compared to bent axis design).	
Applications			
Universal use, as a fixed displacement hydraulic pump and especially as a variable displacement hydraulic pump in modern applications of stationary and mobile hydraulics.			

Axial piston pumps in bent axis design

Hydraulic pumps in bent axis design (see Fig. 5.26) have, similarly to the previous case, the pistons (2) led out in the cylinder block (3), to which the rotary motion is transmitted, and are axially placed with respect to the axis of rotation. In these converters, however, the whole pistons block is deflected by the angle α to the horizontal axis (axis of the drive shaft (1)). The connection between the drive shaft and the pistons block is performed by means of the carrier plate (4). The pistons are connected to the carrier plate by ball joints (5). In the case of a converter in bent axis design, conical, spherical, or cylindrical pistons with a connecting rod are used. The suction and discharge are solved by means of the control plate (6) with kidney-shaped grooves, and in most cases, it is of spherical shape [11].

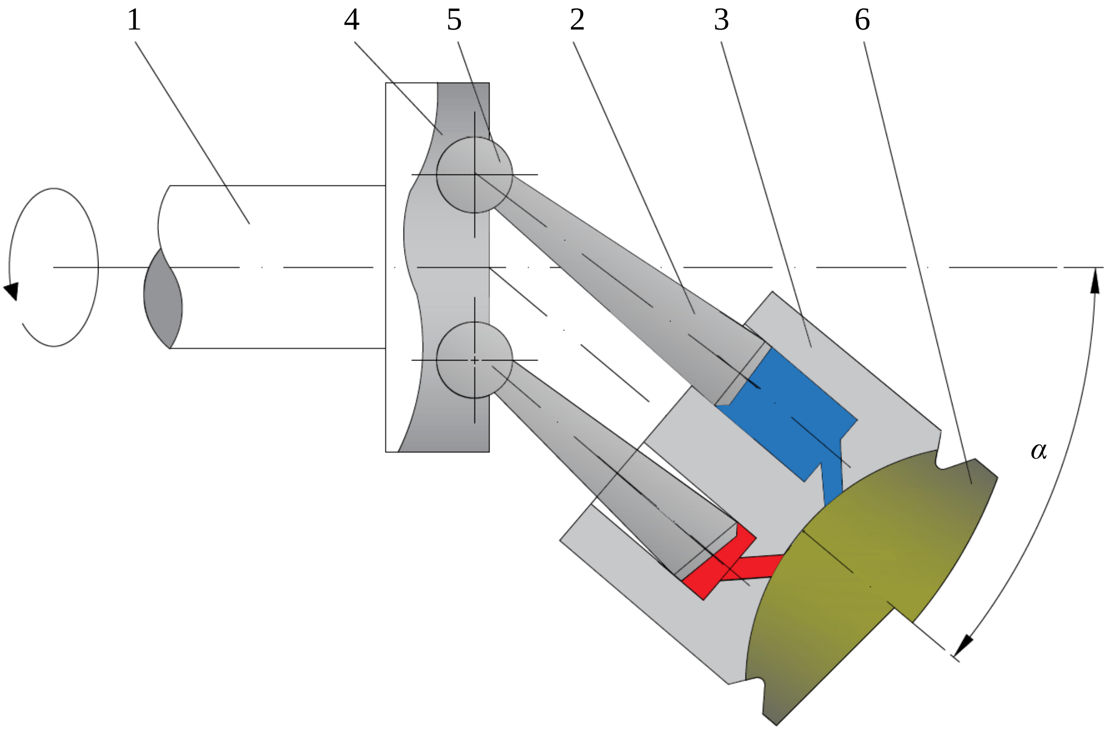


Fig. 5.26 Axial piston hydraulic pump in bent axis design

1 – drive shaft, 2 – piston, 3 – cylinder block, 4 – carrier plate, 5 – ball joints, 6 – control plate

It is possible to determine the piston stroke h as follows:

$$h = D \cdot sin\alpha \,, \tag{5.35}$$

where h [m] is the piston stroke, D [m] is the diameter of the pitch circle of pistons, and α [°] is the angle of inclination of the block in relation to the vertical axis.

The theoretical geometric stroke volume V_g of the hydraulic pump is given by the formula [12]:

$$V_g = \frac{\pi \cdot d^2}{4} \cdot h \cdot z \,, \tag{5.36}$$

where V_g [m^3] is the geometric stroke volume, d [m] is the piston diameter, and z [-] is the number of pistons.

Also, these hydraulic pumps can be divided into fixed displacement pumps (for one direction of flow) with the angle α up to 45° or variable displacement pumps (for one or two directions of flow) with the angle $\alpha = \pm$ 25°. The function principle is the same as in the case of pumps in swashplate design, but the change in geometric stroke volume is achieved by changing the inclination angle α of the block, which in the case of bidirectional flow can be inclined to both sides. The change of the block position is usually realized by a hydraulic servo cylinder controlled by a controller.

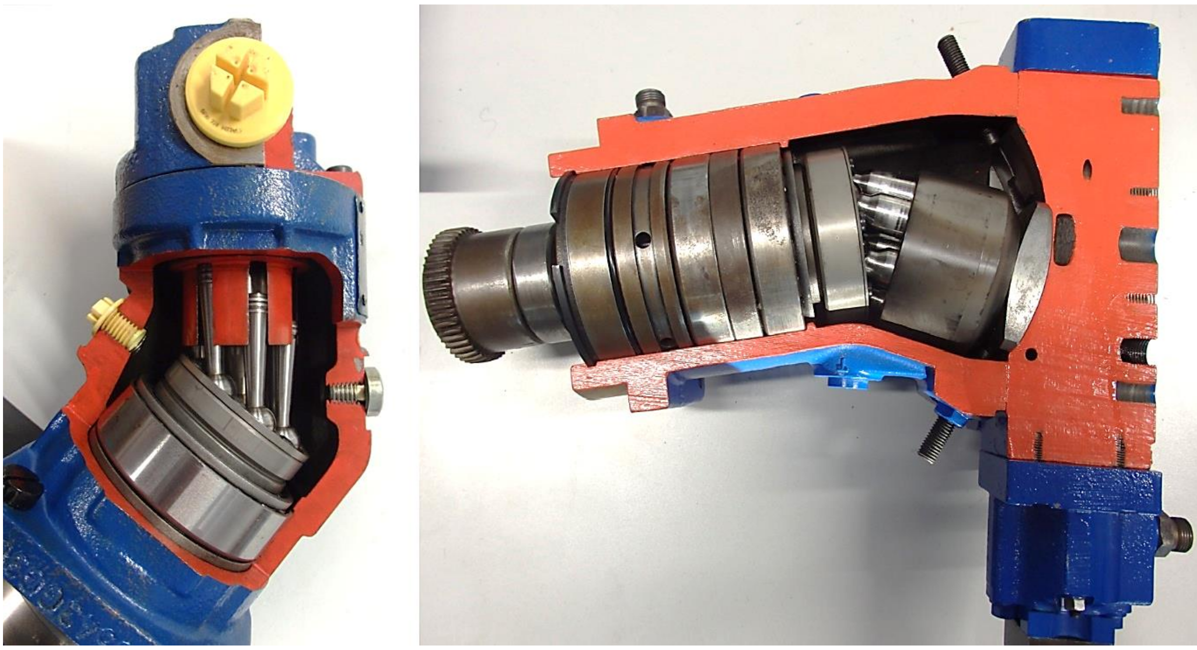


Fig. 5.27 An example of hydrostatic axial converter in bent axis design

The bent construction of the piston block does not allow a design with a continuous shaft. The bent axis design with an inclined block enables work at higher speeds and has lower demands on liquid purity compared to the swashplate design. They can usually be used as motor-generators with higher total efficiency and lifetime (only small transverse forces act on the pistons, most of the force effects are captured by the massive bearings and the central pin).

Tab 5.10 Basic parameters and properties of axial piston pumps in bent axis design

Speed n [min^{-1}]	Geometric stroke volume V_g [cm^3]	Flow (at n = 1 500 min^{-1}) Q [dm^3 · min^{-1}]	Nominal pressure p_n [MPa]
Up to 6 000	100 ÷ 3 500	70 ÷ 5 250	35 ÷ 40
Operating viscosity ν [mm^2 · s^{-1}]	Starting viscosity ν [mm^2 · s^{-1}]	Minimum viscosity ν [mm^2 · s^{-1}]	Total efficiency η_T [-]
10 ÷ 100	1 000 ÷ 1600	5 ÷ 10	0.91 ÷ 0.93
Advantages		Disadvantages	
High pressures, possibility to change geometric stroke volume (suitable for control and regulation), very good efficiency, lower transverse forces on pistons - higher durability, lower demands on liquid cleanliness.		Higher price, high adjustment forces during control, no possibility of continuous shaft design, more difficult cooling.	
Applications			
Universal use as fixed and variable displacement hydraulic pumps, especially in closed hydraulic systems of mobile machines (excavators, railway technique).			

Axial piston hydraulic motors

Axial piston hydraulic motors can also be available in swashplate and bent axis designs. The basic functional principles and design are practically identical to those of hydraulic pumps, which broadly applies to the mentioned advantages and disadvantages of their use. The function of the motor is, of course, opposite to the hydraulic pump. A pressure liquid is fed to the piston surfaces, which results in the piston movement in the cylinder block. The resulting pressure force from the piston F_p always acts in the piston axis. At $\alpha \neq 0$ (the plate or block inclination by an angle α) this force does not act perpendicularly to a given element. Therefore, it is necessary to distribute the force; the standard component of the F_N force is captured by the inclined plate (or by the carrier plate). Using the tangential component of the force F_T, the torque is derived and transmitted to the output shaft of the hydraulic motor (see Fig. 5.28 and Fig. 5.29) [3], [30].

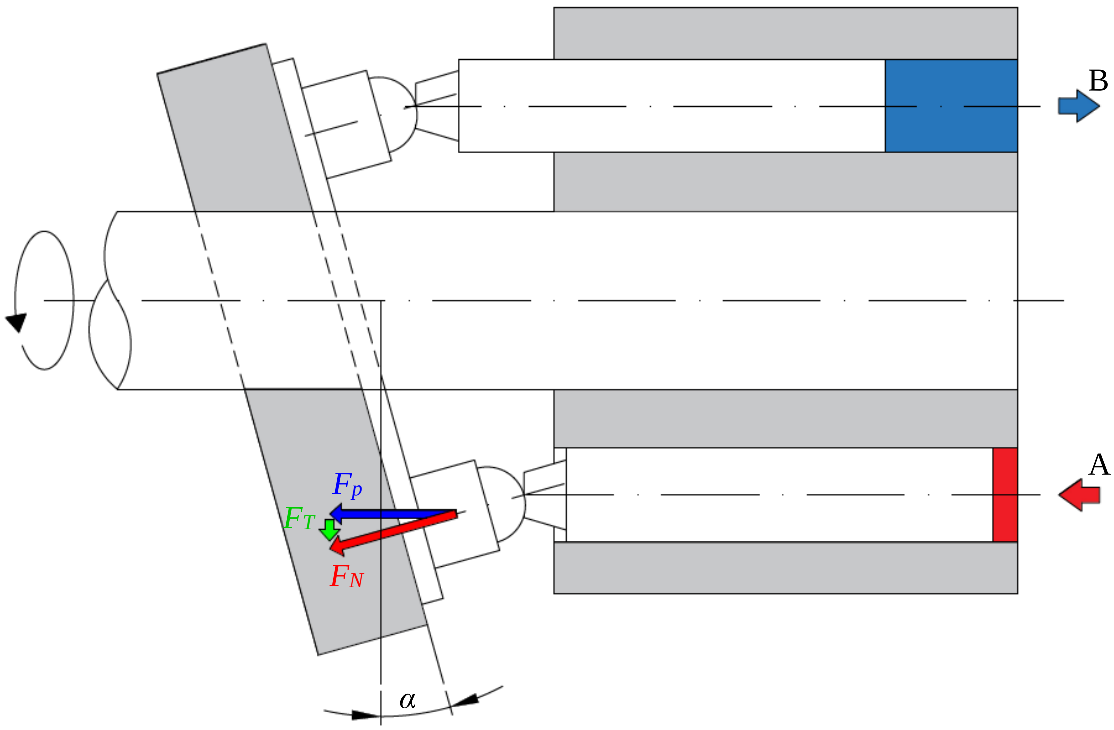


Fig. 5.28 Force conditions of axial piston hydraulic motor in swashplate design

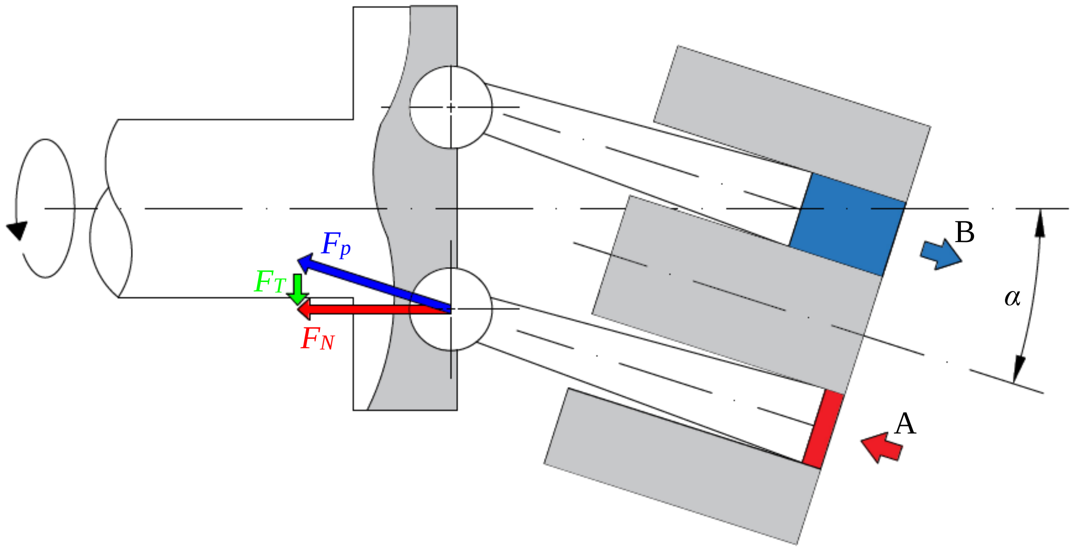


Fig. 5.29 Force conditions of axial piston hydraulic motor in bent axis design

Hydraulic motors can be in fixed or variable displacement versions, with the possibility of continuous regulation of output speed or torque. The swashplate design is mainly used in the fixed displacement version and is generally characterized by worse start-up parameters and lower efficiency. Hydraulic motors in bent axis design are more often used in the variable displacement version. Compared to hydrogen pumps, they have a lower regulation range and the minimum inclination of the block α is often limited to a value of approx. 5° (due to the limitation of the maximum speed). Hydraulic motors used in closed circuits are also equipped with flushing blocks or stop (deceleration) valves in order to protect the motor from excessive speed increase and possible cavitation.

The basic parameters of axial piston motors in swashplate design, including their applications, are shown in Tab 5.11.

Tab 5.11 The basic parameters of axial piston motors in swashplate design

Minimum speed n_{min} [min^{-1}]	Maximum speed n_{max} [min^{-1}]	Torque M [N · m]	Nominal pressure p_n [MPa]
25 ÷ 200	1 500 ÷ 3 000	140 ÷ 5 500	28 ÷ 35
Geometric stroke volume V_g [cm^3]	Operating viscosity v [mm^2 · s^{-1}]	Total efficiency η_T [-]	
32 ÷ 1 000	17 ÷ 130	0.92	
Applications			
Usually as fixed displacement hydraulic motors.			

The basic parameters of axial piston motors in bent axis design, including their applications, are shown in Tab 5.12.

Tab 5.12 The basic parameters of axial piston motors in bent axis design

Minimum speed n_{min} [min^{-1}]	Maximum speed n_{max} [min^{-1}]	Torque M [N · m]	Nominal pressure p_n [MPa]
100 ÷ 150	800 ÷ 5 000	25 ÷ 4 500	35 ÷ 50
Geometric stroke volume V_g [cm^3]	Operating viscosity ν [mm^2 · s^{-1}]	Total efficiency η_T [-]	
5 ÷ 2 000	17 ÷ 130	0.95	
Applications			
Usually as fixed and variable displacement hydraulic motors.			

5.1.9 Radial piston hydraulic pumps and motors

The last group of rotary converters mentioned here are the radial piston converters. As the name suggests, they use pistons, which are placed radially in relation to the axis of rotation, for energy transfer. Both radial piston hydraulic pumps and motors (the use of which is more common) are produced, and in both cases, they can be in fixed or variable displacement designs. The change in geometric volume is usually achieved by changing the eccentricity. Radial piston converters can be further divided according to the piston placement into converters with pistons in the rotor (inside impinged radial piston pumps) and converters with pistons in the stator (outside impinged radial piston pumps).

Radial piston pumps with eccentric cylinder block (inside impinged radial piston pumps)

This type of hydraulic pump was very widespread in the past. Today they are seldom used and have been replaced by axial piston hydraulic pumps in most applications. The rotary motion of the drive shaft is transmitted by means of a carrier to the rotor (1) (see Fig. 5.30), which rotates around the static central pin. The pistons (2) are radially guided in cylinders in the rotor body. The external guidance of the pistons is given by the inner surface of the stator (3), against which the pistons are supported by means of the slippers (4). There is a joint connection between the slippers and the pistons. The pistons are cylindrical and perform a linear motion in the rotor body. There are channels in the central pin, which are separated by a fixed barrier separating the suction and discharge spaces. The suction and discharge ports from the space under the pistons lead into these spaces. Also, these constructions always have an odd number of pistons; each piston performs one stroke (suction and discharge) per revolution of the drive shaft. In order to ensure the stroke of the pistons, the rotor and the stator must be placed eccentrically to each other. The movement of the pistons must be forced by the centrifugal force during rotation and by the eccentricity of the rotor with respect to the stator. The slippers, similarly to the axial converters in swashplate design, are provided with a groove for liquid supply and friction reduction [6], [12], [23].

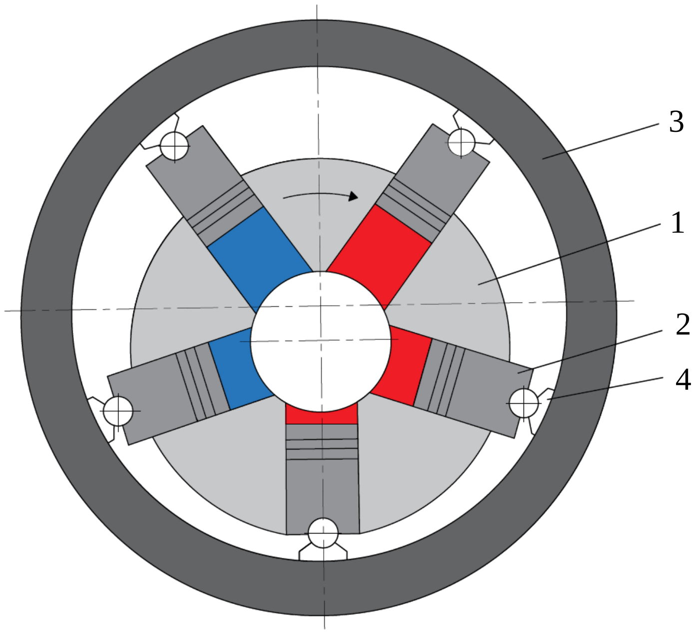


Fig. 5.30 Radial piston pump with cylinder block (inside impinged radial piston pumps)

1 – rotor, 2 – piston, 3 – stator, 4 - slipper

The theoretical geometric stroke volume V_g of these converters is given by the formula [12]:

$$V_g = \frac{\pi \cdot d^2}{4} \cdot 2 \cdot e \cdot z \, , \tag{5.37}$$

where d [m] is the piston diameter, e [m] is the eccentricity, and z [-] is the number of pistons.

Tab 5.13 Basic parameters and properties of radial piston pumps with eccentric cylinder block (inside impinged radial piston pumps)

Speed n [min^{-1}]	Geometric stroke volume V_g [cm^3]	Flow (at n = 1 500 min^{-1}) Q [dm^3 · min^{-1}]	Nominal pressure p_n [MPa]
Up to 2 000	16 ÷ 1 015	48 ÷ 1 218	21 ÷ 35
Operating viscosity ν [mm^2 · s^{-1}]	Starting viscosity ν [mm^2 · s^{-1}]	Minimum viscosity ν [mm^2 · s^{-1}]	Total efficiency η_T [-]
15 ÷ 110	200	10	0.9
Advantages		Disadvantages	
High pressures, high efficiency, possibility to change geometric stroke volume.		High price, less compact compared to axial piston hydraulic pumps.	
Applications			
As fixed and variable displacement hydraulic pumps of stationary drives, in modern systems, they are replaced by axial piston hydraulic pumps.			

The change in the geometric stroke volume can be achieved by changing the eccentricity. This is done by changing the position of the stator relative to the rotor axis either by a set screw or a hydraulic servo valve. The basic parameters and properties of these pumps are given in Tab 5.13.

Radial piston hydraulic pumps with eccentric shaft (outside impinged radial piston pumps)

These hydraulic pumps are used only as fixed displacement pumps and use valve liquid distribution, as shown in Fig. 5.31 and Fig. 5.32 [11]. The pistons (1) are guided in cylinders, and placed in the stator (2). The eccentric placement is ensured by means of a drive shaft (3) with the eccentric (5). The pistons perform a linear movement during the stroke, the piston extension is realized by the eccentric, and the return movement is ensured by the spring (4). The suction space is delimited by a radial groove in the eccentric (5). During the suction stroke (the discharge valve (9) is closed), the liquid is delivered into the space (6) through the drilling in the piston (7) and the suction valve (8). During the discharge, the suction valve is closed, and the liquid is forced out of the space (6) by the piston through the discharge valve (9) to the output of the hydraulic pump [6], [12], [23].

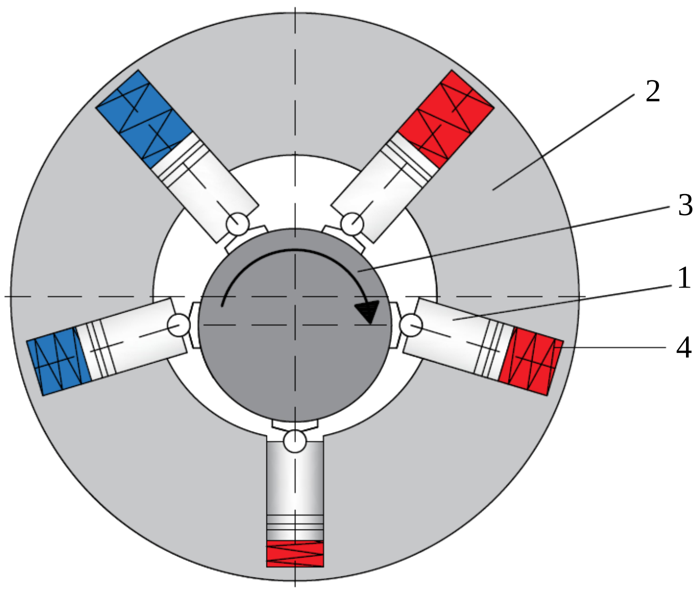


Fig. 5.31 Radial piston pump with eccentric shaft (outside impinged radial piston pumps)

1 – piston, 2 – stator, 3 – drive shaft, 4 – spring

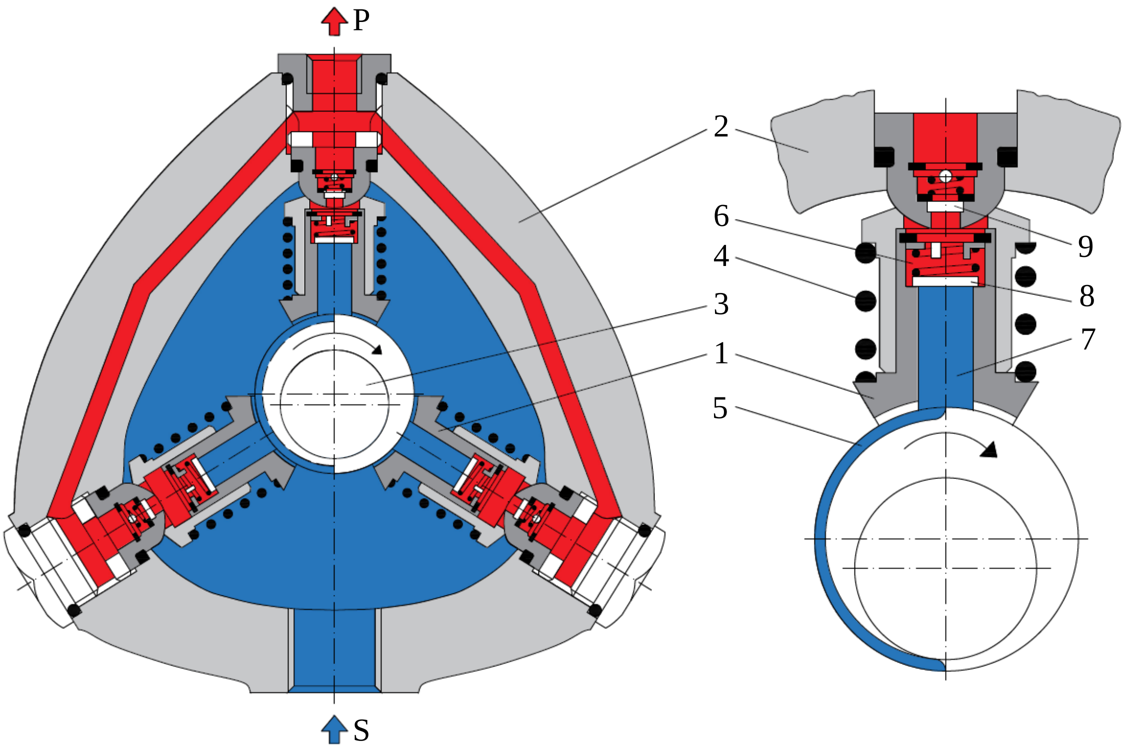


Fig. 5.32 Radial piston pump with eccentric shaft (outside impinged radial piston pumps)

1 – piston, 2 – stator, 3 – drive shaft, 4 – spring, 5 – radial groove in eccentric, 6 – space under piston, 7 – piston drilling, 8 – suction valve, 9 – discharge valve

The geometric stroke volume of the hydraulic pump is calculated similarly to the previous design. The basic parameters and properties of these pumps are given in Tab 5.14.

Tab 5.14 Basic parameters and properties of radial piston pumps with eccentric shaft (outside impinged radial piston pumps)

Speed n [min^{-1}]	Geometric stroke volume V_g [cm^3]	Flow (at n = 1 500 min^{-1}) Q [dm^3 · min^{-1}]	Nominal pressure p_n [MPa]
Up to 3 000	0.6 ÷ 140	2 ÷ 200	21 ÷ 70
Operating viscosity ν [mm^2 · s^{-1}]	Starting viscosity ν [mm^2 · s^{-1}]	Minimum viscosity ν [mm^2 · s^{-1}]	Total efficiency η_T [-]
12 ÷ 100	1 000	10	0.9
Advantages		Disadvantages	
Very high pressures, very good efficiency.		High price, less compact compared to axial piston hydraulic pumps.	
Applications			
As fixed displacement hydraulic pumps for the highest pressures in stationary equipment.			

Radial piston hydraulic motors with eccentric shaft (outside impinged radial piston motors)

The principle of their function is similar to that of hydraulic pumps. The pistons are usually hydrostatically balanced the liquid from the working space flows through the bore in the piston into the space between the slipper and the eccentric, resulting in reduced friction Fig. 5.33. The liquid distribution to the individual pistons is solved by a rotary slide valve [11].

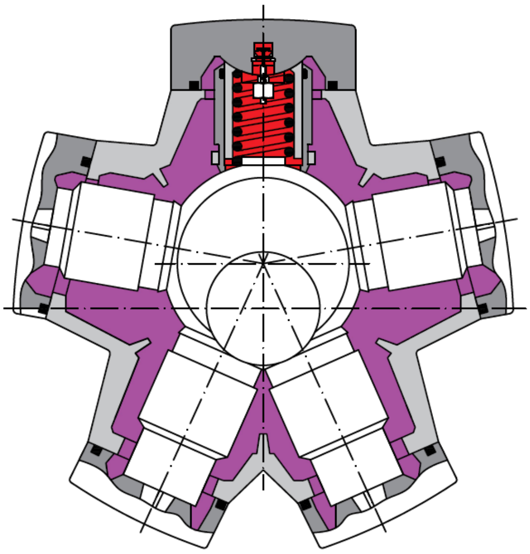


Fig. 5.33 Radial piston hydraulic motor with eccentric shaft (outside impinged radial piston motor)

In some designs, it is possible to change the eccentricity of the eccentric and, therefore, the geometric stroke volume of the motor. They are usually used as fixed displacement slow-speed and high-torque motors. The basic parameters and properties of radial piston motors with eccentric shaft are given in Tab 5.15.

Tab 5.15 Basic parameters and properties of radial piston motors with eccentric shaft (outside impinged radial piston pumps)

Minimum speed n_{min} [min^{-1}]	Maximum speed n_{max} [min^{-1}]	Torque M [N · m]	Nominal pressure p_n [MPa]
0.1 ÷ 10	150 ÷ 3 000	20 ÷ 32 000	14 ÷ 30
Geometric stroke volume V_g [cm^3]	Operating viscosity ν [mm^2 · s^{-1}]	Total efficiency η_T [-]	
10 ÷ 6 000	20 ÷ 150	0.97	
Advantages		Disadvantages	
High pressures, good efficiency, can be used as low-speed, high-torque and high-speed.		High price, less compact compared to axial piston hydraulic motors.	
Applications			
Fixed displacement hydraulic motors of stationary and mobile machines.			

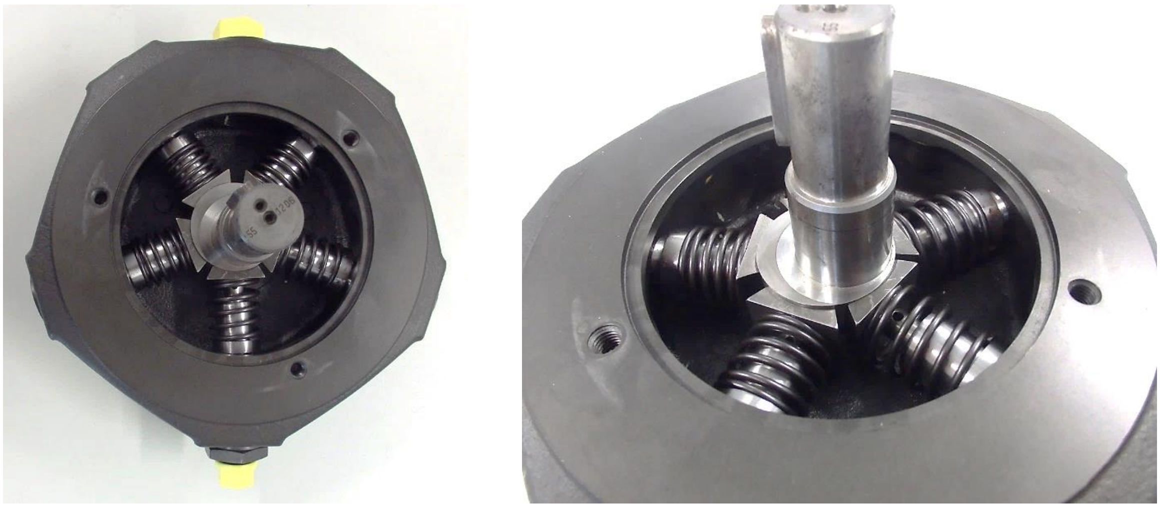


Fig. 5.34 An example of radial piston hydraulic motor with eccentric shaft (outside impinged radial piston motor)

Multi-stroke radial piston motors

Basically, it is a constructional alternative to a converter with pistons in the rotor. The curved path is performed on the inner surface of the stator (1) (see Fig. 5.35). The pistons (2) are equipped with rollers (3), which roll along the curved path. The rotor (4) with the cylinders is connected to the output shaft (5). The liquid is fed from the front by a distributor pin (6) under the pistons. The radially positioned pistons are pushed against the inner stator surface. This causes the decomposition of the pressure force of the acting piston, and its tangential component will provide the torque [11], [29].

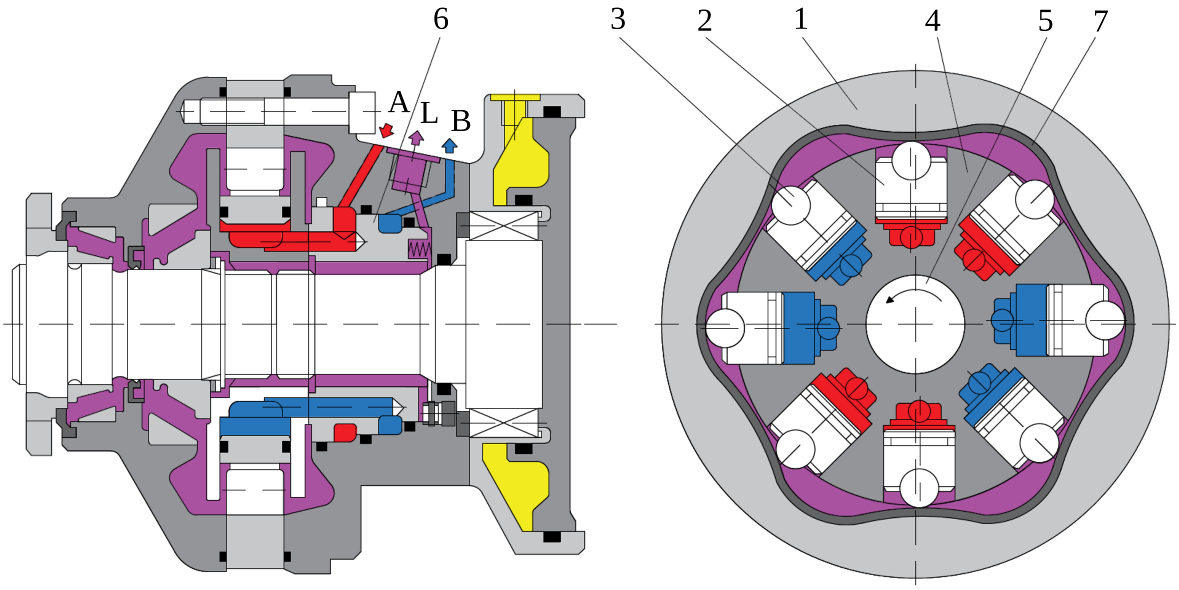


Fig. 5.35 Multi-stroke radial piston motor

1 – stator, 2 – piston, 3 – roller, 4 – rotor, 5 – driven shaft, 6 – distribution pin, 7 – curved path

These hydraulic motors are sometimes referred to as multi-stroke because each piston performs multiple strokes per revolution of the output shaft. They are used mainly as slow-speed motors and can achieve the highest torques. The basic parameters and properties of multi-stroke radial piston motors are given in Tab 5.16.

Tab 5.16 Basic parameters and properties of multi-stroke radial piston motors

Minimum speed n_{min} [min^{-1}]	Maximum speed n_{max} [min^{-1}]	Torque M [N · m]	Nominal pressure p_n [MPa]
0.1 ÷ 2	20 ÷ 300	1500 ÷ 1 000 000	17 ÷ 45
Geometric stroke volume V_g [cm^3]	Operating viscosity ν [mm^2 · s^{-1}]	Total efficiency η_T [-]	
200 ÷ 50 000	20 ÷ 150	0.98	
Advantages		Disadvantages	
High pressures, high efficiency, low-speed, high-torque.		High price, less compact compared to axial piston hydraulic pumps.	
Applications			
Low-speed high-torque fixed displacement hydraulic motors for direct drive of, for example, gates, wheel excavators, mixing drums.			

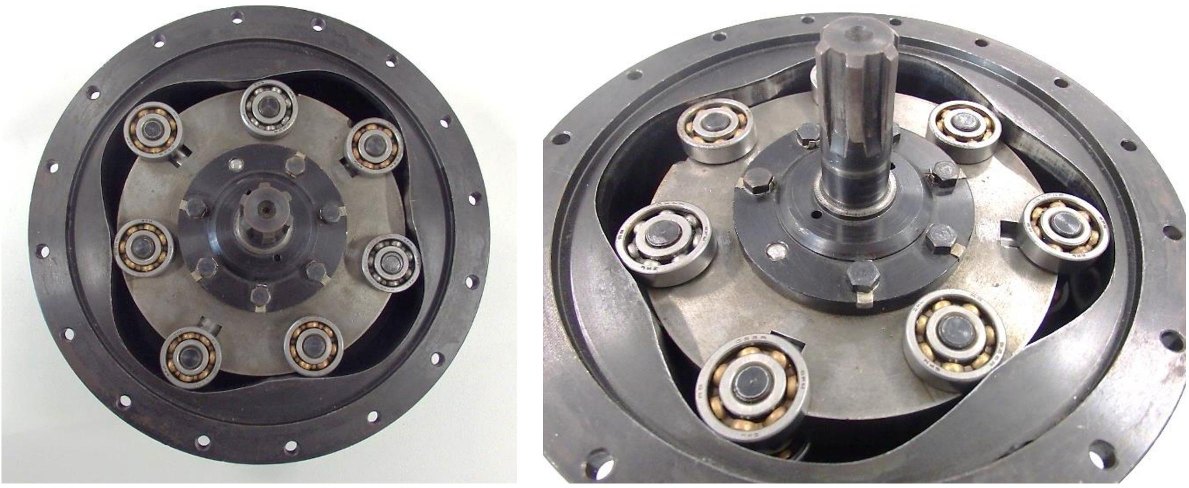


Fig. 5.36 An example of multi-stroke radial piston hydraulic motor

5.1.10 Pump controls

The control of hydraulic pumps means the control of their geometric stroke volume. This leads to the desired change in flow, pressure, or power of a given hydraulic pump.

Control is a simpler way of operating. In the case of the vane and radial piston converters it is the control of the stator position relative to the rotor; in the case of axial piston converters it is the control of the plate or block inclination. Depending on the control method, the control can be manual, mechanical, hydraulic, electrical, or electro-hydraulic. Further, depending on the degree of control, the control can be divided into direct and indirect (pilot).

In the case of direct control, the control mechanism has one stage. It is usually mechanical (e.g., by a lever or set screw), electromechanical (by electric servo motor) or hydraulic (by hydraulic motor).

The indirect control consists of two or more stages. The first control stage can be mechanical (e.g., spring preload, the position of lever or pedal), electrical (depending on the incoming electric current) or hydraulic (by liquid pressure). The second (last) stage is always hydraulic, using one or two linear hydraulic servomotors, which are usually located directly in the body of the hydraulic pump.

Regulation is a higher level of control that is characterized by feedback. It is defined by an additional control device that operates on mechanical or electro-hydraulic principles. It works automatically and can monitor courses of the required parameters (pressure, flow, power, etc.).

Constant pressure control

The purpose of this control is to maintain constant pressure in a hydraulic system independently of load and volumetric flow. In this case, a constant pressure value p_{max} is adjusted. When it is reached, there will be a sharp drop in the volumetric flow rate to a minimum or zero value, as is shown in Fig. 5.37. After unloading the hydraulic pump, the maximum volumetric flow rate increases immediately. The control principle has already been described for the vane hydraulic pumps (see Fig. 5.20, Fig. 5.21 and Fig. 5.22).

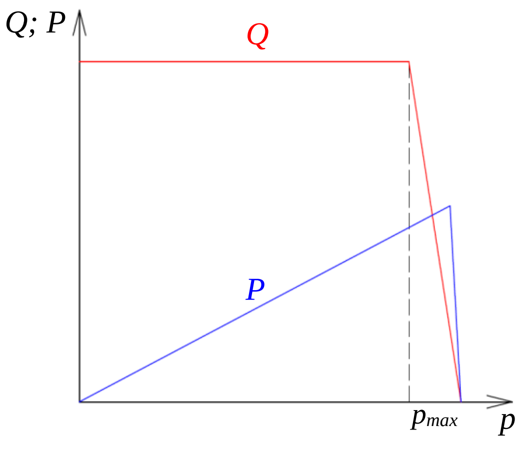


Fig. 5.37 Q - p characteristic constant pressure control

In theory, the liquid pressure can be fed directly from the discharge of the hydraulic pump to the control cylinder, where it acts against the spring force. In this case, however, the spring preload must be high, and it is difficult to obtain the required properties over the whole control range. The principle of mechanical control was described in chapter 5.1.7. A better solution is the single-edge control (see Fig. 5.38 (left)), where the maximum pressure is adjusted at the pressure valve (4). The orifice (5) creates resistance when the required pressure is reached. An example of double-edge control is shown in Fig. 5.38 (right) [12].

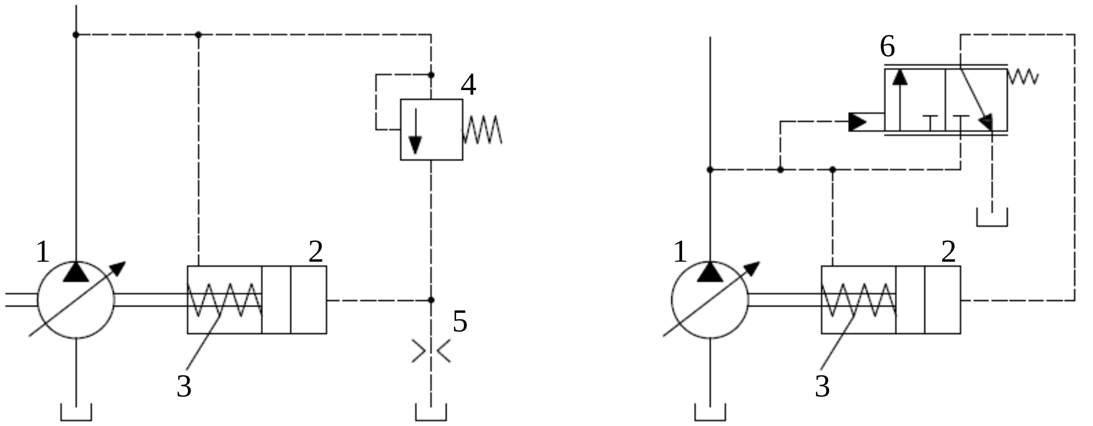


Fig. 5.38 Simplified schematic of a hydraulic pump with constant pressure control, single-edge control (left), double-edge control (right)

1 – piston, 2 – servomotor, 3 – spring, 4 – pressure valve, 5 – orifice, 6 – control valve

In modern hydraulic systems, constant pressure controls marked DR and DRG are currently used. An example of DR constant pressure control of a hydraulic pump is shown in Fig. 5.39.

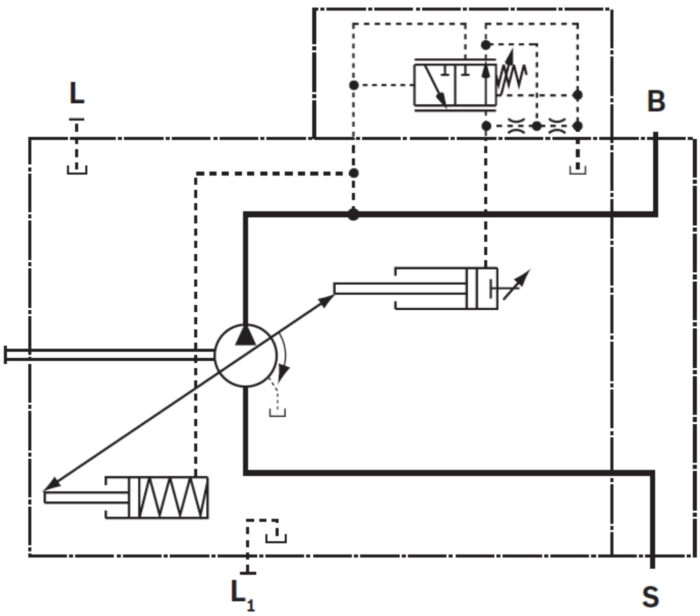


Fig. 5.39 DR constant pressure control of hydraulic pump (Bosch Rexroth) [37]

An example of DRG constant pressure control of a hydraulic pump with double-edge control and the possibility of remote control is shown in Fig. 5.40.

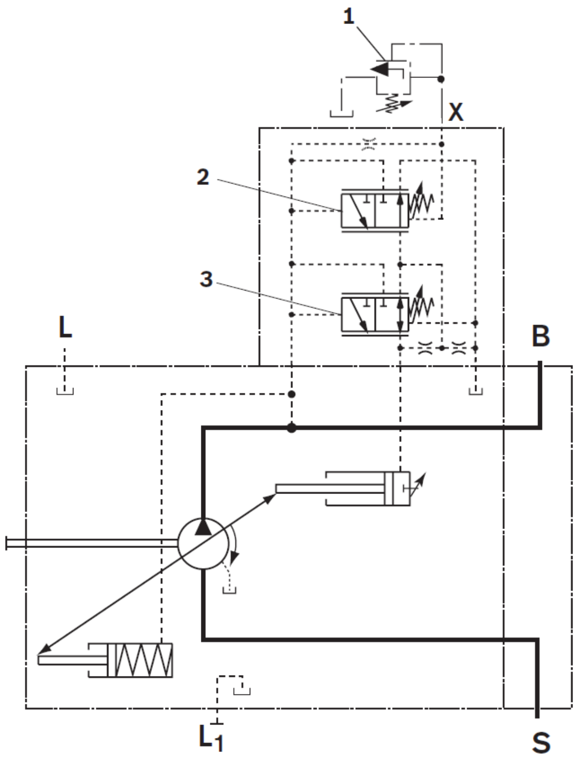


Fig. 5.40 DRG constant pressure control of hydraulic pump (Bosch Rexroth) [37]

The constant pressure control can be used for hydraulic pumps, where the geometric stroke volume decreases with the increasing pressure, or for hydraulic motors, where the geometric stroke volume of the motor increases with the increasing pressure. This type of control is mainly used for the speed control of a hydraulic motor loaded with constant torque.

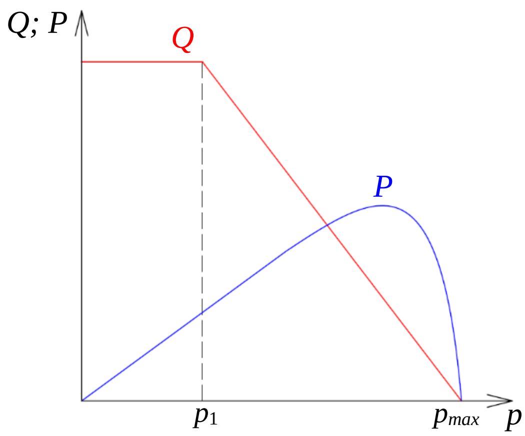


Fig. 5.41 Q - p characteristic pressure compensated control

Pressure compensated control

This is a modified version of the previous control. The pressure controller is adjusted to a gradually decrease in the flow rate from the adjusted pressure p_1 up to the maximum value p_{max}, as is shown in Fig. 5.41. This type of control is often used, for example, in hydraulic presses. When the pressing unit moves towards the pressed material, a higher speed is suitable, but high pressure is not required. During the pressing process, the velocities are usually low, but an increase in pressure (pressing force) is required.

Constant flow control

The purpose of this control is to ensure the constant volumetric flow of the hydraulic pump independently of its speed and load (see Fig. 5.42). For example, when using internal combustion engines, the speed fluctuation can be caused by a change in the fuel supply, and this results in changes in the volumetric flow rate of the hydraulic pump resulting in changes in the speed of the hydraulic motor.

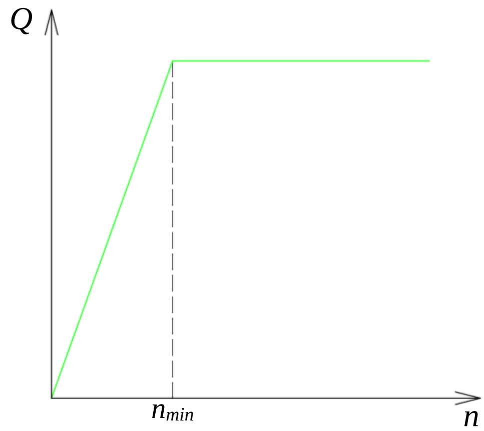


Fig. 5.42 Q - n characteristic constant flow control

The principle will be explained using a vane hydraulic pump with mechanical-hydraulic control (see Fig. 5.43). The pressure controller from Fig. 5.21 is supplemented at the output of the hydraulic pump by the orifice (9). The pressure gradient on the orifice acts against the spring (7) force on the controller slide (8). If, for example, the input speed of the hydraulic pump drops, the pressure gradient on the orifice is changed, and this causes a change in the force balance on the controller spool. The slide is moved to the left, thus connecting the liquid to the surface of the larger piston (5) via the bore in the slide. The stator (1) is also shifted to the left, resulting in an increase in the geometric stroke volume of the hydraulic pump. This ensures a constant flow rate as the decrease in input speed is compensated by an increase in the geometric stroke volume according to the equation $Q = V_g \cdot n$. It is evident that at zero eccentricity (as the stator position is drawn in the figure), the hydraulic pump delivers no flow to the system. Eccentric positioning of the stator relative to the rotor is necessary for the liquid flow. Then the description of the function given here can be accepted. The decrease in speed will actually increase the pressure on the larger piston (5) and thus increase the geometric volume of the hydraulic pump [30].

The size of the required constant flow rate can be controlled either by changing the orifice cross-section or by changing the spring preload of the controller spool.

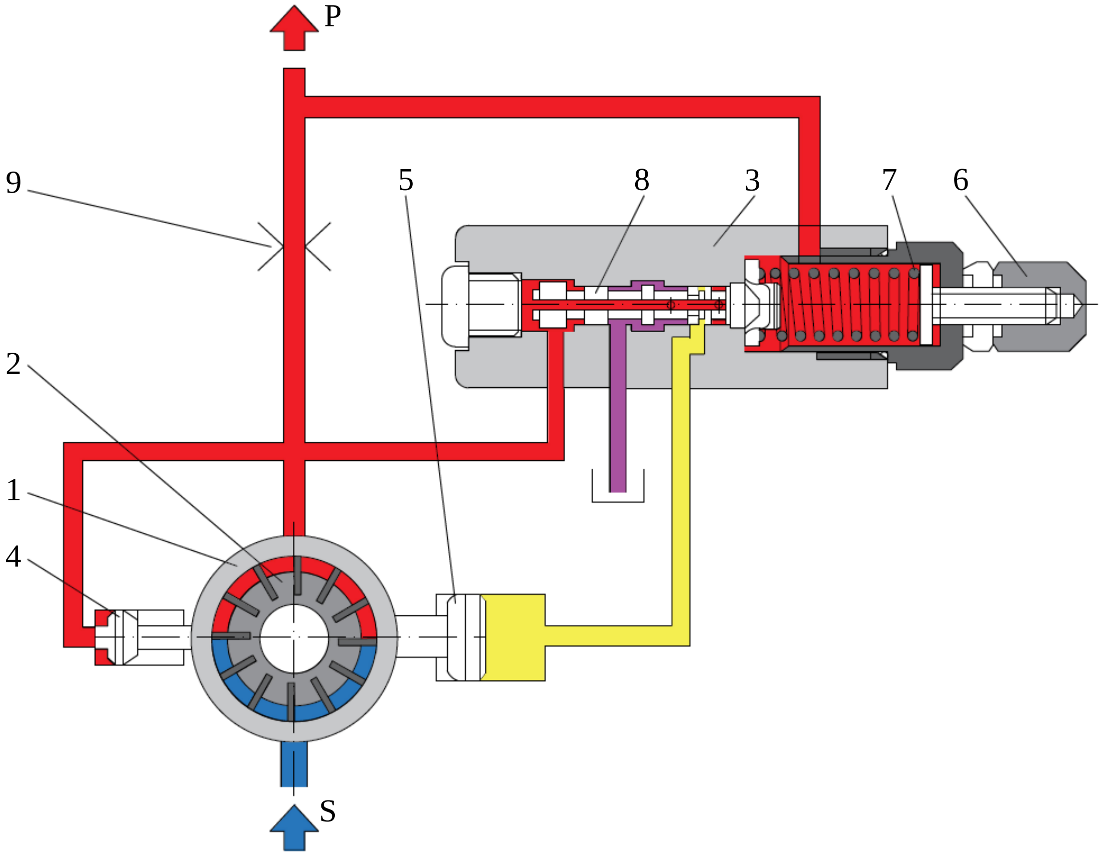


Fig. 5.43 Vane hydraulic pump with mechanical-hydraulic constant flow control

1 – stator, 2 – rotor, 3 – pressure controller, 4 – left (smaller) piston, 5 – right (larger) piston, 6 – set screw, 7 – spring, 8 – spool, 9 – orifice

A simplified circuit of a hydraulic pump with single-edge control is shown in Fig. 5.44 (left), with double-edge control in Fig. 5.44 (right) [12].

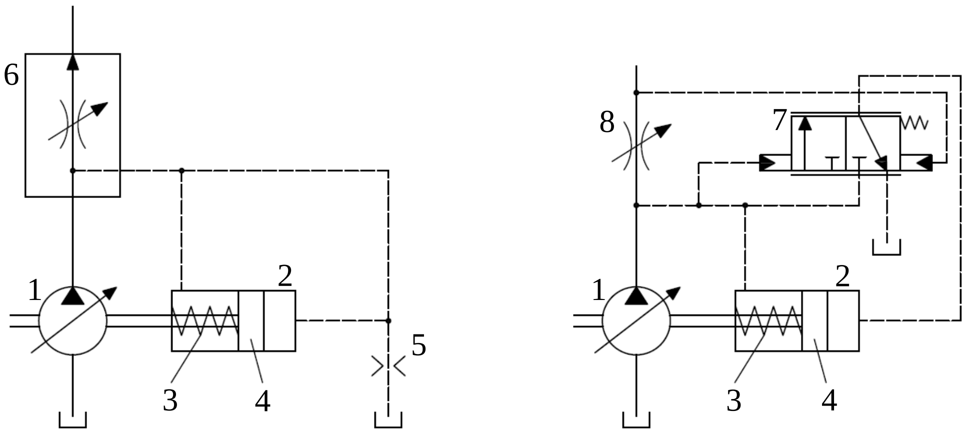


Fig. 5.44 Hydraulic pump with constant flow control, single-edge control (left), double-edge control (right)

1 – hydraulic pump, 2 – servomotor, 3 – spring, 4 – piston, 5 – orifice, 6 – 3-way flow control valve, 7 – control valve, 8 – throttle valve (or orifice)

An example of a commonly used constant flow control marked FR (N) is shown in Fig. 5.45.

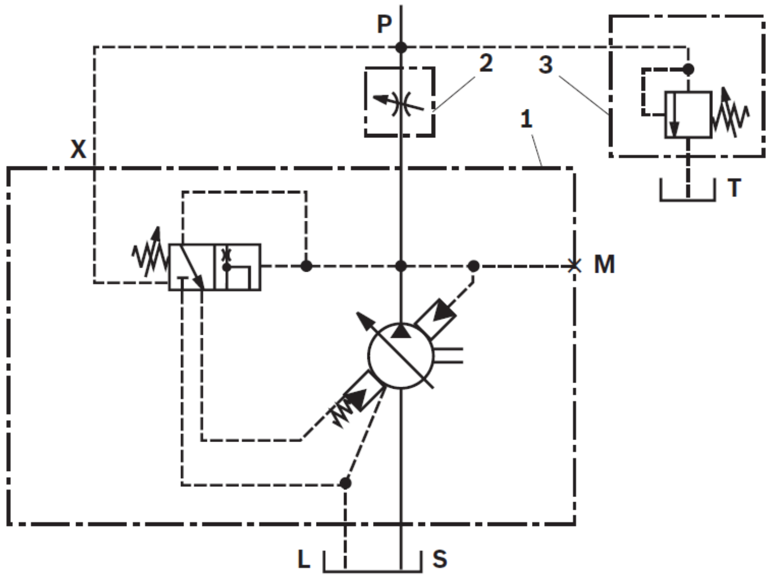


Fig. 5.45 FR (N) constant flow control of hydraulic pump (Bosch Rexroth) [39]

Control of constant pressure and flow

In this case, the hydraulic pump is equipped with two controllers, as shown in Fig. 5.46. The flow controller (1) ensures a constant flow which is proportional to the pressure gradient on the control throttle valve (3). The maximum pressure is adjusted on the pressure controller (2). When this pressure is exceeded the system switches to constant pressure control, see the characteristic in Fig. 5.47 [12].

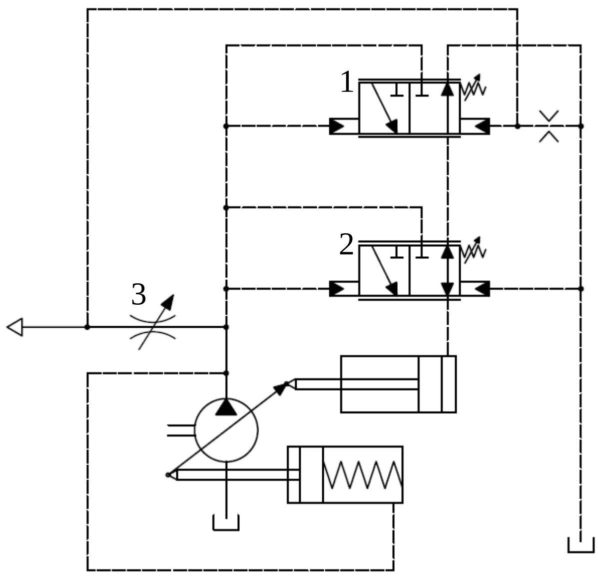


Fig. 5.46 Hydraulic pump with control of constant pressure and flow

1 – flow controller, 2 – pressure controller, 3 – control valve for flow adjustment

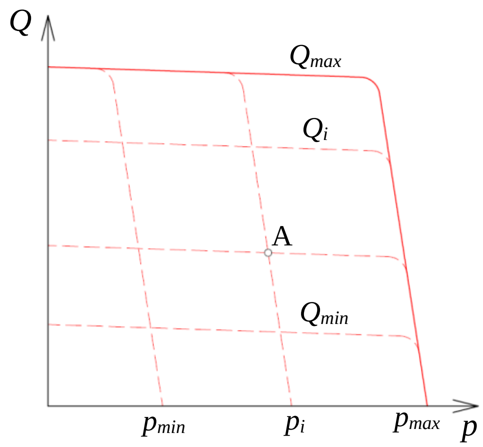


Fig. 5.47 FR (N) *Q* - *p* characteristic constant pressure and flow

The use of this control in practice means that the flow rate does not depend on the load; the throttle valve with the flow controller works as the 2-way flow control valve. The use of such control is mainly in mobile devices. It is energy saving, and sometimes it is also called Load Sensing control [40].

An example of a hydraulic pump with control of constant pressure and flow, designated DFR/DFR1, is shown in Fig. 5.49.

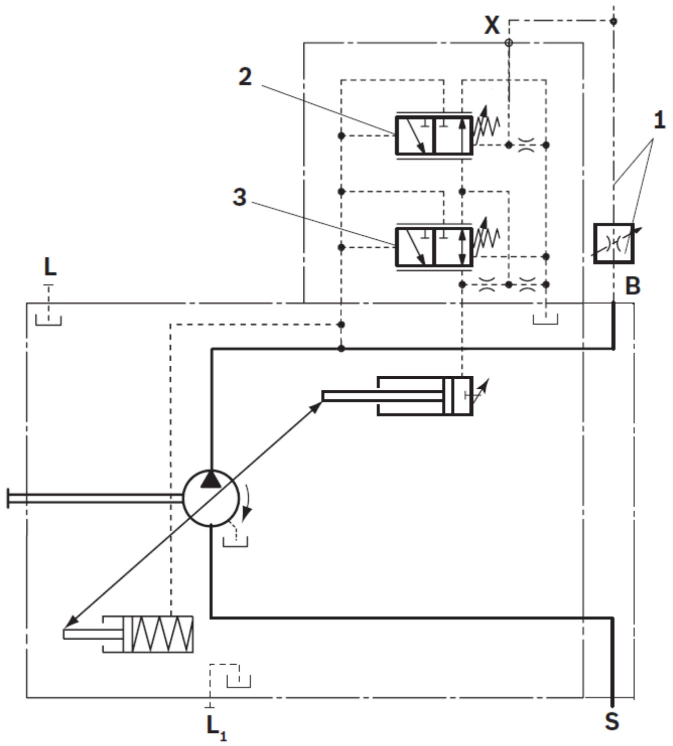


Fig. 5.48 DFR/DFR1 control of constant pressure and flow of hydraulic pump (Bosch Rexroth) [37]

Constant power control

Constant power control (see Fig. 5.49 (left)) is used in applications where maximum pressure and flow are not required simultaneously. This allows using an electric motor (internal combustion engine) with less power in the system, which reduces acquisition and operating costs. In the simplest version, the controller is equipped with two (or three) springs of different lengths. The ideal course of the characteristic has the shape of a hyperbola Fig. 5.49 (right). In fact, this characteristic consists of two straight lines whose slope corresponds to the stiffness of the springs. First, the liquid pressure in the cylinder acts against the longer spring; after its compression to the level of the shorter spring, it acts against both springs simultaneously [12].

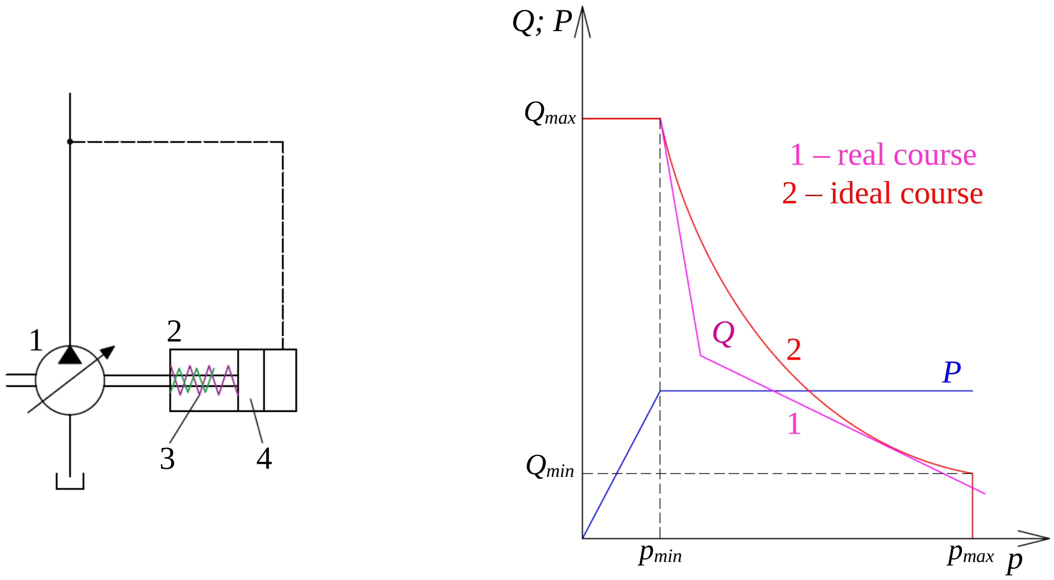


Fig. 5.49 Hydraulic pump with control of constant pressure and flow (left), characteristic of hydraulic pump (right)

1 – hydraulic pump, 2 – servomotor, 3 – springs, 4 - piston

In this case, the device can only move along the power curve, but it is not possible to achieve parameters in the area below the curve. However, if the application requires the system's working point to be in any area, it is necessary to use the so-called superior power limiter.

An example of constant power control of a hydraulic pump, designated DFLR, is shown in Fig. 5.50.

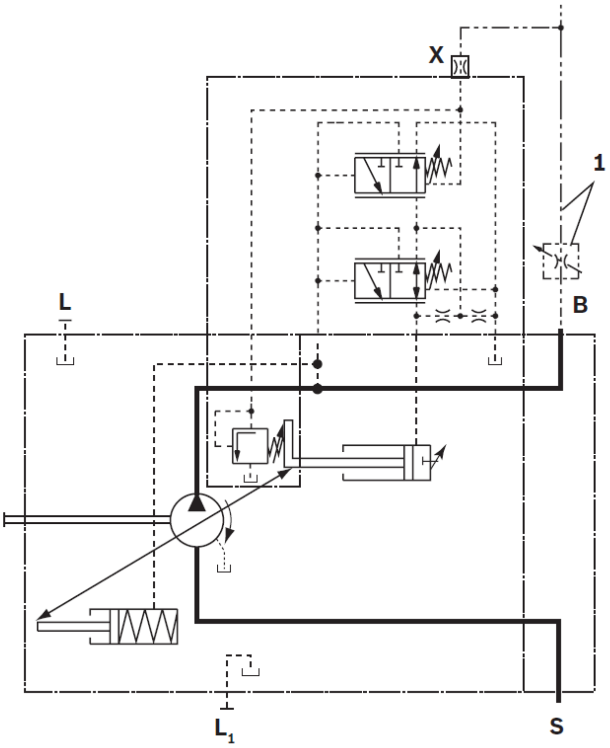


Fig. 5.50 DFRL constant power control of hydraulic pump (Bosch Rexroth) [37]

This type of control is very often used in hydrostatic drives of mobile machines.

5.2 Rotary actuators

Rotary actuators are only hydraulic motors in practice. The input liquid pressure energy is converted to the output mechanical energy of the rotating shaft. They are produced in several construction designs, which differ in their function and have different output swivel angles. They can be used as swivel and reversing devices,, e.g., for control of flaps, jaws, swivel arms, and clamping fixtures. Other possible applications are,, e.g., in manipulating and lifting equipment or for controlling ship rudders.

Vane rotary actuators

They are also sometimes referred to as vane hydraulic motors. They can be produced with single (see Fig. 5.51 (right)) or double vanes. The liquid is fed from the pressure branch into the inner space of the actuator, and the input and output of the actuator are separated in the housing (1) by the vane (2). The vane is connected to the output shaft (3). Due to the liquid acting on the vane, a pressure force is generated, which causes the rotary motion of the vane in the housing and torque is obtained on the output shaft. If constant pressure is ensured at the actuator input, a constant output torque is also obtained throughout the swing range. The rotation speed depends on the input flow rate. The resulting swivel angle is limited by the construction, where the specific value can be adjusted by mechanical stops [23].

The vane rotary actuator with double vanes is similar in design, except that the inner space is divided into four parts. The swivel angle is half compared to the design of the single vanes, but the torque is double.

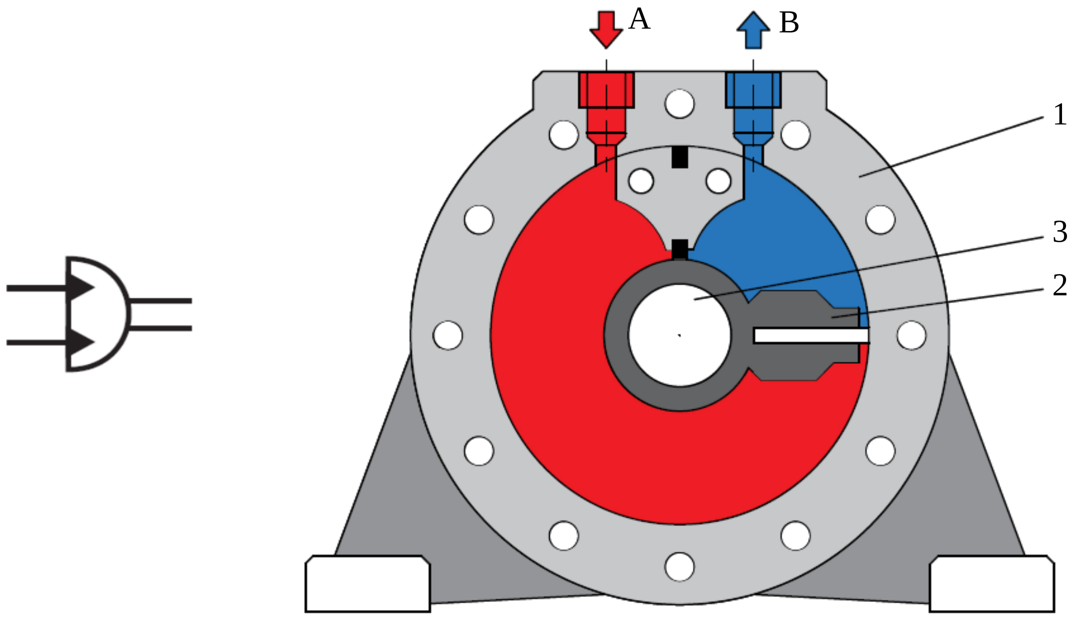


Fig. 5.51 Vane rotary actuator with single vanes

1 – housing, 2 – vane, 3 – driven shaft

In the case of the vane rotary actuator with single vanes, the vanes sealing around the perimeter and on the face surfaces is problematic. This reduces the total efficiency, which is only $\eta_t = (60 \div 70)$ %.

Tab 5.17 Basic parameters of vane rotary actuator with single vanes

Swivel angle α [°]	Nominal pressure p [MPa]	Torque M [N · m]	Total efficiency η_T [-]
270	do 21	up to 24 000	0.6 ÷ 0.7

In-line piston/rotary actuator with rack and pinion drive

In this construction, the actuator (see Fig. 5.52) consists of the housing (1) in which the piston with the rack (2) is guided. The piston in the housing performs a linear motion, which is transmitted to the pinion drive (3), connected to the output shaft (4), by means of gearing. The output swivel angle depends on the gearing length and can be greater than 360°. The torque size depends on the input pressure and the piston area; the rotation speed is based on the input volumetric flow. The swivel angle is adjusted by mechanical stops in the actuator body [11], [12], [20]. Compared to vane actuators, sealing problems are eliminated, which significantly increases the total efficiency. The basic parameters are given in Tab 5.18.

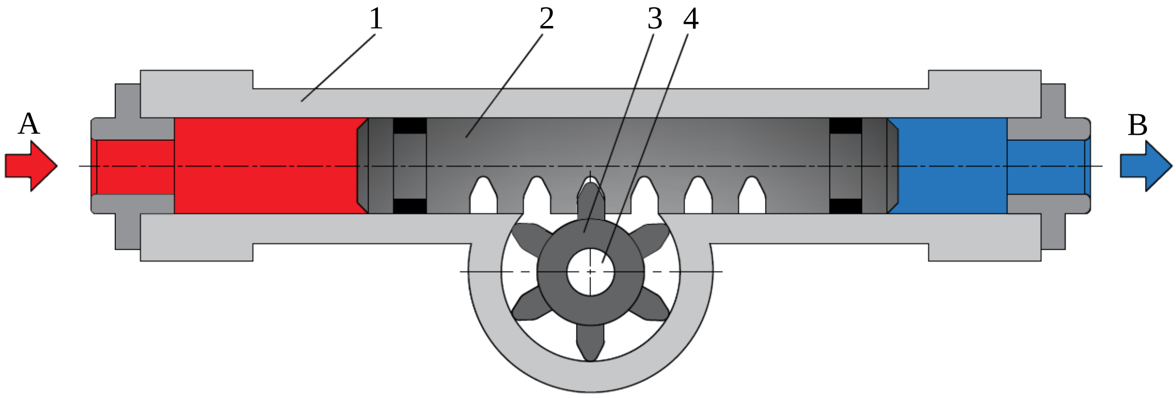


Fig. 5.52 In-line piston/rotary actuator with rack and pinion drive

1 – housing, 2 – piston with rack, 3 – pinion drive, 4 – output shaft

Tab 5.18 Basic parameters of in-line piston/rotary actuator with rack and pinion drive

Swivel angle α [°]	Nominal pressure p [MPa]	Torque M [N · m]	Total efficiency η_T [-]
even more than 360°	do 20	up to 150 000	0.9

These actuators have relatively small dimensions and are mainly used in lifting and manipulation equipment.

In addition to the above two constructions of actuators, there are also sometimes rotary piston/rotary actuator with drive pivot operated by means of a thread or parallel piston/rotary actuator.

5.3 Hydraulic cylinders

Hydrostatic converters with a linear movement of the output component are linear hydraulic motors. In practice, the name “hydraulic cylinders” is used for these converters. The input liquid pressure energy is transformed into the output mechanical energy in the motor housing. The output element of the motor performs a linear movement. Hydraulic cylinders can transfer relatively large forces even with relatively small dimensions. These are the most commonly used hydraulic motors and are used in a wide variety of different applications.

The basic parts of a hydraulic cylinder are shown in Fig. 5.53. The cylinder housing (1) is fixed by means of the front (2) and rear (3) cover. The piston (4) connected to the piston rod (5) moves inside the motor. The linear motion of the piston and piston rod is obtained by means of the liquid pressure, which is fed into the housing by the inputs (6) and (7). The piston is equipped with a guide (8) and a movement seal (9). In the front cover, the piston rod guide (10), the piston rod seal (11) and the wiper (12) are sealed by the O-rings (13) [30].

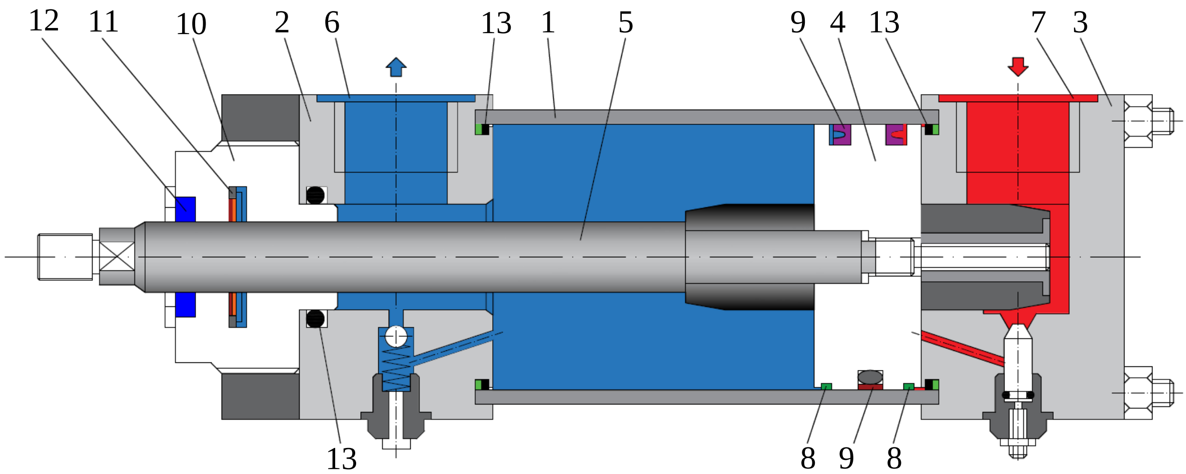


Fig. 5.53 Hydraulic cylinder

1 – housing, 2 – front cover, 3 – rear cover, 4 – piston, 5 – piston rod, 6, 7 – inputs to motor, 8 – piston guide, 9 – piston seal, 10 – piston rod guide, 11 – piston rod seal, 12 – wiper, 13 – O-rings

The design of hydraulic cylinders depends on the required function and operating parameters. The basic division of these motors is:

- according to the force derivation,
  - single-acting – they derive the force from the liquid pressure in one direction only (the return movement is derived from the spring force or an external force),
  - double-acting – they derive the force from the liquid pressure in both directions.
- according to the piston rod design,
  - single-rod cylinder – the piston rod is led in one direction only (they can be single-acting or double-acting, sometimes referred to as differential hydraulic motors),

    - double-rod cylinder – the piston rod is led in both directions (they are usually double-acting).
- by number of working stages,
    - single-stage – classic hydraulic cylinders with one piston,
    - multi-stage – multi-piston motors whose movement or force effect is composed of several parts (telescopic motors, tandem motors).
- according to construction design,
    - bolt,
    - welded,
    - screwed.
- according to the mounting method (cylindrical caps and lugs, flanges, foots, ball joints)
- according to the damping used (motor with or without damping).

Hydraulic cylinders are produced by a large number of manufacturers, often smaller companies. Among the main manufacturers operating in the Czech Republic, it is possible to mention Bosch Rexroth, Parker, Eaton, Hydraulics, or Charvát Group, whose technical data and catalogue data were used in the creation of this chapter.

5.3.1 Basic calculation relationships of hydraulic cylinders

An example of a double-acting single-rod hydraulic cylinder, which is the most commonly used, is given to determine the equations. The output quantities of hydraulic cylinders are the force *F* and the velocity *v*. In order to determine the movement velocity of the motor piston rod, the general continuity equation for an ideal liquid is applied:

$$Q = A \cdot v \Rightarrow v = \frac{Q}{A}. \tag{5.38}$$

where Q [$m^3 \cdot s^{-1}$] is the volumetric flow, A [m^2] is the area and v [$m \cdot s^{-1}$] is the velocity.

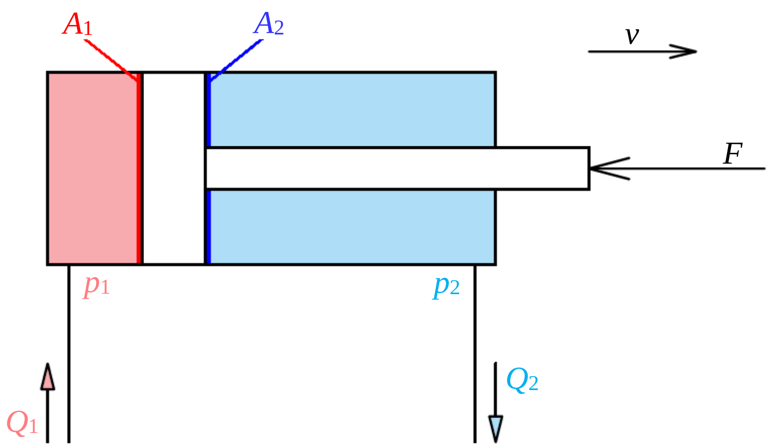


Fig. 5.54 Double-acting single-rod hydraulic cylinder

For the piston rod extension (see Fig. 5.54), it is possible to determine the theoretical extension velocity v_t of the piston rod as follows:

$$v_t = \frac{Q_1}{A_1}, \tag{5.39}$$

where v_t [m · s^{-1}] is the theoretical velocity of the piston rod extension, Q_1 [m^3 · s^{-1}] is the volumetric flow at the motor input, and A_1 [m^2] is the piston area.

Similarly, as with other converters, some energy is dissipated in hydraulic cylinders. The real output velocity of the piston rod v_1 will be with consideration of volumetric losses:

$$v = \frac{Q_1}{A_1} \cdot \eta_{vol} , \tag{5.40}$$

where v [m · s^{-1}] is the real extension velocity of the piston rod, and η_v [-] is the volumetric efficiency of the hydraulic cylinder.

In general, the volumetric efficiency of a hydraulic cylinder can be defined by the following formula:

$$\eta_{vol} = \frac{v}{v_t}. \tag{5.41}$$

Flow losses can occur between the internal parts separated by the piston or at the interface between the piston rod and the outside environment. Modern hydraulic cylinders use seals that are made of high-quality materials, and there is minimal leakage. Flow losses are neglected in most cases. The calculation of the force of the hydraulic cylinder is based on Pascal's law:

$$p = \frac{F}{A} \Rightarrow F = p \cdot A . \tag{5.42}$$

where p [Pa] is the pressure, F [N] is the force, and A [m^2] is the area.

The piston of the double-acting hydraulic cylinder separates the internal spaces in the motor into two parts. During the piston rod extension, the space on the piston is filled with liquid, and at the same time, a different liquid volume must flow out of the space on the piston rod side. For an ideal hydraulic cylinder, the equation of force balance can be written as follows:

$$F_t = p_1 \cdot A_1 - p_2 \cdot A_2 , \tag{5.43}$$

where F_t [N] is the theoretical force, p_1 [Pa] is the input pressure to the hydraulic cylinder (on the piston side), p_2 [Pa] is the output pressure from the hydraulic cylinder (on the piston rod side), A_1 [m^2] is the piston area, and A_2 [m^2] is the annulus area on the piston rod side.

In order for the piston rod to extend, resistance to movement must be overcome. The real force F of a hydraulic cylinder will be lower, as it is necessary to include mechanical and pressure losses in the calculation:

$$F = (p_1 \cdot A_1 - p_2 \cdot A_2) \cdot \eta_{mh} , \tag{5.44}$$

where F [N] is the real force of the hydraulic cylinder and η_{mh} [-] is the mechanical-hydraulic efficiency of the hydraulic cylinder.

It is possible to express the mechanical-hydraulic efficiency by the formula:

$$\eta_{mh} = \frac{F}{F_t}. \tag{5.45}$$

Mechanical losses of straight hydraulic cylinders are caused by seal friction and the guidance between the piston and piston rod. The magnitude of passive (frictional) resistances depends mainly on the selected seal material and the accuracy of machining (roughness) of the internal surfaces of the motor body. Large passive resistances can lead to jerky motion at low motor speeds (so-called stick-slip effect). Pressure losses are mainly caused by local resistances in the input and output channels of the motor. The mechanical-hydraulic efficiency of a hydraulic cylinder generally increases with increasing pressure. Then, the total efficiency of a linear hydraulic motor can be determined as the ratio of the mechanical power and the hydraulic power (i.e., the power input of the motor):

$$\eta_T = \frac{P_m}{P_h} = \frac{F \cdot v}{p_1 \cdot Q_1 - p_2 \cdot Q_2}, \tag{5.46}$$

where η_T [-] is the total efficiency of the hydraulic cylinder, P_m [W] is the mechanical power, P_h [W] is the hydraulic power, and Q_2 [$m^3 \cdot s^{-1}$] is the volumetric output flow from the hydraulic cylinder.

The total efficiency η_T of hydraulic cylinders can be up to 95 % (u servo motors up to 98 %).

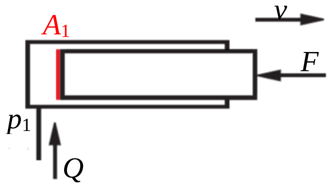
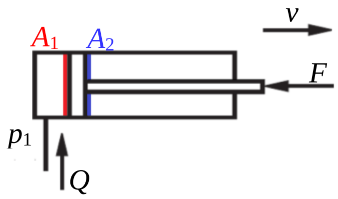
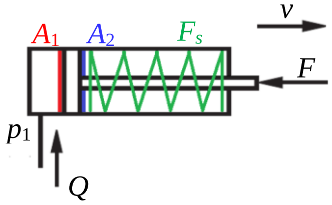
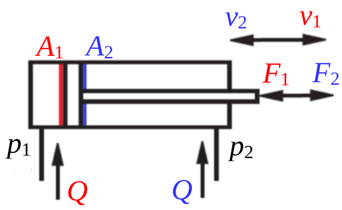
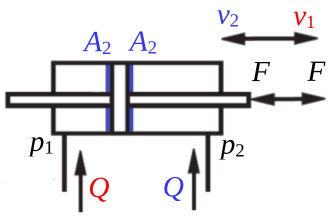
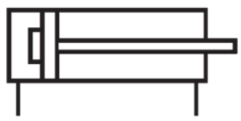
5.3.2 Basic designs of hydraulic cylinders

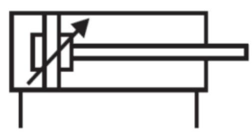
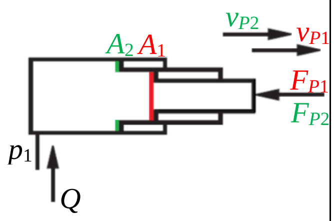
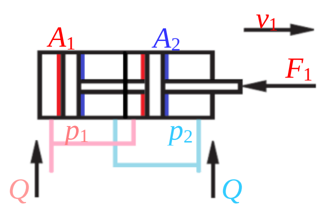
The primary division of hydraulic cylinders is into single-acting and double-acting. Single-acting hydraulic motors have the liquid supply on only one side of the motor. They generate a force in one direction, and reverse motion is achieved by an external force from the load or weight, or by the action of a spring built into the motor body. Double-acting hydraulic motors have a liquid supply on both sides of the motor and generate a force effect in both directions. The magnitude of the resultant force is given by the pressure of the working liquid and the area on which the liquid acts. Hydraulic cylinders are commonly manufactured for pressures up to 40 MPa, diameters of 250 mm and strokes of approx. 4000 mm. For special motor designs, the individual parameters can be higher. The maximum velocity of the piston rod is 0.5 $m \cdot s^{-1}$. The motor stroke h is the length by which the piston rod of the motor can be extended. In typical cases, the motor stroke is approximately equal to the motor length.

In some applications, the motor stroke is required to be greater than its retracted length. In these cases, telescopic hydraulic motors can be used. The telescopic hydraulic motor is composed of several stages. The individual motor stages are gradually extended, starting with the maximum diameter. The total stroke of the motor is approximately given by the multiplication of the length in the retracted state and the number of motor stages. A typical example of the use of telescopic motors is when tipping the body of trucks. In cases where a higher force effect is required and for operational or design reasons it is not possible to increase the pressure or piston diameter of the motor, a tandem motor can be used. In this design, the pistons and piston rods are arranged in series in the motor housing. The greater force effect is given by the sum of the individual surfaces on which the liquid acts. The disadvantage is that the length of the motor increases.

The basic designs of hydraulic cylinders, including their hydraulic graphical symbols are given in Tab 5.19 [1].

Tab 5.19 Basic designs of hydraulic cylinders, output parameters and hydraulic graphical symbols

Design of motor	Output parameters	Graphical symbol
Plunger cylinder The reverse movement is ensured by an external load, the seal is only at the plunger exit from the motor (lower friction), higher weight.	$F = p_1 \cdot A_1$ $v = \dfrac{Q}{A_1}$	
Single-acting cylinder The reverse movement is ensured by an external load.	$F = p_1 \cdot A_1$ $v = \dfrac{Q}{A_1}$	
Single-acting cylinder with spring The reverse movement is ensured by the spring, it is necessary to overcome the spring resistance in order to generate the force effect.	$F = p_1 \cdot A_1 - F_s$ $v = \dfrac{Q}{A_1}$	
Double-acting single-rod cylinder It generates a force effect in both directions, different velocities and forces in both directions are given by the area ratio A_1/A_2. It is called a differential motor.	$F_1 = p_1 \cdot A_1 - p_2 \cdot A_2$ $v_1 = \dfrac{Q}{A_1}$ $F_2 = p_2 \cdot A_2 - p_1 \cdot A_1$ $v_2 = \dfrac{Q}{A_2}$	
Double-rod cylinder The force effect is in both directions, piston rods can be of the same diameter (then there is the same velocity and force in both directions), and of different diameters (the calculation of output parameters is similar to that of the acting single-rod cylinder).	$F = (p_1 - p_2) \cdot A_2$ $v_1 = v_2 = \dfrac{Q}{A_2}$	
Double-acting single-rod cylinder with non-adjustable damping on piston side.		

Design of motor	Output parameters	Graphical symbol
Double-acting single-rod cylinder with non-adjustable damping in both end positions.		
Single acting telescopic cylinder The reverse stroke is ensured by an external load. The motor consists of individual stages that have different areas. The telescopic motor can also be in a double-acting design, then there would be two inputs at the motor graphical symbol.	$F_{P1} = p_1 \cdot A_{P1}$ $v_{P1} = \frac{Q}{A_{P1}}$ $F_{P2} = p_1 \cdot A_{P2}$ $v_{P2} = \frac{Q}{A_{P2}}$	
Tandem cylinder Higher force effect.	$F_1 = p_1 \cdot (A_1 + A_2) - 2 \cdot (p_2 \cdot A_2)$ $v_1 = \frac{Q}{A_1 + A_2}$	

Abbreviations in table:

F – force, F_1 – force during piston rod extension, F_2 – force during piston rod retraction, A_1 – piston area, A_2 – annulus area on piston rod side, p_1 – input pressure to cylinder, p_2 – output pressure from the cylinder, v_1 – extension velocity, v_2 – retraction velocity.

5.3.3 Basic parts and construction designs of hydraulic cylinders

The **housing** of hydraulic cylinders consists of a seamless steel pipe. The inner surface must be machined to the required roughness. Surface unevenness may damage the integrity of seals during operation and thus limit their lifetime. The demands on the quality of the internal surfaces increase even more with increasing working liquid pressure. For low-pressure systems up to approx. 10 MPa, machining to roughness of at least $Ra = 0.4$ µm is required, for higher pressures at least $Ra = 0.2$ µm. Finishing methods for machining internal surfaces are grinding followed by honing, boring and rolling. The materials used must have sufficient tensile strength, be corrosion resistant and, for welded structures, have guaranteed weldability. For less demanding applications, steels of the usual quality can be used, but stainless steels of higher grades are more often used [12].

For thin-walled pipe ($r_2/r_1 \leq 1.18$), it is possible to calculate the limiting wall thickness s_2 according to the equation:

$$s_2 = \frac{p \cdot D}{2 \cdot \sigma_d}. \tag{5.47}$$

where s_2 [mm] is the cylinder wall thickness, σ_d [MPa] is the allowable tensile stress of the material, r_1 [mm] is the outer radius of the pipe, and r_2 [mm] is the inner radius of the pipe.

For thick-walled pipe ($r_2/r_1 > 1.18$), it is possible to calculate the limiting wall thickness s_2 according to the equation:

$$s_2 = \frac{D}{2}\left(\sqrt{\frac{\sigma_d + p\cdot(1-2\mu)}{\sigma_d - p\cdot(1+\mu)}}\right), \tag{5.48}$$

where D [mm] is the outer pipe diameter, p [MPa] is the maximum pressure in the hydraulic motor, σ_d [MPa] is the allowable tensile stress of pipe material, and μ [-] is Poisson's ratio, for steels $\mu = (0.27 \div 0.30)$.

The bottom (cover) of the hydraulic motor can be compared to a circular plate, supported continuously on the circumference, and loaded with a continuous load, where its thickness s_1 can be calculated according to the equation:

$$s_1 = 0.405 \cdot D \cdot \sqrt{\frac{p}{\sigma_d}}. \tag{5.49}$$

The steel **piston** is usually equipped with grooves around its perimeter for the placement of movement seals and piston guidance. The piston is used to transmit the force and also as an auxiliary guide of the piston rod. In the case of hydraulic motors manufactured with smaller diameters, the piston and piston rod may be made from one piece. For larger diameters, a bolted connection is used.

The **piston rod** is used to transmit the force to the driven mechanism. During operation, the surface of the piston rod is in contact with the outside environment. The piston rod is made of steel, with a tough core. The surface of the piston rod must be hard, wear-resistant, corrosion-resistant, and resistant to environmental influences. Stainless steels are used. Hardness and tensile strength are ensured by surface hardening, while internal stresses are removed by subsequent tempering. Corrosion resistance is obtained by galvanic chrome plating, copper plating, or by welding anti-corrosion materials. The surface of the piston rod is ground, and the required quality (minimum Ra = 0.2 µm) is achieved by polishing or superfinishing. This modified surface reduces friction and increases seal lifetime.

An important element of hydraulic cylinders is the piston and piston rod **guidance**. The piston rod guidance is usually bronze in the form of a replaceable sleeve or a welded bronze layer in which spiral grooves are formed to prevent drag pressure that overloads the piston rod seal. It must be sufficiently dimensioned; its length is chosen with regard to the working pressure, the method of motor mounting, the magnitude of the radial force, etc., in the following range [12]:

$$L = (0.8 \div 1.2) \cdot d\ , \tag{5.50}$$

where L [mm] is the guidance length and d [mm] is the piston rod diameter.

The piston guidance can also be bronze or made of highly resistant plastic, and its length is correspondingly shorter. In modern hydraulic cylinders, the guidance is replaced by guide rings and belts.

The piston rod of a hydraulic cylinder can be stressed in compression or tensile during operation. In the case of compressive loading, it is necessary to consider the magnitude of the loading compressive force F, related to the so-called piston rod slenderness λ_p. When the critical force F_{CR} (or critical stress) is exceeded, the piston rod is bent and is stressed not only in compression but also in bending. In this case, it is buckling stress (buckling bending), and it is necessary to **check the piston rod for buckling strength** according to Table 1.4.

Tab 5.20 Methods of checking for buckling strength depending on the piston rod's slenderness

Slenderness of the piston rod	Method of checking
$\lambda_p \leq 40$	check for simple compression or tensile stress
$40 < \lambda_p < \lambda_m$	area of non-elastic buckling, Tetmajer's check for buckling strength
$\lambda_p > \lambda_m$	area of elastic buckling, Euler's check for buckling strength

The piston slenderness is defined as follows:

$$\lambda_p = \frac{l_{red}}{i}, \tag{5.51}$$

where λ_p [-] is the piston rod slenderness, l_{red} [m] is the reduced length of the piston rod, and i [m] is the radius of gyration of the piston rod cross-sectional area.

The reduced length l_{red} of the piston rod depends on the mounting type of a given hydraulic motor. The determination of the reduced piston rod length for conventional hydraulic motor mountings is shown in Fig. 5.55.

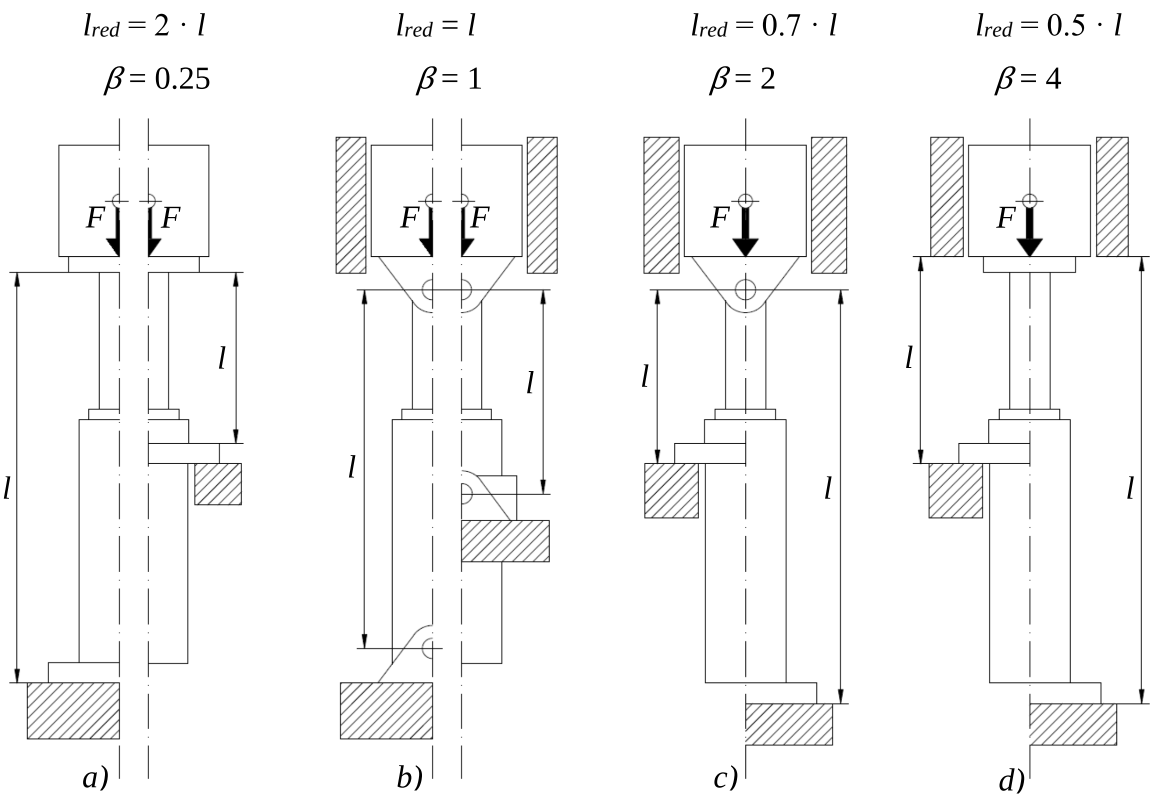


Fig. 5.55 Mounting possibilities of hydraulic cylinder

a) the motor housing is fixed, the end of the piston rod is free, b) the motor and piston rod are equipped with pivots, the piston rod pivot is movable in the motor axis, c) the motor housing is fixed, the piston rod end is equipped with a pivot that is movable in the motor axis, d) the motor housing and piston rod are fixed, the piston rod end is movable in the motor axis

The radius of gyration i of the piston rod cross-sectional area is given by the formula:

$$i = \sqrt{\frac{J}{A}}, \tag{5.52}$$

where J [m^4] is the moment of inertia of the piston rod cross-sectional area to the axis of symmetry of the piston rod and motor, and A [m^2] is the piston rod cross-sectional area.

According to the calculated value of the piston rod slenderness, the further calculation procedure for the buckling check is determined, as is shown in Table 1.4. In the above table, there is a limiting piston rod slenderness λ_m, which can be determined from the equation:

$$\lambda_m = \pi \cdot \sqrt{\beta} \cdot \sqrt{\frac{E}{\sigma_u}}, \tag{5.53}$$

where λ_m [-] is the limiting piston rod slenderness, β [-] the coefficient dependent on the piston rod mounting, the values of which are given in Fig. 5.55, E [MPa] is the Young's modulus of elasticity of a piston rod material, and σ_u [MPa] is the elasticity limit stress of a piston rod material.

Tetmajer's check

It is valid for non-elastic buckling areas, the loading force F (or tensile stress σ_t) is checked, the value of which must be less than the value of the critical force F_{CR} (critical stress σ_{CR}).

The Tetmajer's critical stress is calculated from the equation [41]:

$$\sigma_{CR} = a_s - b_s \cdot \lambda_p \, , \tag{5.54}$$

where σ_{CR} [MPa] is the critical stress, and a, b [MPa] are the material constants depending on the steel grade, the values of which are given in the engineering tables.

The critical force F_{CR} is defined as follows:

$$F_{CR} = \sigma_{CR} \cdot A \, , \tag{5.55}$$

where F_{KR} [N] is the critical force, and A [mm^2] is the piston rod area.

The safety factor S is usually chosen in the range of $S = (2 \div 4)$, whereby the ratio between the critical, and the loading force is given by the relationship:

$$S = \frac{F_{CR}}{F} \, , \tag{5.56}$$

where S [-] is the safety factor, and F [N] is the loading force.

Alternatively, it is possible to perform the check using the critical stress, as follows:

$$S = \frac{\sigma_{CR}}{\sigma_t} \, , \tag{5.57}$$

where σ_t [MPa] is the tensile stress.

The tensile stress is given by the ratio of the loading force to the piston rod area:

$$\sigma_t = \frac{F}{A} \, . \tag{5.58}$$

The loading force F is given by the sum of the static and dynamic forces.

Euler's check

It is used for areas of elastic buckling. It must be true that the loading force F is less than the critical force F_{CRe}. The Euler's critical force is given by the equation [41]:

$$F_{CRe} = \frac{\pi^2 \cdot E \cdot J}{l_{red}^2} \, . \tag{5.59}$$

The safety factor is chosen in the range of $S_e = (2 \div 4)$ and is expressed as follows:

$$S_e = \frac{F_{CRe}}{F} \, . \tag{5.60}$$

The covers are connected to the sides of the cylindrical body of the hydraulic motor. The covers are usually made of steel or cast iron, the front cover is on the piston rod side (motor head), and the rear cover is often referred to as the bottom of the motor. Each cover has holes for the connection of a hydraulic line with working liquid, whose cross-section must correspond to the required flow rate. According to the mounting method of the cover to the motor body, the following constructions of straight hydraulic motors can be distinguished.

Tie rod construction

The motor covers are tightened with tie rods and nuts and are pretensioned to the force corresponding to the working pressure. The tie rod construction is not used for motors of longer length; the working pressures are max. 25 MPa. The advantage of motors of this design is their disassembly, which enables a modular solution and, therefore, makes them suitable for serial production.

Welded construction

In general, motors of welded construction are cheaper. They can be used in a wide range of operating pressures. For higher working pressures, it is necessary to ensure sufficient weld quality. The cover can either be welded directly to the motor body or bolted to a flange that is welded to the motor body.

Screwed construction

Motors of screwed construction are used for middle to high pressures. The covers are screwed directly on the motor body, or they are screwed to a flange which is threaded to the motor body. The motors of this construction are demountable.

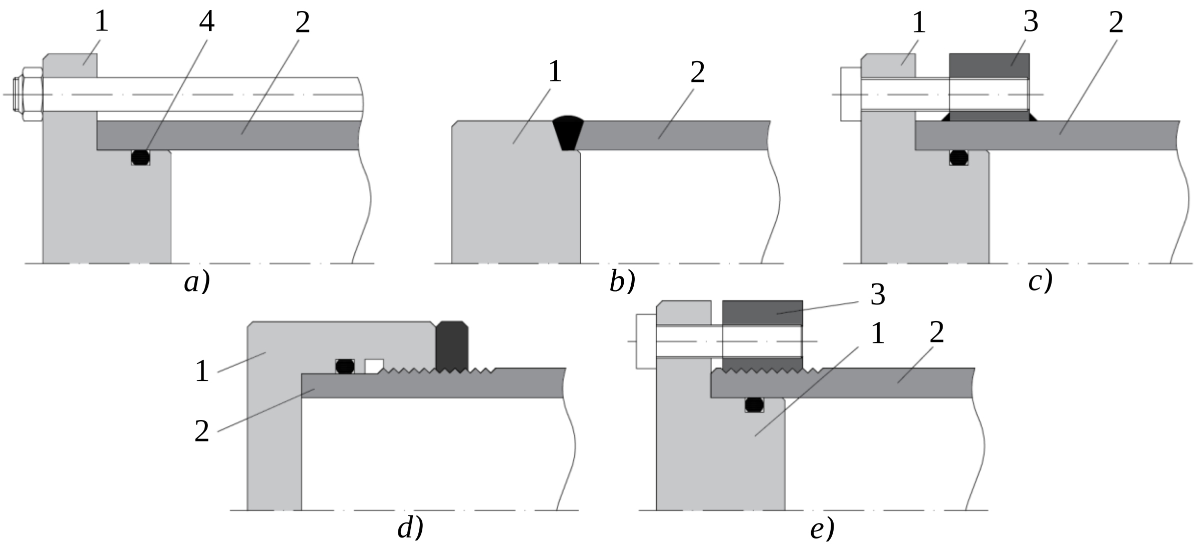


Fig. 5.56 Possible construction solutions of hydraulic cylinders – connection of the motor body to the cover,

a) tie rod connection between motor body and cover, b) cover is welded to motor body, c) cover is screwed to flange welded to motor body, d) cover is screwed on the motor body, e) cover is screwed to flange screwed to the motor body.

1 – cover, 2 – motor body, 3 – flange, 4 – seal (O-ring)

Examples of possible construction solutions for hydraulic cylinders are shown in **Chyba! Nenalezen zdroj odkazů.** [1], [6], [42].

5.3.4 Types of connections of hydraulic cylinders

Depending on the application and the required function, different methods of mounting hydraulic cylinders are used. The motor mounting can be fixed or swivelled [3], [12], [30].

Fixed mounting:

- foot mounting – the feet are secured by supports to absorb the force acting on the piston rod of the hydraulic motor (see Fig. 5.57*a*),
- front flange mounting – mounting suitable for vertically mounted motors, the mounting should be chosen regarding the application of a higher force for the given movement direction of the hydraulic motor; the correct mounting method is shown in Fig. 5.57*b*,
- rear flange mounting – mounting suitable for vertically mounted motors, the correct mounting method is shown in Fig. 5.57*c*.

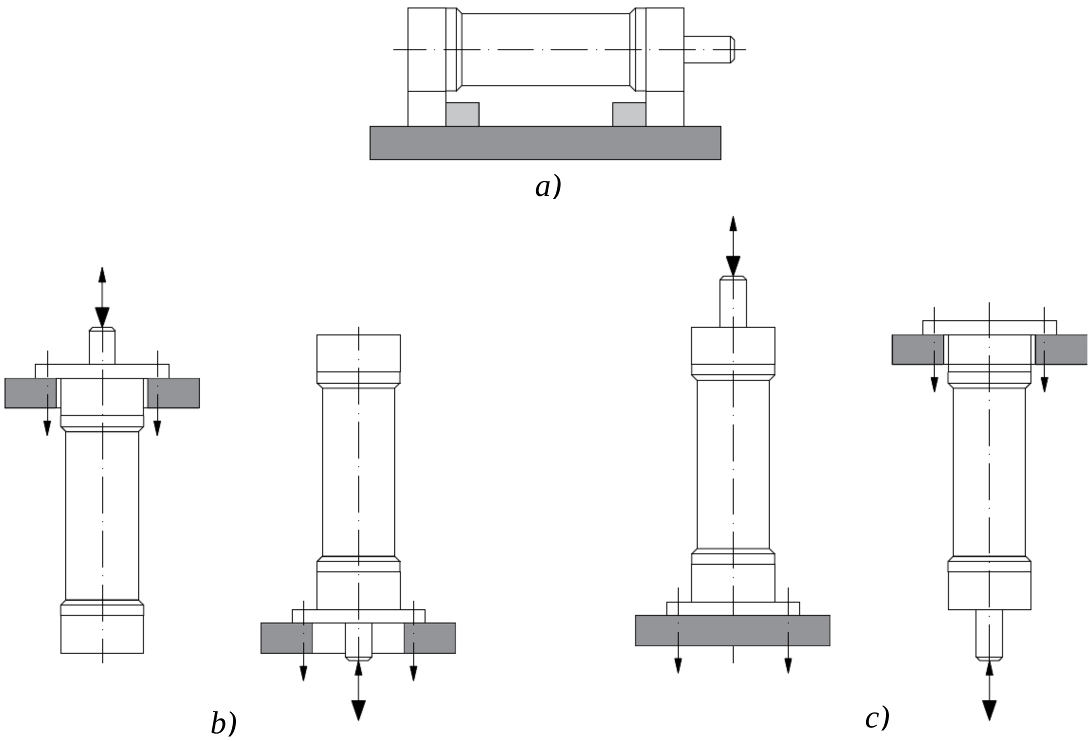


Fig. 5.57 Fixed mounting of hydraulic cylinders

a) foot mounting, b) front flange mounting, c) rear flange mounting

Swivel mounting:

- plain clevis bearing at both ends – it enables deflection in a plane perpendicular to the eye axis (see Fig. 5.58*a*),
- swivel clevis at both ends – it also enables deflection in the plane of the pin axis (see Fig. 5.58b),
- centre trunnion mounting – the pin is welded to the sleeve, which is connected to the motor body (see Fig. 5.58*c*). Alternatively, the sleeve can be mounted to the flange,
- on ball journal – no radial forces are transmitted, for example, for telescopic motors (see Fig. 5.58*d*).

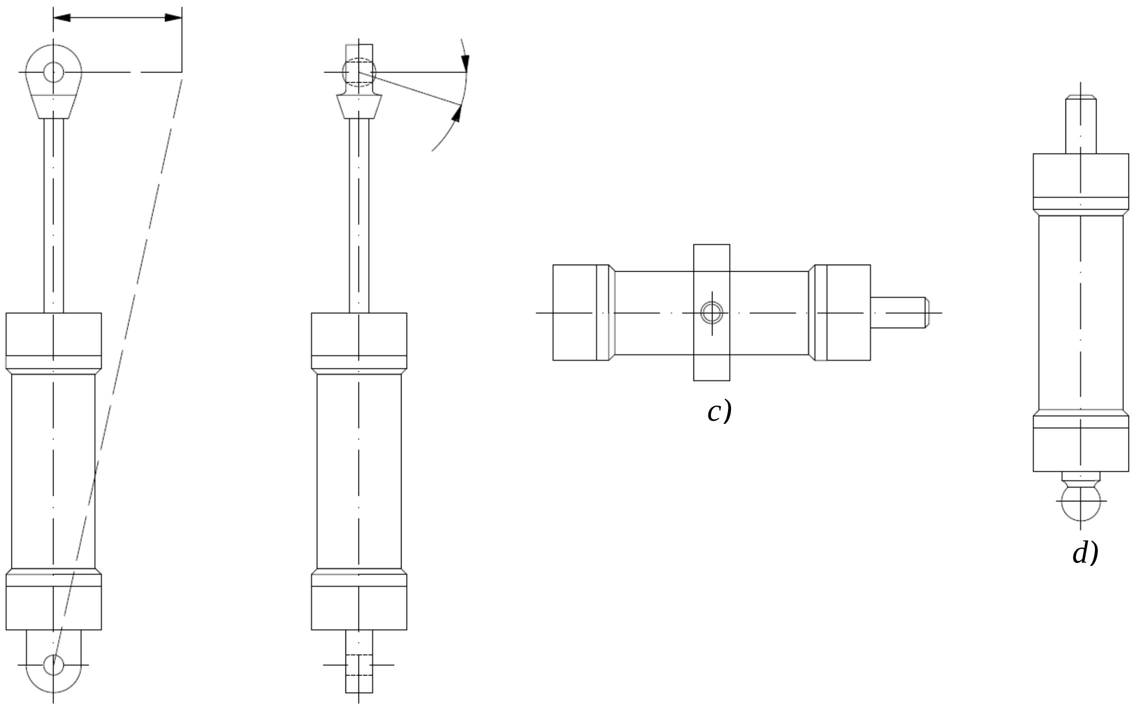


Fig. 5.58 Swing mounting hydraulic cylinders

a) plain clevis bearing at both ends, b) swivel clevis at both ends, c) centre trunnion mounting, d) ball journal

The piston rod of a hydraulic cylinder is usually fitted with an external thread or drilled with an internal thread (less common). It is possible to connect an eye with a sliding bearing, an eye with a spherical bearing or, for example, a fork with a cylindrical pin to the end of the piston rod. The combination of the swivel mountings of hydraulic cylinders and the piston rod ends allows the motors to be deflected in different planes and eliminate some misalignment of the pins [43].

5.3.5 Cushioning of end positions of hydraulic cylinders

The piston and piston rod move with relatively high kinetic energy in the body of the hydraulic motor. Stopping the piston movement at the stroke end by the piston impact with the cover would cause increased stress on the motor, shortening its service life and consequent damage to the motor, especially at higher movement velocities and with large connected inertial masses. For this reason, the cushioning of hydraulic motors in end positions is applied (see Fig. 5.59).

The piston (1) is equipped tapered cushioning bush (2) that enters the bore (3) in the cover (4). Between the piston and the cover, the liquid in the space (5) is closed. From this space, the liquid is discharged through the bore (6) and the throttle valve (7), to the output of the hydraulic motor (8). The damping effect can be adjusted on the throttle valve (the smaller the flow area of the throttle valve, the greater the damping effect). When the piston moves in the opposite direction, the space (5) under the piston is filled through the check valve (9) and the bore to the check valve (10).

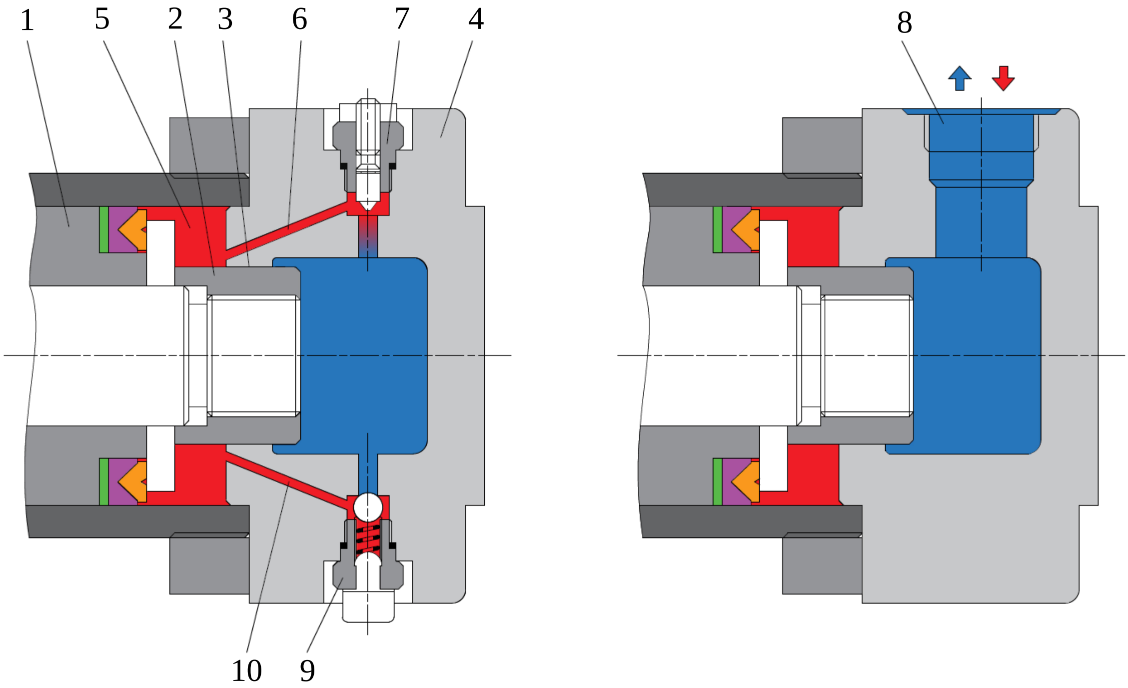


Fig. 5.59 Adjustable cushioning of piston at cylinder base

1 – piston, 2 – tapered cushioning bush, 3 – cover bore, 4 – hydraulic cylinder cover, 5 – liquid space, 6 - bore to the throttle valve, 7 – throttle valve, 8 – output of the cylinder, 9 – check valve, 10 – bore to the check valve

The cushioning can be applied at one or both ends of the hydraulic motor. The cushioning effect can be constant (an orifice is used instead of the throttle valve) or, more often, adjustable by means of the throttle valve. By means of constructional modifications of the cushioning bush, it is possible to change the piston velocity on the travel path. It is recommended to use motors with damping at movement velocities higher than 0.1 $m \cdot s^{-1}$.

5.3.6 Special designs of hydraulic cylinders with control block

Linear hydraulic motors with implemented control blocks (see Fig. 5.60) are used on equipment for component and material testing in research and development departments and also in testing rooms of many industrial sectors.

Materials, components and structural elements which are to be tested are subjected to mechanical loads in order to check their function, suitability for specific applications, degree of wear and lifetime. Hydraulic drives are used to ensure the required mechanical load, especially when creating a highly dynamic load profile. To ensure excellent drive control quality and the requirement for highly dynamic motion, a linear double-rod hydraulic cylinder in servo quality with low frictional resistance is used.

These hydraulic motors are produced for nominal pressures up to 28 MPa, working strokes of (20 ÷ 200) mm, velocities up to 5 m · s^{-1} and a maximum frequency of 50 Hz. The hydraulic motor usually includes a piston rod position measuring system with SSI output and resolution up to 0.5 μm.

Fig. 5.60 Special version of linear hydraulic motor with Bosch Rexroth control block [44]

The hydraulic control block (see Fig. 5.61) consists of a main control valve (servo valve) which ensures the desired velocity and force profile, two accumulators, check valves and diagnostic elements.

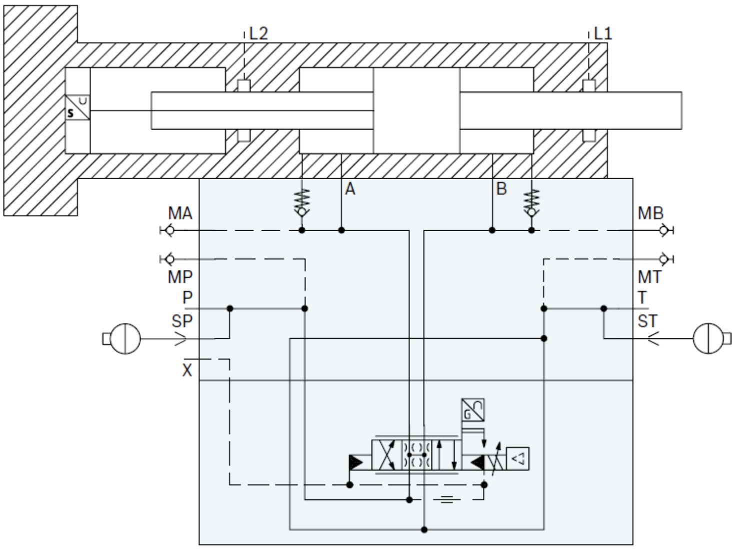


Fig. 5.61 Schematic diagram of a hydraulic motor with control block [44]

6. Seals in hydraulics

The purpose of the seals is to prevent liquid leakages from a system, or to reduce leakages between functional parts within the system. Due to the fact that hydraulic equipment often operates at high pressures, seals are an important part of most hydraulic components. The correct selection of seals has a significant impact on the operational reliability, lifetime and efficiency of hydraulic equipment. The selection of the seal is primarily based on the size of the sealed gap and the pressure size. For some internal parts of hydraulic elements, metal-to-metal seals are used. A very fine degree of fit is chosen, and high surface and manufacturing accuracy is required with clearances in the order of units of μm (a typical example are spools of directional valves). In this case, there are flow losses, but the leakages are so small that they can be tolerated in terms of losses. Moreover, for moving parts, a small leakage is desirable because it allows the formation of a lubricating film and prevents dry friction.

The basic requirements for seals are:

- perfect or high tightness,
- s compatibility with working liquid and material of a sealed component,
- long service life (resistance to wear and degradation of material properties during ageing),
- low friction coefficient (for the sealing of moving parts).

The seals must meet the above requirements over the whole range of working pressures and temperatures. In addition, high reliability, simple assembly and replaceability, and low cost are required for seals. Depending on the location and required sealing function, the seals are divided into:

- stationary – for sealing components that do not move relative to each other,
- moving – for sealing mutually moving parts.

For stationary and moving seals, a further distinction is made as to whether they are used to seal static pressure or dynamic pressure (which is variable over time).

Based on the required function and operating conditions, a suitable sealing material is selected.

The main manufacturers of hydraulic sealing technology include Hennlich, Parker, Trelleborg and SKF, whose technical and catalogue data have been used for this chapter.

6.1 Materials of sealing elements

In hydraulic systems, elastomers and thermoplastic elastomers are most commonly used as seal materials. Elastomers are flexible (elastic), while thermoplastic elastomers are flexible only up to a certain temperature and gradually change to a plastic (inelastic) state. There are a large number of sealing materials, they often differ only in partial properties, and their parameters are variable according to the composition of the mixtures used in their production. Letter designations are used to designate groups of materials in accordance with the ISO standard. In practice, we can often see the trade names of individual materials. The basic parameters of elastomers are temperature resistance and hardness. The hardness of rubber, vulcanised, or thermoplastic materials are measured by one of the Shore methods. Manufacturers' catalogues usually state the Shore A (ShA) hardness value. The most commonly used materials for sealing elements in hydraulic systems, including their designations and basic properties, are shown in Tab 6.1 [11], [12], [16], [23], [45].

Tab 6.1 Materials of sealing elements and their basic parameters

Material designation	Business name	Material type	Hardness [ShA]	Temperature resistance [°C]	Note
NBR	Perbunan Europrene Breon	Acrylnitril-Butadien	70 ÷ 90 (40 ÷ 90)	-30 ÷ 100 (short-term to 130)	the most commonly used elastomer in basic hydraulic circuits
HNBR	Therban	Hydrogenated acrylonitrile butadiene	70	-30 ÷ 150	similar properties to NBR, use for higher temperatures
FPM	Viton Fluorel	Fluor rubber	75 ÷ 80 (60 ÷ 90)	-20 ÷ 200 (+230)	usually used as a replacement for NBR in more difficult applications
EPDM	BunaAP Dutral	Ethylen-Propylen-Dien	70 (60 ÷ 80)	-45 ÷ 155	
AU, EU	Polyuretan Hythane	Polyester uretane Polyether uretane	93	-35 ÷ 110	highly wear-resistant elastomer that does not tolerate hot water and steam
PTFE	Teflon	Polytetrafluorethylene	98	-200 ÷ 250	thermoplastic, very low friction coefficient, used in combination with an elastic element
SBR	Cariflex	Styren-Butadien	30 ÷ 95	-25 ÷ 100	
IIR	Butyl Polysar	Butyl rubber	40 ÷ 80	-25 ÷ 120	

The compatibility of sealing materials with working liquids is very important for hydraulic systems. An orientation overview is given in Tab 6.2, the suitability of use is shown by the coloured marks [11], [12], [16], [23]. If two different colours are indicated, the application of the given material may be conditional or based on the diversity of product mixtures.

Tab 6.2 Compatibility of sealing materials with working liquids

Material designation	Mineral oils	Non-flammable liquids				Ecological liquids			
	HH ÷ HG	HFAE	HFB	HFC	HFDR	HETG	HEPG	HEES	HEPR
NBR	🟢	🟢	🟢	🟢	🔴	🟡	🟡🔴	🟡	🟡
HNBR	🟡	🟡	🟡	🟡	🔴		🟡	🟡🔴	🟡
FPM	🟡	🟡	🟡	🟡🔴	🟢	🟢	🟢	🟢	🟢
EPDM	🔴	🔴	🔴	🟢	🟡	🔴	🔴	🔴	🔴
AU, EU	🟡	🟡🔴	🟡🔴	🟡🔴	🔴	🟡	🟡🔴	🟡	🟡
PTFE	🟢	🟢	🟢	🟢	🟢	🟢	🟢	🟢	🟢
SBR	🔴	🔴	🔴	🟢	🔴				
IIR	🔴	🔴	🔴	🟡	🟢				
🟢 suitable for use, 🟡 can be used, 🔴 must not be used									

Suitable sealing materials for individual hydraulic components are standardly specified in catalogues. It is always necessary to follow manufacturers' specifications for the selection of sealing materials.

6.2 Stationary seals

Stationary seals are used to seal gaps between components that do not move relative to each other. These seals usually require perfect tightness without leakages. Stationary seals are used in hydraulic equipment,, e.g. for sealing flanges and screw connections, seals between the cover and body of linear motors, seals between individual elements in valve blocks, seals in pipe fittings and connecting parts, seals in housings and covers of aggregates, etc. Almost all commonly specified materials are used for stationary seals.

O-rings

They are the most commonly used seals in hydraulic equipment. They are characterized by their simple construction, easy installation and low price. They are flexible rings of circular shape, the characteristic dimensions of which are the inner diameter d_1 and the diameter of the cord cross-section of the ring d_2 (see Fig. 6.1).

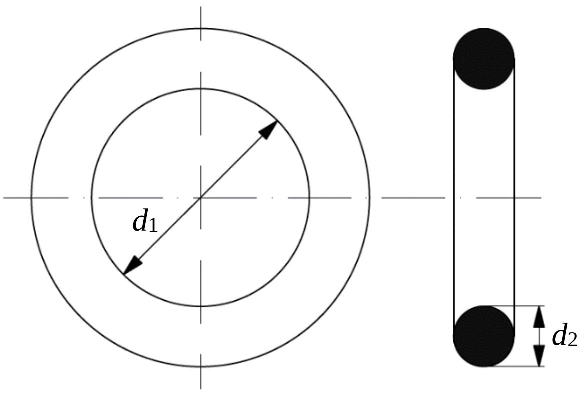


Fig. 6.1 Shape and dimensions of O-ring

The O-rings are inserted into the slots of the rectangular profile. When the ring is placed in the slot, the cord diameter exceeds the slot height. After careful insertion of the counterpart, the ring is deformed by (6 ÷ 26) %; by this compression, the ring obtains a preload, as is shown in Fig. 6.2 (left). The magnitude of the deformation depends on the slot depth and is defined by the selected diameter of the ring cord. In order to achieve the sealing effect and to ensure the required ring service life, the ring surface must not be damaged. For this reason, the ring is lubricated with oil or vaseline and the counter piece must be fitted with a run-in with rounded edges. As the system pressure increases, the ring is pressed against the opposite wall of the slot. Its cross-section is deformed, and the sealing capacity increases, as is shown in Fig. 6.2 (right).

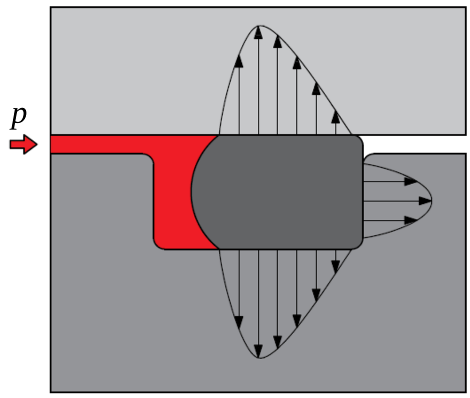


Fig. 6.2 O-ring mounting in the rectangular slot, ring deformation after counterpart inserting (left), ring deformation after applying pressure (right)

When the system reaches the limit pressure, the ring starts to be pressed into the sealed gap; see Fig. 6.3 (left). This phenomenon is called extrusion and can lead to permanent ring damage. Extrusion cannot be completely prevented, but an increase in the limit pressure can be achieved, for example, by reducing the thickness of the sealed gap or by using a ring made of a harder material.

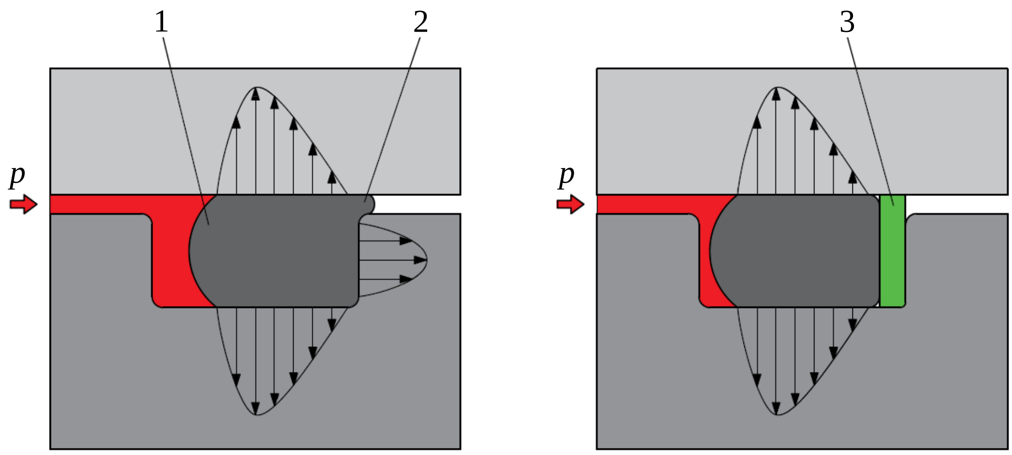


Fig. 6.3 Ring pressing into a sealing gap (left), use of support ring (right)

1 – O-ring, 2 – pressing the ring into a sealed gap, 3 -support ring

However, reducing the gap thickness may not be possible in production, or it would be too expensive, and rings made of harder materials have a greater permanent compressive deformation and, thus, shorter service life. The best solution to increase the bearing capacity of the O-ring is to use a support ring made of plastic material, as shown in Fig. 6.3 (right).

The diameters of the ring cords are manufactured in the range $d = (1 \div 16)$ mm, for inner ring diameters in the range $d_1 = (1 \div 665)$ mm. The hardness of O-rings used in hydraulic systems is in the range of (70 ÷ 93) ShA and is chosen depending on the size of the maximum working pressure (see Tab 6.3) [46]. O-rings are mainly used for sealing static pressure.

Tab 6.3 Selection of O-ring material hardness depending on maximum working pressure [45]

Hardness [ShA]	Maximum pressure [MPa]
70	10
80	20
90	50

X-ring

Classic O-rings cannot be used as seals in applications where large dynamic pressure changes occur. Pressure changes cause pulsating movements of the O-ring in the slot and its rotation (it may even twist the ring), which leads to increased wear and reduced service life of the ring. X-rings can be used in such applications. The characteristic dimensions are the inner diameter d_1 and the size h of the square cross-section, which is in the shape of the letter “X” Fig. 6.4 (left).

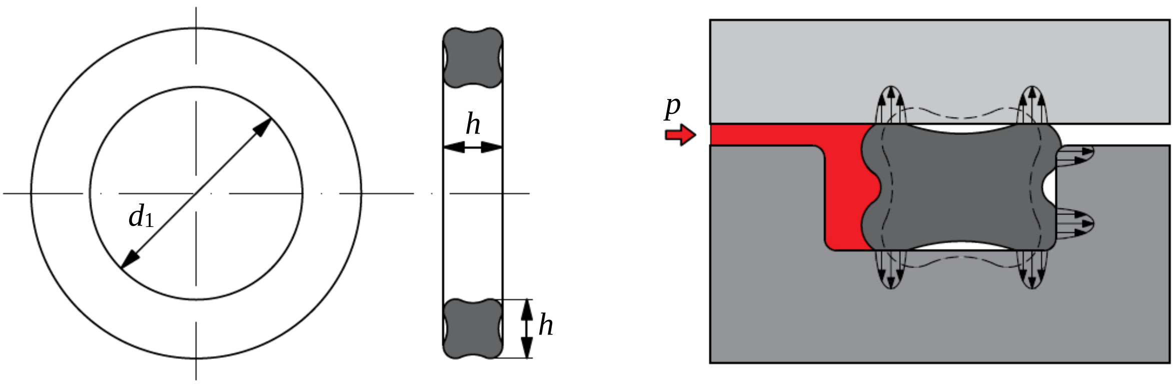


Fig. 6.4 Shape and characteristic dimensions of X-ring (left), use of X-ring (right)

X-rings are also placed in rectangular slots, and the principle of their function is similar to that of O-rings. Their shape with four sealing edges increases the sealing effect (see Fig. 6.4 (right)) and prevents rotation of the X-ring in the slot under dynamic loading. For higher working pressures, they are used in combination with a support ring to prevent extrusion. As standard, they are made of NBR or FPM rubber for pressures up to 40 MPa [47].

X-rings can also be used as motion seals in less demanding applications. Due to their shape, a small amount of liquid enters between the sealing edges, which creates a lubricating layer and reduces friction. The required initial deformation (preload) is not as large as with O-rings, which reduces friction and increases the lifetime of the seal, especially for seals of moving parts. X-rings are also used to seal the oscillating and rotary motion of shafts and spindles.

Profiled sealing rings

These are rings of various profiles used to seal threaded connections in hydraulic systems Fig. 6.5 (left). They are shaped in such a way that there is only minimal deformation of the seals in the connection, ensuring reliable sealing and long service life even in systems with high pressures up to 60 MPa. The hardness of the ring is usually (80 ÷ 85) ShA. Profiled rings are used together with O-rings, e.g. in ES-4/ES-4VA (VOSS), EO2-FORM (Parker) or WALFORM (Eaton) fittings, and their shapes vary according to the design of individual manufacturers.

Flat metal rings

These are flat steel rings (washers) under screw heads, which have a vulcanized sealing edge made of elastic material on the inner diameter. When the screw is tightened, the cutting edge is deformed by the opposing areas, thus producing a sealing effect. They are used to seal screw and flange connections Fig. 6.5 (right).

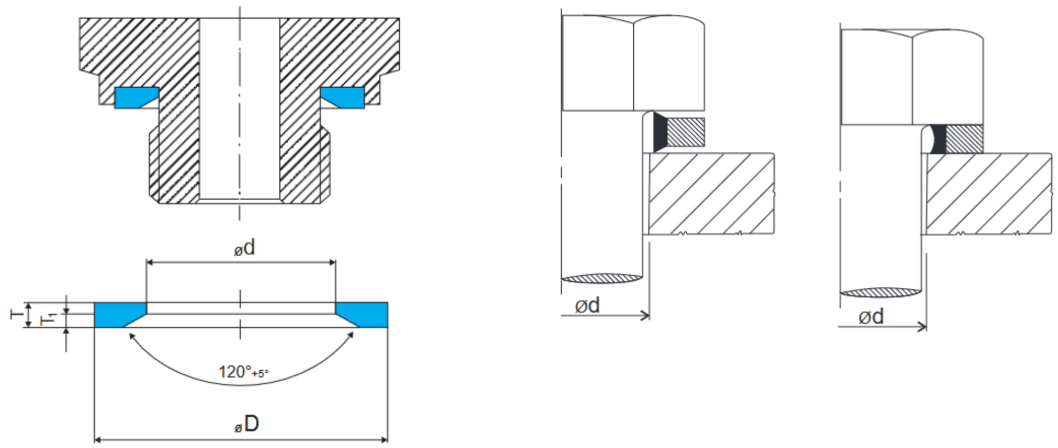


Fig. 6.5 Profile HTR sealing rings from Hennlich company [48] (left), flat steel ring – bonded seals from Hennlich company [49] (right)

6.3 Movement seals

They are used to seal mutually moving parts; a typical example is the seal between the piston and piston rod of linear hydraulic motors. They are more demanding as they must ensure the required tightness in the stationary state even when the sealed component is moving. During movement, the sealing elements are subjected to higher stresses, which increases seal wear. In hydraulic systems, there is also a requirement that a thin layer of the lubricating film remains behind the sealing element during movement, which ensures lubrication of the sliding surfaces and reduces friction.

Most often, various types of sealing sleeves are used as movement seals, which can be single-element or multi-element, supplemented with thrust, support, or anti-extrusion rings. For higher pressures, combined seals are used, which consist of several different sealing elements. Wiper and guide rings can also be included in the movement seals.

Latch sealing sleeves

The basic design of a latch sealing sleeve with a U-shaped profile is shown in Fig. 6.6. The sleeve profile is terminated on one side by a support area (1), which can be flat or rounded. On the other side of the sleeve, there are latches (2), which can be the same length (symmetrical), or one of the latches is longer. The sleeve is placed in the slot with a small overlap; after the pressure liquid is supplied, the shape of the sleeve is deformed and fills the sealed space. Similar to O-rings, the sealing effect of the sleeve increases with the increasing system pressure. The highest sealing effect is at a point called the heel of the sleeve (3); at the same time the highest wear of the sleeve is at this point.

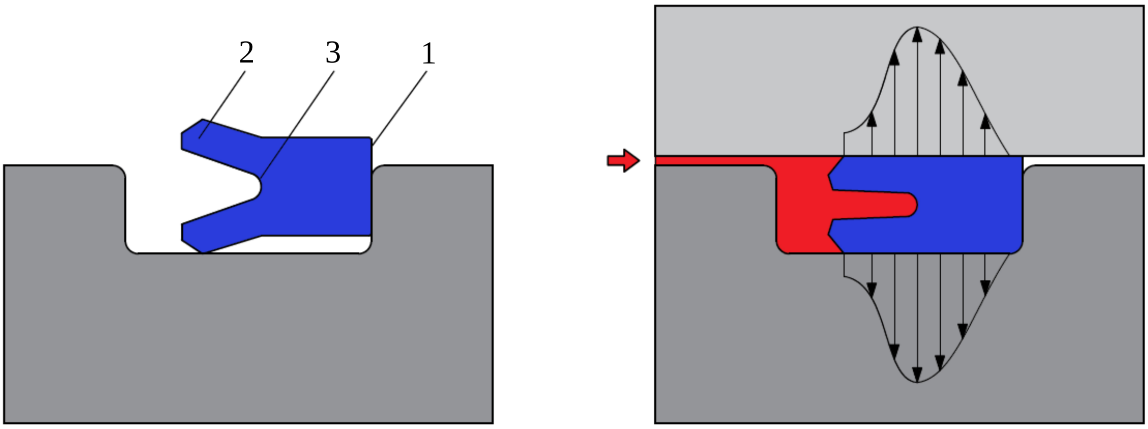


Fig. 6.6 Latch sealing sleeve with U-shaped profile

1 – support area, 2 – latches, 3 – heel of the sleeve

These sleeves can be used for sealing of both the piston and the piston rod. Most types of cuffs are unyielding, and when used for piston seals, the sleeve cannot be inserted over the outer diameter. In this case, the piston must be dismountable. The sleeves are single-acting seals, that is, they generate a sealing effect in one direction only. If sealing in both directions is required, two sleeves must be used, which are placed in the mirror image, with the supporting parts facing each other (usually in separate slots).

They are manufactured from NBR rubber for lower pressures up to approx. 16 MPa and movement velocities up to a maximum of 0.5 $m \cdot s^{-1}$. For higher pressures, PU (polyurethane) is used for pressures up to 28 MPa and maximum movement velocities of 1 $m \cdot s^{-1}$, or for pressures up to 40 MPa and a maximum velocity of 0.5 $m \cdot s^{-1}$. They are used for light to medium working conditions [50], [51].

To increase the sealing effect Fig. 6.7, in some cases, the sleeve (1) in connection with a thrust ring (2) (made of harder material) is used. Hard rubber support rings (3) are used to prevent the sleeve from being pressed into the sealed gap.

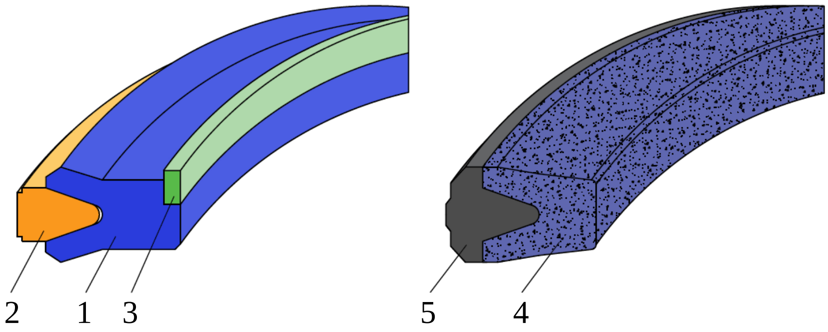


Fig. 6.7 Latch sealing sleeve with thrust and support ring (left), textile rubber sleeve connected with soft rubber part (right)

1 – latch sealing rubber sleeve, 2 – thrust ring, 3 – support ring, 4 – textile rubber sleeve, 5 – soft rubber part

The sleeves can be all-rubber or textile-rubber. The all-rubber design is not very suitable for sealing moving parts. It does not ensure sufficient lubricating film formation, and the linear hydraulic motor is prone to jerky movement at low speeds (stick-slip effect). In the textile-rubber design, the sleeve body is reinforced with a textile insert. The textile insert stiffens the sleeve, and increases the resistance to the cuff material being pushed into the sealed gaps, but mainly by absorbing liquid, it facilitates the formation of a lubricating film during start-up. For sealing the piston rod, textile-rubber sleeves are usually used, which are connected to a soft rubber part (usually made of NBR rubber); see Fig. 6.7 (right). The soft part (5) guarantees the preload of the latches, transmits the pressure from the liquid to them evenly and, at the same time, protects the inner part of the sleeve (4) from wear. Use for pressures of max. 30 MPa and movement velocities of max. 0.5 m $\cdot$ s^{-1} [50], [51], [*52*].

This seal type is designed for the highest pressures up to 60 MPa and is equipped with a ring made of hard elastomer (PTFE), which reinforces the heel of the sleeve. Suitable for moderate to heavy operating conditions [12].

Terrace sleeves

The latches of the terrace sleeves are more open, and the support part is significantly smaller compared to the previous version of the sleeves. They are characterized by a profile in the shape of the letter V. They are only used as multiple elements, mounted in sets between the thrust and support rings. They are characterized by high reliability and can be used for sealing both piston and piston rod. A typical example of a multi-element terrace sleeve for sealing the piston rod is shown in Fig. 6.8 (left).

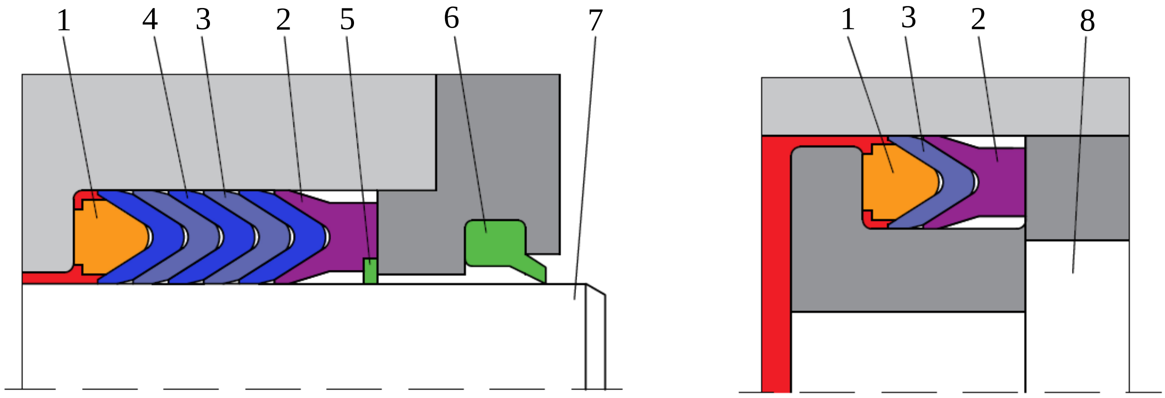


Fig. 6.8 Seal with terrace sleeves, seal of the piston rod (left), piston seal (right)

1 – thrust ring, 2 – support ring, 3 – textile-rubber sleeve, 4 – rubber sleeve, 5 – anti-extrusion ring, 6 – wiper, 7 – piston rod, 8 - piston

The seven-element seal consists of the thrust ring (1), which is smaller in order to allow liquid access to the sleeves. The seal further consists of the support ring (2), which may be equipped with an anti-extrusion ring (5), the terrace textile-rubber (3) and the rubber (4) sleeves, which are alternately placed between the support ring and the thrust ring. The rubber sleeves are usually made of NBR rubber, and the thrust ring is made of hardened fabric; the support ring for smaller diameters is made of thermoplastic, and for larger diameters, it is usually a textile-rubber ring. Terrace sleeves for sealing the piston rod usually have five or seven elements. They are commonly used for pressures up to 40 MPa and maximum movement velocities of 0.5 $m \cdot s^{-1}$, in a special design for the heaviest working conditions for pressures up to 70 MPa and velocities of 0.15 $m \cdot s^{-1}$ [12], [30], [50], [51], [52].

Terrace sleeves for piston sealing are three-element, as shown in Fig. 6.8 (right). They consist of the thrust ring (1), the support ring (2) and the terrace sleeve (3). The thrust ring is usually made of plastic (acetal), the sleeve can be rubber (PU) or for higher pressures, textile-rubber, and the support ring is usually textile-rubber. The terrace sleeves are also single-acting seals. In the case of a double-acting linear hydraulic motor, a special seal must be used for both movement directions. The maximum allowable working pressures and movement velocities are the same as for the seal of the piston rod.

Combined seal

These seals consist of several sealing elements of different designs. They are usually double-acting seals with a sealing effect in both movement directions. They are characterized by a high sealing effect even in the stationary state and allow sealing larger sealing gaps. They are used for seals of the piston and piston rod. An example of a combined seal is shown in Fig. 6.9 (left). The O-ring (1) has the function of the thrust ring, where even at rest, it creates a preload that is transmitted to the shaped sealing element (2). The sealing effect is also increased with the increasing pressure due to the O-ring deformation.

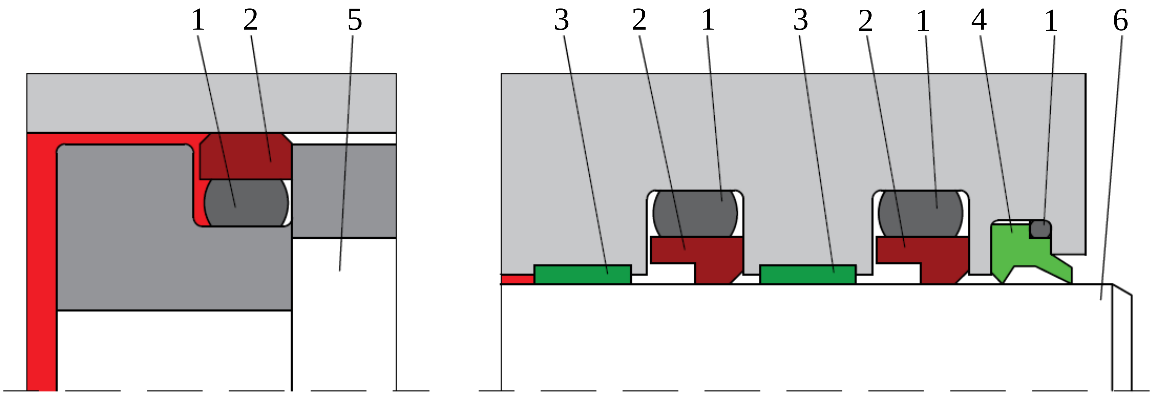


Fig. 6.9 Combined seal, two-element piston seal (left), an example of a combined seal of piston rod (right)

1 – O-ring, 2 – shaped sealing element, 3 – guide rings, 4 – wiper, 5 – piston, 6 – piston rod

For the piston sealing, the O-rings are usually made of NBR or FPM and, in some versions, can have a cross-section other than circular. The sealing element 2 is always made of a material with high wear resistance. Based on the material used, maximum working pressures and movement speeds are given in Tab 6.4. When PTFE material is used, the element is usually filled with bronze powder, anthracite, or glass wool to increase its lifetime. These elements can have different shapes depending on the operating conditions. They are produced in diameters of (12 ÷ 400) mm and are also suitable for one-piece pistons (simple assembly) [50], [51].

When the piston rod is sealed, the O-rings are also made of NBR or FPM. The sealing element has a different shape (see Fig. 6.9 (right)), which is adapted to the pressure load (the pressure liquid acts on the seal of the piston rod from one side only). The material of the sealing element is TPE or PTFE. The values of the maximum operating parameters are in accordance with Tab 6.4.

Combined seals are usually used with guide belts made of hardened fabric or PTFE and bronze.

Tab 6.4 Maximum operating parameters for materials of combined seal [50], [51]

Seal materials	Maximum pressure [MPa]	Maximum movement velocity [$m \cdot s^{-1}$]
PU	20	1
	25	0,5
TPE (thermoplastic)	25	1
	30	0,5
PTFE (thermoplastic)	25	4
	30	2
	35	1
PA (polyamide - thermoplastic)	50	2

An example of a compact piston seal with integrated guide rings is shown in Fig. 6.10. Flexible part (1) is made of softer rubber (NBR), and the sealing element (2) of harder, wear-resistant rubber (PU). The integrated guide rings (3) are usually plastic (acetal, polyacetal). It can be used for pressures up to 25 MPa and movement velocities of 0.5 m · s^{-1}, or up to 40 MPa for movement velocities of 0.15 m · s^{-1} [52], [52].

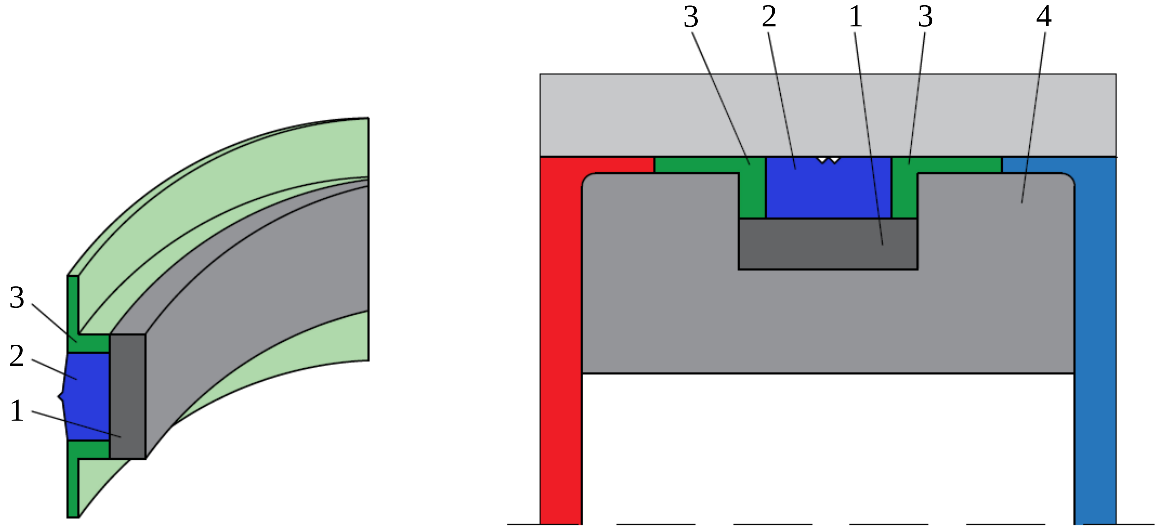


Fig. 6.10 Compact piston seal with integrated guide rings

1 – flexible part, 2 – sealing element, 3 – guide rings, 4 - piston

Guide rings and belts

Guide rings or belts define the position of the piston inside the cylinder and, in the case of the piston rod, are used as bearings to absorb radial forces. They must also ensure that the metallic surfaces of the piston, piston rod and motor housing do not come into contact.

The guide rings (belts) are made of hardened fabric (fabric and polyester resin), PTFE and bronze, or polyacetal [54]. The required length (width) of the ring can be calculated according to the equation [12]:

$$L_1 = \frac{(2 \div 4) \cdot F_r}{p_{sA} \cdot d}, \tag{6.1}$$

where L_1 [mm] is the ring length, F_r [N] is the radial force acting on the piston rod, p_{sA} [N · mm^{-2}] is the allowable specific pressure, and d [mm] is the piston rod diameter.

Values of the allowable specific pressure p_{sA} depend on the used material and temperature, as is shown in Tab 6.5. The table also shows the maximum movement velocities of the individual materials and the range of diameters of the piston and piston rod for which they are produced.

Tab 6.5 Values of the allowable specific pressure p_{sA} of materials of guide rings [53]

Ring material	Allowable specific pressure p_{sA} [N·mm^{-2}]			Maximum movement velocity	Piston rod diameter	Piston diameter
	-20 °C	+23 °C	+80 °C	[m · s^{-1}]	[mm]	[mm]
hardened fabric	110	115	58	5	15 ÷ 355	25 ÷ 680
PTFE and bronze	20	20	9	5	8 ÷ 900	10 ÷ 900
polyacetal	60	70	31	1	16 ÷ 80	32 ÷ 90

An example of the use of guide rings on the piston rod is shown in Fig. 6.9 (right), the piston guide is shown in Fig. 6.11, or by means of integrated guide rings in Fig. 6.10.

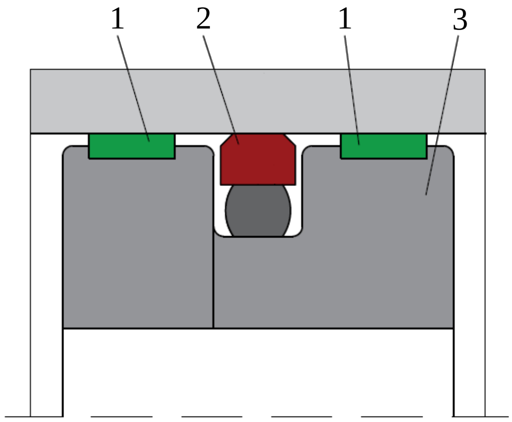


Fig. 6.11 Example use of guide rings for piston guide

1 – guide rings, 2 – combined seal, 3 - piston

Wipers

The wiper is used to wipe impurities from the surface of the piston rod during its movement. It is always placed as the last element on the front cover of the linear hydraulic motor. Primarily, it must prevent the entrance of impurities from the external environment into the system during the retraction of the piston rod. In this case, it is a single-acting element, where the ring has one wiper edge in the outer part, as is shown in Fig. 6.12 (left). In some applications (especially for operation in a dusty environment), it is advisable to use a double-acting wiper. The latter is different in shape, with two wiper edges (inner and outer); see Fig. 6.12 (right). The inner edge is used to wipe the lubricating film from the piston rod surface as the piston rod is extended.

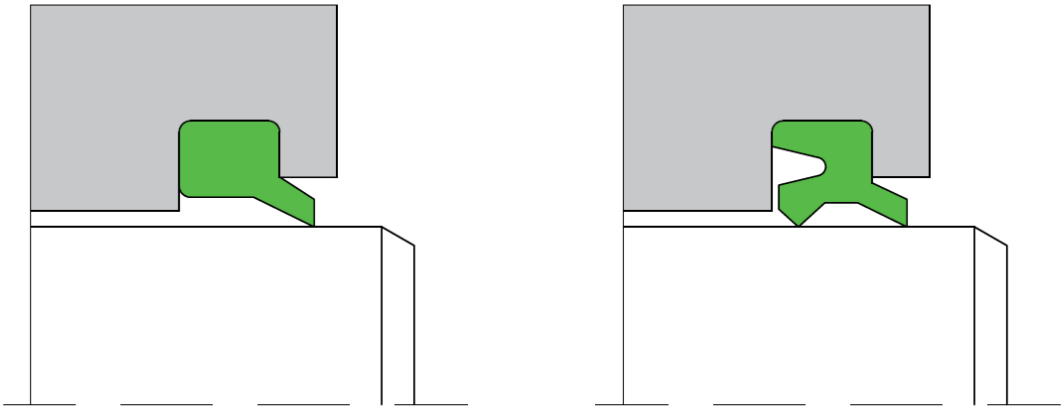


Fig. 6.12 Wiper, single-acting (left), double-acting (right)

The wipers are available in many shapes. For lighter conditions, they can be made of NBR, but usually, a material with higher hardness and wear resistance is used (PU, TPE, PTFE and bronze together with O-ring). Therefore, they can also be used in a combined design (see Fig. 6.9 (right)). In some cases, they are produced with metal reinforcement for pressing into a slot [55], [56].

An example of the use of all sealing elements in a linear hydraulic motor is shown in Fig. 6.13.

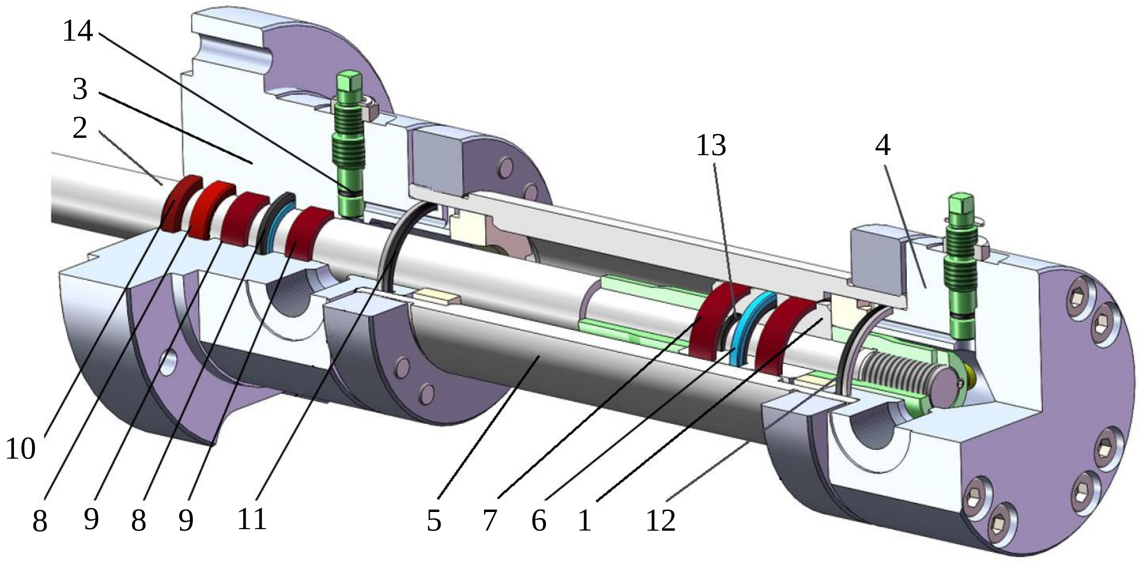


Fig. 6.13 Example use of sealing elements in linear hydraulic motor [57]

1 – piston, 2 – piston rod, 3 – front cover, 4 – back cover, 5 – motor body, 6 – piston movement seal, 7 – piston guide belts, 8 – movement seal of piston rod, 9 – guide belts of piston rod, 10 - wiper, 11 – immovable (stationary) seal of front cover, 12 – immovable (stationary) seal of back cover, 13 – immovable (stationary) seal of piston and piston rod, 14 – O-ring

Seal lifetime

The seal lifetime is limited. In addition to sudden defects such as pinching of the O-ring during insertion into the sealing gap, abrasive damage due to impurities in the liquid, or puncture of the sleeve by pressure, gradual wear of the seal cannot be completely avoided. This results in an increase in leakages and can lead to a rupture of the seal.

Wear of sealing elements increases with pressure size, pressure pulsations and sudden pressure changes. For movement seals, the greatest wear generally occurs during motor start-up and run-out. This is aggravated by using sealing elements made of soft materials that can adhere to the sealing surface, where the necessary lubricating film is not ensured. It is important to maintain the required quality of the sealing surfaces (surface roughness of the contact surfaces of the piston and piston rod with the cylinder body). With a new seal, there may be a limiting friction at which the wear of the sealing elements is several times greater. Temperature also has a significant effect on wear. Low temperatures cause the hardening of rubber materials, which can lead to increased leakages. On the other hand, elastomers soften at high temperatures, and their properties and sealing effect are changed. Undissolved air in the liquid also affects the lifetime, where air bubbles expand and distort the surface of the sealing elements as they are propagated through the sealed gap.

In order to ensure the required lifetime of the sealing elements and therefore the reliability of hydraulic equipment, it is necessary to select the correct seal type and material for the given application, to ensure correct installation and to observe the operating parameters.

7. Elements for flow limiting and flow direction control

The main function of these elements is to limit the liquid flow or to specify the direction of the liquid flow. Elements from this group are part of most hydraulic systems. Depending on the construction design, these elements can be seat or spool. In the case of seat valves, depending on the shape of the sealing element, there are versions with a ball, cone, needle, or plate. In the spool valve design, a cylindrical or flat spool valve is usually used. The movement of the spool in the valve body can be linear or rotary. The basic construction elements of valves for flow control and flow direction control are shown in Fig. 7.1 [1], [3].

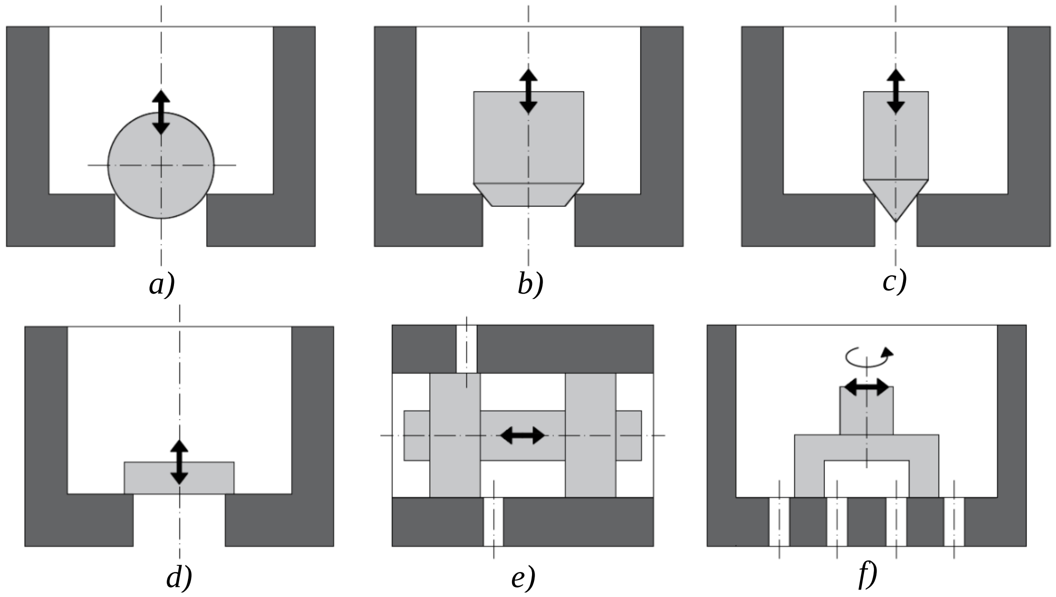


Fig. 7.1 Basic construction elements of valves for flow control and flow direction control

a) seat construction with ball, b) seat construction with cone, c) seat construction with needle, d) seat construction with plate, e) spool construction – cylindrical spool with linear motion, f) spool construction – flat spool with linear or rotary motion

These are valves that are in the product range of practically every company producing elements of hydraulic circuits. It is possible to mention the manufacturers Bosch Rexroth, Hydac, Danfoss, Argo Hytos, Parker, Eaton, HAWE Hydraulik, whose technical data, and parameters were used in the creation of this chapter.

7.1 Check valves

Non-return valves are also commonly referred to as check valves. They allow the liquid flow in only one direction, in which usually minimum resistance to the liquid flow is required. In the second direction, the check valves must be perfectly tight. These are purely seat construction elements, where check valves with a ball or cone are most commonly used, depending on the shape of the sealing element. The tightness of these valves is ensured by the seat construction and the back pressure of the liquid or spring. In the case of the valve without a spring, the mounting position must be vertical, with the seat located at the bottom part. The graphical symbol of the check valve without spring is shown in Fig. 7.2 (top left). If the valve with spring is used, the mounting position is arbitrary. The spring has only a minimum preload in the standard valve design, and the opening pressure is in the range of (0.05 ÷ 0.5) MPa. Graphical symbol of the check valve with spring is shown in Fig. 7.2 (bottom left).

The check valve with a ball is only used for small volumetric flows and clearances up to approx. 8 mm (ball valves are sensitive to oscillations at higher clearances). For higher volumetric flows and clearances, valves with a cone are used. The cone has a better sealing ability and, at the same clearance, has a lower weight compared to the ball. The weight has an effect on the natural frequency, when compared to the cone, the weight of the ball would increase at the same clearance. Thus the natural frequency of the ball would decrease, and there would be a risk of its oscillation. The design of the cone is lightened, and the shape of the cone allows better guidance in the valve body. An example of a check valve with a cone is shown in Fig. 7.2 (right). The cone (1) is compressed into the valve seat (2) by the spring (3). The cone is guided in the valve body (4). Free liquid flow through the valve is only possible in the direction from left to right (A→B). The liquid pressure delivered to input A acts on the cone face, and after overcoming the spring resistance, the cone moves to the right. This allows the liquid flow through the valve seat and the holes in the cone body (5). In the opposite direction (B→A) the liquid flow is impossible. The liquid delivered to input B pushes the cone towards the seat, and the sealing effect of the cone in the seat only increases with the increasing liquid pressure [5], [11].

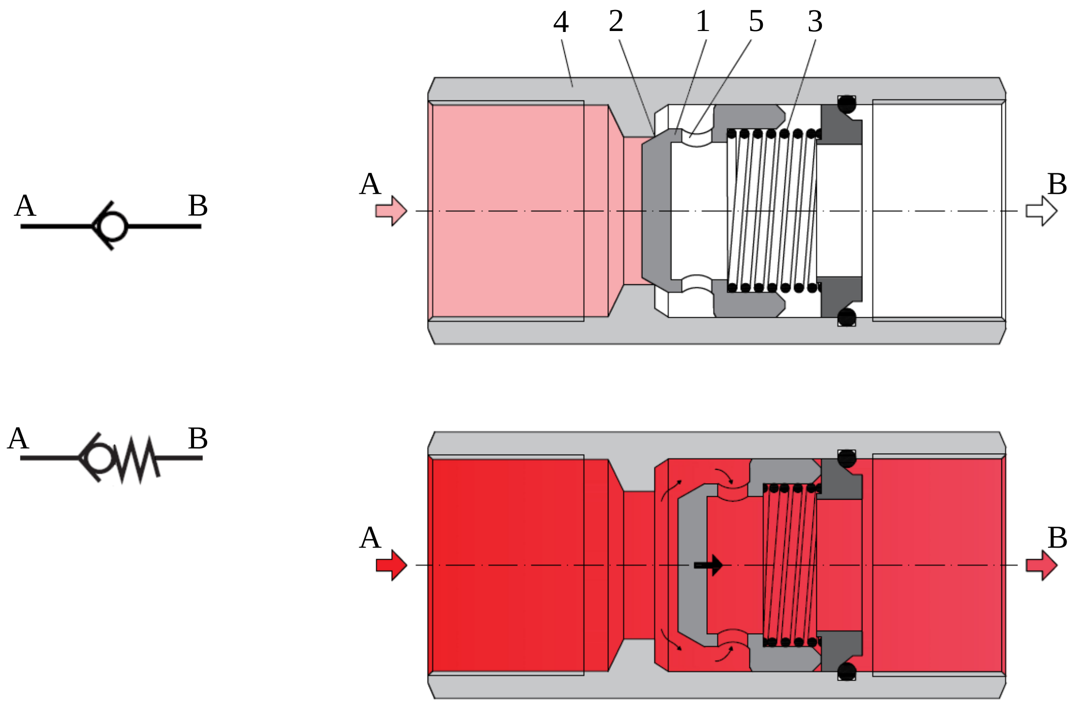


Fig. 7.2 Check valve, graphical symbol of check valve without spring (top left), graphical symbol of check valve with spring (bottom left), an example of a check valve with cone and spring (right)

1 – cone, 2 – seat, 3 – spring, 4 – valve body, 5 – holes in cone body

An example of Δp - Q characteristic of a check valve, where Δp is the pressure gradient on the valve and Q is the volumetric flow through the valve, is shown in Fig. 7.3. The individual courses of the characteristic correspond to different spring preload - different opening pressures of the valve.

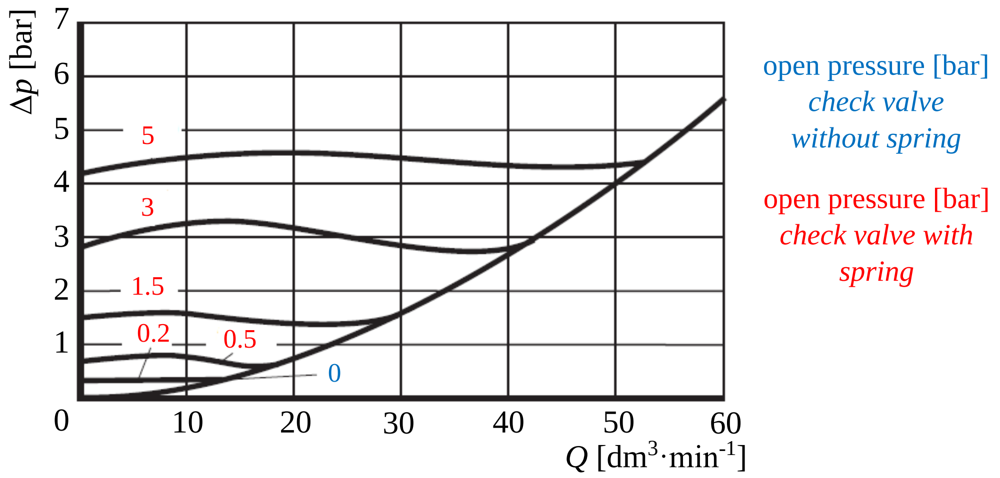


Fig. 7.3 Δp – Q characteristic of check valve

Check valves are used in most hydraulic systems and, in some cases, can be directly integrated into other components. Selected examples of their use in a hydraulic circuit are shown in Fig. 7.4. Two hydraulic pumps and a linear hydraulic motor (1) are placed in the circuit. The hydraulic pump (2) is a high pressure hydraulic pump with a smaller geometric stroke volume. The hydraulic pump (3) has a higher geometric volume but only works with lower pressures. While the piston rod of the hydraulic motor is extended without any external resistance (rapid displacement), both hydraulic pumps supply the volumetric flow to the hydraulic motor together via the check valve (5). When the resistance on the piston rod increases (e.g., during pressing), the hydraulic pump (3) is unloaded by means of the relief valve (9), and the check valve (5) protects the low-pressure hydraulic pump (3) against high pressure from the branch of the hydraulic pump (2). Often, the check valve is connected in parallel to another element, thus putting that element out of action when the liquid flows in the opposite direction. In the mentioned circuit. It is possible to control the piston rod extension velocity by means of the throttle valve (10). The velocity control is not needed for the reverse movement (retraction) of the piston rod. The throttle valve is taken out of operation by bypassing through the check valve (6). The circuit uses a two-stage electrically controlled directional valve (4). The check valve (7) is connected in parallel to the return line filter (11). The spring of the check valve is preloaded to the maximum allowable pressure drop across the filter. The check valve (7) thus prevents the filter element from rupturing during its clogging.

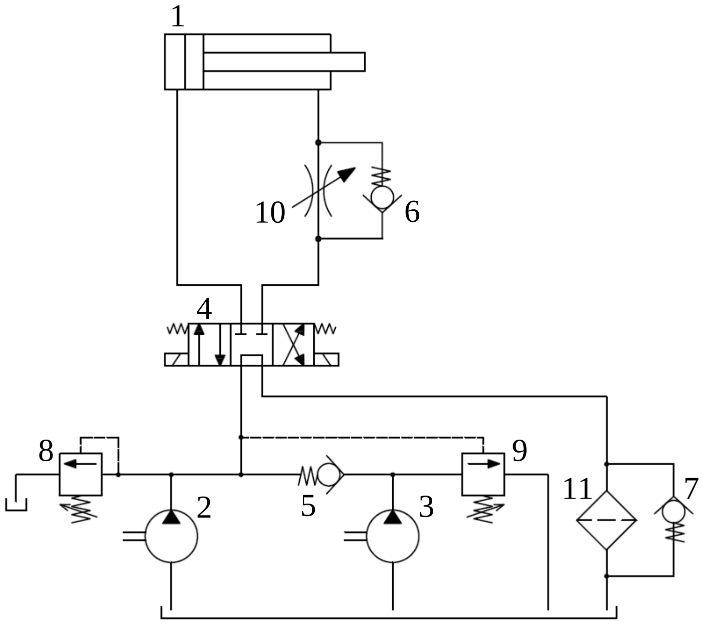


Fig. 7.4 Use examples of check valves

1 – linear hydraulic motor, 2 – high pressure hydraulic pump with smaller geometric stroke volume and higher pressure, 3 – hydraulic pump with higher geometric stroke volume and smaller pressure, 4 – directional valve, 5 ÷ 7 – check valves, 8 and 9 – relief valves, 10 – throttle valve, 11 – return line filter

Check valves are commonly manufactured for volumetric flows up to 4000 $dm^3 \cdot min^{-1}$ and working pressures up to 45 MPa.

7.2 Pilot operated check valve

It is a check valve of seat design, which can be direct, or pilot operated. The possibilities of graphic symbols are shown in Fig. 7.6. A piloted operated check valve (also known as single-acting hydraulic lock) also allows free flow in only one direction (A→B), but in contrast to the conventional check valve, it is equipped with a control input channel X. When an external pressure signal is applied to the X input of the valve, the liquid through the valve is also allowed in the opposite direction (B→A).

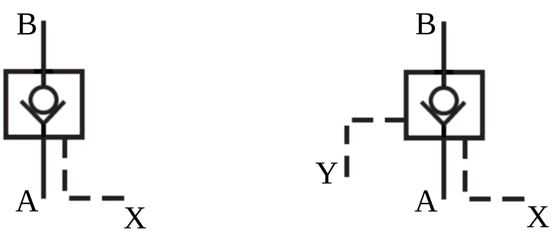


Fig. 7.5 Pilot operated check valve, graphical symbol for valve with internal drain port of the control piston (left), graphical symbol for valve with external drain port of the control piston (right)

The design and function principle of the pilot operated check valve are explained using a simplified section in Fig. 7.6 (left) [5], [6]. The cone (1) is compressed by the spring (3) into the valve seat (2). If the valve is in the function of the check valve, it allows the liquid flow only in the direction A→B. By feeding the liquid to the A input, the cone is moved upwards after overcoming the spring resistance. In this way, inputs A and B are connected. The liquid from the space above the cone is pushed out through the drilled holes (5) in the cone Fig. 7.6 (right).

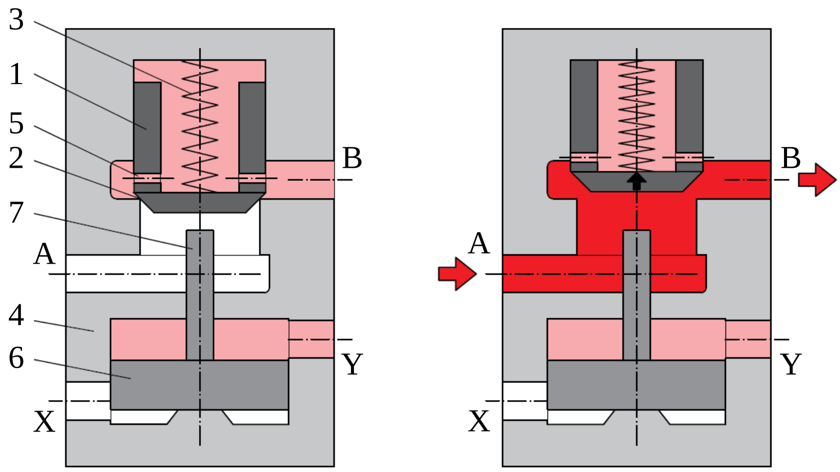


Fig. 7.6 Function principle of pilot operated check valve (A→B)

1 – cone, 2 – seat, 3 – spring, 4 – valve body, 5 – holes in cone body, 6 – control piston, 7 – pin of the control piston

The liquid flow through the valve in the opposite direction (B→A) is impossible because after the liquid supply to inlet B, the cone is pushed into the seat by the spring and the liquid pressure Fig. 7.7 (left). In the lower part of the operated check valve, there is the control piston (6), which is connected to the pin (7), which is brought out against the cone. In this case, the leakage flow of the control piston is externally discharged through the Y outlet, which corresponds to the graphic symbol shown in Fig. 7.5 (right). If it is necessary to enable the flow through the valve in the direction (B→A), an external pressure signal (control pressure) is applied to the piston area A_p through the X input Fig. 7.7 (right). By acting the control pressure p_X on the piston area, the piston is moved upwards, and the force required to move the cone out of the seat is generated. The force is transmitted from the piston to the cone by means of the pin.

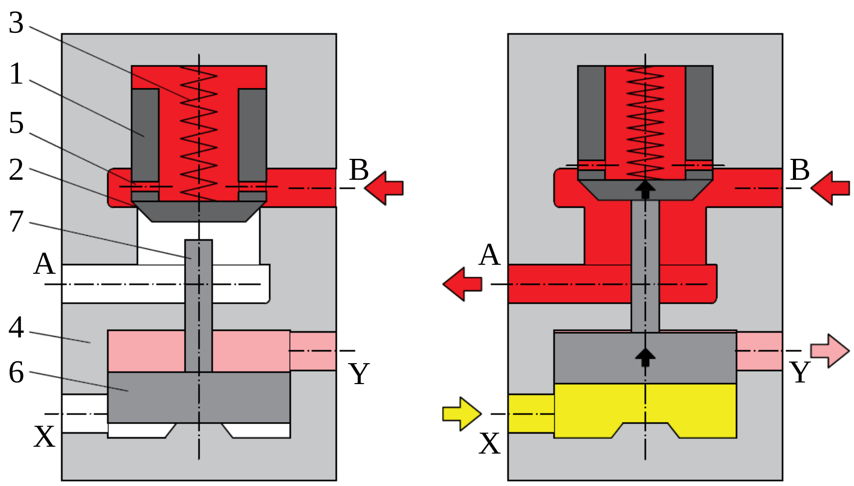


Fig. 7.7 Function principle of pilot operated check valve (B→A)

1 – cone, 2 – seat, 3 – spring, 4 – valve body, 5 – holes in cone body, 6 – control piston, 7 – pin of the control piston

To determine the magnitude of the required control pressure p_X, it is necessary to determine the forces acting on the cone. We will consider that there is the liquid pressure p_B in the space above the cone. In the space below the cone is the liquid pressure p_A, ate cone area A_c and the pin area A_{pin} Fig. 7.8.

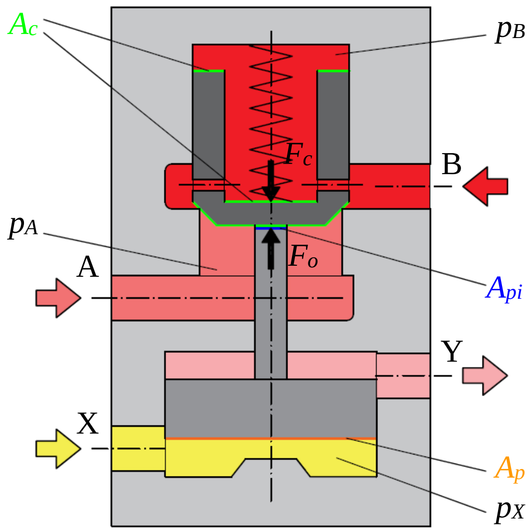


Fig. 7.8 Determine the magnitude of the required control pressure p_X

Then, the closing force acting on the cone F_c consists of the pressure force from the liquid above the cone and the spring force F_s:

$$F_c = p_B \cdot A_c + F_s \, . \tag{7.1}$$

The opening force F_o consists of the pressure force from the control pressure below the control piston and the pressure force from the fluid in the A channel below the cone:

$$F_o = p_X \cdot A_p + p_A \cdot (A_c - A_{pin}). \tag{7.2}$$

To pick up the cone, it is obvious that $F_o > F_c$. From the balance of forces on the pilot operated check valve, it is possible to determine the minimum control pressure p_X:

$$p_X \cdot A_p + p_A \cdot (A_c - A_{pin}) = p_B \cdot A_c + F_s\,,$$

$$p_X \cdot A_p = p_B \cdot A_c + p_A \cdot (A_{pin} - A_c) + F_s\,,$$

$$p_X = \frac{p_B \cdot A_c + p_A \cdot (A_{pin} - A_c) + F_s}{A_p}. \tag{7.3}$$

If input A is without pressure, the minimum control pressure is calculated according to:

$$p_X = \frac{p_B \cdot A_c + F_s}{A_p}. \tag{7.4}$$

A design example of a direct pilot operated check valve with internal drain port of the control piston is shown in Fig. 7.9 [30]. Pilot operated check valves are used for lower volumetric flow rates up to approx. 300 $dm^3 \cdot min^{-1}$ and pressures up to 31.5 MPa. The ratio of the areas of the cone and the control piston is usually $A_c : A_p = 1{:}2$.

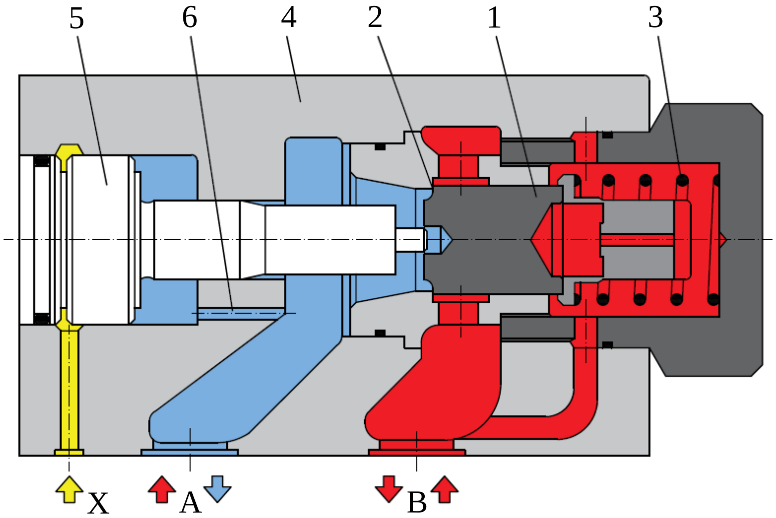


Fig. 7.9 Direct pilot operated check valve with internal drain port of the control piston

1 – cone, 2 – seat, 3 – spring, 4 – valve body, 5 – control piston, 6 – internal drain port of the control piston

For higher pressures (45 MPa) and higher flow rates (up to 6,400 $dm^3 \cdot min^{-1}$) indirectly pilot operated check valves are used. An example of the construction is shown in Fig. 7.10 [11]. A hole is drilled in the head of the main cone (1), into which a smaller auxiliary cone (ball) (3) is inserted. By acting the control pressure signal from the X input on the surface area of the control piston (6), the auxiliary cone is firstly extended (opened). The liquid begins to flow from the space above the auxiliary cone to output A. This will gradually decrease the pressure at input B (and, therefore, the closing pressure of the liquid acting on the surface of the main cone) and decrease the required opening control pressure. The force of the control piston overcomes the resistance of the main cone, pushes it out of the seat (5) and allows free liquid flow through the valve in the direction (B→A). The required size of the control pressure is primarily based on the ratio of the areas of the control piston and the auxiliary cone, which in this case is significantly higher compared to the direct operated check valves.

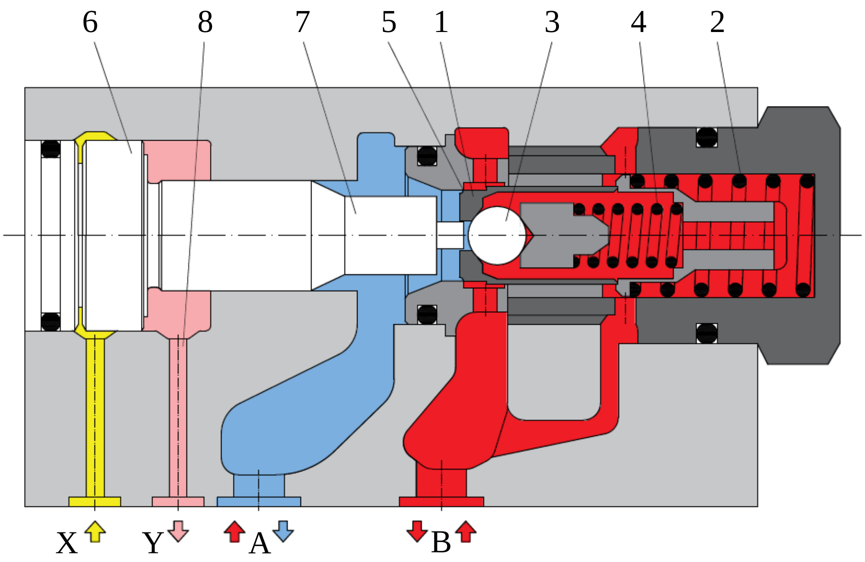


Fig. 7.10 Indirectly pilot operated check valve with external drain port of the control piston

1 – main cone, 2 – spring of main cone, 3 – auxiliary cone (ball), 4 – spring of auxiliary cone, 5 – valve seat, 6 – control piston, 7 – pin of control piston, 8 – external drain port of the control piston

Pilot operated check valves are used in applications with linear hydraulic motors in cases where the piston rod is loaded by an external force even when the hydraulic motor is inactive, and it is necessary to ensure its constant position (typical examples are various types of lifting equipment). In these cases, spool directional valves cannot be used to fix the position of the piston rod of the hydraulic motor because the motor could be subject to jerky movement due to leakages on the directional valve. Examples of the use of the pilot operated check valve are shown in Fig. 7.11. The circuit with double-acting linear hydraulic motor (1) is shown in Fig. 7.11 (left). The piston rod of the hydraulic motor is loaded by a load (tensile force). The lowering velocity of the load can be controlled by means of the throttle valve (2). To control the movement direction of the piston rod, a three-position directional valve (3) is located in the circuit. When the directional valve is in the middle position, the piston rod of the hydraulic motor does not move, which is ensured by the pilot operated check valve (4). For the load lifting (piston rod retraction), it is necessary to activate the right position of the directional valve. The liquid pressure overcomes the cone resistance of the pilot operated check valve, and the liquid flows freely into the annulus space of the linear hydraulic motor (through the check valve (5)). For the load lowering, the directional valve is repositioned to the left position. The liquid from the hydraulic pump is supplied to the piston area of the hydraulic motor, the X control input of the pilot operated check valve to this line is connected. The pressure control signal applied to the control piston of the pilot operated check valve causes the cone to lift from the valve seat (4). This allows the liquid flow from the hydraulic motor's annulus area through the throttle valve, the pilot operated check valve and the directional valve, to the tank. In this case, the pilot operated check valve can be used with an internal drain port of the control piston.

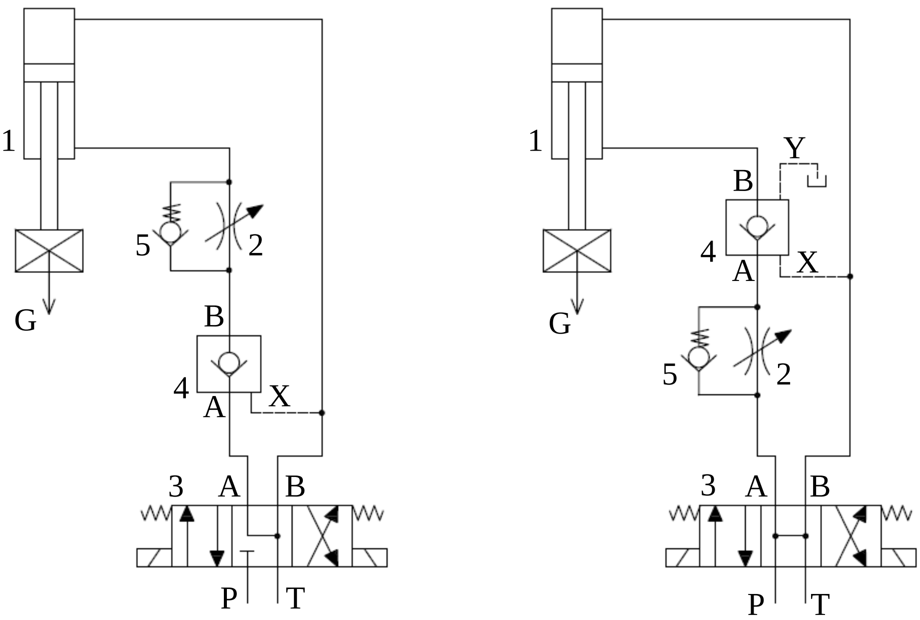


Fig. 7.11 Use examples of pilot operated check valve

1 – linear hydraulic motor, 2 – throttle valve, 3 – directional valve, 4 – pilot operated check valve, 5 – check valve

A similar circuit with a linear hydraulic motor is shown in Fig. 7.11 (right), except that the throttle valve is located between the pilot operated check valve (4) and the directional valve (3). The function of the circuit is practically identical. In this case, when lowering the load of the hydraulic motor, the opening of the pilot operated check valve in the flow direction from B → A is influenced by the setting of the throttle valve (and the size of its resistance). In this arrangement, the pilot operated check valve with the external Y drain port of the control piston must be used. In the two circuits shown in Fig. 7.11, a directional valve was used which in the middle position connects the outputs A and B to channel T. This is necessary to ensure the correct and safe operation of the pilot operated check valve.

The pilot operated check valve with the internal connection of input A to the pin side of the control piston can be used when there is no pressure load on input A. If input A is pressure loaded, the design with the external drain port Y is used.

If it is necessary to ensure a constant position of the piston rod for both movement directions during the inactivity of a linear hydraulic motor, a **double pilot operated check valve** is used. The double pilot operated check valve is an element which consists of two pilot operated check valves (it is also sometimes referred to as the double check valve). Check valves can be direct operated at lower operating flows and pressures. These valves are more sensitive to oscillations, especially in circuits with throttle valves. A solution to this may be the use of the pilot operated check valves with an auxiliary cone. A simplified graphical symbol of the double pilot operated check valve is shown in Fig. 7.12 (top left); the detailed version of this symbol is shown (bottom left) [11].

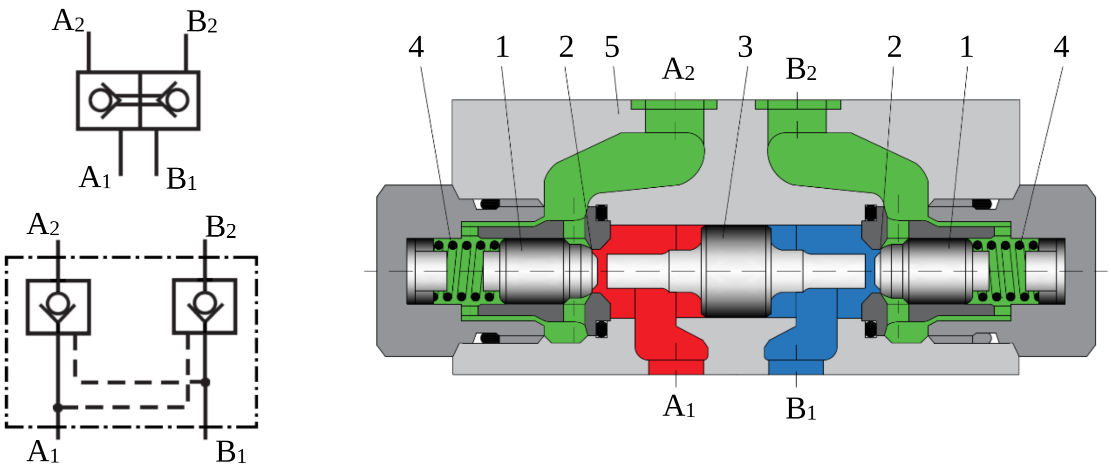


Fig. 7.12 Double pilot operated check valve, simplified graphical symbol (top left), detailed graphical symbol (bottom left), an example of design solution of double pilot operated check valve (right)

1 – cone, 2 – seat, 3 – control piston, 4 – spring, 5 – valve body

An example of a construction of the double pilot operated check valve is shown in Fig. 7.12 (right). The movable control piston (3) is placed between the two operated check valves. The movement direction and the generation of the control (opening) force of the piston depend on the pressure signal fed from input A1 or B1. The double pilot operated check valve is used especially in circuits with linear hydraulic motors. As its name indicates, it serves to "lock" the piston rod position of the hydraulic motor, which can be loaded by an external force in both directions.

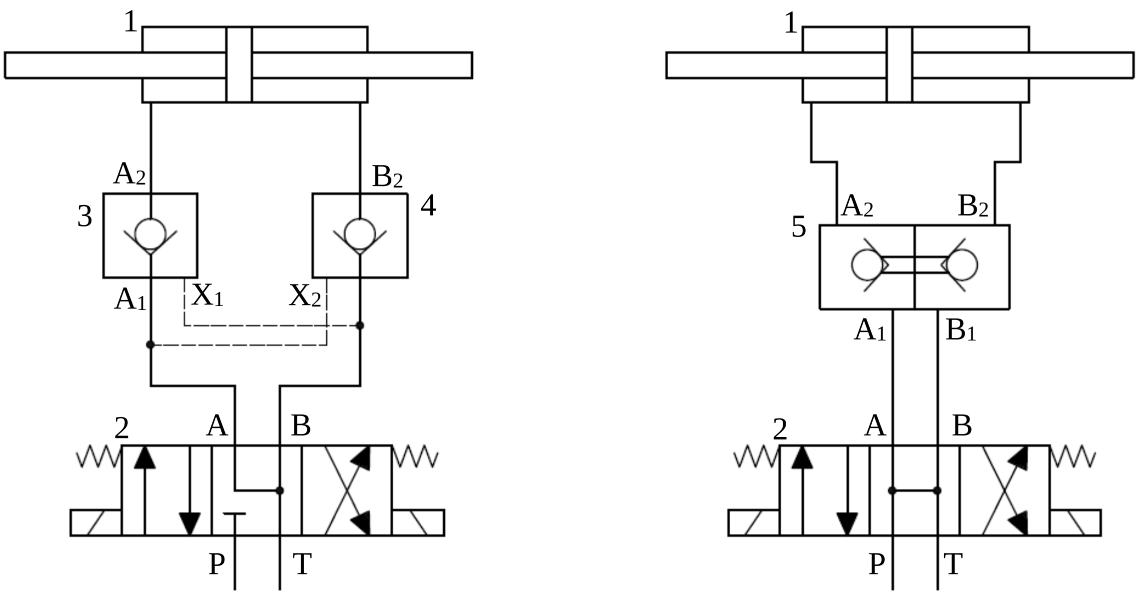


Fig. 7.13 Use an example of double pilot operated check valve in circuit

1 – double rod hydraulic cylinder, 2 – directional valve, 3 and 4 – pilot operated check valves of double pilot operated check valve, 5 - double check valve

The function of the double pilot operated check valve can be explained using the circuit shown in Fig. 7.13 (left). In the circuit, there is placed the linear double-rod hydraulic cylinder (1), which can be loaded in both directions by external forces. The control of the movement direction of the hydraulic motor is ensured by means of the directional valve (2). In order to lock the piston position in the body of the hydraulic motor, the double pilot operated check valve, which consists of two pilot operated check valves (3) and (4), is used in the circuit. While the middle position of the directional valve is active, the liquid flow through the check valves in the direction (A2 → A1) and (B2 → B1) is closed without leakages and the piston of the hydraulic motor remains at rest.

To move the hydraulic motor piston to the right, the directional valve must be moved to the left position. The liquid from the hydraulic pump flows through the pilot operated check valve (3) in the direction (A1 → A2) to the left input of the hydraulic motor. At the same time, a pressure control signal is fed to the control input X2 of the pilot operated check valve (4), which enables the liquid flow through this valve in the direction (B2 → B1). To move the piston of the hydraulic motor to the left, it is necessary to activate the right position of the directional valve. The function is similar, the liquid flows through the pilot operated check valve (4) in the direction (B1 → B2), the pressure control signal is fed to the control input X1, thus allowing the liquid flow through the valve (3) in the direction (A2 → A1).

The double pilot operated check valve is usually placed between throttle valves and a directional valve. In order to ensure the reliable operation of the double pilot operated check valve, a directional valve must be used below the double pilot operated check valve, which has outputs A and B connected to the T-channel in the middle position. This means that if the piston rod of the hydraulic motor is locked in a given position, there must be no pressure signal at the inputs X1 and X2 of the double pilot operated check valve. The middle position of the directional valve will ensure that in the rest position of the piston rod of the hydraulic motor the liquid is drained from these inputs into the tank.

7.3 Shut-off valves

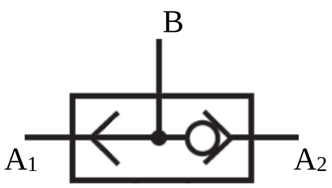
Shut-off valves are designed to limit the liquid flow. An absolute tightness in both directions is required in the closed state. In the open state, the valves ensure minimum resistance to liquid flow. In hydraulic systems, they are located in suction lines, are a part of safety and shut-off blocks of accumulators, and can be used to temporarily limit the function (out of service) of a circuit part, or,, e.g., as drain valves of tanks. Ball valves (ball cocks) (see Fig. 7.14 (right)), which are usually manually operated, are the most commonly used shut-off valves.

Fig. 7.14 Shut-off valve, a graphic symbol of shut-off valve (left), ball valve design [11] (right)

7.4 Logic (shuttle) valve

These are seat valves with two inputs A1 and A2, and one output B. They represent the logical function “or”. As is shown in Fig. 7.15 (right), A ball or cone moves in the valve body, which connects the A1→B or A2→B ways depending on the size of the input pressures. They are used,, e.g., as shuttle valves for sectional directional valves with Load Sensing system in mobile hydraulics. An example of the use of a logic valve is shown in Fig. 8.16.

A similar function to these logic valves can also be obtained by using two check valves.



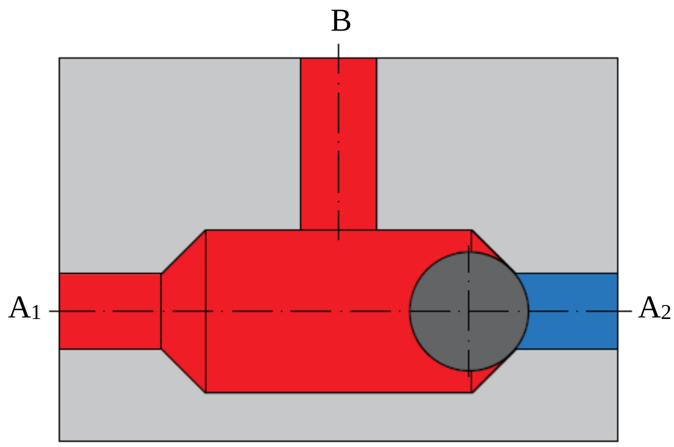


Fig. 7.15 Logic valve with the function „or“, graphical symbol of a valve (left), a simplified section of a valve (right)

7.5 Directional valves

Directional valves are used to control the flow direction or to limit the liquid flow. Most often, the motion direction of the hydraulic motor is controlled by means of the directional valves. Each directional valve has two or more functional positions. The individual positions of the directional valve represent different connections of internal ways (inputs and outputs) allowing the liquid flow. The position change of the directional valve is achieved by an external force and depends on the selected control type. Each directional valve can be characterized by:

- according to number of working positions – with 2, 3 or more spool positions,
- according to number of ways – with 2, 3 or more spool service ports,
- according to operational modes – with mechanical, electrical, hydraulic, or pneumatic control, or a combination of individual methods (serial, parallel, or serial-parallel control),

An example of graphical symbols of hydraulic (spool) directional valves is shown in Fig. 7.16. The number of positions of the directional valve is given by the number of square areas in the graphical symbol. In each position, the corresponding connection of the ways is shown.

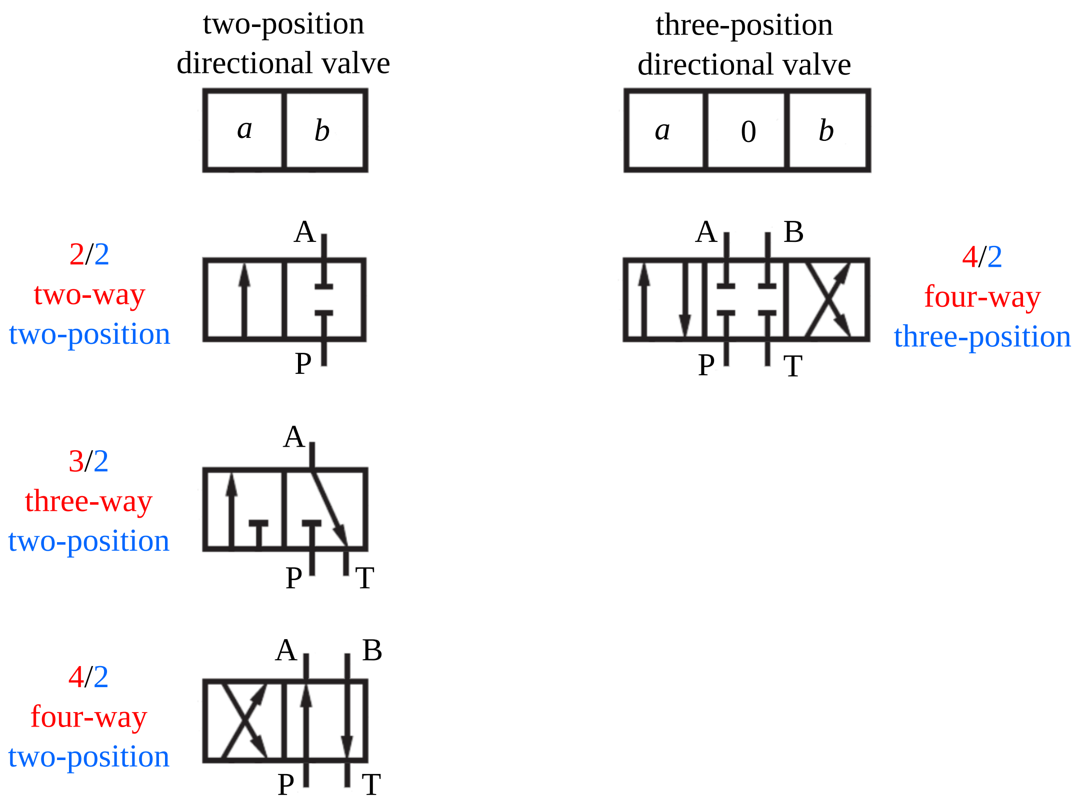


Fig. 7.16 Examples of graphical symbols of directional valves

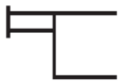
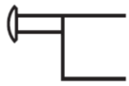
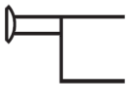
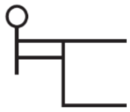
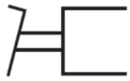
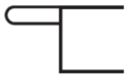
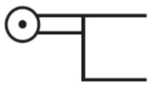
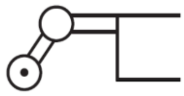
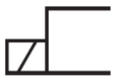
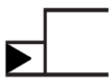
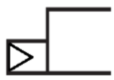
For the directional valves, the shortened designation is subsequently used using two numbers is separated by a slash. The first number indicates the number of ways and the second number the number of positions of the directional valve. For example, 3/2 means a three-way two-position directional valve, 4/3 is a four-way three-position directional valve, etc. Hydraulic directional valves use capital letters to indicate the ways. At the bottom of the symbol, the letter P indicates the input connected to the pressure source (usually the connection from the hydraulic pump). The letter T is the input connected to the low-pressure line (most often the connection to the tank). Other inputs would be designated by the letters R, S, U, etc. At the top of the symbol, the outputs are designated by the letters A and B (usually the connection to the hydraulic motor). In the case of a multi-way directional valve, the other inputs would be designated C, D, etc.

The designation of ways of the directional valve is specified in hydraulic circuits only for one position - the so-called neutral position or for the position that is active after the system starting.

In some cases, small letters are used to indicate the positions of the directional valve. The designation is according to the alphabet a, b, etc., in the order of the positions from left to right. For directional valves with more than two positions, the middle (neutral) position is designated by the symbol 0. An example of the position designation is shown in Fig. 7.16. Such position designation is mainly suitable for the description and the function control of the directional valve.

The term control of the switchgear means how the position change of the directional valve will be obtained. The basic control uses the **mechanical** principle of force transmission to the internal moving part of the directional valve. The control can be manual, usually using a button or lever, foot control using a pedal, or control by pulleys or push rods. Springs are also very often used. They ensure, most often, the return of the directional valve to the basic position. The control of the directional valves is indicated in the graphical symbols on the sides of the end positions. The symbols corresponding to the individual operating modes are shown in Tab 7.1.

Tab 7.1 Basic operating modes of directional valves and their graphical symbols

Mechanical control			
Push button		Push button	
Pull-out knob		Push button/pull-out knob	
Lever		Pedal	
Push rod		Spring	
Roller shaft		Roller lever	
Electrical control			
By electromagnet (1 winding)		By electromagnet (2 windings which act in opposition to each other)	
Hydraulic control		Pneumatic control	
By liquid pressure		By gas pressure	

The **electrical** control is realized by electromagnets, which consist of a core, coil, and armature. Using electrical control, the directional valve can be easily operated from a distance and is also suitable for automatic working cycles. The position change of the directional valve is caused by the armature movement on which the electromagnetic force acts. AC or DC electric current is used to control the electromagnets.

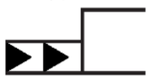
DC electromagnets have a longer lifetime, manufacturers state (40 ÷ 50) million switches. They can remain permanently under current (current flows through them without current peaks) without being damaged. The adjustment time is usually (25 ÷ 60) ms, and the number of switches is up to 15,000 per hour. These electromagnets require a rectifier and are generally more expensive compared to AC electromagnets [3], [5].

AC electromagnets are faster, the adjustment time is usually (10 ÷ 30) ms, but this may not be an advantage for this hydraulic directional valve (pressure peaks may occur in the system during fast adjustment). Current peaks occur in the electromagnets when the directional valve is repositioned. When the movement of the directional valve is blocked, these magnets are quickly overheated and burned. For similar reasons, the switching frequency is also limited (approximately 7500 switches per hour). Therefore, the main advantage is the lower price and the possibility of controlling by current directly from the electrical network without using a rectifier [3], [5].

Electromagnets are usually cooled by air or oil, and depending on the cooling medium used, they are different in their construction. For air-cooled electromagnets, the internal space must be sealed against liquid filling from the directional valve. The seal is subject to wear, which can lead to liquid leakages. Impurities and moisture from the atmosphere can enter the interior of the electromagnet. The advantage of the air-cooled electromagnets is a lower price compared to the oil-cooled construction and the possibility to change the electromagnet during operation without leakage of the working liquid. In the case of oil-cooled electromagnets, the space of the electromagnet is connected to the low-pressure part of the directional valve. In addition to cooling, the oil also ensures lubrication of the internal moving parts and the movement damping of the electromagnet armature - this reduces noise during the repositioning of the directional valve. The chosen cooling method affects the operation, reliability, and lifetime of the electromagnet [3], [5].

Hydraulic and pneumatic controls use fluid pressure to reposition the hydraulic directional valve. In the case of hydraulic directional valves, a combination of several of the above control methods is very often used. Examples of combined controls are shown in Tab 7.2. In the case of parallel control, it is possible to change the position of the directional valve by one or the other operation mode (usually, the second control mode is mechanical using springs or as an emergency control by means of a button).

Tab 7.2 Examples of combined controls of directional valves and their graphical designation

Parallel control			
By push button or electromagnet		By spring or electromagnet	
Serial control			
Two-stage hydraulic control		Two-stage electro-hydraulic control	
	Series-parallel control		
	By spring or electro-hydraulic control		

In the case of serial control, both of these operation modes are simultaneously involved in changing the position of the directional valve. In this case, it is referred to as the two-stage control of the directional valve. A typical example of the two-stage control of the hydraulic directional valves is the electro-hydraulic control. The last possibility is the so-called series-parallel control of directional valves [43].

7.5.1 Spool directional valves

These are the most commonly used directional valves, which are characterized by a relatively simple construction. The body of the directional valve is a ductile iron casting, in which the channels and chambers are pre-cast. There is a movable spool in the body, which ensures the required connection of the ways (channels) of the directional valve. Depending on the spool movement, the directional valve designs are distinguished with a rotary movement of the spool (less common) or with a sliding movement of the spool. An example of a mechanically operated spool directional valve with the sliding movement of the spool and its corresponding graphic symbol are shown in Fig. 7.17 [5], [11]. In the body (4) of the directional valve, there is a movable cylindrical spool (2). The initial neutral position is ensured by a pair of springs (3), which are placed on the spool´s sides and act on its faces. In this case, it is a so-called closed centre directional valve. The P input from the hydraulic pump is closed in the neutral position of the directional valve. Applying a tension or compression on the control lever (1), the spool position is changed, and, therefore, the connection of the ways (channels) of the directional valve is changed.

For example, when the lever is pushed to the right, a tensile force acting on the left spool face will be generated through the control mechanism. The left spring will be compressed, and the spool will move to the left. In this position, the pressure input P will be connected to the output A, and, at the same time, input B to the return line T. This allows the liquid flow through the directional valve in the direction from (P → A), and, at the same time, from (B → T).

When the control lever is pulled to the left, a pressure force will be applied to the spool. The right spring will be compressed, and moving the spool valve to the right will allow the liquid flow through the directional valve in the direction from (P → B), and, at the same, time from (A → T). If no force is acting on the control lever (when it is released), then one of the pairs of springs that has been compressed will move the spool back to the basic central position.

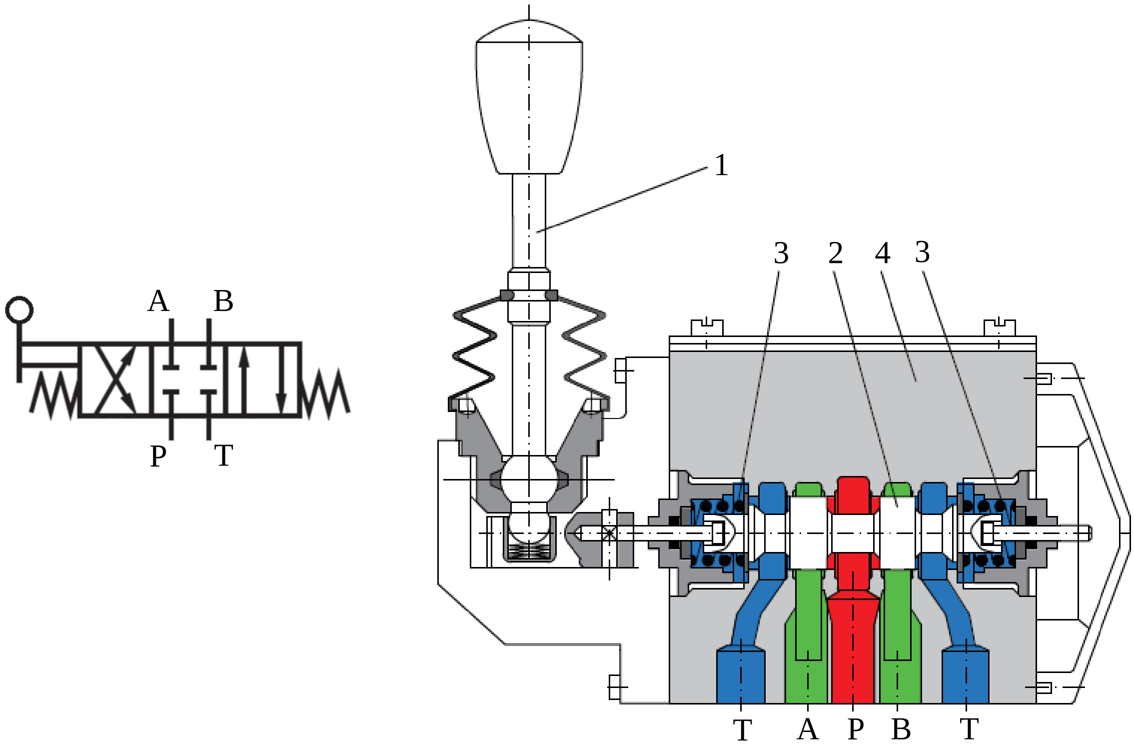


Fig. 7.17 Mechanically (by lever) control spool directional valve with sliding spool movement

1 – control lever, 2 – cylindrical spool, 3 – springs, 4 – body of directional valve

The spool is usually cylindrical, made of low carbon steel, cemented, and hardened on the surface to increase its strength. The spool is the control element in the directional valve, and its movement in the body must be ensured with minimum friction. The clearance between the spool and the body of the directional valve is usually $s = (4 \div 10)$ µm. Such precise placement is necessary to minimize flow losses, so-called leakages in the directional valve. In addition to the clearance between the spool and the body, the pressure gradient between the individual channels of the directional valve also has a major influence on the size of the leakages.

The size of the loss flow Q_l of a directional valve is given by the equation [5]:

$$Q_l = \frac{\pi \cdot d_s \cdot s}{12 \cdot \eta \cdot l} \cdot \Delta p \,, \tag{7.5}$$

where d_s [m] is the spool diameter, s [m] is the gap thickness between the body of the directional valve and the spool, η [Pa·s] is the liquid dynamic viscosity, l [m] is the gap length, and Δp [Pa] is the pressure gradient (pressure difference) causing flow through the gap.

The leakage reduction of the directional spool valve can be achieved by increasing the gap length l, i.e., by a width overlap of the spool with respect to the chamber width. In general, there are three possibilities of spool overlap of the directional valves - positive, zero and negative overlaps; see Fig. 7.18 [20].

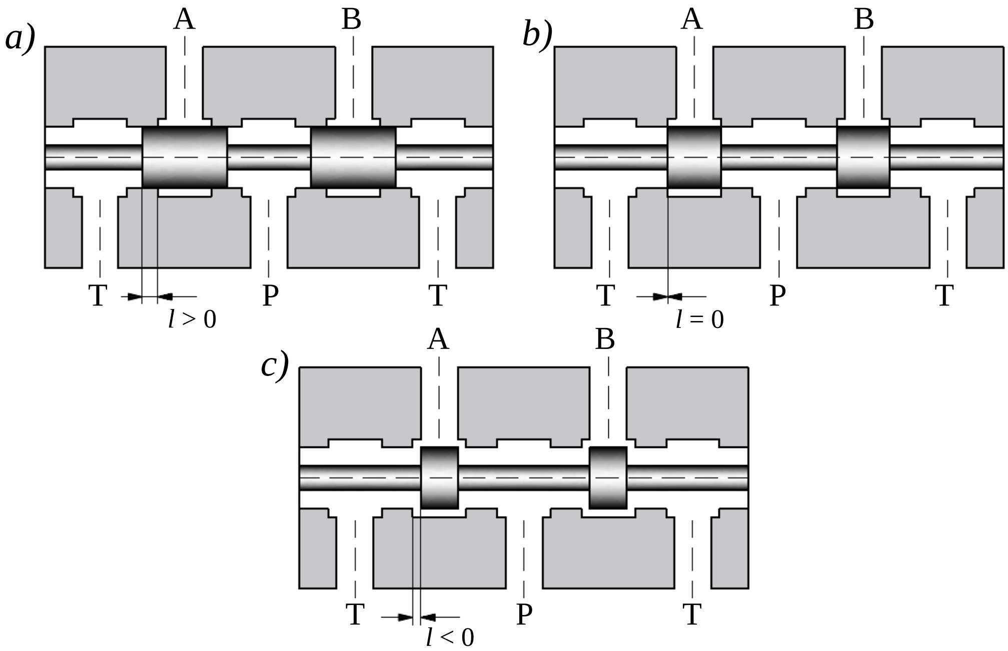


Fig. 7.18 Possibilities of spool overlaps of directional valves

a – positive overlap, b – zero overlap, c – negative overlap

The positive overlap exhibits the lowest leakages in the neutral (middle) position. However, pressure shocks (peaks) may occur when the spool is repositioned. This is due to the greater length of the spool overlapping the slot in the body of the directional valve. This phenomenon can be eliminated by chamfering the spool edges or by triangular slots in the spool control edges. The length of the overlap depends on the size of the directional valve and the spool stroke. The size of the directional valve is defined by its nominal flow rate, and the spool stroke depends on the control type used. For example, in the case of electric control, the spool stroke is limited by the armature stroke of the electromagnet. The positive overlap is usually used in closed centre directional spool valves.

In the case of the zero spool overlap, the pressure shocks (peaks) are not so significant during repositioning the directional valve, but leakages in the neutral position are higher. In addition, this construction solution is more difficult to precision manufacture and is practically only used for proportional valves and servo valves.

The negative overlap has a smaller length of the spool part overlapping the slot in the body of the directional valve. This overlap eliminates pressure peaks when the directional valve is repositioned.

It is obvious from the function principle of spool directional valves and the constructional arrangement of the spool in the body that these directional valves will always have leakages, even if all the ways of the directional valve are closed (as is the case, for example, with the middle position of the directional valve shown in Fig. 7.17).

By simply changing the construction design of the cylindrical spool, inserted into the same body of the directional valve, different connections of the ways of the directional valve can be achieved. The most commonly used spool directional valves and their possible spool shapes are shown in Fig. 7.19.

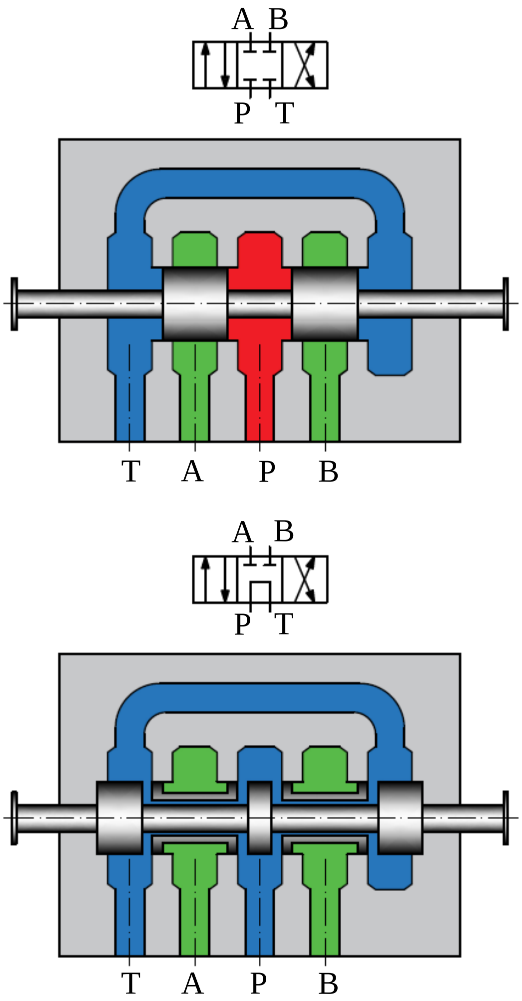


Fig. 7.19 Different spool shapes of directional valves, graphical symbol and spool type „E" (top left), graphical symbol and spool type „J" (top right), graphical symbol and spool type „G" (bottom left), graphical symbol and spool type „H" (bottom right)

The designation of the spool types of directional valves varies among individual manufacturers; however, the established (frequent) designation is by means of capital letters. The symbols and shapes of the "E" and "J" spool types of the closed centre directional valve are shown in Fig. 7.19 above. In the same figure below, the symbols and shapes of the "G" and "H" spool types of the open centre directional valve are shown. In the case of the open centre directional valve, the P input from the hydraulic pump is connected to the return line (T channel) in the neutral position.

There are a large number of possible combinations of internal ways connection of the directional valve. An example from the catalogue sheet of the Bosch Rexroth 4WE6J6X directional valve is shown in Fig. 7.20.

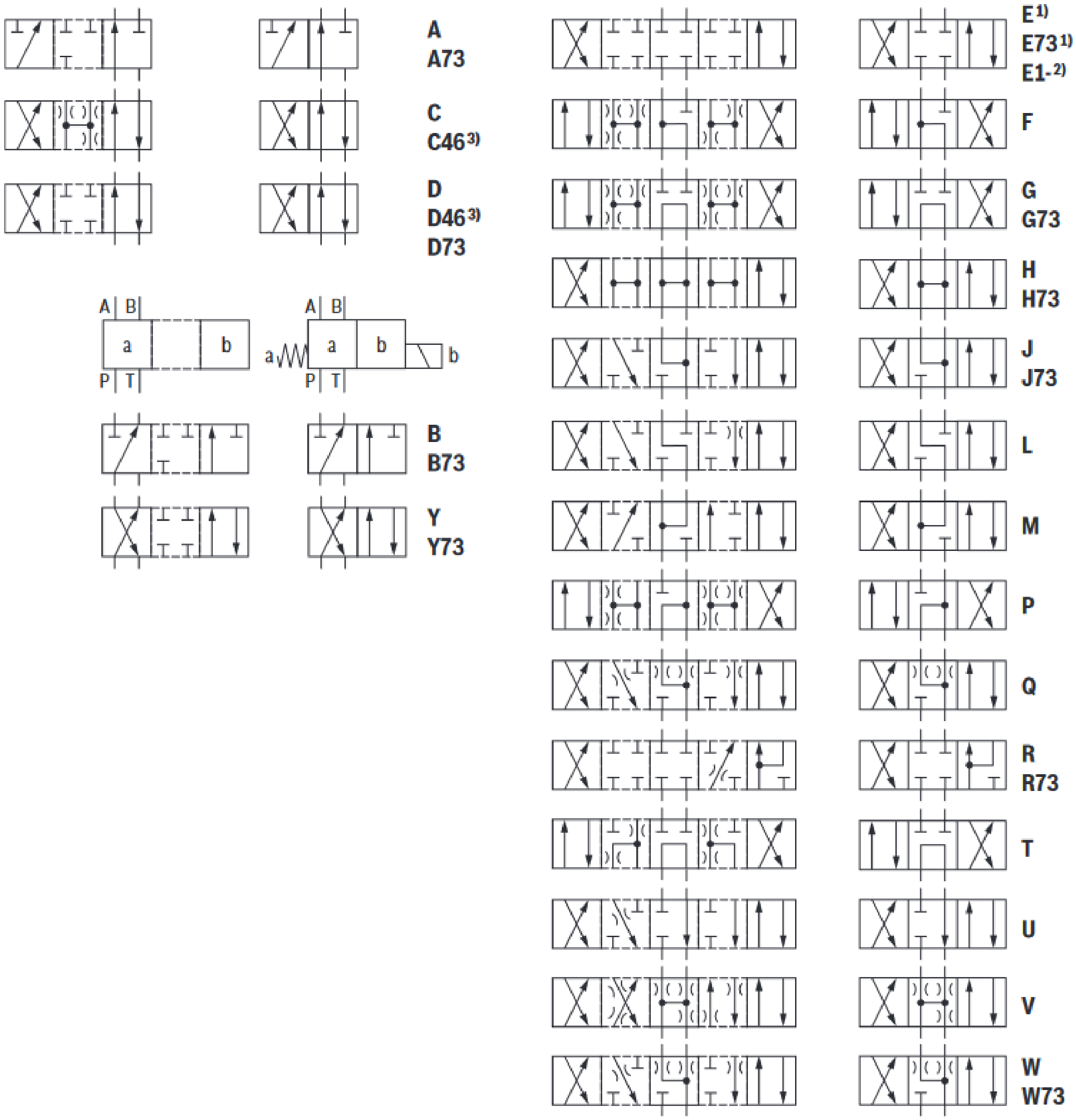


Fig. 7.20 An example of possible versions of directional valve 4WE6J6X from Bosch Rexroth company [58]

Directional spool valves can be direct operated (single-stage) or pilot operated (two-stage). Direct operated directional valves are manufactured up to a clearance of 10 mm for volumetric flow rates up to approx. 160 $dm^3 \cdot min^{-1}$ and pressures of 35 MPa. The control is usually mechanical (manual) or electric by electromagnets. An example of a single-stage electrically operated directional spool valve is shown in Fig. 7.21 [29]. It is a closed centre directional valve. In the basic position, the P input is closed. By feeding a control signal to the left control electromagnet (3), the armature (5) of this electromagnet is extended, and the directional valve is moved to the left position (the spool (2) is moved to the right against the spring (8)). The directional valve will remain in this position as long as the control signal (electric current) on the electromagnet is active. By changing the spool position, the ways of the directional valve will be connected, allowing the liquid flow through the directional valve in the direction (P → B) and simultaneously (A → T).

When the control signal on the electromagnet (3) is interrupted, the spool of the directional valve is moved to the left due to the spring (8) force. The directional valve will return to the middle position.

By feeding the control signal to the right control electromagnet (4), the directional valve is moved to the right position (the spool (2) is moved to the left against the spring (7)). This connects the ways of the directional valve in the direction (P → A) and simultaneously (B → T).

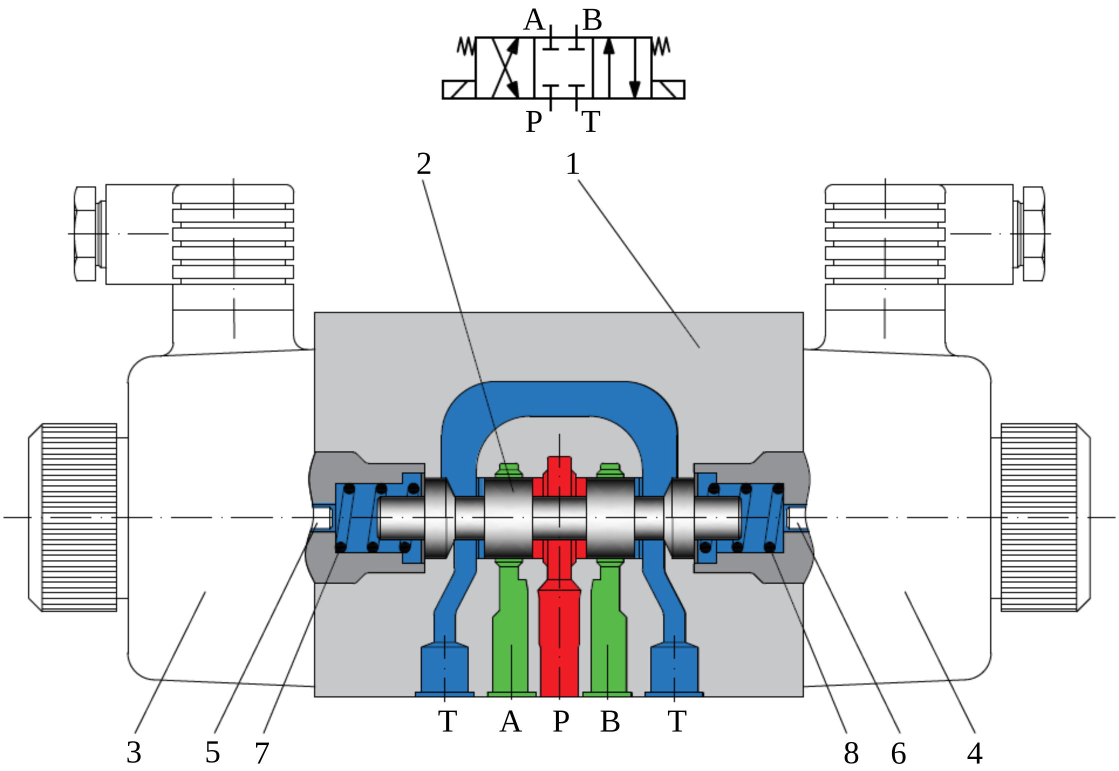


Fig. 7.21 Single-stage 4/3 directional spool valve operated by electromagnets

1 – valve body, 2 – spool, 3 and 4 – control electromagnets, 5 and 6 – electromagnet armature, 7 and 8 – springs

For directional valves with larger diameters, higher adjustment forces are required to change the spool position. In these cases, two-stage pilot operated directional spool valves are used. The first control stage consists of a directional control valve of smaller clearance, whose control is usually mechanical or electrical. The second power stage of the directional valve consists of a higher slide clearance and is hydraulically (or pneumatically) operated. Two-stage directional spool valves are commonly manufactured for volumetric flow rates of 1100 $dm^3 \cdot min^{-1}$. An example of an electro-hydraulically operated two-stage directional spool valve is shown in Fig. 7.22 [11]. It is a closed centre directional valve. The pressure input P is closed in the basic position of the directional power valve.

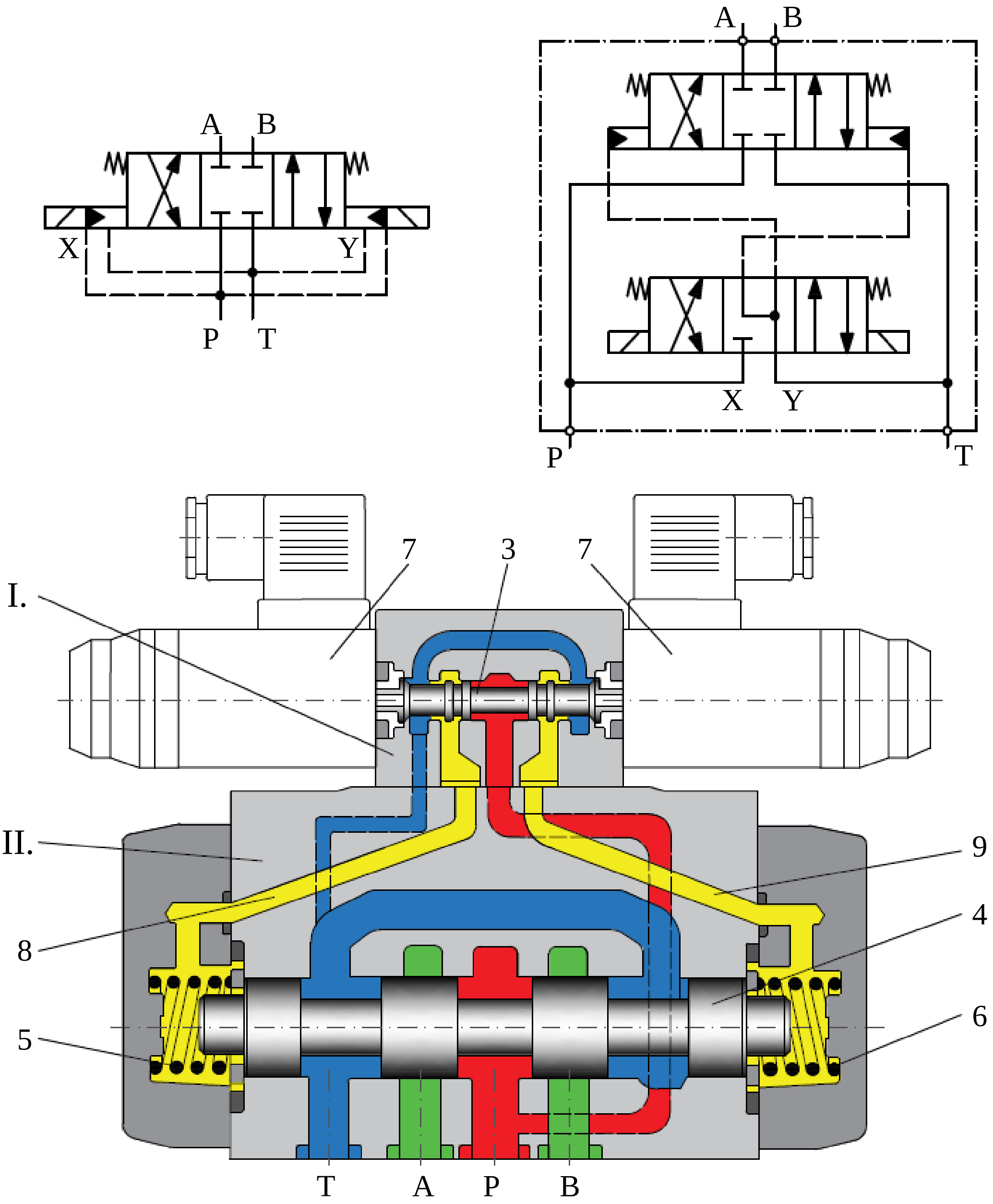


Fig. 7.22 Two-stage (pilot operated) electro-hydraulically operated 4/3 directional valve

I –directional control valve (1st stage), II –directional power valve (2nd stage), 3 – spool of directional control valve, 4 – spool of directional power valve, 5 and 6 – springs of directional power valve, 7 – control electromagnets of control stage, 8 and 9 – hydraulic control of power stage

By feeding a control signal to the left electromagnet (7) of the directional control valve, its spool (3) is moved to the right. This allows the liquid flow through the directional control valve into the space (9) of the hydraulic control of the directional power valve. The liquid pressure acting on the right spool face (4) will ensure that this spool is moved to the left. This changes the connection of the ways of the directional power valve and the liquid can flow through the directional valve in the direction (P → A) and simultaneously (B → T). The detailed circuit diagram of this directional valve is shown in Fig. 7.22 (top right) and its simplified graphical symbol is shown (top left). The pressure input for the pilot valve can be either internal from the P input or external (X input). The drain liquid can be fed internally to T or externally (Y inlet).

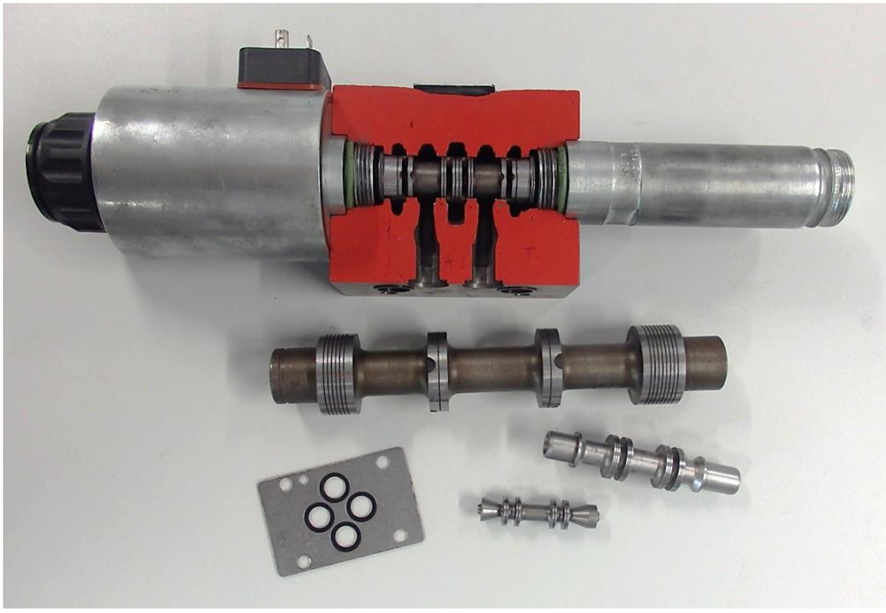
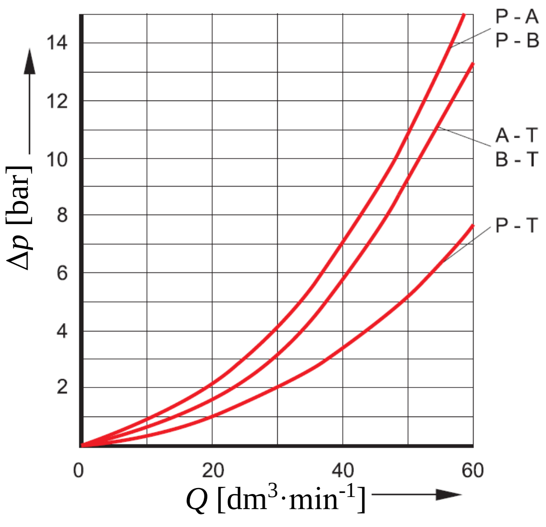


Fig. 7.23 An example of spool directional valve and various types of spool

Static characteristics

Static characteristics of directional valves are commonly specified in manufacturers' product catalogues. Fig. 7.24 (left) shows a static Δp – Q characteristic. This characteristic depends on the liquid viscosity and is determined experimentally, while the individual curves correspond to the different connections of the ways of the directional valve. The pressure gradient Δp represents a pressure drop corresponding to the given volumetric flow through the directional valve. When determining the pressure drop of the directional valve, it is necessary to keep in mind that in most cases, liquids flow through the directional valve in both directions (e.g., from P → A and simultaneously from B → T in a 4/3 directional valve).



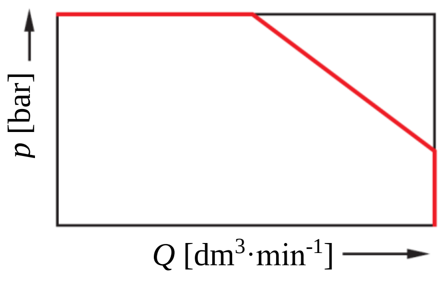


Fig. 7.24 Static Δp – Q characteristic of directional valve (left), static p – Q characteristic of directional valve (right) [11]

The static $p - Q$ characteristic of a directional valve is shown in Fig. 7.24 (right). This is a power characteristic. In this case, the pressure p is the system working pressure. This curve represents the power limitation of working parameters (pressure and flow) of the directional valve. The reasons for the limitation are mainly the pressure and hydrodynamic forces acting on the spool of the directional valve during the liquid flow [5], whose values are already so large that the electromagnet's force cannot overcome them. The power limitation is used for better orientation during the selection of a directional valve.

7.5.2 Directional poppet valves

The second construction group of directional valves are the directional poppet valves. The movable element for flow limitation in the valve body is a ball or a cone that fits into the seat of the directional valve. The poppet design of the directional valves, as opposed to the spool valve construction, ensures tightness for certain flow directions.

An example of a single ball poppet directional valve is shown in Fig. 7.25 [30]. This is 3/2-way electromagnetically operated directional valve. In the basic position, the ball (1) is pushed by the spring (2) force into the left seat of the directional valve. The pressure input P is connected to the A output, while the T output of the directional valve is closed without leakages. The liquid can flow only in the direction (P → A).

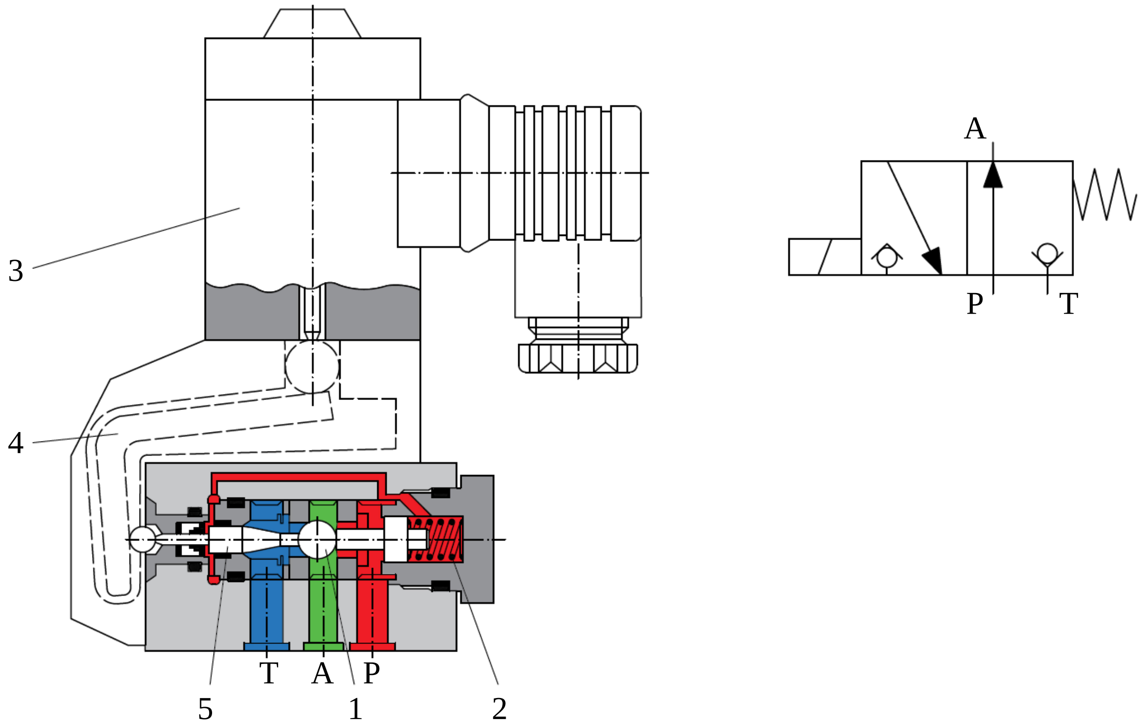


Fig. 7.25 Electromagnetically operated poppet valve as a single ball valve

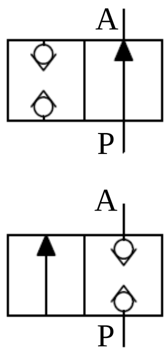
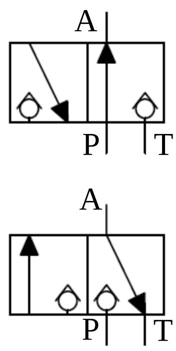
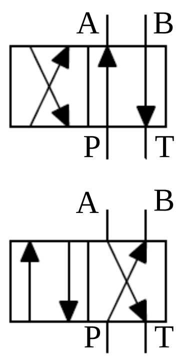
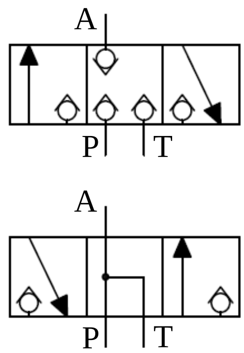
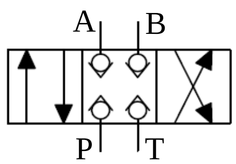
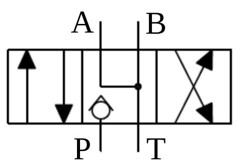
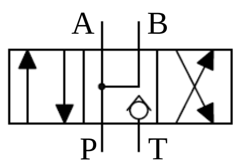
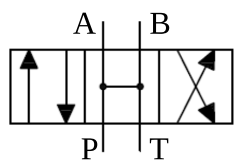
1 – ball, 2 – spring, 3 – electromagnet, 4 – lever, 5 – piston

After applying the control signal to the electromagnet (3), it exerts a force on the lever (4), which controls the piston (5) against the spring force. The ball moves to the right seat of the directional valve and closes the input P. At the same time, channels A and T are connected. In this case, the liquid flows through the directional valve in the direction A → T.

The directional poppet valves are produced in both single-stage and two-stage versions. The single-stage directional valves are usually electromagnetically operated, used only for low volumetric flow rates up to 36 $dm^3 \cdot min^{-1}$, but for high pressures up to 63 MPa. The two-stage design of the directional poppet valves is usually operated electro-hydraulically, with applications for volumetric flow rates 4000 $dm^3 \cdot min^{-1}$ and pressures to 50 MPa.

The possible design of directional poppet valves is given in Tab 7.3. The way closing of the directional poppet valve is marked in the graphical symbol by the ball symbol sitting in the seat.

Tab 7.3 Different versions of directional poppet valves

2/2 directional valve	3/2 directional valve	4/2 directional valve	3/3 directional valve
			
4/3 directional valve			
			

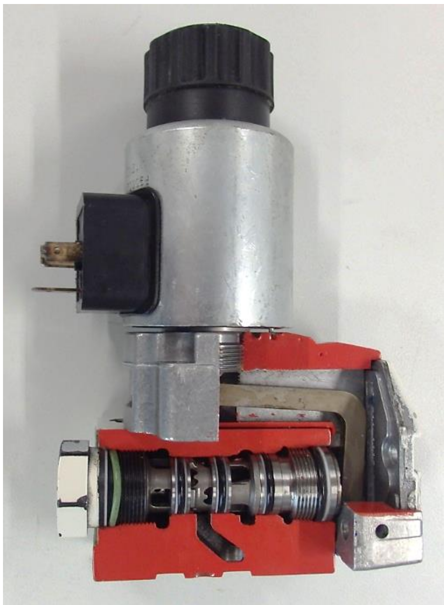


Fig. 7.26 An example of seat directional valve in cut

7.5.3 Group spool directional valves

These are sectional and monoblock directional valves. In the case of the group section directional valve shown in Fig. 7.27, the directional valve consists of more individual sections. The section may also include other hydraulic elements such as pressure valves, throttle valves, etc.

Fig. 7.27 Group three-section directional valve from Bucher Hydraulic company [59]

An example of a circuit with a five-section modular directional valve for use with a fixed displacement hydraulic pump is shown in Fig. 7.28. The first section on the left is an unloading section with a central relief valve; the other two sections consist only of a 4/3-way directional valve with a throttle valve to change the movement velocity. The fourth section is used to drive a linear hydraulic motor with a requirement to reduce the input pressure, including a pressure locking in channel A with an overpressure protection function. The fifth section has the same function as sections 2 and 3. The last section of the directional valve consists of the closing plate - i.e., the so-called free section.

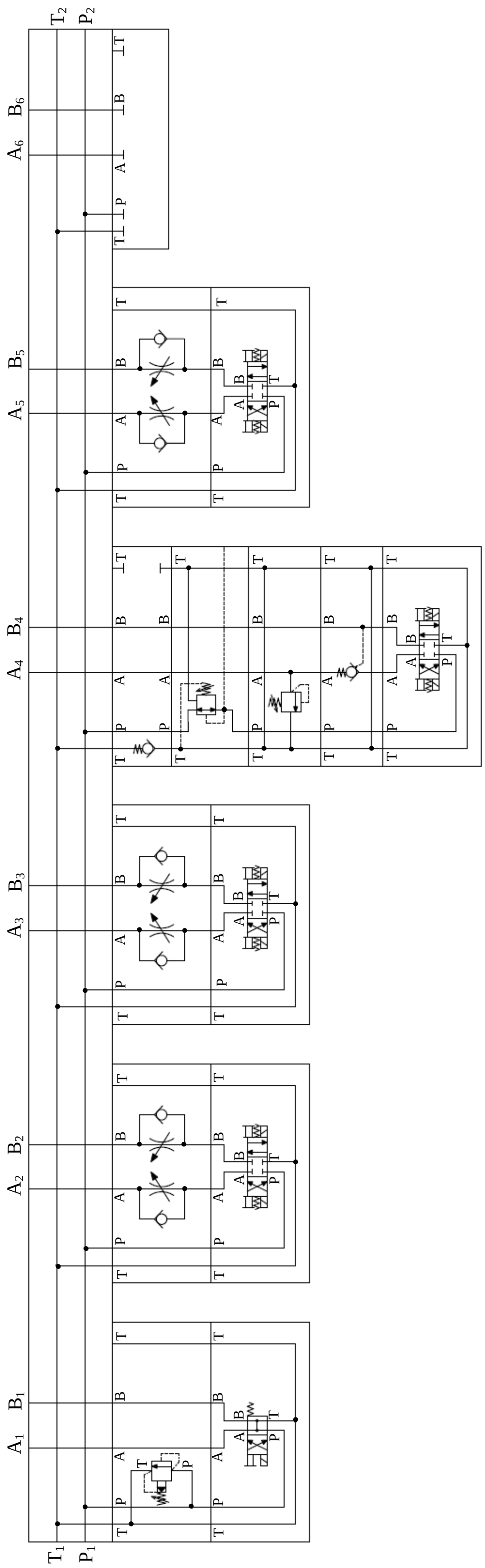


Fig. 7.28 Hydraulic five-section valve block for use with fixed displacement hydraulic pump

Monoblock group directional valves have a single body in which the spools and control elements of directional valves are integrated. An example of a Bosch Rexroth monoblock group directional valve is shown in Fig. 7.29.

Fig. 7.29 Monoblock mobile directional valve from M0-40 Bosch Rexroth company

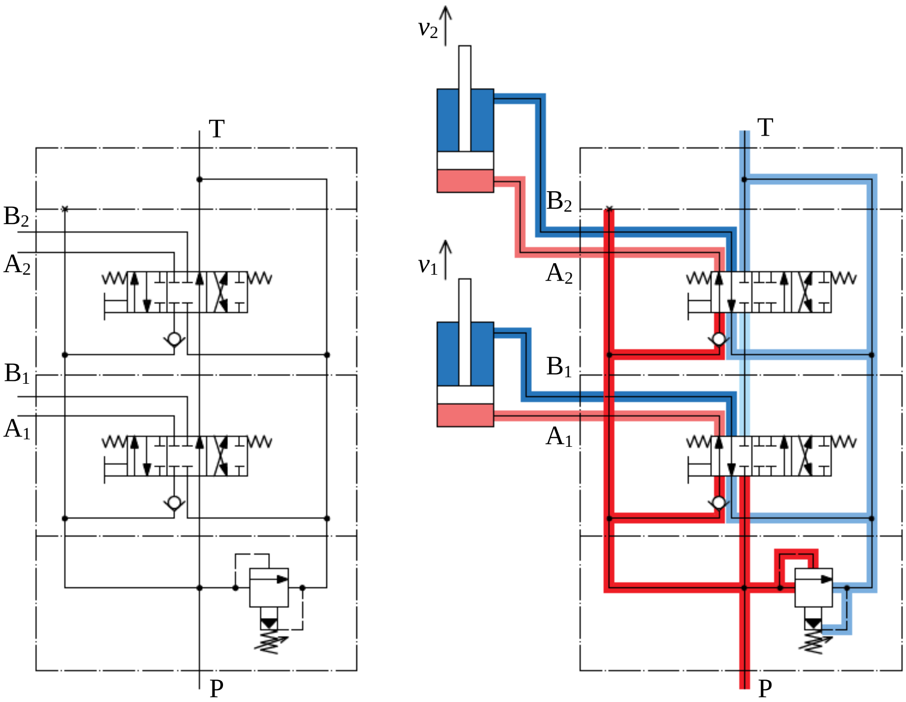


Fig. 7.30 Parallel arrangement of open centre group directional valves

The arrangement of group directional valves can be serial, parallel, or tandem and is given by the connection of two adjacent elements. In Fig. 7.30 is shown an example of parallel connection of elements. In a parallel arrangement, each consumer (hydraulic motor) connected to the outputs A and B can be independently operated. When more consumers operate simultaneously, the liquid flow is divided according to their resistances. The relief valve is used to limit the maximum pressure. Check valves at the input to individual sections of the directional valve to prevent the load from sinking when multiple consumers are simultaneously operated. Connection in circuit when both left position of direction valves are activated is shown in Fig. 7.30 (right). Fluid pressure is supplied to inputs of both hydraulic cylinders

The tandem arrangement Fig. 7.31 allows only the operation of individual appliances independently. The simultaneous operation of more consumers is not possible because when one directional valve is repositioned, the liquid supply from the pressure line P is interrupted at the directional valves connected behind it (which is shown in right figure).

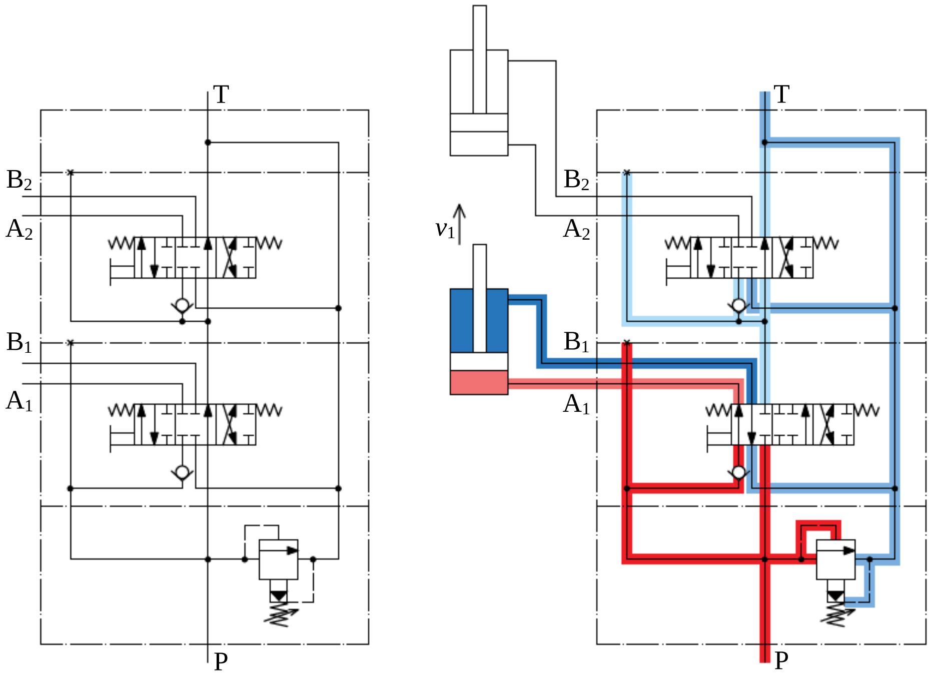


Fig. 7.31 Tandem arrangement of open centre group directional valves

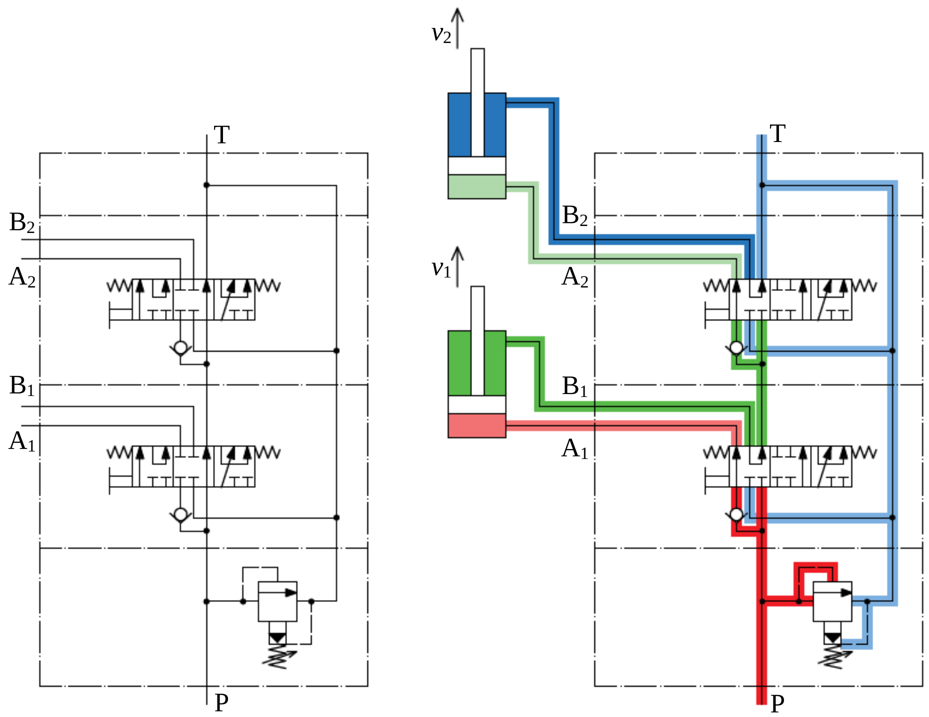


Fig. 7.32 Series arrangement of open centre group directional valves

The series arrangement Fig. 7.32 allows the operation of individual consumers individually as well as the simultaneous operation of more consumers. However, the pressure impact of the connected consumers is summed during simultaneous operation. The liquid drain is guided through one common channel T.

Due to their compactness and small size, group directional valves are mainly used in mobile machines (in mobile hydraulics).

7.5.4 Use of directional valves in circuits

Directional valves are usually used to control the movement direction of a hydraulic motor. The open centre directional valve (see Fig. 7.33) is used in combination with a constant flow source.

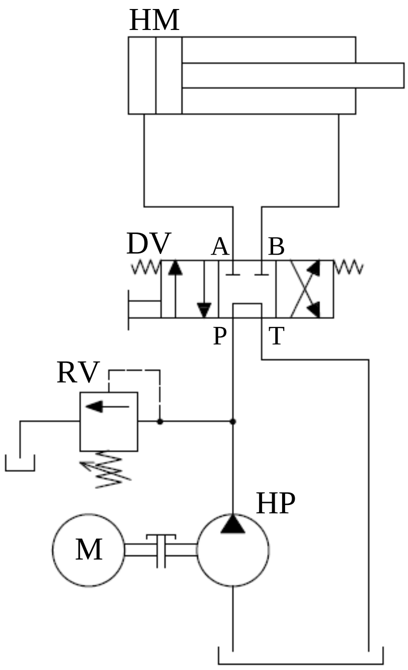


Fig. 7.33 An example of using open centre directional valve

DV – directional valve, HP – hydraulic pump, HM – hydraulic motor, RV – relief valve

The P and T ways are connected in the basic position of the directional valve DV. The liquid from the hydraulic pump HP flows through the directional valve into the tank and the hydraulic pump is unloaded. After repositioning the directional valve to one of the extreme positions, the volumetric flow from the hydraulic pump is fed to the hydraulic motor HM, which is connected between outputs A and B. The relief valve RV is used to limit the maximum system pressure.

In the case of using a closed centre directional valve, as is shown in Fig. 7.34 (left), the liquid supply from the hydraulic pump HP is closed in the basic position of the directional valve DV1. When the hydraulic motor HM is inactive due to the increase in pressure behind the hydraulic pump, the relief valve RV would open, and the liquid would flow through the relief valve back into the tank. The pressure at the output of the hydraulic pump would correspond to the maximum system pressure (adjusted at the relief valve), and there would be a significant energy dissipation. Therefore, it is suitable to unload the hydraulic pump when the hydraulic motor is inactive. This is possible, e.g., by using the unloading directional valve DV2. During the time when the middle position of the main directional valve DV1 is active, a control signal is fed to the electromagnet of the directional valve DV2, which is moved to position b. The liquid from the hydraulic pump flows through the directional valve DV2 into the tank, the output pressure of the hydraulic pump is only high enough to overcome the resistance to the liquid flow. The hydraulic pump is unloaded. When a signal is applied to one of the control electromagnets of the directional valve DV1, the control signal on the directional valve DV2 is interrupted, and the spring force moves this directional valve to the position a. The liquid flows through the directional valve DV1 to the hydraulic motor.

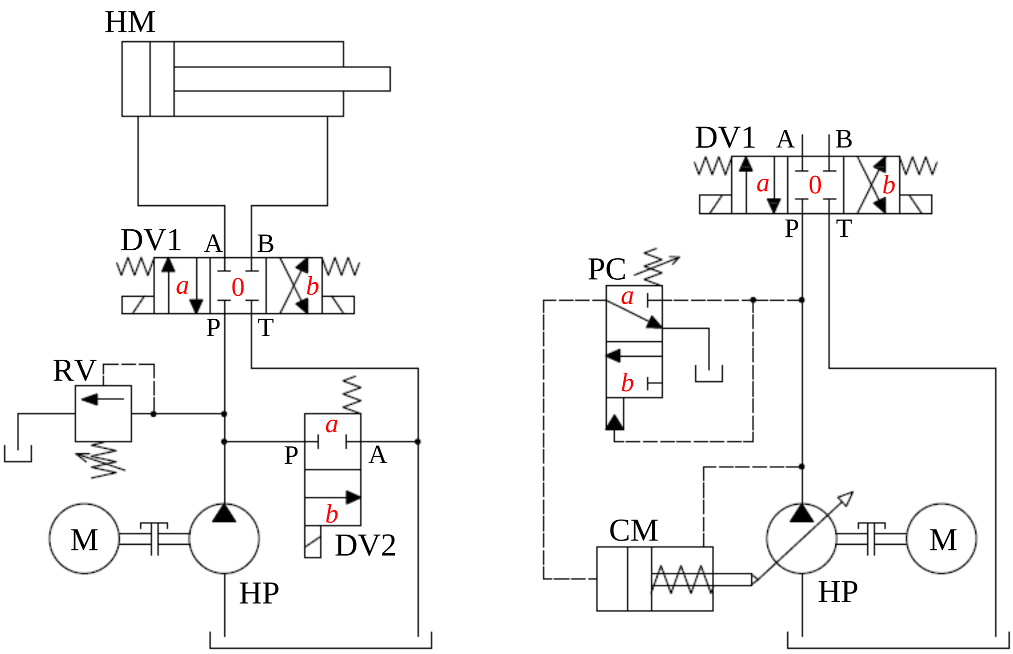


Fig. 7.34 An example of using a closed centre directional valve, circuit with an unloading directional valve (left), circuit with a hydraulic pump with constant pressure control (right)

DV1 – main directional valve, DV2 – unloading directional valve, HP – hydraulic pump, HM – hydraulic motor, RV – relief valve, PC – pressure controller, CM – control mechanism of hydraulic pump

Another possibility of using a closed centre directional valve is in combination with a constant pressure source. An example of this use is shown in Fig. 7.34 (right). The circuit is equipped with the hydraulic pump HP with constant pressure control. The pressure controller PC is shown as a 3/2-way directional valve. The maximum (constant) system pressure is adjusted by the spring preload of this controller. If the main directional valve DV1 is in the middle position (the supply P from the hydraulic pump is closed), the liquid pressure will move the spool of the pressure controller to position b. The pressure in the piston space of the control mechanism CM of the hydraulic pump will increase, and the geometric stroke volume of the hydraulic pump will be decreased to zero value. The hydraulic pump does not supply any additional liquid flow to the system until the directional valve DV1 is repositioned.

A two-section modular construction for manipulation with linear hydraulic motors is shown in Fig. 7.35. The left section consists of the central pressure relief valve (1), the throttle valves (2) for adjusting the velocity of the linear hydraulic motor and the double pilot operated check valve (4) for locking the position of the piston rod of the hydraulic motor. The modular building is closed by the 4/3-way directional valve (5). The right section is used to control the linear hydraulic motor by means of the 4/3-way proportional directional valve (6), where the pilot operated directional poppet valves (7) are implemented in the working branches A and B, in order to lock the pressure (position) of the linear hydraulic motor.

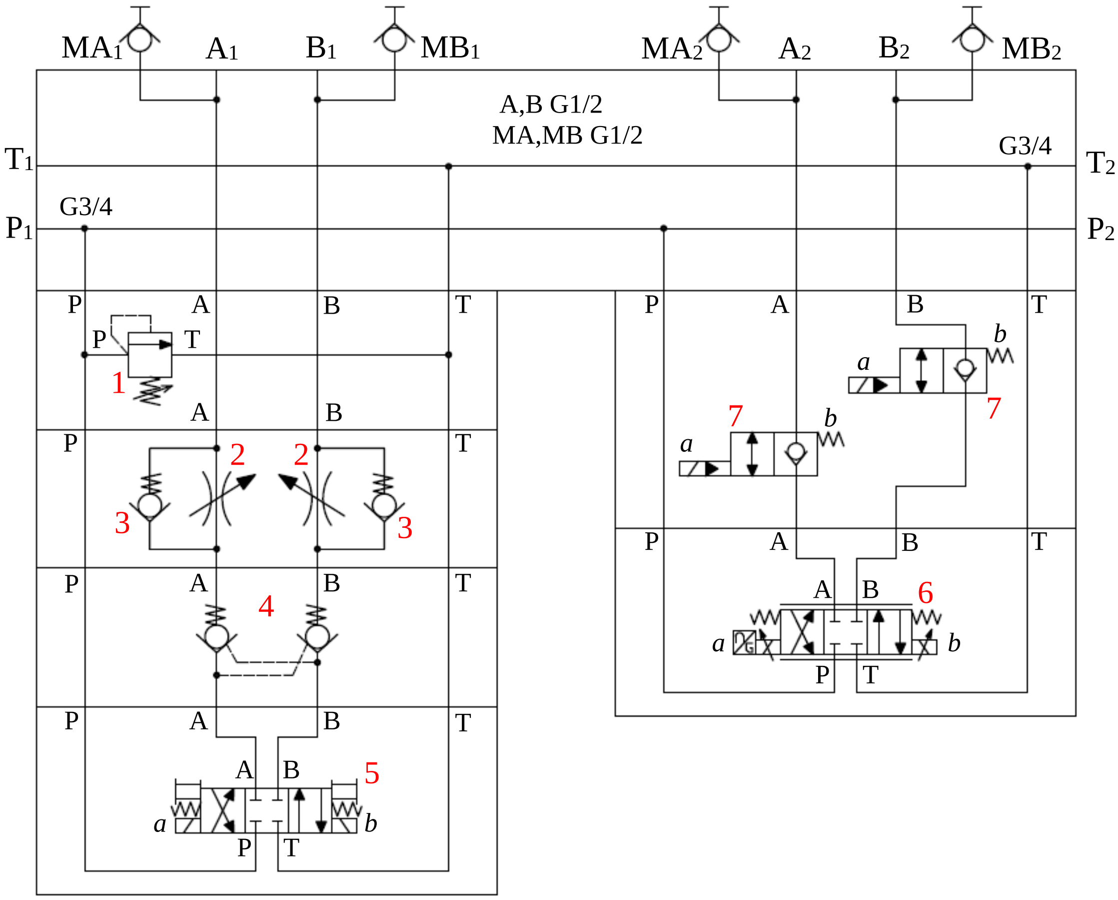


Fig. 7.35 An example of using a closed centre directional valve for a circuit with hydraulic pump with constant pressure control

1 – relief valve, 2 – throttle valve, 3 – check valves, 4 – double pilot operated check valve, 5 – directional valve, 6 – proportional directional valve, 7 – directional poppet valves

8. Flow control valves

In general, these are elements that increase resistance to fluid flow. The change in resistance is achieved by changing the flow area. An increase in resistance is reflected in a hydraulic circuit by a subsequent change in pressure and always leads to energy losses. The flow area can be constant, represented by orifices and nozzles, or variable, represented by throttling valves, proportional distributors, or flow dividers. When fluid flows over the flow area (flow throttling), a part of the pressure energy is converted to heat energy, and this usually results in undesirable warming of the fluid.

Elements belonging to this group can be described as nonlinear resistance to motion. The flow through the throttle cross-section can be calculated from the equation:

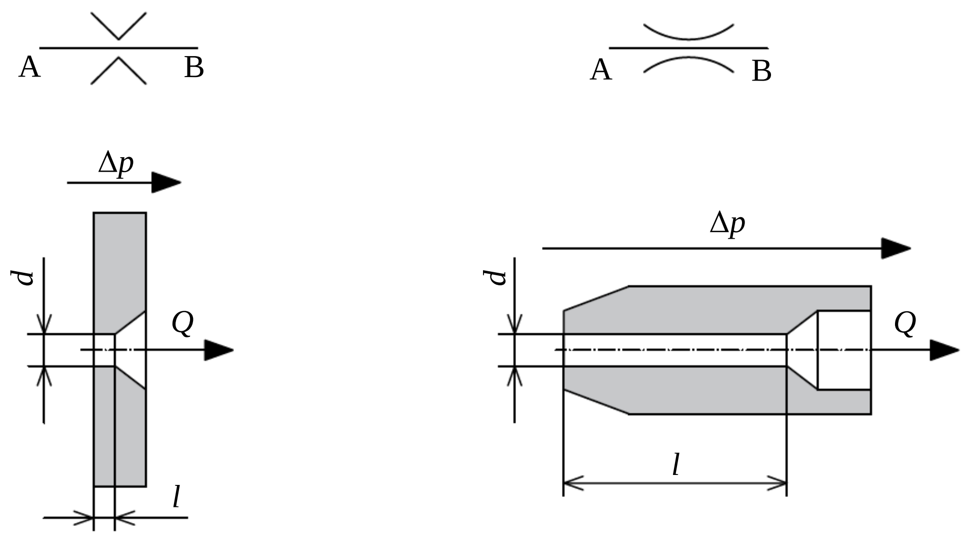
$$Q = \mu \cdot A \cdot \sqrt{\frac{2 \cdot \Delta p}{\rho}}, \tag{8.1}$$

where Q [$m^3 \cdot s^{-1}$] is the volumetric flow, A [m^2] throttling point, Δp [Pa] pressure loss (pressure difference in front of and behind a throttle element), ρ [$kg \cdot m^{-3}$] is the fluid density, and μ [-] is the flow coefficient.

The size of the flow coefficient μ depends on the geometry and shape of the edges of the flow area, the back pressure behind the element and the Reynolds number (character of the flow). It is usually in the range $\mu = (0.6 \div 0.8)$.

8.1 Orifices and throttles

It is one of the simplest elements for controlling the flow rate. They are characterized by a constant flow area and create a constant resistance in a hydraulic system. The difference between the orifice (Fig. 8.1 left) and the nozzle (Fig. 8.1 right) is due to the difference in the ratio of their length to the inner diameter *l/d*.



$\frac{l}{d} = 0{,}2 \div 2$

$\frac{l}{d} > 20$

Fig. 8.1 Constant resistance flow elements, orifice (left), throttle (right)

Orifices can be used in hydraulic systems to create a pressure gradient (maintain a constant volumetric flow at a constant pressure gradient). They are used to dampen pressure peaks and are part of simple hydraulic regulators,, e.g. in pilot operated pressure relief valves or variable displacement pumps. The throttles, together with flap, act as a mechanical-hydraulic converter in servo valves or are used in other control elements. The above-mentioned *l/d* ratio also defines the dependence of the characteristic of a given element on the liquid viscosity and, therefore, on the temperature. It can be concluded that the smaller the *l/d* ratio, the smaller the viscosity dependence, which is also reflected in the graphical symbol. The same assumption applies to other elements designed to control the flow rate [3], [42].

Dependence of the volumetric flow Q of orifice or throttle on the pressure loss Δp is shown in Fig. 8.2.

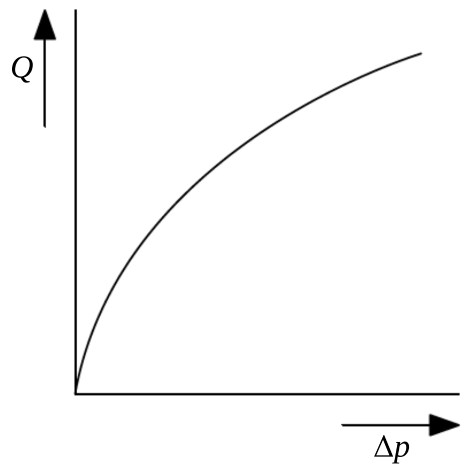


Fig. 8.2 Dependence of the volumetric flow Q of orifice or the throttle on the pressure loss Δp

8.2 Throttle valves

In the case of throttle valves, it is possible to continuously change the resistance to fluid flow, this is realized by changing the size of the flow area A. Throttle valves are usually used in circuits to control the velocity (speed) of hydraulic motors. While the dependence of the volumetric flow Q on the flow area A is linear, the dependence on the pressure loss Δp is non-linear, as shown in Fig. 8.3. The flow coefficient μ, as in the case of orifices, depends on a number of factors and usually takes values $\mu = (0.68 \div 0.78)$. The volumetric flow through the throttle valve can be calculated according to equation (8.1).

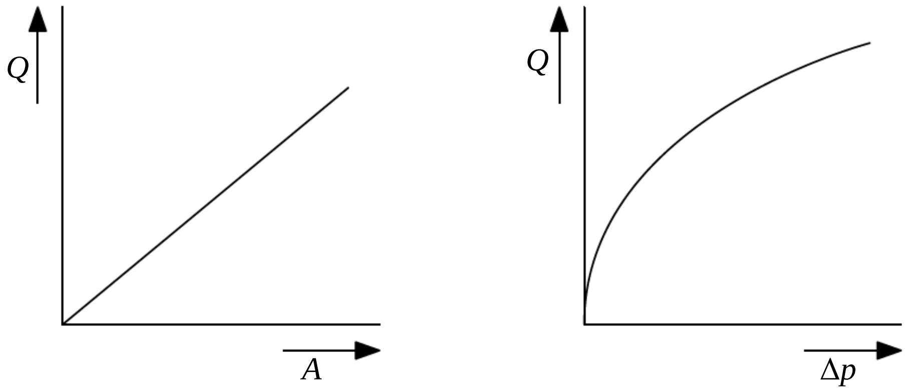


Fig. 8.3 Dependence of the volumetric flow Q of the throttle valve on the flow area A and the pressure loss Δp

The flow area size of the throttle valve depends on its design and the size of the control parameter h. The basic designs of throttle valves include the spool valve with a rectangle longitudinal slot, the spool valve with a triangle longitudinal slot, the spool valve with an overlapping circular hole (i.e., clearance and circumference throttles) and the needle valve. A change in the control parameter h can be achieved by rotary or linear movement on the moving part of the valve. The dependence of the flow area A on the control parameter h for different throttle valve designs is shown in Fig. 8.4 [3], [11].

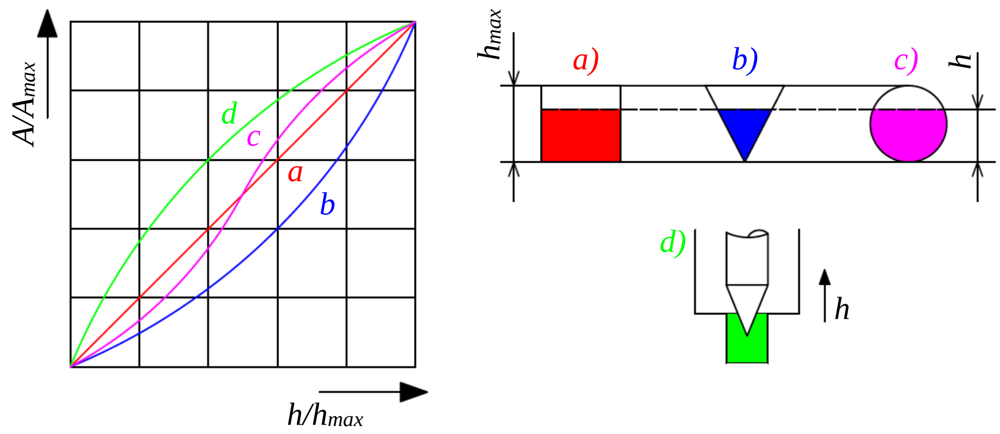


Fig. 8.4 Dependence of the flow area A on the control parameter h for different throttle valve designs

a) spool valve with rectangle longitudinal slot, b) spool valve with triangle longitudinal slot, c) spool valve with overlapping circular hole, d) needle valve

Static characteristics of throttle valves are shown in Fig. 8.5. The dependence $Q = f(\Delta p)$ is shown for different constant values of the throttle valve opening ratio φ, which is given by the ratio of the current and maximum opening (control parameter h).

$$\varphi = \frac{h}{h_{max}} \tag{8.2}$$

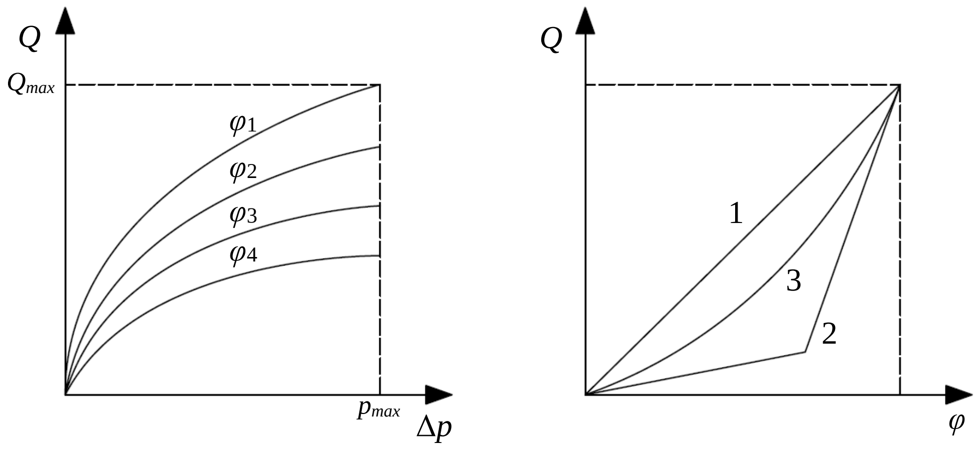


Fig. 8.5 Static characteristics of throttle valves

1 – linear, 2 – broken, 3 - progressive

In terms of construction, throttle valves are manufactured either to be mounted directly in the pipe (see Fig. 8.6 and Fig. 8.7) or for mounting on a connection plate or cube (see Fig. 8.10). The throttle valve shown in Fig. 8.6 allows flow throttling in both flow directions. The change in the flow area of the valve (1) is caused by the rotary motion of body (2), which results in a linear movement of the valve's moving part (3) through the thread. In practice, throttle valve with a parallel-arranged check valve (so-called throttle check valve), which serves as a bypass in one flow direction, is more commonly used (see Fig. 8.7).

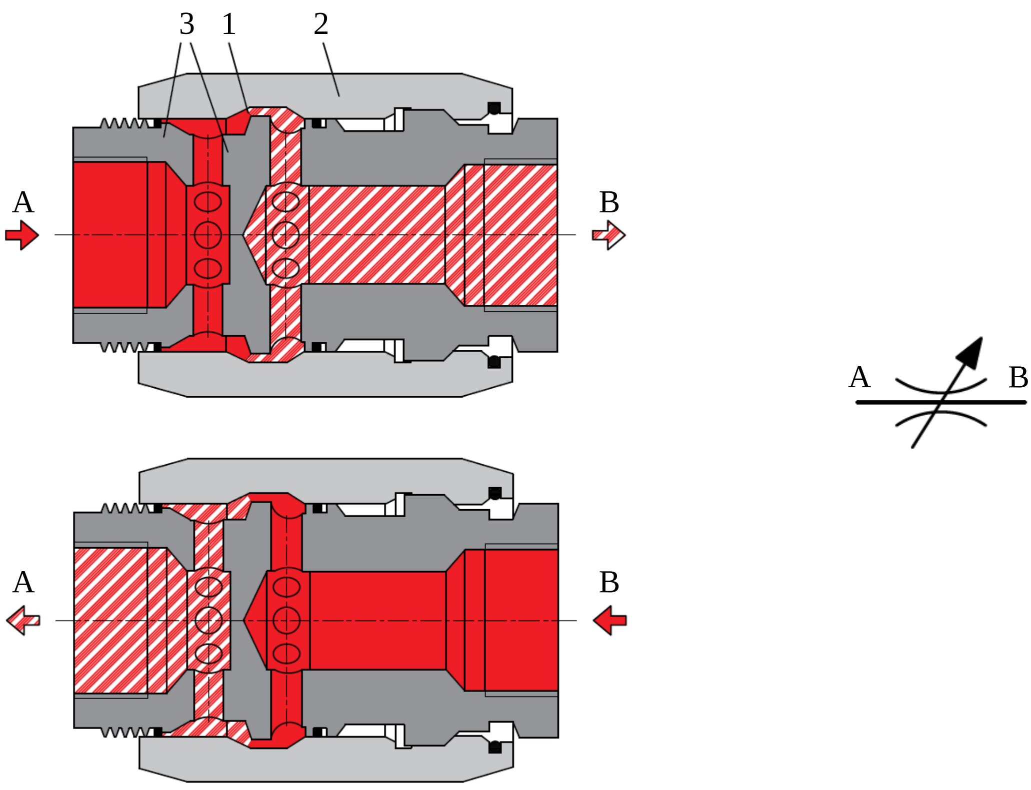


Fig. 8.6 Throttle valve design for pipe mounting

1 - the flow area of the valve, 2 – valve body, 3 – valve moving part

In terms of design, the throttle check valve is similar to the valve in the previous case. However, a check valve is built inside the throttle valve, which defines the direction of flow throttling. In the case of liquid flow in the direction from A to B, see Fig. 8.7 (top), the liquid pressure acts on the surface of the check valve cone (1) and it produces a pressure force that is greater compared to the spring (2) force. Therefore, the cone is pushed out of the seat (3). In this flow direction, the liquid flows through both the flow area of the throttle valve (4) and the check valve, and there is no flow control. In the opposite flow direction (i.e., from B to A) Fig. 8.7 (bottom), the liquid pushes the cone (1) of the check valve into the seat (2) and the liquid flow is possible only around the circumference of the throttle valve (4). In this direction, the flow is controlled. In this case, the check valve is located parallel to the throttle valve. The design can also be without a spring for the check valve. The throttle check valve is used to control the velocity or speed of hydraulic motors in one direction only.

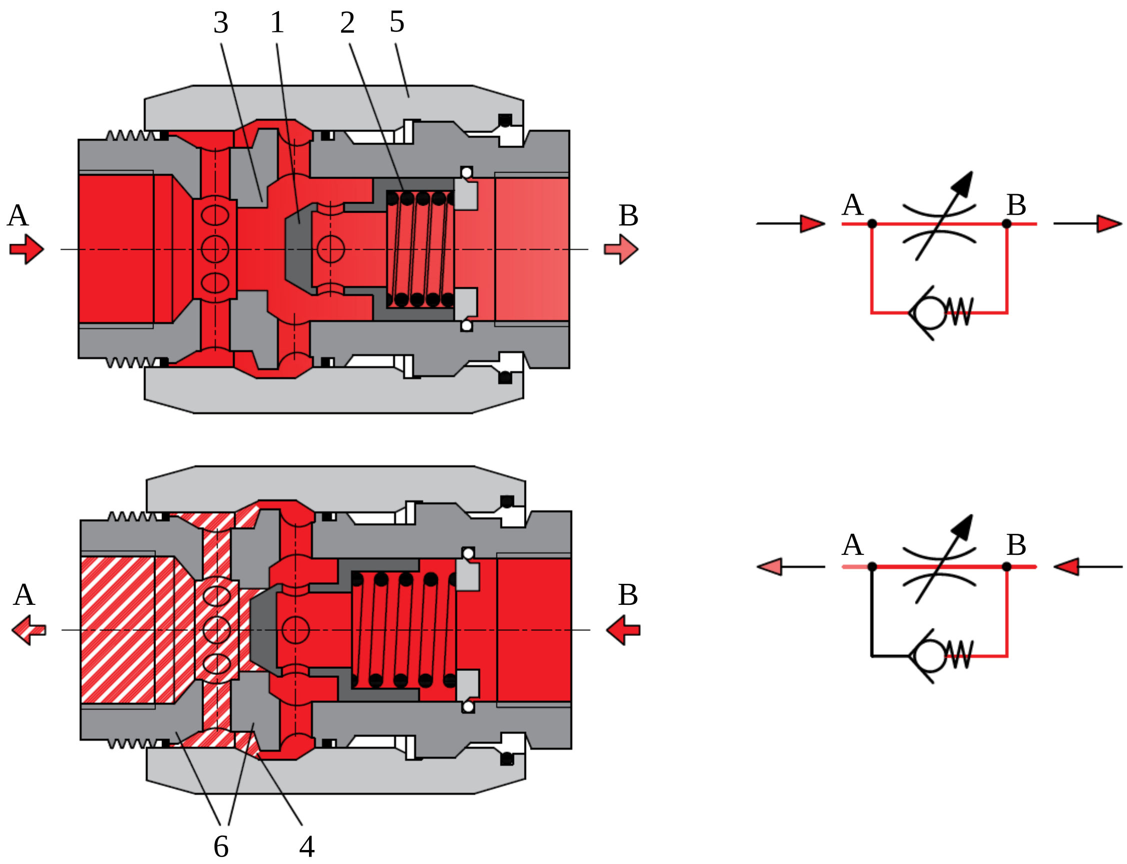


Fig. 8.7 Throttle check valve

1 – check valve cone, 2 – spring, 3 – check valve seat, 4 – flow area of the throttle valve, 5 – valve body, 6 – throttle valve moving part

Depending on the orientation of the check valve, the flow throttling can be at the entrance to thee working spaces of the hydraulic motor (Fig. 8.8) or at the output of the hydraulic motor (Fig. 8.9).

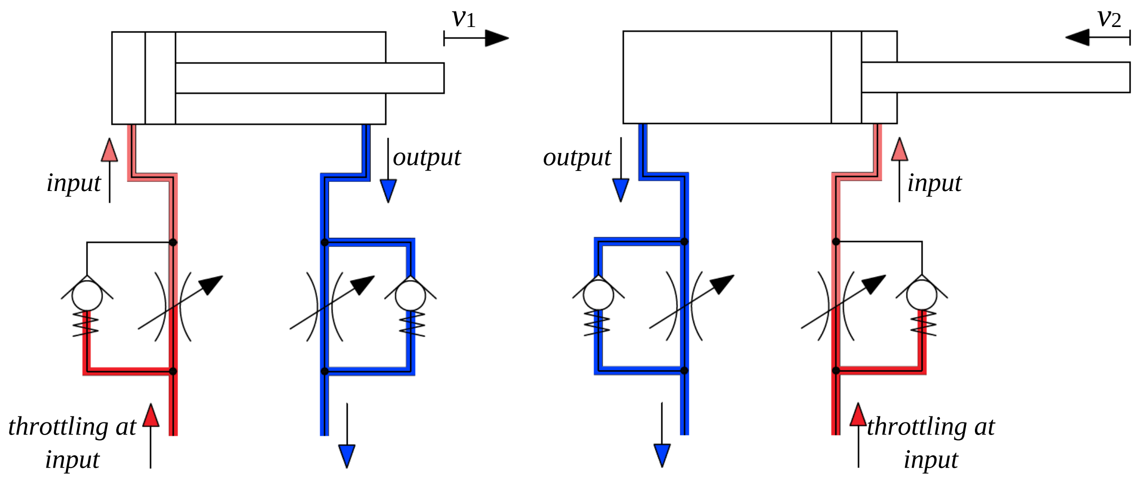


Fig. 8.8 Connection of throttle valves, throttling at the entrance (input) of a hydraulic motor

It is more common to throttle the liquid flow at the outlet of the hydraulic motor. It is characterized by more continuous starting of the hydraulic motor and is used in cases where it is necessary to manipulate a mass load of the hydraulic motor operating in dynamic mode. The disadvantage of this connection is the danger of pressure multiplication on the piston rod side of the differential hydraulic cylinder. In the case of inlet throttling, there is no risk of pressure multiplication due to the location of check valves, and there are less seal resistances and their wear. However, this connection is not suitable for hydraulic motors operating in dynamic mode with a non-negligible mass load on the piston rod.

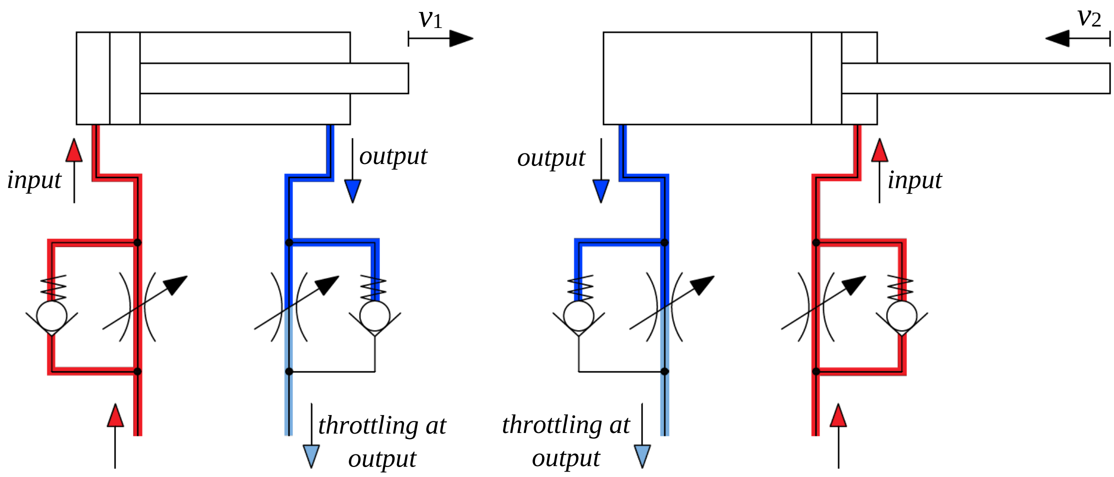
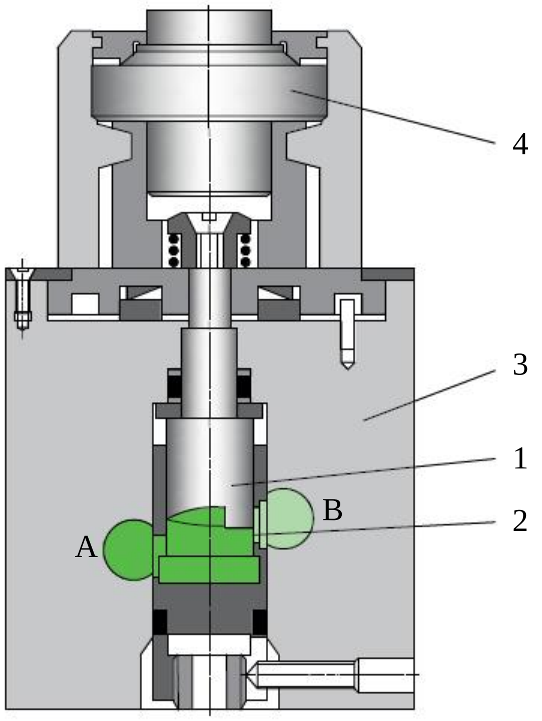


Fig. 8.9 Connection of throttle valves, throttling at the outlet (output) of a hydraulic motor

In the previously mentioned throttle valves, the flow rate was dependent on the temperature change and, thus also, on the viscosity change. The throttle valve with temperature compensation (also known as fine throttle valve; see Fig. 8.10) has a helical spool (1) that overlaps only a narrow rectangular slot (2), and the flow through this valve is only slightly dependent on the viscosity change [30].



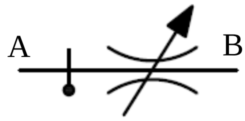


Fig. 8.10 Throttle valve with temperature compensation

1 – spool, 2 – rectangular slot, 3 – valve body, 4 – control mechanism

Throttle valves are usually used when the load on a given hydraulic motor is constant or only slightly variable.

8.3 Throttle valves with pressure gradient stabilisation

The above-mentioned pressure-dependent throttle valves cannot maintain a constant flow of liquid under variable load on the hydraulic motor. When the force (torque) on the hydraulic motor is changed, its velocity (speed) is also changed. In applications where it is necessary to maintain a constant velocity or speed of a hydraulic motor independently of changes in pressure gradient, throttle valves with pressure gradient stabilization can be used. Sometimes these valves are also referred to as flow controllers. These elements consist of the throttle valve, which performs the function of an adjustable measuring orifice and the pressure compensator arranged in parallel (or in series) with respect to the throttle valve.

8.3.1 2-way flow control valve

It is a two-way throttle valve with pressure gradient stabilization (or 2-way flow control valve). The possibilities of graphical symbols are shown in Fig. 8.11. The detailed symbol of the 2-way flow control valve is shown in Fig. 8.11 (left). Its simplified symbol, in temperature-compensated design, is shown in Fig. 8.11 (in the middle). Compared to the pressure-dependent throttle valve, 2-way flow control valve can only regulate flow in one direction of flow. In practice, this valve can be supplemented by a parallel connected check valve, which in some designs is already part of this valve, as shown in Fig. 8.11 (right).

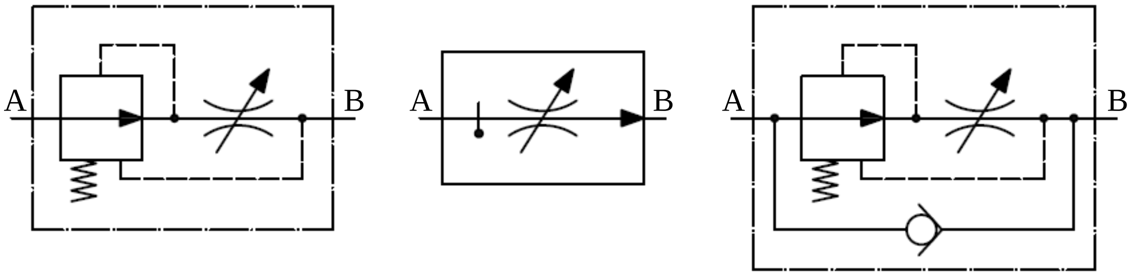


Fig. 8.11 Graphical symbols of 2-way flow control valve, a detailed symbol (left), a simplified symbol in temperature-compensated design (in the middle), detailed symbol in version with check valve (right)

The valve function is demonstrated in Fig. 8.12. The 2-way flow control valve consists of the series-connected pressure compensator PC (its function is similar to a pressure reducing valve) and the throttle valve TV. The throttle valve basically performs the function of an orifice with adjustable hydraulic resistance. The volumetric flow Q_{HG} supplied by the hydraulic pump HG is divided into the flow Q_1, which continues through the 2-way flow control to the hydraulic motor HM, and the flow Q_{RV}, which flows through the relief (pressure) valve RV. It is obvious that some of the pressure energy is dissipated on the relief valve, as is the case in a circuit with pressure-dependent throttle valves. When liquid flows through the 2-way flow control valve, the pressure loss Δp_{PC} is created on the spool edges of the pressure compensator PC, and the pressure loss Δp_{TV} on the throttle valve.

The volumetric flow Q_{HG} supplied by the hydraulic pump HP is given by formula:

$$Q_{HG} = Q_1 + Q_{RV} \,. \tag{8.3}$$

The pressure loss Δp_{PC} on the spool edges of the pressure compensator PC is defined:

$$\Delta p_{PC} = p_1 - p_2 \,. \tag{8.4}$$

The pressure loss Δp_{TV} on the throttle valve TV:

$$\Delta p_{TV} = p_2 - p_3 \,. \tag{8.5}$$

The spool faces of the pressure compensator have the same areas A_{2PC} and A_{3PC}, while the spring is inserted on the side of the area A_{3PC}, which exerts the force F_s when compressed. The pressure loss on the throttle valve Δp_{TV} is fed as feedback to the slide faces of the pressure compensator PC (the pressure p_2 between the pressure compensator and the throttle valve is fed to the area A_{2PC}, and the pressure p_3 between the throttle valve and the hydraulic motor is fed to the area A_{3PC}. The pressure gradient on the throttle valve Δp_{TV} acts against the spring force F_p and sets the spool valve in the equilibrium position that ensures the required flow [5], [6].

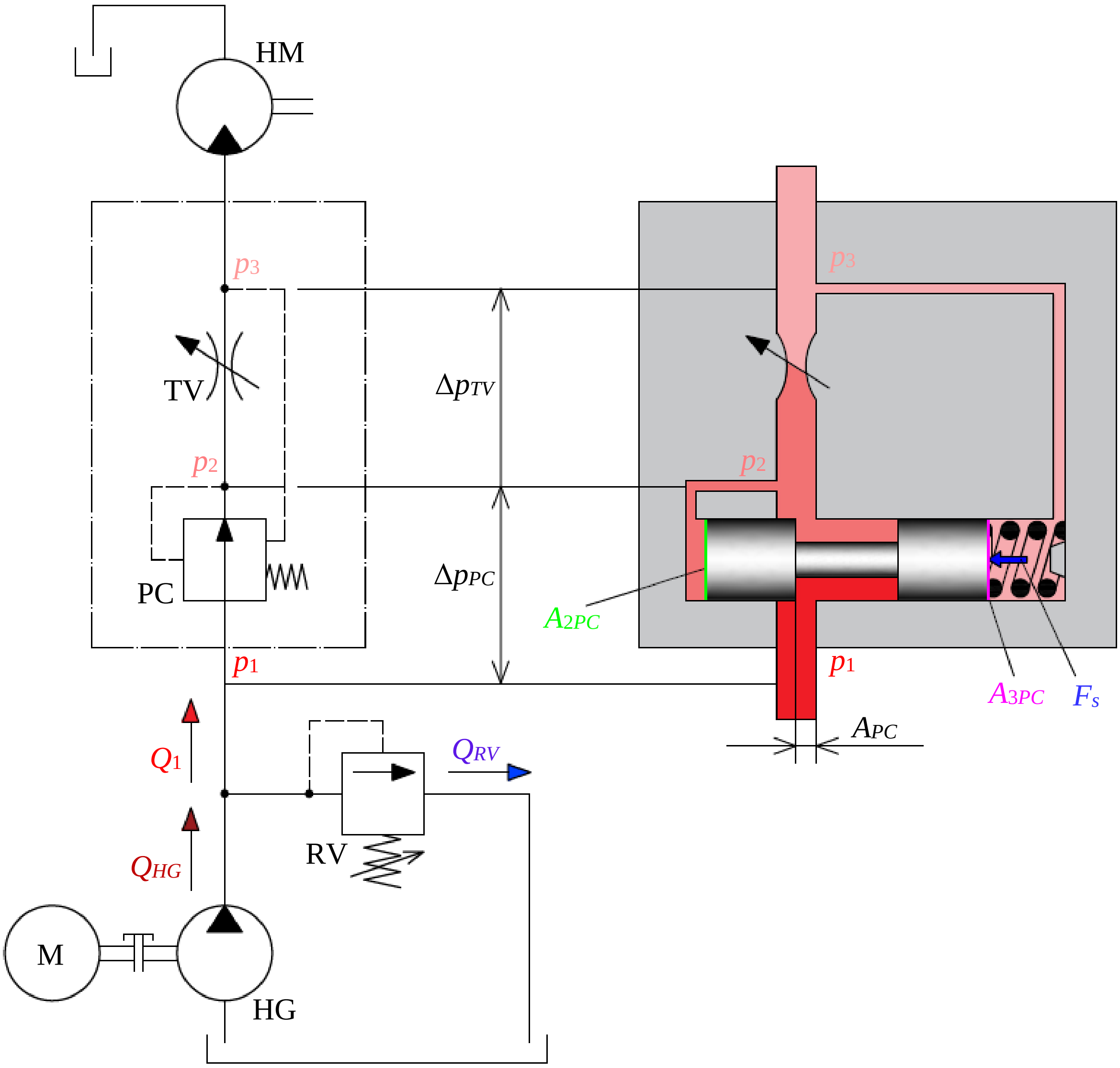


Fig. 8.12 Function principle of 2-way flow control valve

HG – hydraulic pump, RV – relief valve, PC – pressure compensator, TV – throttle valve, HM – hydraulic motor, M – electric motor

It is possible to express the force balance on the spool of the pressure compensator PC as follows:

$$p_2 \cdot A_{2PC} = p_3 \cdot A_{3PC} + F_s \, . \tag{8.6}$$

Since the spool areas of the pressure compensator are the same (i.e., $A_{2PC} = A_{3PC} = A_{PC}$), it is possible to modify equation (8.6) as follows:

$$A_{PC} \cdot (p_2 - p_3) = F_s$$

$$A_{PC} \cdot \Delta p_{TV} = F_s$$

$$\Delta p_{TV} = \frac{F_s}{A_{PC}} = const. \tag{8.7}$$

Since the spool stroke is relatively small and the increment of the spring force is insignificant in relation to its preload, the resulting spring force F_s can be considered constant during the control processes. Then, according to equation (8.7), the pressure gradient Δp_{TV} on the throttle valve will also be constant.

If the load on the hydraulic motor is changed, the force balance on the spool of the pressure compensator is disturbed. For example, if the load on the hydraulic motor is increased, the volumetric flow Q_1 through the valve is decreased. At the same time, the pressure gradient on the throttle valve Δp_{TV} is decreased, which disturbs the balance of forces on the spool and causes it to shift to the left. In this way, the flow area A_1 of the pressure compensator increases, which leads to an increase in the flow rate Q_1. As soon as the original flow rate Q_1 is reached, which corresponds to the original value of the pressure gradient on the throttle valve Δp_{TV}, the spool movement of the pressure compensator will stop (the spool is in the equilibrium position again). Note that this valve operates with a permanent deviation of (2 ÷ 5) % from the original flow value. If the pressure-dependent throttle valve was used in this application, a change in load on the hydraulic motor would cause a change in the pressure gradient across the throttle valve. This would result in a change in the flow size and, thus, a change in the speed of the hydraulic motor (assuming that the input pressure p_1 is constant).

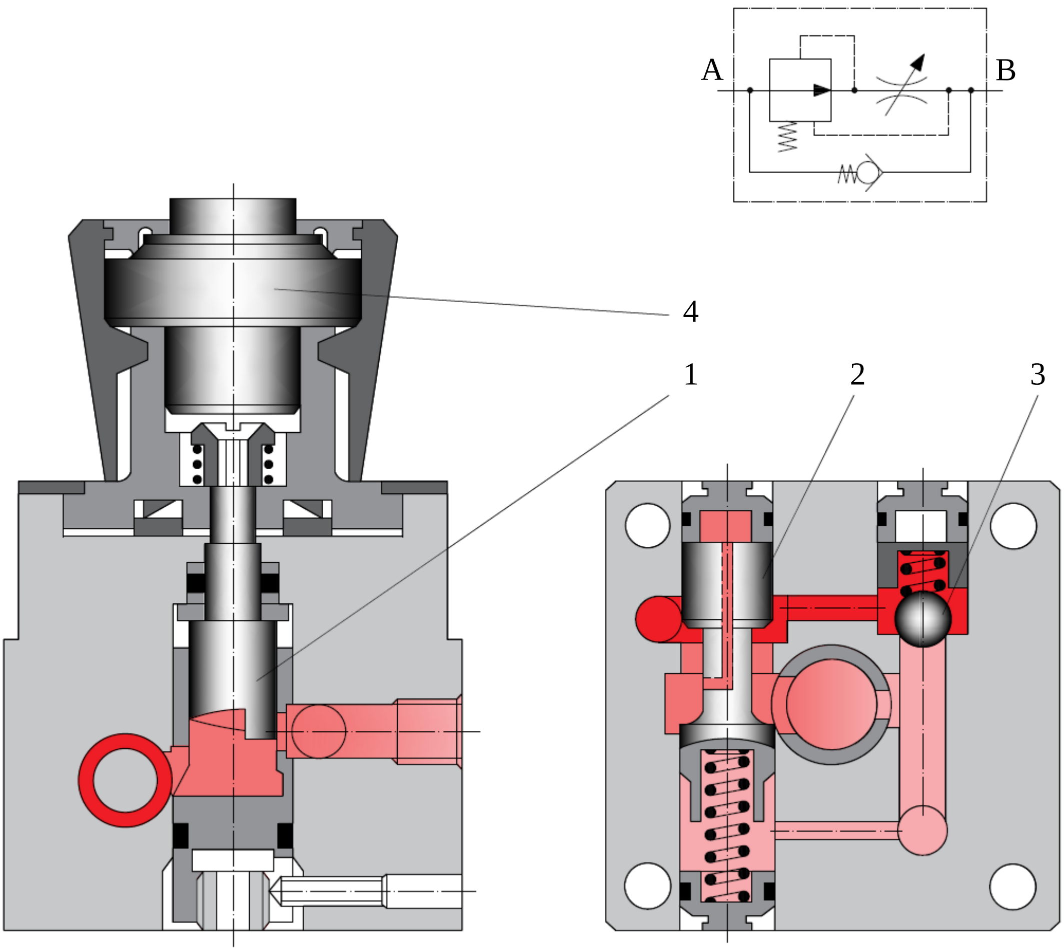


Fig. 8.13 Cross-section of 2-way flow control valve

1 – throttle valve, 2 – pressure compensator, 3 – check valve, 4 – control mechanism

A cross-section of a 2-way flow control valve is shown in Fig. 8.13. The flow rate is adjusted by changing the flow cross-section of the throttle valve [11].

The load Q - Δp characteristic of the 2-way flow control valve is shown in Fig. 8.14. The volumetric flow through this valve is almost constant, independently of the load pressure.

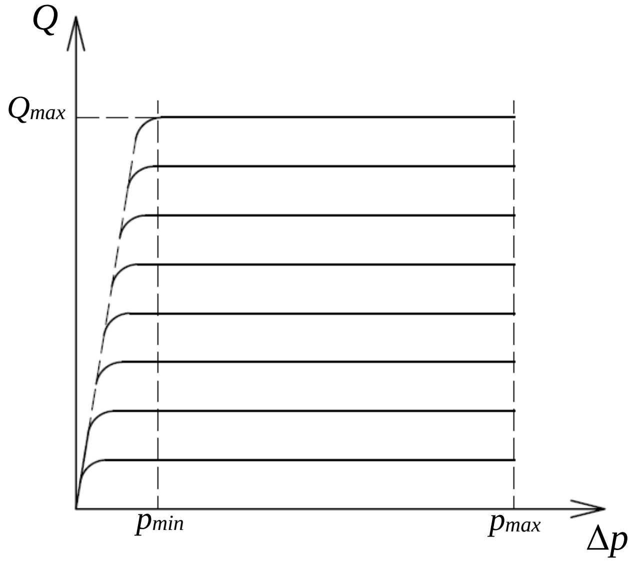


Fig. 8.14 Load Q - Δp characteristic of 2-way flow control valve

In general, with these valves, the order of connection of the elements (i.e., the pressure compensator and the throttle valve) does not matter, but the control signals must always be correctly fed to the spool faces. 2-way flow control valves are used in practical applications,, e.g., for synchronization of linear or rotary hydraulic motors, to ensure a constant speed of rotary hydraulic motors in conveyors with a hydraulic drive and as so-called flow valves used to protect against the high velocity of descent of linear hydraulic motors. The 2-way flow control valve is also a part of every modern section of the mobile hydraulic distributor.

8.3.2 3-way flow control valve

In addition to 2-way flow control valves, 3-way flow control valves Fig. 8.15 are also used in practical applications. These valves are similar in design, the third way connects the internal space of the pressure compensator to the tank, or the third way can be used as a pressure loaded way to drive a consumer (very often,, e.g., a fan). In practice, this enables energy savings in a system, especially when working with an unloaded hydraulic motor. This system of pressure compensator and hydraulic load sensing from the consumer is called as Load-Sensing system. It is used in combination with a fixed displacement hydraulic pump, and is often used, especially in mobile hydraulic applications.

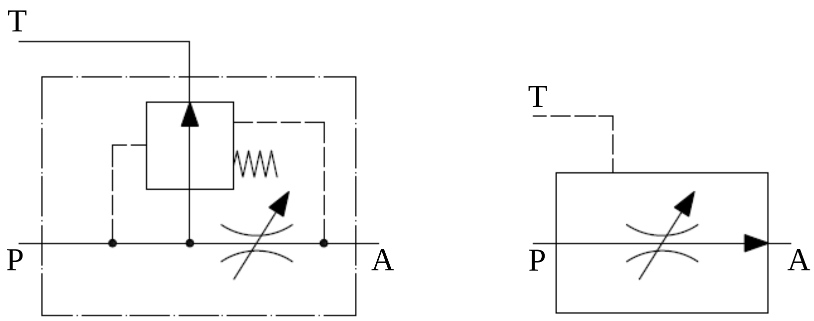


Fig. 8.15 Graphical symbols of 3-way flow control valve, detailed symbol (left), simplified symbol (right)

Fig. 8.16 (left) shows the proportional valve (3) (in the function of the throttle valve) with the 2-way flow control valve (2). The purpose of the proportional valve is to change the volumetric flow to the consumer independently of the change in the pressure gradient. The logic (shuttle) valve (4) ensures that the pressure signal to the right side of the pressure compensator (green line) is always supplied from branch A or B, depending on which branch has a higher pressure. This pressure p_2, together with the spring, acts on the right area A_{PC} of the spool compensator, against which the pressure p_1 acts from the left side of the compensator spool in front of the proportional valve. The force analysis on the pressure balance shows that the area A_{PC} and the spring force F_s are constant, so the pressure gradient on the proportional valve must also be constant and, therefore, equal to the spring force related to the area A_{PC}.

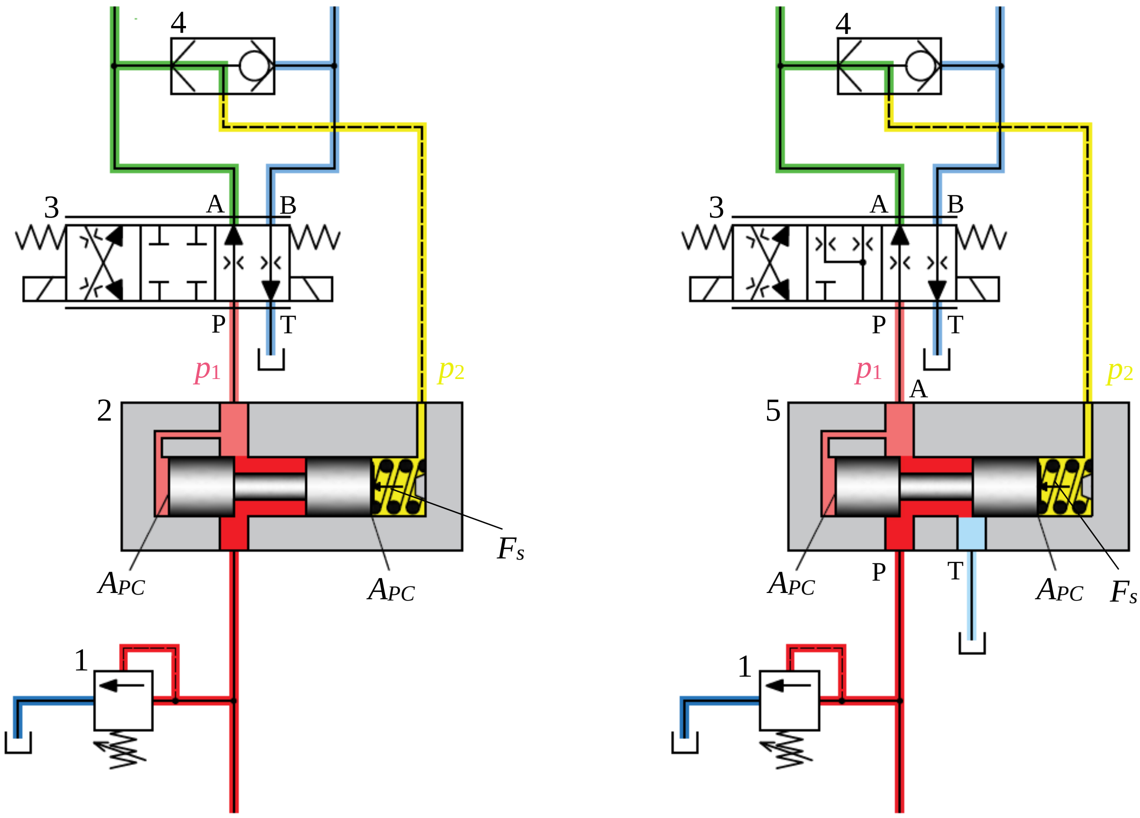


Fig. 8.16 Proportional valve with 2-way flow control valve (left), proportional valve with 3-way flow control valve

1 – relief valve, 2 – 2-way flow control valve, 3 – proportional valve, 4 – logical (shuttle) valve, 5 – 3-way flow control valve

Fig. 8.16 (right) shows the proportional valve (3) (again in the function of the throttle valve) with the 3-way flow control valve (5) as a partially energy-saving technical solution for a throttle or proportional valve in combination with a fixed displacement hydraulic pump. The 3-way flow control valve has a third path leading to the drain T, or this branch can be pressure loaded. The proportional valve maintains a constant pressure gradient corresponding to the spring force F_s in relation to the area A_{PC}. If the consumer does not use the total flow of the hydraulic pump, then the liquid overflow from the hydraulic pump flows through the T-branch to the tank, but only with a pressure gradient equal to the sum of the pressures on the consumer and the pressure gradient on the throttle element. This solution is very often used in mobile hydraulics, especially in agricultural equipment.

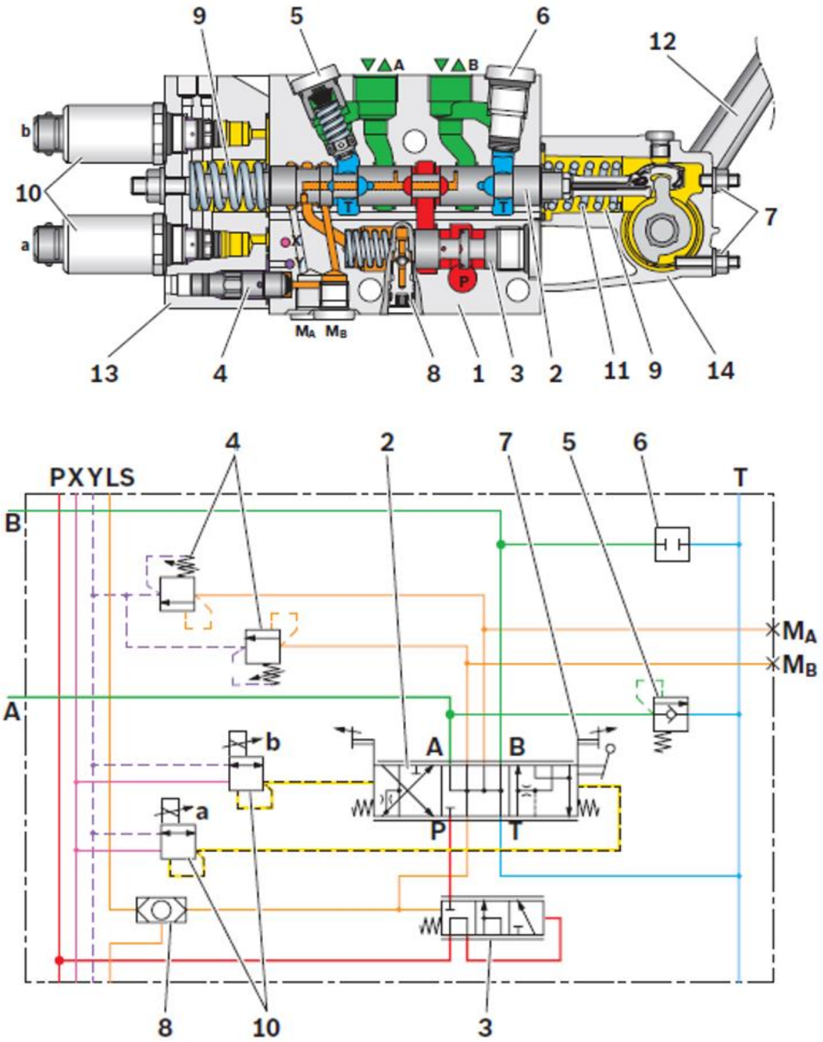


Fig. 8.17 Hydraulic mobile control block M4-1 with input 2-way flow control valve and Load Sensing system [60]

1 – cast iron body, 2 – main (control) valve spool, 3 – 3-way flow control valve, 4 – Load Sensing pressure relief valve, 5 – secondary relief valve with liquid suction function, 6 – plug, 7 – mechanical limitation of spool stroke, 8 – Load Sensing shuttle valve, 9 – spring, 10 – pressure reducing valves for control of main spool, 11 – compression spring, 12 – control lever, 13 – cover plate of side A, 14 – cover plate of side B

Fig. 8.17 shows the M4-12 mobile directional valve, which is used as a sectional Load Sensing directional valve in combination with a fixed or variable displacement hydraulic pump, mainly in the field of forestry and construction machinery, where the most demanding requirements for performance, function and operational reliability are required.

The main slide gate (2), which controls the flow through the valve to the consumer, is controlled by the lever (12) or electro-hydraulically by means of two proportional reducing valves (10). The spool stroke and, therefore, the maximum flow can be limited by means of the mechanical stroke limiter (7). In order to ensure a constant flow through the valve independently of the pressure gradient on the consumer, the pressure compensator (3) is located at the input to the channel P of the main spool (2), which is controlled by the pressure from the working branches A and B through the main spool (2) of the valve. Load Sensing signal is fed through the shuttle valve (8) to the working hydraulic pump. The hydraulic directional valve also contains Load Sensing relief valves (4), which serve to limit the maximum pressure in the working branch A or B. The function of the directional valve in combination with the pressure compensator corresponds to that of a pressure reducing valve. This means that when the required pressure is reached in a given branch of the section, the pressure compensator is closed and the liquid flow to the consumer is interrupted. The output branch from the directional valve A contains the secondary pressure relief valve (5) with liquid suction. Its purpose is to protect branch A against excessive overpressure due to external force acting on the consumer.

8.4 Flow dividers

In cases where two or more hydraulic motors of the same geometric stroke volume are used in a hydraulic system, and their synchronous operation is required even with different loads on these motors, it is possible to use a so-called flow divider. Flow dividing can be realized by means of flow divider valves or volumetric flow dividers.

8.4.1 Flow divider valves

Flow divider valves are of spool valve design and operate on the principle of flow throttling (see Fig. 8.18). The input flow of liquid Q_1 is divided in the valve into two equally large output flows Q_{2A} and Q_{2B}. The flow divider consists of slide valve (1), which performs the function of a pressure compensator and two identical orifices (2), which perform the function of flow measurement. A change in the load on one of the hydraulic motors leads to a change in the output flow (i.e., $Q_{2A} \neq Q_{2B}$) of the divider and, at the same time, causes a change in the pressure gradient on the orifices. This is accompanied by unequal pressures p_1 and p_2, which are applied to the spool faces of the flow divider. Due to the difference in pressures, the balance of forces on the spool will be disturbed. Therefore, the spool is moved to one side or the other (the spool throttles the branch with the higher volumetric flow and, at the same time, opens the branch with the lower volumetric flow). The spool position thus takes into account the current load on the individual hydraulic motors and the spool is pressure balanced. The accuracy of the spool type flow divider valve depends on the flow rate and the load difference on the motors, and ranges from $(3 \div 5)$ %.

A similar function of synchronising the movement of two hydraulic motors can be achieved by connecting the motors in parallel, together with two 2-way flow control valves. Compared to this variant, the flow divider is a simpler, smaller and cheaper solution. However, as with other elements that operate on the flow throttling principle, higher energy losses must be considered when operating a system using the flow divider.

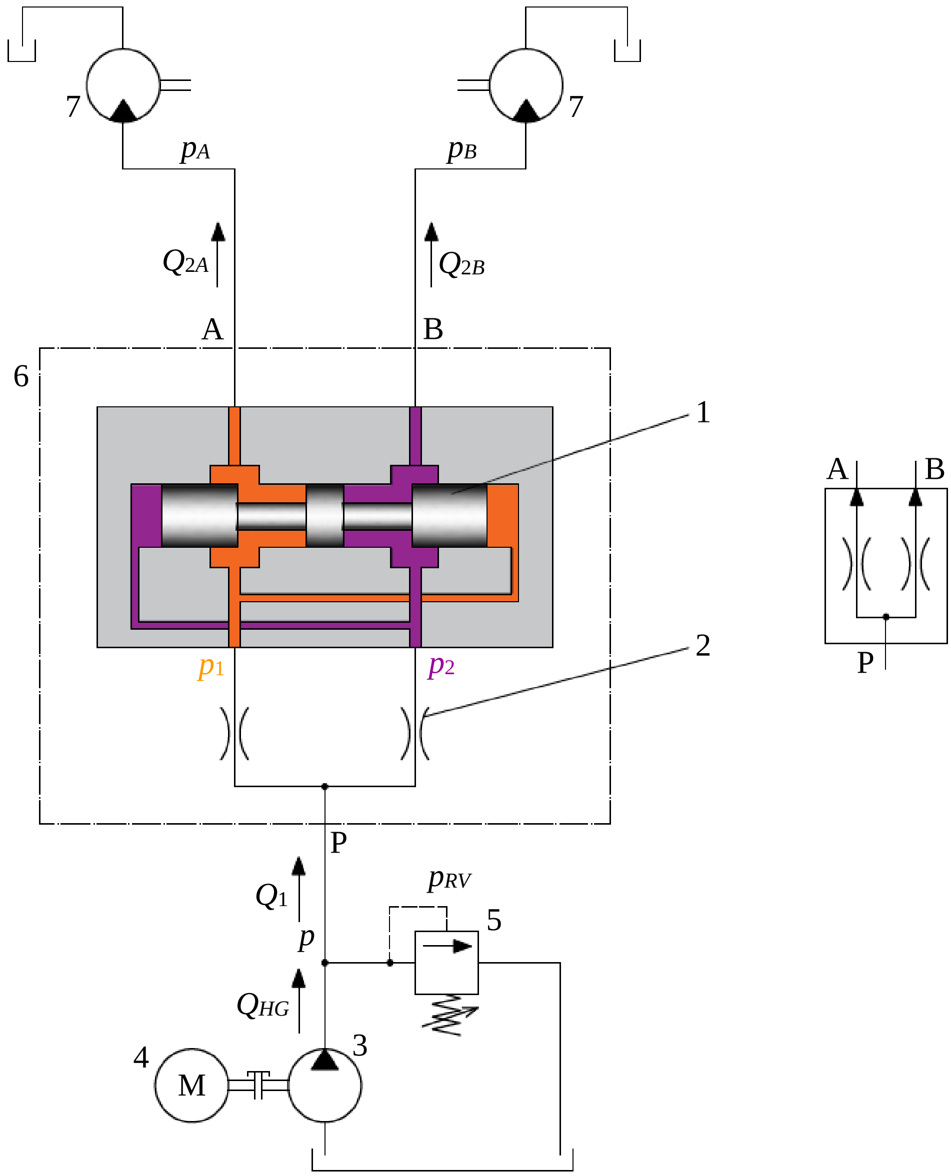


Fig. 8.18 Function principle of flow divider valve

1 – slide valve, 2 – orifice, 3 – hydraulic pump, 4 – electric motor, 5 – relief valve, 6 – flow divider, 7 – rotary hydraulic motors

8.4.2 Volumetric flow dividers

An alternative to the flow divider valve is to use the volumetric principle. Basically, it is a connection of two or more rotary hydraulic motors of the same geometric stroke volume, which are connected by one continuous shaft, as shown in Fig. 8.19. The input flow Q_1 is evenly divided to the individual motors, which rotate at the same speed. Synchronization accuracy and operating parameters are based on the construction and properties of the used converters. The most commonly used are gear converters, which are the cheapest, simplest and characterized by the accuracy of up to (1 ÷ 1.5) % [6].

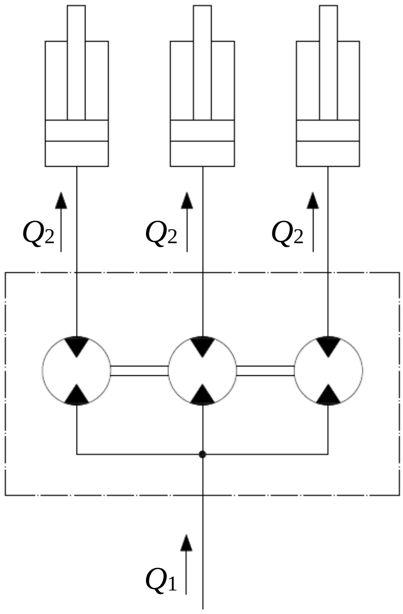


Fig. 8.19 Volumetric flow dividers

In addition, linear synchronised flow dividers can also be used in some applications, see the flow divider (Fig. 8.20). This type of flow divider is used to synchronize the operation of hydraulic motors on presses and lifting equipment. The pressure liquid is fed into the central linear hydraulic motor (1), which consists of four identical hydraulic motors connected to a combined central area. When the central surface is moved, the liquid is precisely displaced from all outputs of the hydraulic motor (1), which are further connected to the linear hydraulic motors (2) located at the corners of the lifting platform. In this way, the same liquid flow to all consumers is guaranteed even under different loads.

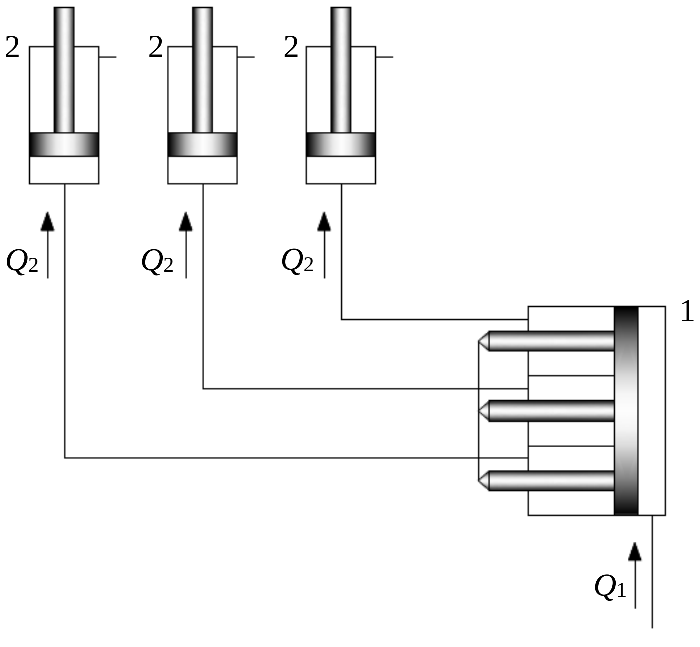


Fig. 8.20 Linear synchronized flow divider

1 – flow divider (central linear hydraulic motor), 2 – consumers (linear hydraulic motors)

9. Pressure control valves

Pressure control valves belong to the basic elements of every hydraulic circuit. In a hydraulic system, they perform a safety function and also serve to control or limit the pressure in a given part of the system. Pressure control valves include pressure relief, constant pressure and pressure reducing valves. In design terms, these valves can be either seat or spool valves. According to the flow rate, they are divided into one-stage (direct operated) and two-stage (pilot operated). In the case of the one-stage valve, the control element is operated directly, usually by a spring. The one-stage valves are used for lower flows, approx. up to 60 $dm^3 \cdot min^{-1}$. For higher flow rates, two-stage (pilot operated) valves are used. The first (pilot) stage is usually controlled by a spring, and the second stage by liquid pressure through the pilot control valve. Leading manufacturers of pressure valves include Bosch Rexroth, Hydac and Danfoss, whose technical data and parameters presented in the product catalogues were used in the preparation of this chapter.

9.1 Pressure relief valves

They are used to protect a system against overload. The pressure relief valve is used to set the maximum allowable pressure in the system or in a given system location. If this pressure is exceeded, the relief valve opens and excess liquid flows through this valve into the tank. The valve must be perfectly tight so that the stiffness of the system is not reduced and energy losses outside its operation are eliminated. For this reason, pressure relief valves of seat design with ball, cone or poppet are used. The pressure relief valves must be sufficiently fast to respond to any pressure peaks in the system. They are typically located parallel to the pressure source (hydraulic pump), are a part of the safety blocks of accumulators, or are located on the annulus side of hydraulic cylinders to protect against possible pressure multiplication. When the liquid flows through the relief valve, energy is always dissipated, and the liquid is intensely heated. The pressure relief valves are assumed to be open (to pass fluid) only for the necessary time. When liquid flows through the pressure relief valve, a pressure gradient (loss) Δp_{RV} is created on this valve. The pressure gradient represents the pressure difference in front of (i.e., p_1) and behind, i.e., p_2) the valve. In the case of a relief valve that is connected to the tank at the outlet, it is possible to consider the pressure p_2 as atmospheric. Then, in relative values, the pressure gradient is equal to the pressure p_1 in front of the valve and corresponds to the pressure p_{RV} set on the relief valve:

$$\Delta p_{RV} = p_1 - p_2 = p_1 - 0 = p_{RV} \,. \tag{9.1}$$

The power loss P_l then corresponds to the volumetric flow Q_{RV} through the relief valve at a given pressure p_{RV}:

$$P_{lRV} = p_{RV} \cdot Q_{RV} \,, \tag{9.2}$$

where P_{lRV} [W] is the power loss during liquid flow through the relief valve, p_{RV} [Pa] is the pressure on the relief valve, and Q_{RV} [$m^3 \cdot s^{-1}$] is the volumetric flow through the relief valve.

Pressure relief valves in hydraulic circuits perform a safety function. They are usually set to a pressure (10 ÷ 30) % higher compared to the maximum operating pressure of the system (e.g., set at constant pressure valve) and are checked regularly.

A one-stage direct operated pressure relief valve and its graphical symbol are shown in Fig. 9.1. The required pressure is set manually by means of the control mechanism (3), which defines the spring preload (2) and its force action on the cone (1). From the other side, the pressure of the liquid acts on the cone. When the pressure of the liquid in the circuit exceeds the allowable value, the force F_L from the liquid overcomes the spring force F_s, the cone moves against the spring which connects the P and T channels and the liquid flows into the tank. Some valves are designed with the damping piston (4), which also serves as a cone guide, improves dynamic properties of the valve and reduces cone oscillations in the seat (6). Cone oscillations can occur, especially at low flow rates and low set pressures. The minimum pressure is defined by the valve manufacturer and usually ranges from (0.15 ÷ 0.45) MPa [11].

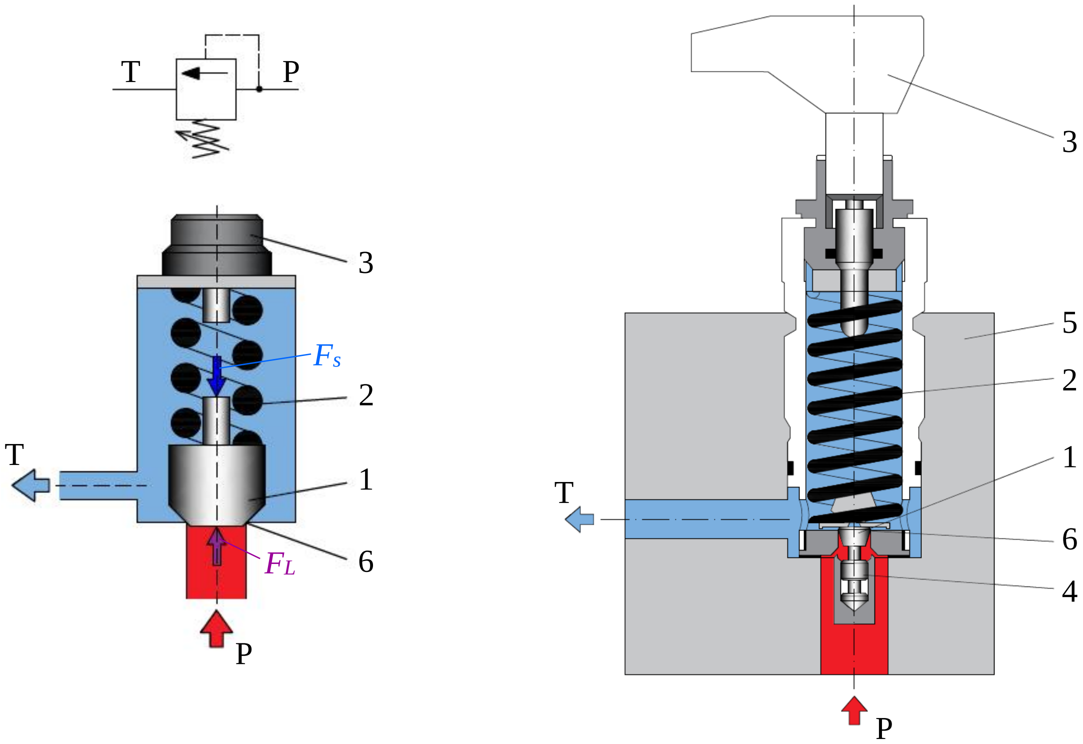


Fig. 9.1 Graphical symbol and force actions on pressure relief valve (left), one-stage direct operated pressure relief valve with cone damping (right)

1 – cone, 2 – spring, 3 – control mechanism, 4 – damping piston, 5 – valve body, 6 - seat

The opening pressure of relief valves should be minimally dependent on the volumetric flow rate and ideally constant over the entire flow range. A static *p – Q* characteristic of the one-stage direct operated pressure relief valve is shown in Fig. 9.2 (left). It is evident that the pressure increases with increasing the volumetric flow rate. This is due to the spring stiffness and valve geometry. The top of the cone can be shaped in some cases, which reduces the valve resistance at higher volumetric flow rates. By modifying this cone shape, the effect of the hydrodynamic force during liquid flow through this valve can be used and thereby reduce the pressure increase at higher volumetric flow rates. The blue curve represents the power limitation of the valve. This is due to local losses in the valve, and the valve cannot be used when the system is operating below the power curve.

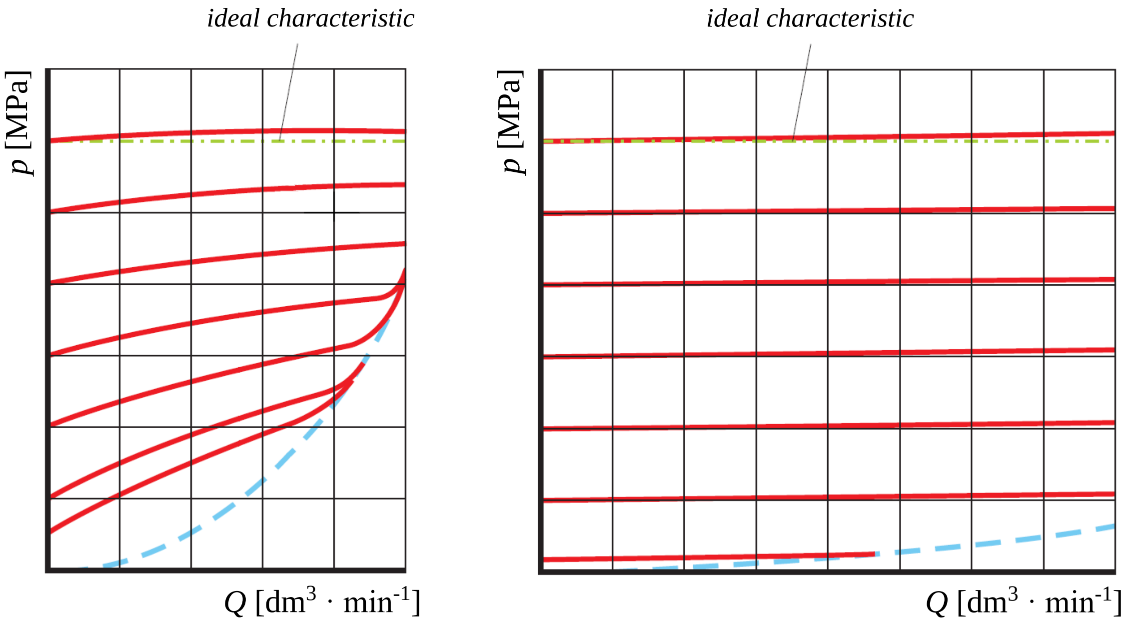


Fig. 9.2 Static p – Q characteristic of one-stage direct operated pressure relief valve (left), characteristic of two-stage pilot operated pressure relief valve (right) [11]

For higher volumetric flow rates, two-stage valves, also called pilot operated valves, are used; see Fig. 9.3 [30]. The first stage consists of a direct operated pilot valve (1) of smaller diameter, which is manually operated (9). In the second stage, there is the power (main) valve (2) of a larger diameter, which is controlled by the liquid pressure. As long as the relief valve is closed (the system pressure is less than the pressure adjusted on the relief valve), the cone (3) of the power valve is pressure balanced (i.e., $p_1 = p_2$). When the pressure p_1 in the system increases above the allowable value, the pressure force of the liquid acting on the cone (5) of the pilot valve overcomes the spring (6) resistance and the cone is pushed out of the seat (11). At this time, the liquid from the system starts to flow through the orifice (7) and the pilot valve (1) into the tank. When the liquid flows through the orifice (7), which acts as a resistance, a pressure gradient Δp_c is created, which changes the force balance on the cone (3) of the power valve (2), because this equation is valid when the fluid flows through the orifice: $p_2 = p_1 - \Delta p_c$. The decrease in the pressure p_2 results in the cone (3) extension of the power stage from the seat (10) (the pressure of the liquid p_1 overcomes the resistance of the spring (4) and the force from the pressure p_2) and the liquid flows through the valve (2) into the tank. The cone (8) is used only to dampen the cone (3) of the power valve.

In the two-stage valve design, only a spring of lower stiffness is required in the power stage. The pressure control is performed only by the pilot valve. In general, two-stage pressure valves have the advantage that they can be unloaded by means of a distributor. In some applications, they are also connected with an unloading distributor, which limits their function as required and allows the liquid flow without resistance.

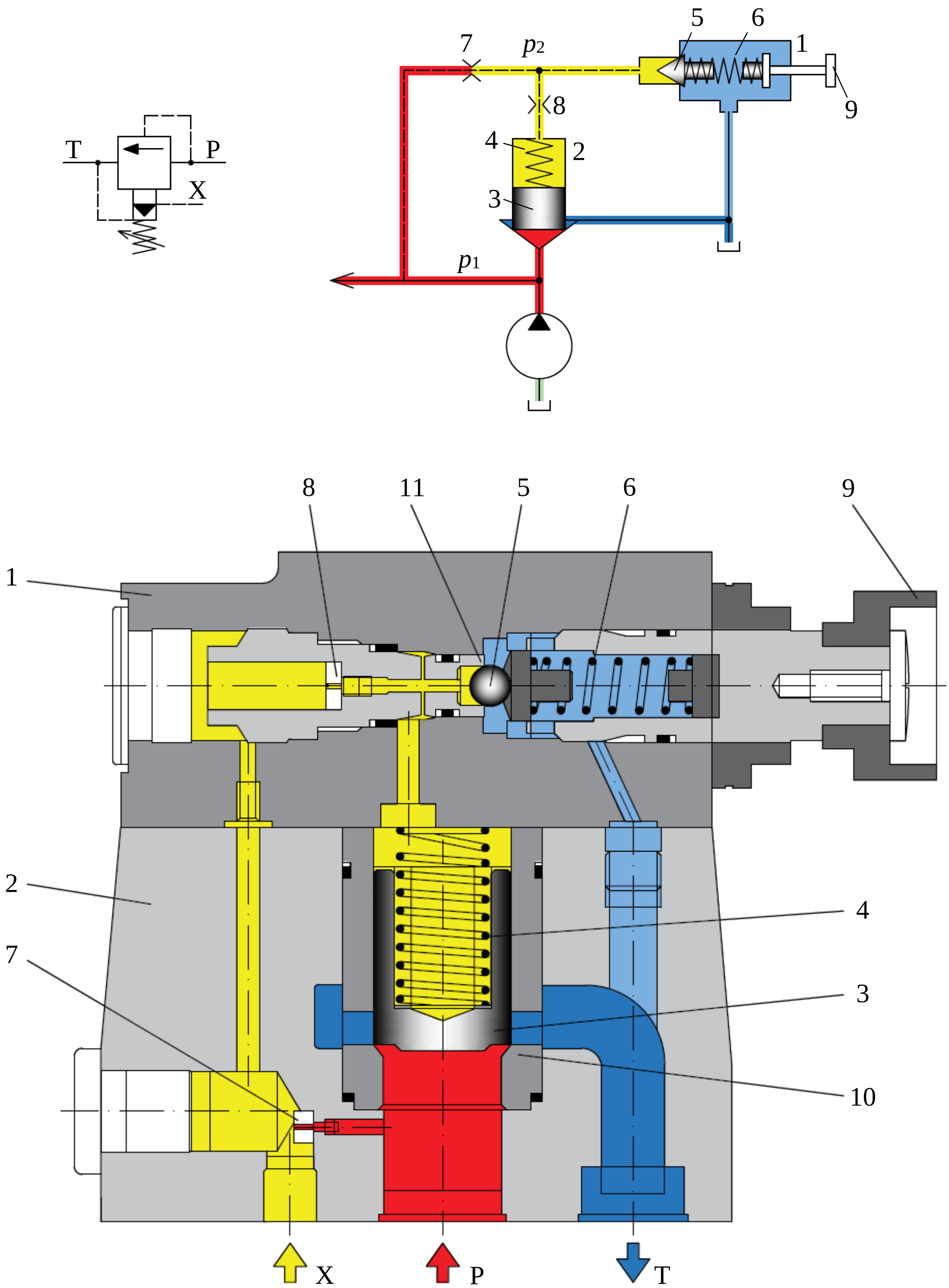


Fig. 9.3 Graphical symbol and construction of two-stage pilot operated pressure relief valve

1 – valve of the first stage (pilot), 2 – valve of the second stage (power), 3 – cone of the power valve, 4 – spring of the power valve, 5 – cone of the pilot valve, 6 – spring of the pilot valve, 7 – control orifice, 8 – damping orifice, 9 – control mechanism, 10 – seat of the second stage valve, 11 – seat of the first stage valve

A static $p - Q$ characteristic of a two-stage pilot operated pressure relief valve is shown in Fig. 9.2 (right). It is evident that compared to the one-stage valves, the magnitude of the opening pressure is less dependent on the flow size, and also the power limitation is less pronounced.

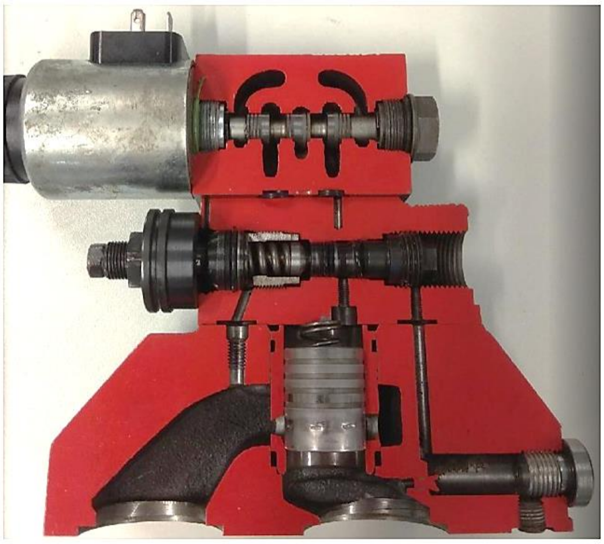


Fig. 9.4 Direct operated pressure relief valve (left), electric discharge pilot operated pressure relief valve (right)

9.2 Hydraulic pressure relief valves (constant pressure valves)

They are principally and constructively similar to pressure relief valves, but they do not perform a safety function in a hydraulic system. Their purpose is to maintain a constant pressure at the input to the constant pressure valve during a continuous liquid flow through this valve. They do not have to be perfectly tight, which makes it possible to use a spool design with these valves. Although leakage occurs on the spool, its movement is more damped, which can be used in control systems. Other properties are similar to pressure relief valves. When these valves are used in hydraulic circuits, the liquid is overflowed through this valve. Thus, also, in this case, it is true that there is heat generation and energy dissipation during the liquid flow through the constant pressure valve. Typical applications of the constant pressure valves are, for example, in circuits with throttle valves, or in filling hydraulic pumps.

A possible solution for the one-stage direct operated constant pressure valve of spool construction is shown in Fig. 9.5. The orifice (3) is used to dampen the spool (1), and the control pressure on the spool edge is internally fed through a separate channel in this case. The control pressure can also be fed externally; this is also reflected in the hydraulic symbol of this element.

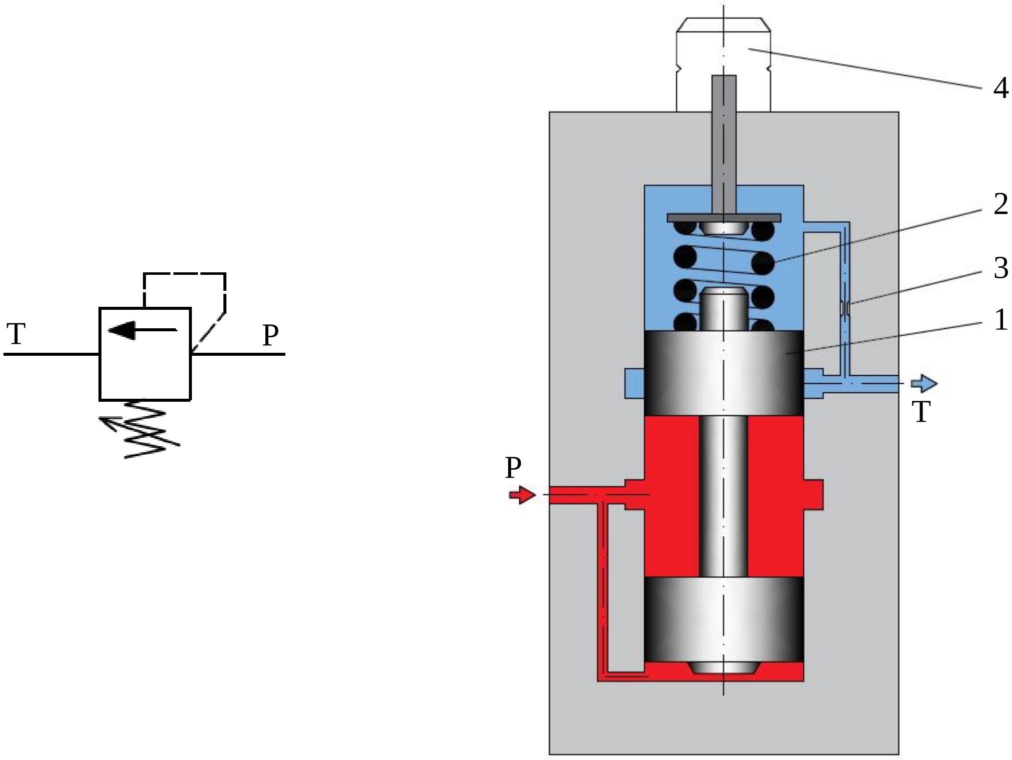


Fig. 9.5 One-stage direct operated constant pressure valve of spool construction

1 – spool of the valve, 2 – spring, 3 – orifice, 4 – control mechanism

The constant pressure valves are also manufactured in the two-stage version. The schematic of a two-stage pilot operated constant pressure valve is shown in Fig. 9.6.

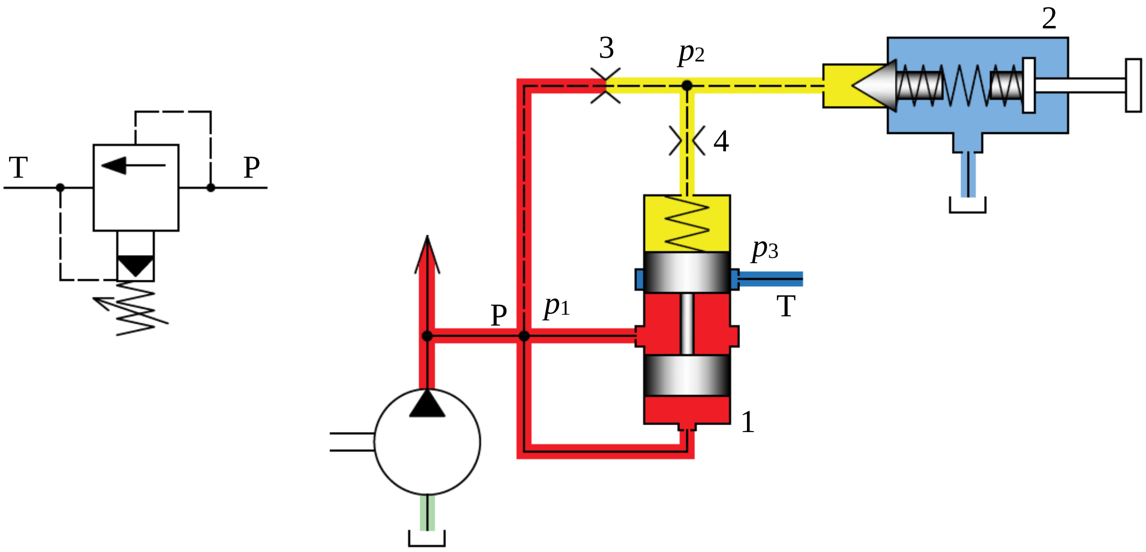


Fig. 9.6 Two-stage pilot operated constant pressure valve of spool construction

1 – power (main) stage, 2 – control valve, 3 – control orifice, 4 – damping orifice

9.3 Pressure reducing valves

As their name indicates, they are used to reduce the input pressure to a lower output pressure and maintain this pressure on the valve output at a constant value. Similarly to pressure relief and pressure constant valves, they are manufactured as direct operated and pilot operated. Pressure reducing valves use spool and seat constructions. Compared to pressure relief and pressure constant valves, they are arranged in series with the consumer. In the basic position, they are open, and the control pressure is detected from the valve output (as can be seen on the hydraulic symbol). 2-way direct operated pressure reducing valve is schematically shown in Fig. 9.7 (left). The required output pressure is adjusted by the spring (1). The liquid flows through the valve in the direction from P to A. The output pressure creates a negative feedback, where the pressure p_2 is fed to the bottom surface of spool (2). If the pressure p_2 is higher compared to the required (adjusted) value, the pressure force of the liquid acting on the spool surface overcomes the spring resistance. The gate valve is subsequently moved up, and the liquid flow through the valve is decreased. Therefore, the valve only passes the amount of liquid that is drawn by the consumer at the required pressure. The valve is fully closed in the case of liquid interruption by the consumer (e.g., when the hydraulic cylinder reaches the end position). Output Y is used to discharge leakage flows caused by spool leakages of the valve.

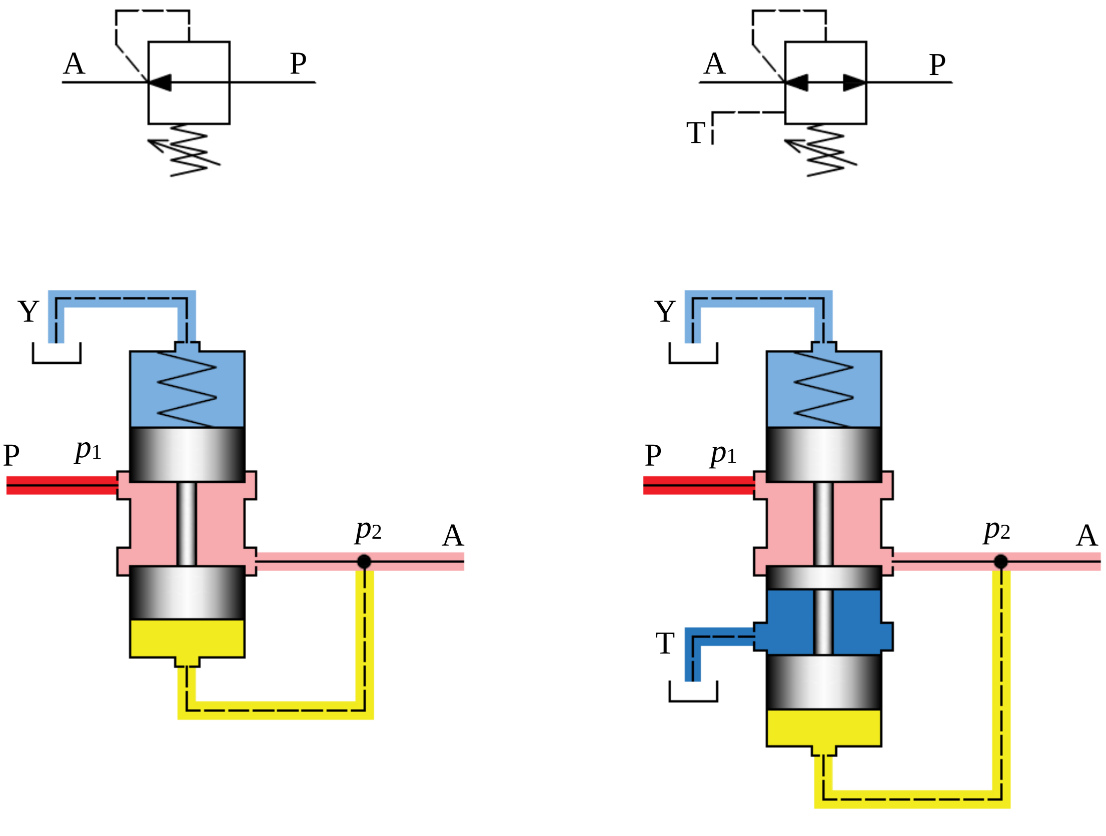


Fig. 9.7 Schematic of 2-way direct operated pressure reducing valve of spool construction (left), 3-way valve (right)

As shown in Fig. 9.7 (right), the pressure reducing valves are also manufactured in 3-way design. The function principle is similar to the previous case. The third valve channel allows the connection of the outlet behind the valve to the tank (channels connected in the direction A→T). It is used in cases where the output behind the valve may generate a back pressure p_2 higher than the input pressure p_1. In this case, the A→T channels are connected, and the pressure reducing valve works as the relief valve in the given direction.

The pressure reducing valves must have a fast output pressure response to flow changes. At the same time, it is necessary to ensure good damping of the spool, which is usually realized by an orifice in the Y port. The pressure loss Δp_{lrv} during the liquid flow through the pressure reducing valve is given by the formula:

$$\Delta p_{lrv} = p_1 - p_2 \ . \tag{9.3}$$

And the power loss P_{lrv} at the pressure reducing valve is calculated as follows:

$$P_{lrv} = \Delta p_{lrv} \cdot Q_{rv} \ , \tag{9.4}$$

where P_{lrv} [W] is the power loss during liquid flow through the pressure reducing valve, Δp_{lrv} [Pa] is the pressure loss on the pressure reducing valve, and Q_{rv} [m^3·s^{-1}] is the volumetric flow through the pressure reducing valve.

According to equation (9.4), it is obvious that the power loss of the pressure reducing valve increases with the volumetric flow and the pressure gradient. It is, therefore, necessary to limit large pressure reductions when using the pressure reducing valve and to limit the operation of the valve to the necessary long time.

A static p_2 - Q characteristic of the pressure reducing valve on Fig. 9.8 (left) expresses the dependence of the output pressure p_2 on the volumetric flow rate Q through the pressure reducing valve at a constant pressure p_1. The dependence of the output pressure p_2 on the input pressure p_1 of the pressure reducing valve at a constant flow rate is shown in Fig. 9.8 (right).

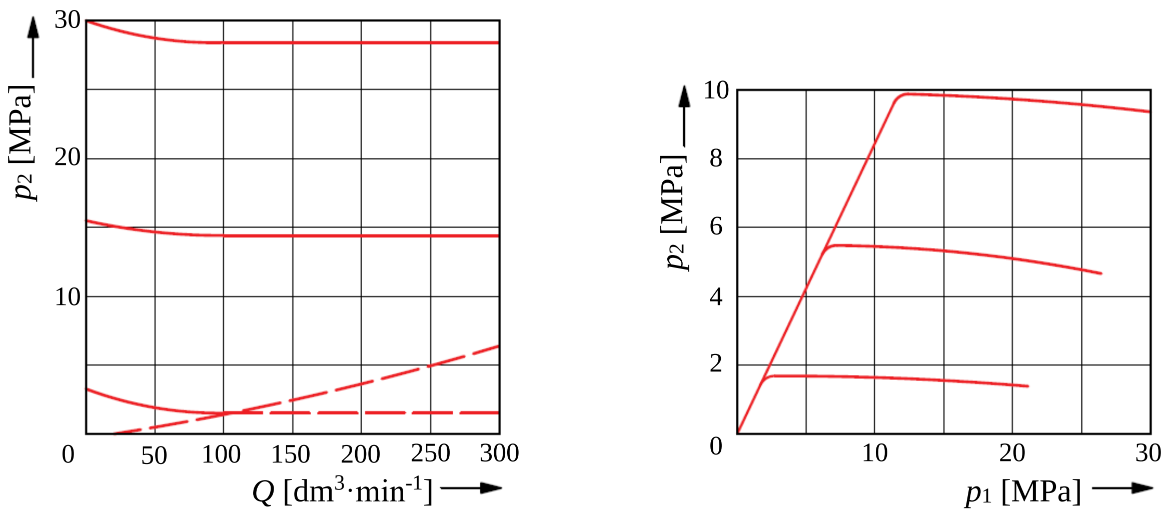


Fig. 9.8 Static characteristics of pressure reducing valves

In addition to these direct operated pressure reducing valves, pilot operated pressure reducing valves are also used in hydraulic systems and can be 2-way or 3-way. Similar to pressure relief valves, they consist of two stages and are mainly used for higher volumetric flow rates.

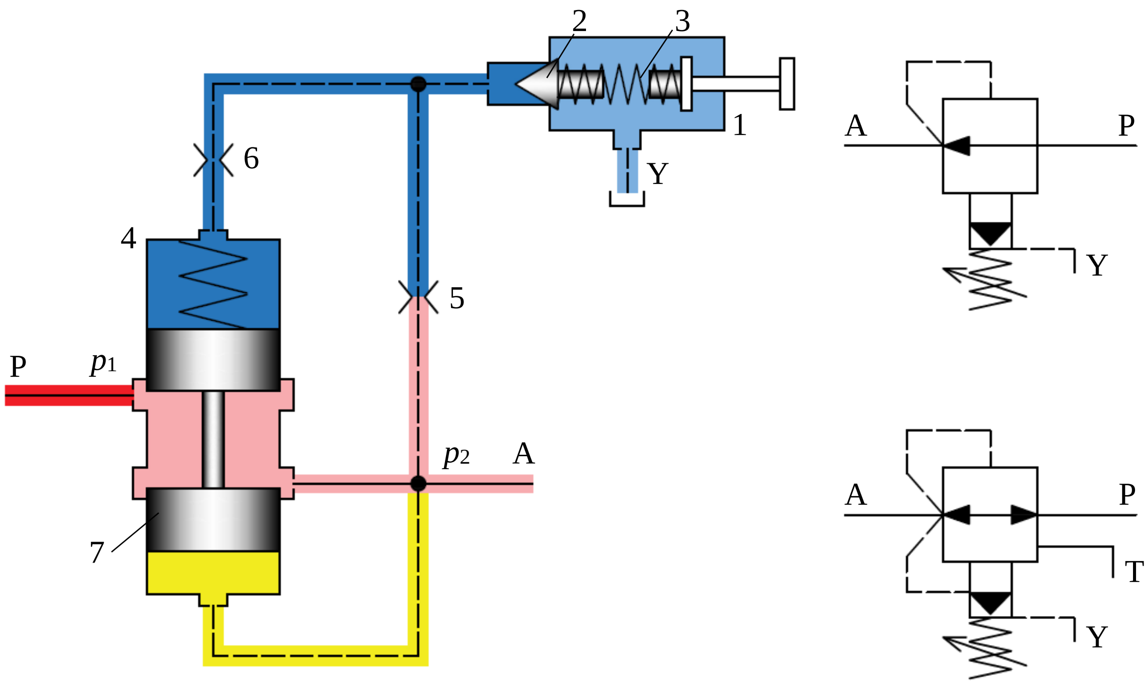


Fig. 9.9 Schematic of 2-way pilot operated pressure reducing valve (left), its graphical symbol (top right), a graphical symbol of 3-way pilot operated pressure reducing valve (bottom right)

1 – relief (pilot) valve, 2 – relief valve poppet, 3 - relief valve spring, 4 – spool of the reducing valve, 5 – control orifice, 6 – damping orifice, 7 - spool

A 2-way pilot operated pressure reducing valve is schematically shown in Fig. 9.9. The first stage consists of the operated ball (or poppet) seat pressure relief valve (1). The reduced pressure is adjusted by the preload of the spring (3). The power part in the second stage consists of the 2-way spool valve (4). When the liquid flows through the orifice (5) (i.e., when the pilot valve is open), a pressure gradient is created by means of the control orifice (5) on the spool (7) of the pressure reducing valve. The orifice (6) is used to dampen the spool movement. The function principle is similar to the previous valves and should be obvious when considering the valve function in Fig. 9.3 and Fig. 9.7.

Fig. 9.10 Seat construction direct operated pressure reducing valve (left), seat construction pilot operated pressure reducing valve (right)

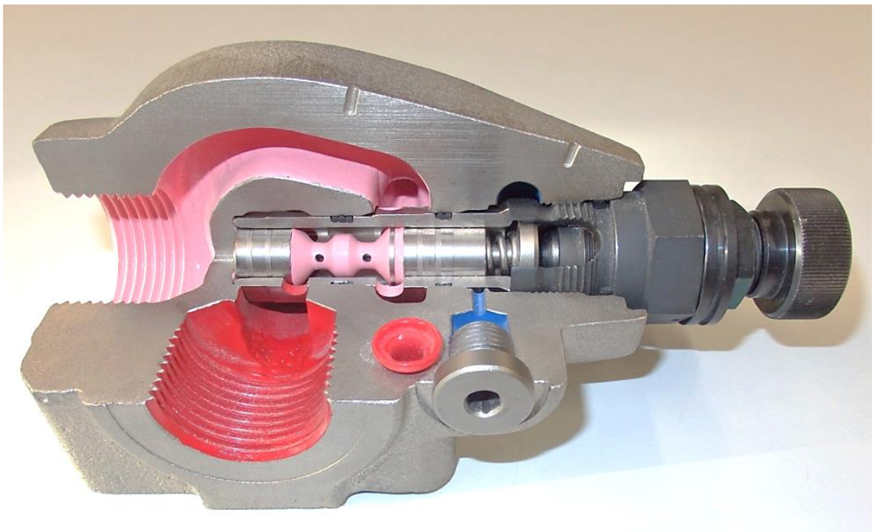


Fig. 9.11 Spool construction pilot operated pressure reducing valve

An example of the use of pressure valves in hydraulic circuits is given in Fig. 9.12 (left). The additional filling hydraulic pump (1) supplies the liquid to the low-pressure branch of the hydrostatic converter with the hydraulic pump (2). The constant pressure valve (3), which is adjusted to low pressure (filling pressure 15 ÷ 20 bar), is connected in parallel to the filling hydraulic pump. The pressure relief valve (4) adjusted to the maximum system pressure is connected in parallel behind the main hydraulic pump. In the circuit shown in Fig. 9.12 (right), there is the linear hydraulic motor (7) in the main branch, which operates at a higher pressure. In the secondary branch, there is a rotary hydraulic motor (6), which operates at a lower pressure. Thus, the working pressure at the input of the rotary hydraulic motor is reduced by means of the pressure reducing valve (5).

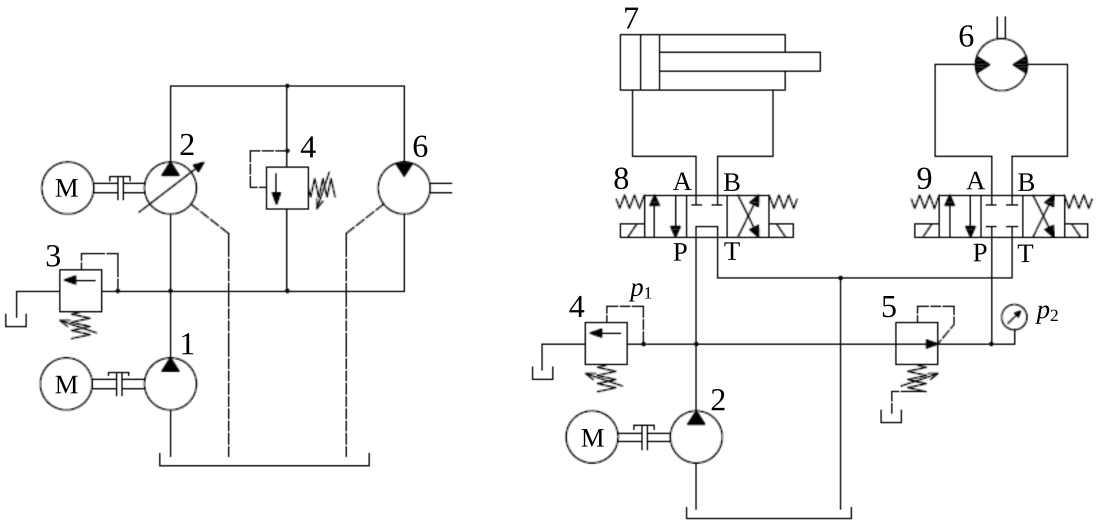


Fig. 9.12 An example of using pressure valves in circuits

1 – filling hydraulic pump, 2 – main hydraulic pump, 3 – pressure valve, 4 – pressure relief valve, 5 – pressure reducing valve, 6 – rotary hydraulic motor, 7 – linear hydraulic motor (cylinder), 8 and 9 – directional valves

9.4 Pressure sequence and unloading valves

These valves belong to the group of pressure valves. Their name is based on the function that they perform in hydraulic systems. They are manufactured in both one-stage and two-stage designs, which is more common. The graphical symbols of these valves are shown in Fig. 9.13.

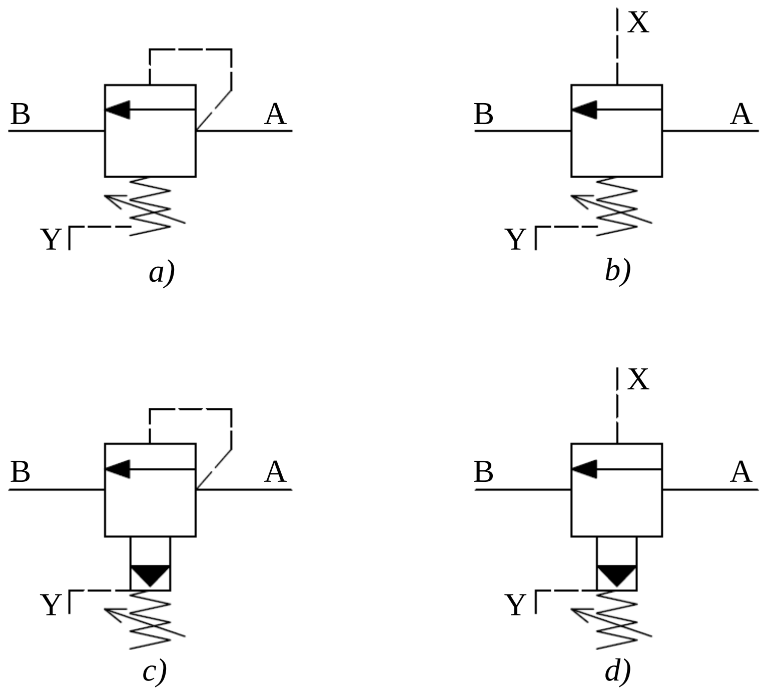


Fig. 9.13 Graphical symbols of pressure sequence and unloading valves
a) direct operated valve with internal pilot pressure oil feed (X), b) direct operated valve with external pilot pressure oil feed (X), c) pilot operated valve with internal pilot pressure oil feed (X), d) pilot operated valve with external pilot pressure oil feed (X)

The pressure input X, which serves as a comparison value in relation to the set pressure on the valve, can be realized internally or it can be externally controlled, and the valve switching is then achieved on the basis of the pressure in another part of the system. These valves are usually used with a parallel connected check valve that can be integrated into the valve construction.

Pressure sequence valves are used to connect pressure to a specific part of the system with respect to system requirements. The pilot operated pressure sequence valve is shown schematically in Fig. 9.14 (right) [6].

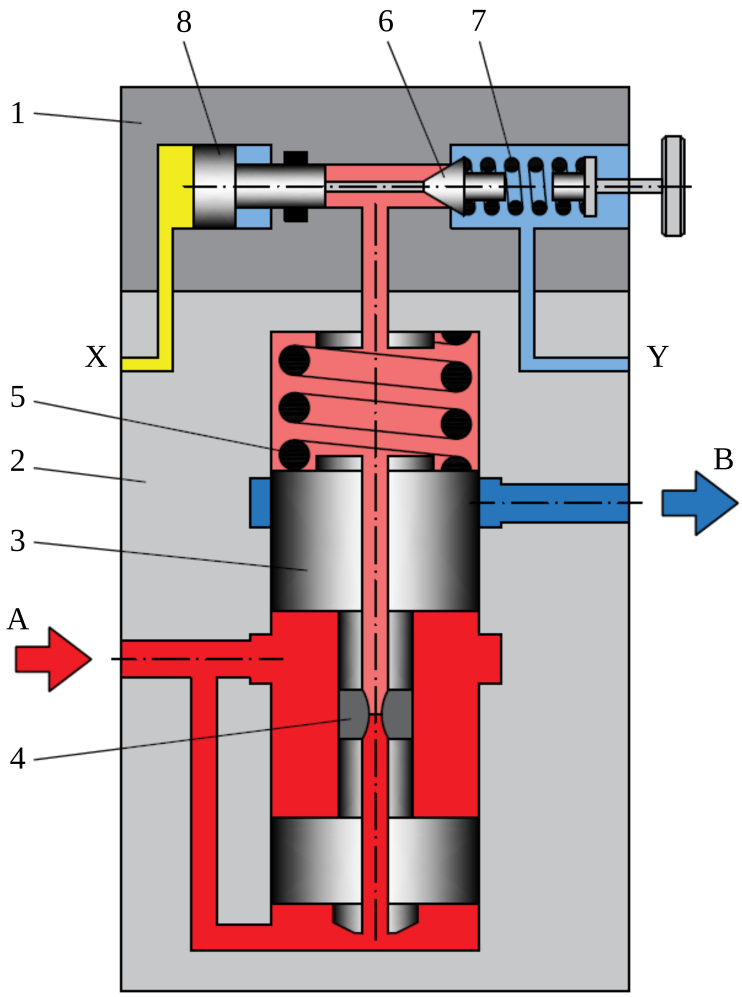


Fig. 9.14 Pilot operated pressure sequence valve with external pilot pressure oil feed (X)

1 – first (control) stage of the valve, 2 – second (power) stage of the valve, 3 – spool, 4 – orifice, 5 –power valve spring, 6 – cone, 7 – control valve spring, 8 - piston

In this case, the spool (3) is drilled, and the control orifice (4), whose function is the same as that of the orifice (7) in Fig. 9.3, is located in this spool. In the first stage of the valve (1), the piston (8) is located against the cone (6) and the spring (7). The external pressure is fed to the piston surface through the X input. If the pressure acting on the piston (8) exceeds the pre-set spring (7) resistance, the cone (6) will be pushed out of the seat. This allows the liquid flow in the direction A→Y through the bore in the spool (3). On the spool (3), due to the resistance of the orifice (4), the pressure balance is disturbed, the power stage spool overcomes the resistance of the spring (5) and channels A and B are subsequently connected.

When using the pressure sequence valves, energy is dissipated on the same principle as described above for the pressure relief valves. Their function can also be replaced using other hydraulic elements.

The possible use of the pressure sequence valve is schematically shown in Fig. 9.15. Two identical hydraulic motors (1) and (2) are located in the circuit, where hydraulic motor (1) is loaded by an external force *F*. The required function of this system is that the piston rod of the (1) must be extended first, and only after reaching the end position the piston rod of the (2) starts to extend. If both motors were connected to a pressure source (3) without using the pressure sequence valve (7), the piston rod of the (2) would first start to move because this piston rod is not loaded. In this system, the pressure p_2 adjusted on the pressure sequence valve (7) must be greater than the pressure p_1 required to extend the piston rod of the (1). In the initial state, no liquid flows through the sequence valve (7) to the hydraulic motor (2) and the volumetric flow rate through the sequence valve is closed. When the piston rod of the hydraulic motor (1) reaches the end position, the pressure p_1 increases to the value adjusted on the sequence valve (7), which is subsequently opened and the liquid flows to the motor (2). The parallel connected check valve (8) ensures bypass of the sequence valve during the reverse movement of the hydraulic motor (2). It is evident that the same function could be achieved by using constant pressure valves, directional valves and switches (sensors) of the piston rod.

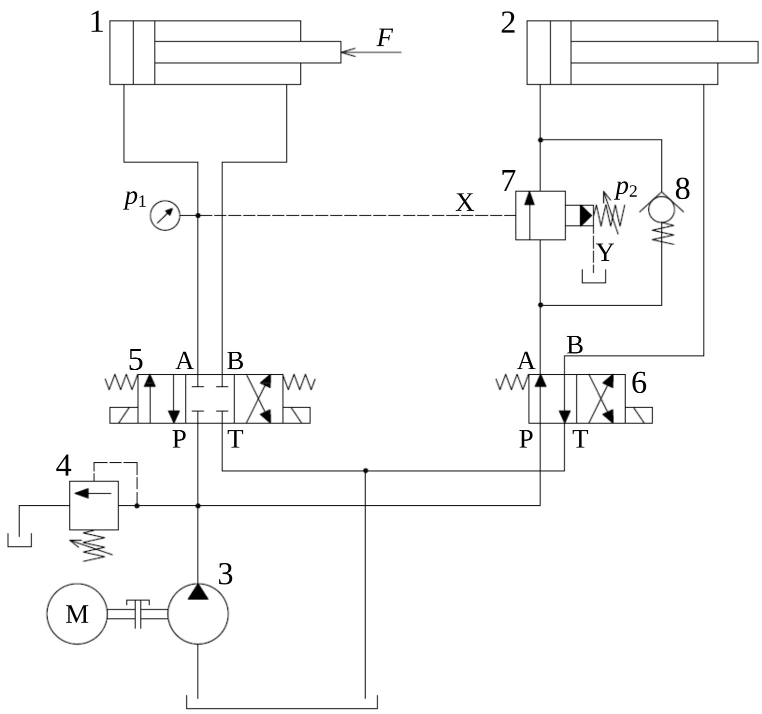


Fig. 9.15 An example of using the pressure sequence valve

1 – linear hydraulic motor loaded by external force, 2 – linear hydraulic motor not loaded, 3 - hydraulic pump, 4 – pressure relief valve, 5 and 6 – directional valve, 7 – sequence valve, 8 – check valve

Unloading valves (often referred to as DA valves in practice) are used to disconnect pressure in specific system parts. The possible applications of these valves are shown in Fig. 9.16 and Fig. 9.17. In the system shown in Fig. 9.16 (left), two hydraulic pumps are used, while a similar principle can be used, for example, in a hydraulic press. The hydraulic pump (1) is high pressure and delivers only a small liquid flow. The hydraulic pump (2) is low pressure and delivers a high liquid flow. The two hydraulic pumps are connected by a common shaft. The opening pressure p_2 is adjusted on the unloading valve (5). At the beginning of the working cycle, the piston rod of the hydraulic motor (3) must approach the workpiece. For the movement of the piston rod, a relatively low pressure p_1 is required, which is lower compared to the pressure adjusted at the unloading valve (5), which remains closed. The flows from both hydraulic pumps are fed to the hydraulic cylinder (3), whose piston rod begins to extend at a higher velocity (rapid displacement). When the piston rod contacts the workpiece, the system resistance increases and the pressure p_1 increases. When $p1 > p2$, the unloading valve is opened and the hydraulic pump (2) is disconnected from the system. This hydraulic pump is now unloaded and the liquid flows from this pump into the tank through the unloading valve with only minimal resistance. For this reason, the extension velocity of the piston rod is decreased, because the liquid is now supplied only to the hydraulic cylinder by a lower volumetric flow from the hydraulic pump (1). The reduction in the velocity of the piston rod movement is not a deficiency. On the contrary, it is usually necessary to apply high forces at relatively low velocities during pressing processes.

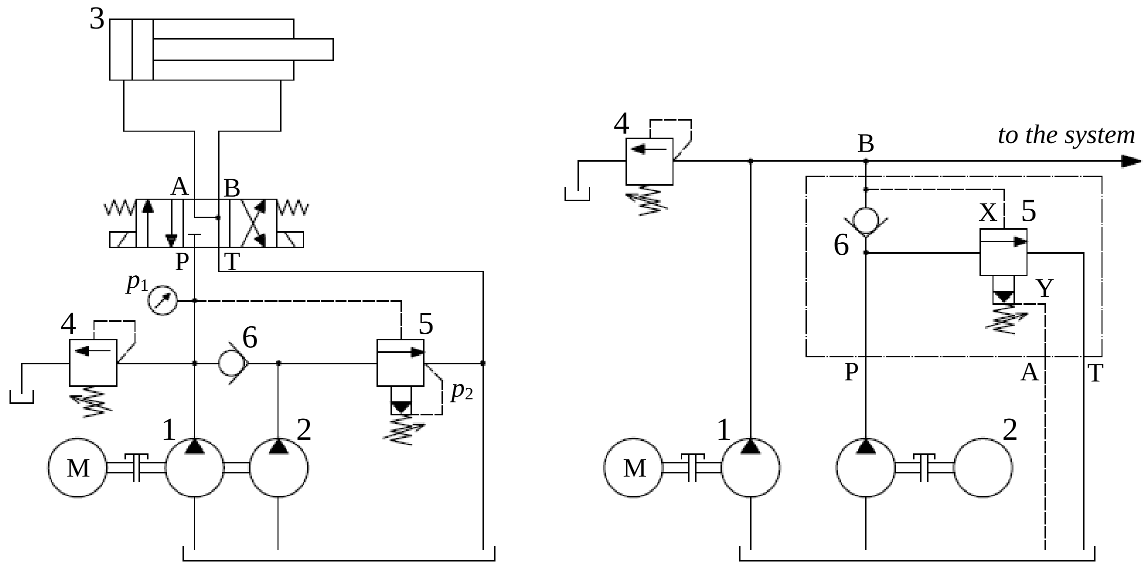


Fig. 9.16 An example of using the pressure unloading valve in a hydraulic press circuit

1 – high pressure hydraulic pump, 2 – low pressure hydraulic pump, 3 – hydraulic motor, 4 – relief valve, 5 – pressure unloading valve, 6 – check valve

The unloading valve is also used in systems with accumulators, as shown in Fig. 9.17 (left). The hydraulic pump (1) fills the accumulator (7). When the pressure in the system reaches the required value, the unloading valve (3) is switched on and disconnects the hydraulic pump from the pressure branch of the system. The unloading valve makes it possible to unload the hydraulic pump without excessive energy dissipation.

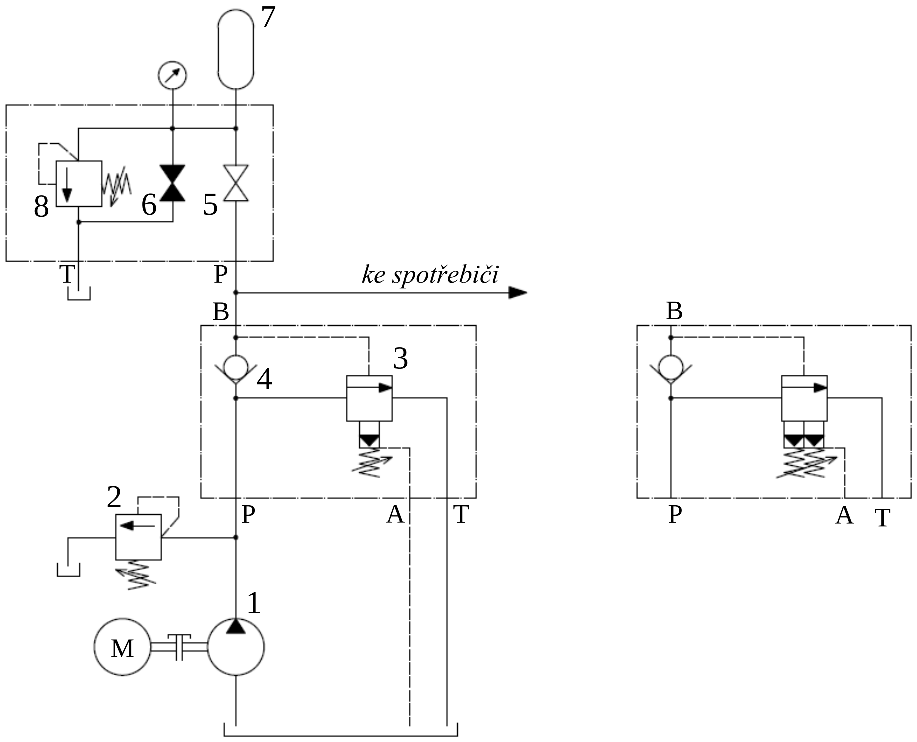


Fig. 9.17 An example of using the pressure unloading valve in an accumulator circuit (left), supply valve – accumulator charging valve (right)

The so-called supply valve (also known as accumulator charging valve) Fig. 9.17 (right), which is more complex and consists of two pressure valves, is very often used in circuits with accumulators. By connecting and disconnecting an accumulator from a pressure source, this valve respects the maximum and minimum pressure requirements in a given system.

10. Hydraulic accumulators

As the name suggests, their main function is to store the liquid's pressure energy. If there is an excess of this energy in a hydraulic system, it can be stored in an accumulator and released back into the system when its consumption increases. In order to store this liquid pressure energy, the accumulator transforms this energy into another energy type (positional energy or, more commonly, the deformation energy of compressed gas or spring). Hydraulic accumulators have a similar function to electric accumulators and are used in hydraulic systems,, e.g. due to energy savings or to eliminate undesirable dynamic phenomena.

Possible functions of the accumulator are:

Coverage of unstable liquid consumption by a hydraulic mechanism**.** This is a common use case of accumulators. The hydraulic pump is commonly dimensioned to the nominal flow rate according to the working cycle of a mechanism. In the case of lower consumption by a given system, the accumulator is filled. In the case of a short-term increase in consumption above the nominal value, the difference in required energy is supplied by the accumulator. In this case, the accumulator enables to use a hydraulic pump with a smaller geometric stroke volume in the system. Therefore, a lower input power of the hydraulic pump is required.

Emergency power source in case of failure of the primary power source. This situation can occur, for example, in the case of a failure of the hydraulic pump or in a power cut. The accumulator will usually complete the work cycle or put the device in a safe position.

Compensation of flow losses (leakages) and **maintenance of constant pressure.** In applications where it is necessary to maintain a constant pressure for a longer period of time (e.g., workpiece clamping, load balancing), it is possible to use an accumulator in the system that can compensate for leakages of a hydraulic motor. The advantage of this connection is also the possibility to unload a hydraulic pump or to use a start/stop system.

Balance of volume changes due to changes in liquid pressure or temperature (volume compressibility and thermal expansion of liquids).

Damping of shocks and pulsations. Hydraulic shocks can be caused by a quick valve repositioning, sudden stops or changes in the orientation of the hydraulic motor movement. In both cases, the excess energy can be absorbed by the accumulator. Flow pulsations can be caused by the unsteady operation of the hydraulic pump or its construction. These pulsations can be compensated by the accumulator, which is connected to the system close to the hydraulic pump.

For example, the accumulator of mobile devices can be used as a **pneumohydraulic spring**. The accumulators are also used in various energy recuperation and **energy-saving systems**, for example for manipulation with heavy loads (load lowering) or in motor vehicles (public bus transport). In these cases, the energy is stored in the accumulator when the device is braking and the energy is released when needed (restarting the device).

It is obvious that accumulators can perform many different functions in hydraulic systems and different accumulator constructions are used depending on the required function and application. The basic division of accumulators can be made according to the method of energy accumulation, i. e. into **mechanical** and **gas** accumulators. In the case of mechanical accumulators, the change of potential energy or spring compression is used. This group includes

weight and **spring** loaded accumulators. In the case of the gas accumulators, the change of the gas internal energy is utilized, where according to the use of the separating element (at the interface between the liquid and the gas) the accumulators are divided into accumulators **with and without** (i.e., with direct contact) **a separating element between the liquid and the gas**. In this case, a **piston**, **bladder** or **membrane** can be used as the separating elements. The basic division of hydraulic accumulators is graphically shown in Fig. 10.1.

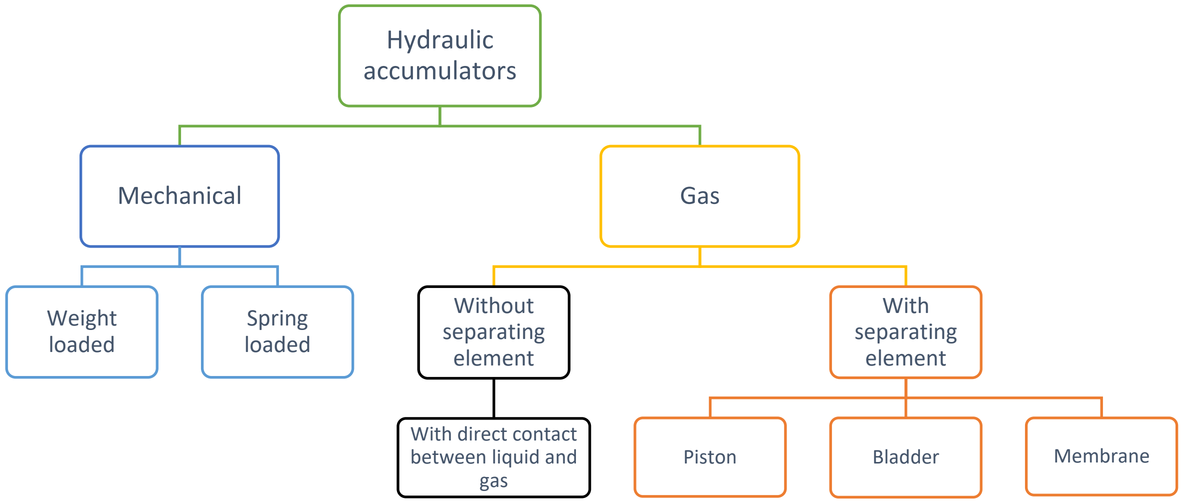


Fig. 10.1 Basic division of hydraulic accumulators

10.1 Weight loaded accumulators

This is the oldest construction type. The accumulator consists of a cylinder and a plunger to which a material load is connected. The liquid pressure energy is accumulated by changing the potential energy of the piston and the load. The advantage of this design is the ability to draw a constant outlet fluid pressure over the entire stroke length (see Fig. 10.2), which distinguishes it from all other types of accumulators. The useful volume of the accumulator is given by its stroke and the cylinder diameter [23], [61].

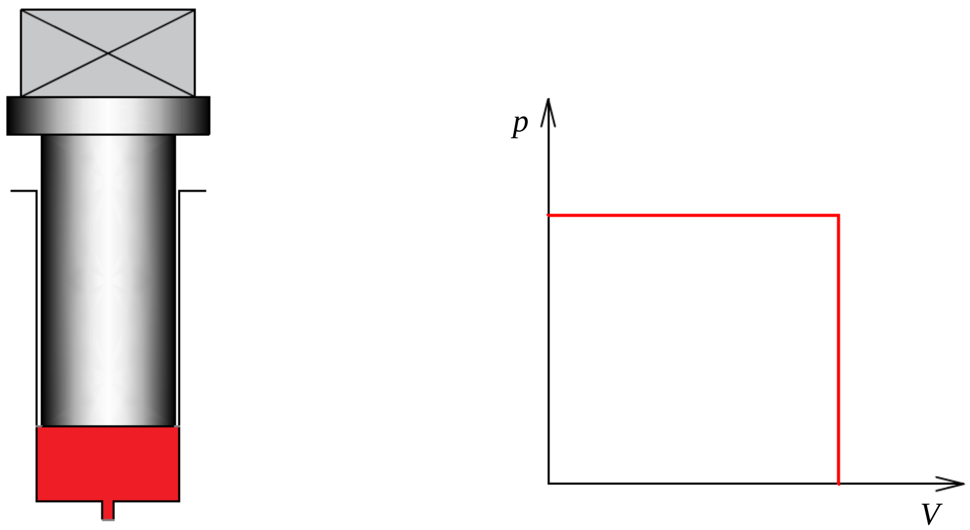


Fig. 10.2 Weight loaded accumulator

The big disadvantage of this accumulator is its larger dimensions and also its weight. Due to the inertia of masses, pressure shocks can occur, especially at higher velocities. The accumulators are designed for working speeds of plunger movement (0.3 ÷ 0.5) $m \cdot s^{-1}$, maximum volumes of 1500 dm^3 and working pressures up to 35 MPa. Their mounting position is vertical. They were previously used, for example, in forming machines, but nowadays they are practically not applied.

10.2 Spring loaded accumulators

The spring loaded accumulator consists of a cylindrical container, piston and spring. When the liquid is fed under the piston area, the spring is compressed by the piston as the pressure of the liquid increases. In this case, the pressure of the liquid is not constant but is given by the spring characteristic, which is time dependent.

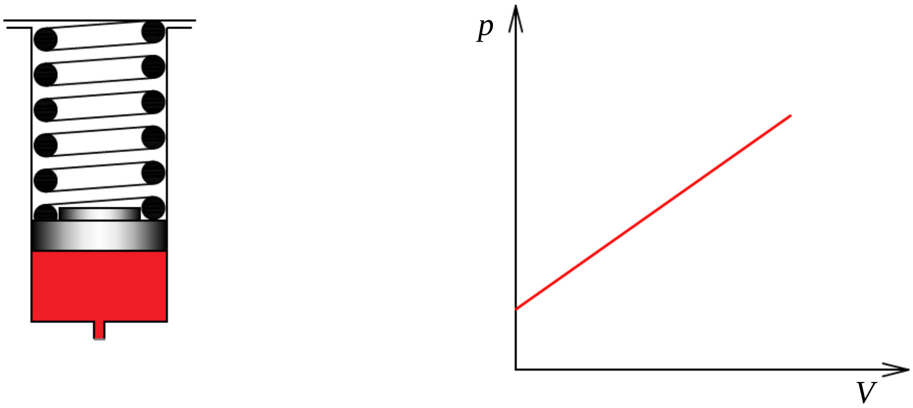


Fig. 10.3 Spring loaded accumulator

They are suitable for small volumes up to 50 dm^3 and lower pressures from 3 to 5 MPa. Their mounting position is arbitrary. They are sometimes used as pressure peak dampers but with lower reliability due to spring cracking. Nowadays, they are practically not used; their function is replaced by gas accumulators in most applications.

10.3 Gas accumulators

Gas accumulators use gas compressibility to accumulate energy. The compressed gas is like a spring and the function is based on the basic laws of thermodynamics. The internal space of the accumulator can be divided into gas and liquid parts, where the interface between the gas and liquid can be (and usually is) defined by a separating element. For safety reasons, the accumulators do not use air, but an inert gas, usually technical nitrogen. When selecting the correct type of accumulator for a given application, in addition to the required flow rate (useful volume of the accumulator) and the working pressure of the device, it is also necessary to consider the velocity of working cycles of the hydraulic system, the used liquid type and, for some types of accumulators, the limitations in the ratio of working and filling pressures. Gas accumulators are currently the most widely used, and according to the type of separating element, there are the following construction designs.

10.3.1 Gas accumulators without separating element between liquid and gas

In this basic design, the liquid and gas are in direct contact without using a separating element. However, the direct contact of the gas with the liquid has significant disadvantages. The gas dissolves (absorbs) in the liquid, thereby affecting its properties. This absorption is even more pronounced at higher pressures. For this reason, these accumulators are practically not used in modern high-pressure oil hydraulics. The possibility of using these accumulators is currently only in low-pressure water hydraulics. In the past, they were also used for high pressures up to 45 MPa and large volumes (even 12,000 dm^3 when more accumulators are connected in parallel) and powers of heavy forming machines, presses and metallurgical equipment.

10.3.2 Piston accumulators

The piston accumulator (see Fig. 10.4) consists of the steel cylindrical container (1) in which the piston (2) moves, which is used as a divider between the gas and liquid parts. The interior of the accumulator is defined by the cover (3) on the gas side and the cover (4) on the liquid side. The gas filling valve (5) is located in the upper cover. Both covers are equipped with a static seal (6). The piston seal (7) is similar to the hydraulic cylinders. The cylindrical container and lids are fixed using the closing nut [11].

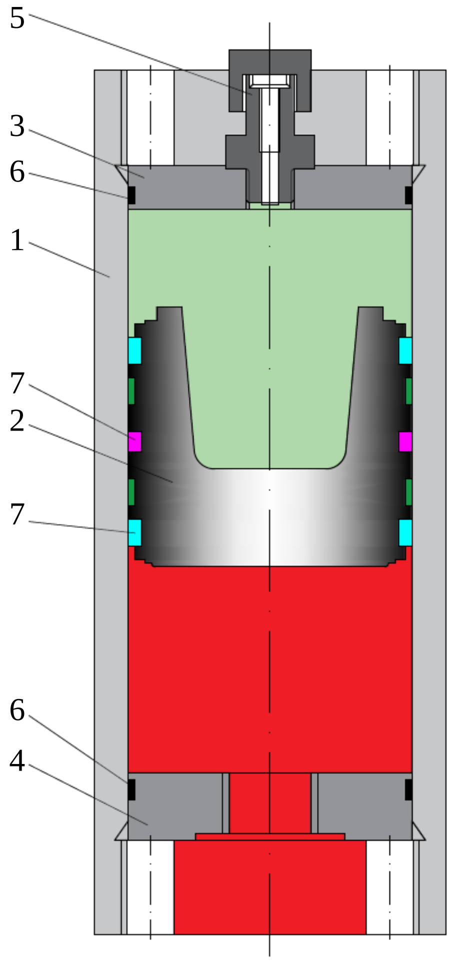


Fig. 10.4 Piston accumulator

1 – cylindrical container, 2 – piston, 3 – cover on the gas side, 4 – cover on the liquid side, 5 – gas filling valve, 6 – static seal, 7 – piston seal

Nitrogen is used as the working gas, which is filled via the check valve using special filling and testing equipment. The piston accumulators are not suitable for water. They are used for oils and synthetic anhydrous liquids. They are used in common applications for pressures up to 35 MPa and volumes up to 1500 dm^3. Manufacturers specify operating pressures $p \geq 55$ MPa for a high-pressure version of this accumulator. A decrease in the pressure of the liquid withdrawn from the accumulator can be limited by connecting additional nitrogen bottles.

In the accumulator, it is necessary to prevent liquid leakages into the gas part. This is ensured by the piston seal, which is designed similarly as in the case of hydraulic cylinders. The seal material is selected according to the type of the used liquid. To prevent damage to the seals during operation, the inner surfaces of the container must be machined (Ra ≤ 0.4 μm). If there is already a small liquid leakage, the seal needs to be replaced. The requirements for the liquid purity are also increased. These factors significantly increase the cost of the accumulator's production, operation and service.

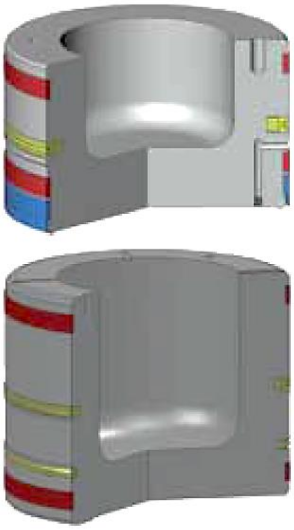


Fig. 10.5 Possible design of accumulator piston (left) [23], piston accumulators product range of Hydac company (right) [62]

With the piston design, it is possible to check the liquid volume in the accumulator or to transmit a signal for other control functions in a hydraulic circuit (e.g. connecting and disconnecting the pump). Mechanical sensing of the piston position is realized by moving the piston rod, which is connected to the piston. This solution requires constructional modifications to the lower cover and increases passive resistances. Other possibilities are to use electrical sensors (with a permanent magnet, inductive sensors) or ultrasonic measurement.

Piston accumulators are less sensitive to quick changes in liquid volume, generally suitable for slower processes and short-term withdrawals of larger liquid amounts. Compared to bladder and membrane accumulators, they are not suitable for damping pressure peaks and pulsations where a very fast response is required. Passive resistances of the seals and also the piston weight act against the piston movement. Although the pistons are of lightweight construction and are made of light alloys (usually aluminium alloys), the reaction time will be much longer than when deforming a flexible bladder or membrane.

The liquid discharge velocity is based on the piston movement velocity and depends on the used seal type. The maximum piston velocity should be up to 0.5 $m \cdot s^{-1}$ when using a standard rubber seal. When the seal is made of a more resistant material (PU, FKM) the velocity is usually higher but not more than 5 $m \cdot s^{-1}$. Pressure shocks can occur at high output velocities of the withdrawn liquid. It is possible to eliminate these shocks by using piston damping in the end positions, similar as in the case of hydraulic cylinders [6], [23].

Their mounting position is arbitrary, but usually, the vertical position is preferred. Piston accumulators are suitable for high pressure changes and, therefore, larger temperature changes. The ratio of the maximum operating liquid pressure p_2 to the gas pre-fill pressure p_0 is practically unlimited. The application of piston accumulators is mainly in theatre technology, in press shops, forges and injection moulding presses. The main manufacturers of piston accumulators are, for example, Hydac, Bolenz &Schäfer and Parker.

In certain applications (with piston or bladder accumulators) it is suitable to increase the gas volume. The volume is increased by adding additional nitrogen bottles when the purpose is to increase the useful volume of the accumulator. A comparison of the usable liquid volume by using an accumulator with an added gas volume is shown in Fig. 10.6.

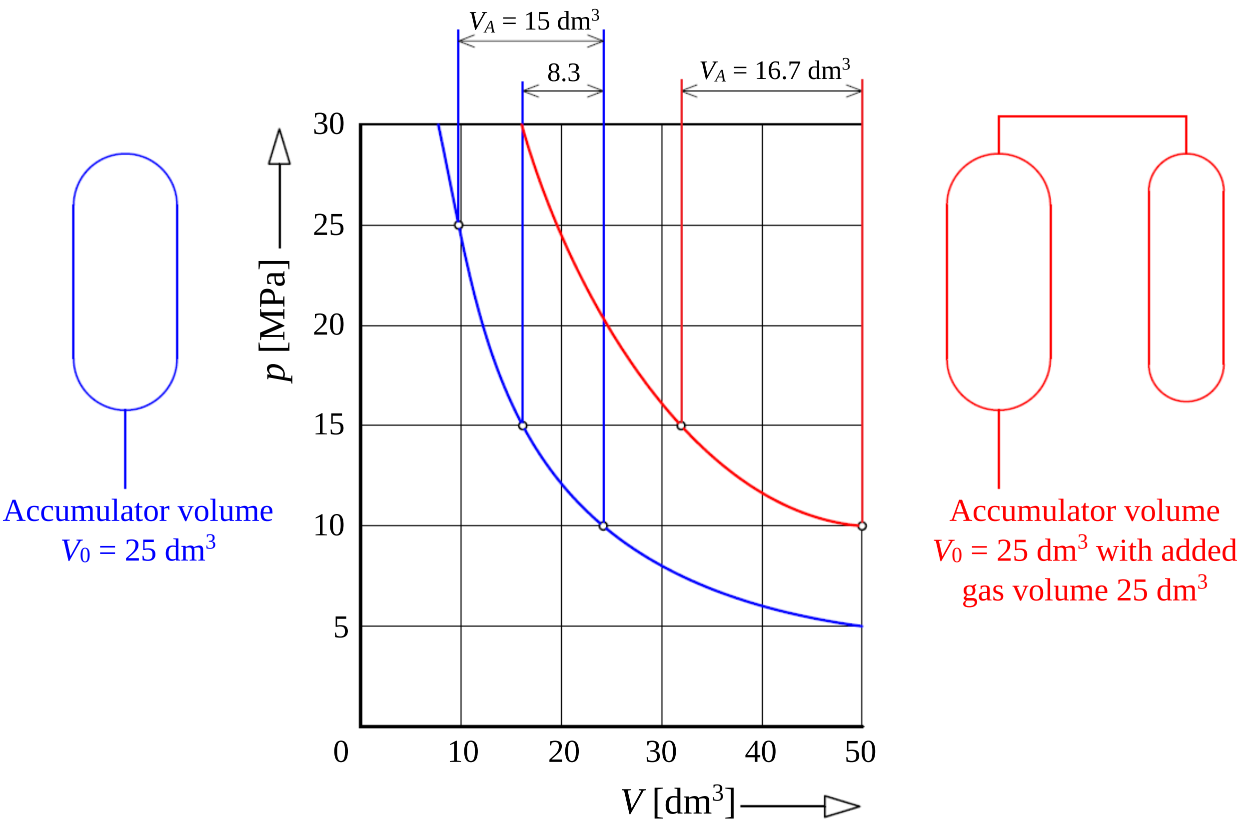


Fig. 10.6 Comparison of usable liquid volume by using an accumulator with added gas volume

10.3.3 Bladder accumulators

This is the universal and the most used accumulator type at the present time. The bladder accumulator (see Fig. 10.7) consists of the steel pressure vessel (1) in which the flexible rubber bladder (2) is placed. Thus, in this accumulator construction, the bladder represents a tight separating component between the gas and the liquid. The valve (3) is used to fill the bladder with gas and is always located on the top of the accumulator. The working fluid flows through the oil inlet (4), which is connected to the hydraulic system through a safety block. The shut-off poppet valve (5) prevents the bladder from being drawn into the pipe and its possible damage when the accumulator is emptied [11].

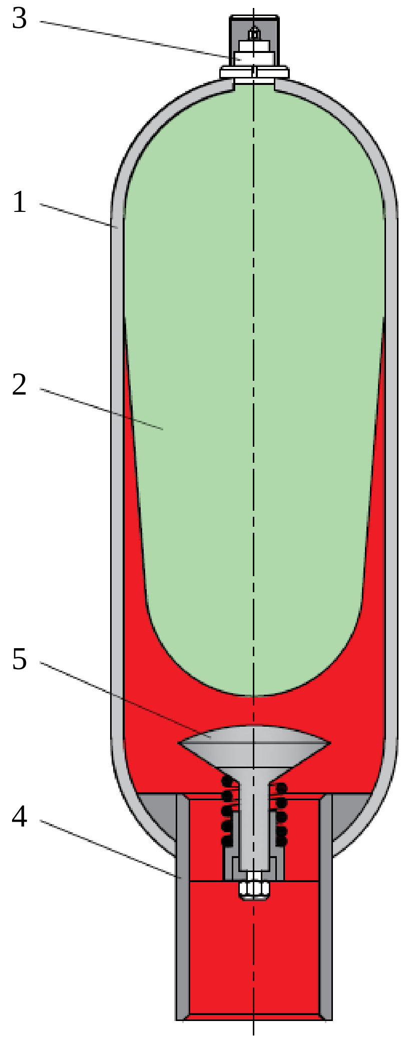


Fig. 10.7 Bladder accumulator

1 – steel pressure vessel, 2 – bladder, 3 – gas valve, 4 – oil inlet, 5 – shut-off poppet valve

Compared to the piston accumulators, there are no passive resistances to overcome during operation, and the inertial masses are practically negligible. This allows bladder accumulators to achieve fast response times and operate at high frequencies. The machining of the inner walls of the steel vessel is only such that the bladder does not rupture. The bladder is perfectly tight and prevents direct contact between the gas and the liquid. In general, these accumulators are more compact, simpler in their construction and are also cheaper and with lower maintenance costs.

The filling and emptying processes of the accumulator are shown in Fig. 10.8. We will now consider the state before the first use of the accumulator, and the hydraulic system is not in operation. The filling equipment is connected to the gas valve which starts to fill the bladder. As the gas pressure increases, the elastic bladder is inflated and fills the inner vessel volume until it starts to act on the poppet valve. After overcoming the spring resistance, the valve is closed, and the gas pressure is further increased to the required filling pressure p_0. The volume V_0 corresponds to the nominal volume of the accumulator.

After starting the hydraulic system, the liquid pressure in the system increases. When the liquid pressure reaches the value of gas filling pressure p_0, the poppet valve opens, and the liquid begins to fill the internal volume of the accumulator. The gas volume is V_1 at the pressure p_1, which is the minimum working pressure of the hydraulic system. When the pressure in the hydraulic part is further increased, the bladder is deformed, and the gas is compressed up to the volume V_2. In this state, the gas pressure is p_2 and, at the same time, it is the maximum working pressure of the hydraulic system. The volume difference $V_1 - V_2$ corresponds to the maximum amount of the removed liquid and is also called the useful volume V_A of the accumulator.

If the pressure decreases below the p_2 value in the hydraulic part, the gas expands, and the bladder pushes the liquid from the accumulator back into the system. In this way, it is possible to cover pressure oscillations in the hydraulic system, to balance unevenness in the liquid intake, etc.

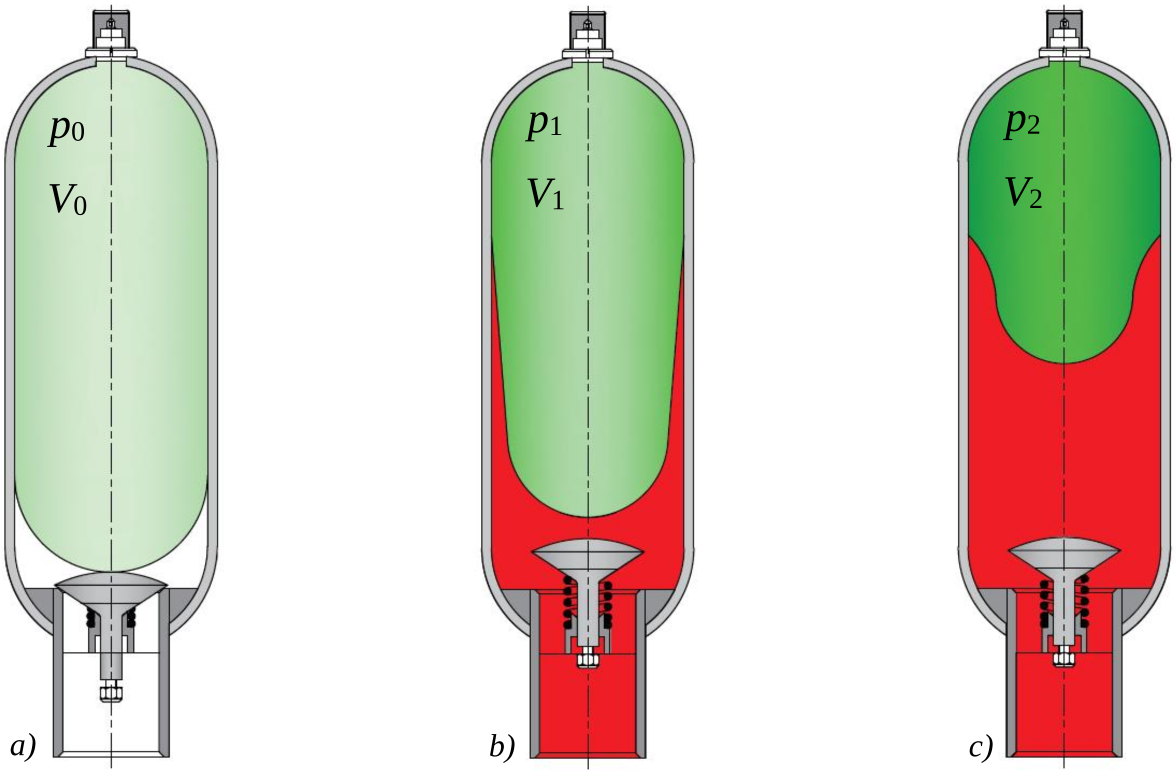


Fig. 10.8 Filling and emptying process of the bladder accumulator

a) filling with gas, b) filling with liquid, c) liquid emptying/withdrawing

When designing a bladder or membrane accumulator, the designer must respect the maximum allowable pressure ratio p_2/p_0. This ratio is based on the limited range of operating temperatures of the elastic bladder or membrane materials. In the case of big and fast pressure changes, a significant decrease in temperature would occur during the gas expansion, which could cause a change in the material properties of the elastic member. The bladder material is usually NBR rubber, FPM rubber (Viton), or IIR rubber (Butyl).

The maximum allowable pressure ratio for bladder accumulators is $p_2/p_0 \leq 4$. The gas filling pressure for these accumulators is chosen to be $p_0 \leq 0.9 \cdot p_1$ in the case that the accumulator is used to compensate for irregular consumption. To dampen pressure peaks and pulsations, the gas filling pressure is chosen as $p_0 = (0.6 \div 0.9)\ p_1$.

Standard bladder accumulators are manufactured with a volume of up to 200 dm^3 and for working pressures of up to 55 MPa. The maximum flow rate is 30 $dm^3 \cdot s^{-1}$, in the special "High - flow" version up to 120 $dm^3 \cdot s^{-1}$. They are used for fast processes and high emptying velocities. Their mounting position is vertical, with the hydraulic valve at the bottom of the accumulator.

Fig. 10.9 Bladder of accumulator (left), bladder accumulator in cut (right) [63]

10.3.4 Membrane accumulators

Membrane accumulators (see Fig. 10.10) are the last type of accumulator used. The spherical steel pressure container (1) is either of welded or bolted construction (for higher pressures). The container contains a rubber membrane (2) which is equipped with a metal ring (3) which prevents the membrane from being drawn into the inlet port, similar to the poppet valve in bladder accumulators. The membrane acts as a separating element between the gas and the liquid.

The flexible membrane allows fast reaction times. The function principle is similar to bladder accumulators. Changes in pressure cause deformation and movement of the membrane. However, this deformation is considerably limited, which also limits the useful volume of membrane accumulators. The membrane lifetime is significantly reduced in the case of large deformations. For this reason, membrane accumulators are normally produced only up to a volume of 4 dm^3. The working pressures are up to 35 MPa for the welded construction and up to 75 MPa for the bolted construction. The maximum pressure ratio $p_2/p_0 = (6 \div 8)$ for the welded construction and $p_2/p_0 = 10$ for the bolted construction [6], [11], [23].

Membrane accumulators are used only for small liquid volumes, usually to dampen pressure shocks and pulsations. Their mounting position is arbitrary.

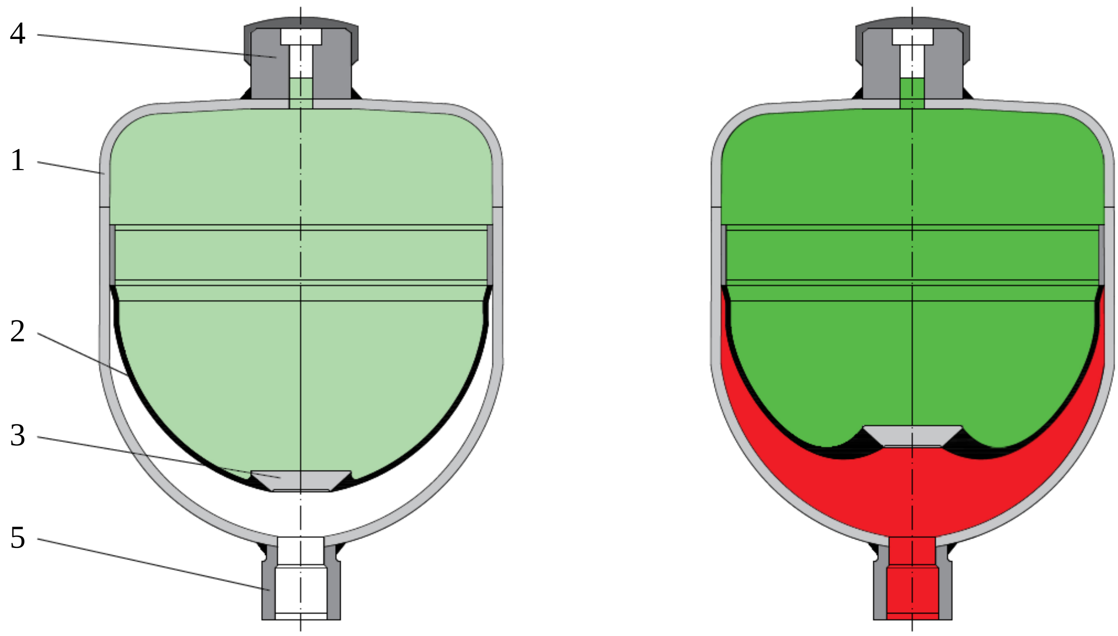


Fig. 10.10 Membrane accumulators

1 – steel pressure container, 2 – rubber membrane, 3 – metal ring, 4 – gas valve, 5 – oil inlet

Fig. 10.11 Membrane accumulator in cut

10.4 Accessories of accumulators

Safety elements, filling and testing equipment and fasteners are among the basic accessories of accumulators.

10.4.1 Safety and isolating control block

This block is connected to the liquid side and must be part of every accumulator that falls into the category of pressure vessels (according to Decree of the Czech Labor Safety Office and the Czech Mining Office No. 18/1979 Coll. and Act No. 174/1968 Coll. in valid versions). The basic design of the block with manual operation is shown in Fig. 10.12 (above left). It consists of a valve block (1) into which the main shut-off valve (2), drain valve (3) and relief valve (4) are built. The main shut-off valve (2) is manually operated and connects or disconnects the accumulator from a hydraulic system. The manually operated drain (shut-off) valve (3) has an unloading function and connects the accumulator (5) to the tank if necessary. The relief valve (4) is always a part of the block, it must be certified, the maximum allowable pressure is adjusted by a given accumulator manufacturer and the relief valve is sealed. The outputs M_1 and M_2 are used to connect the manometer.

Fig. 10.12 (bottom left) shows a block with an additional electrically operated valve (7), which is used for automatic unloading of the accumulator (5), for example, during machine shut-down or power cut. The other elements of the block perform the same function as in the previous case. The connection of the safety block to the accumulator is shown in Fig. 10.12 (right).

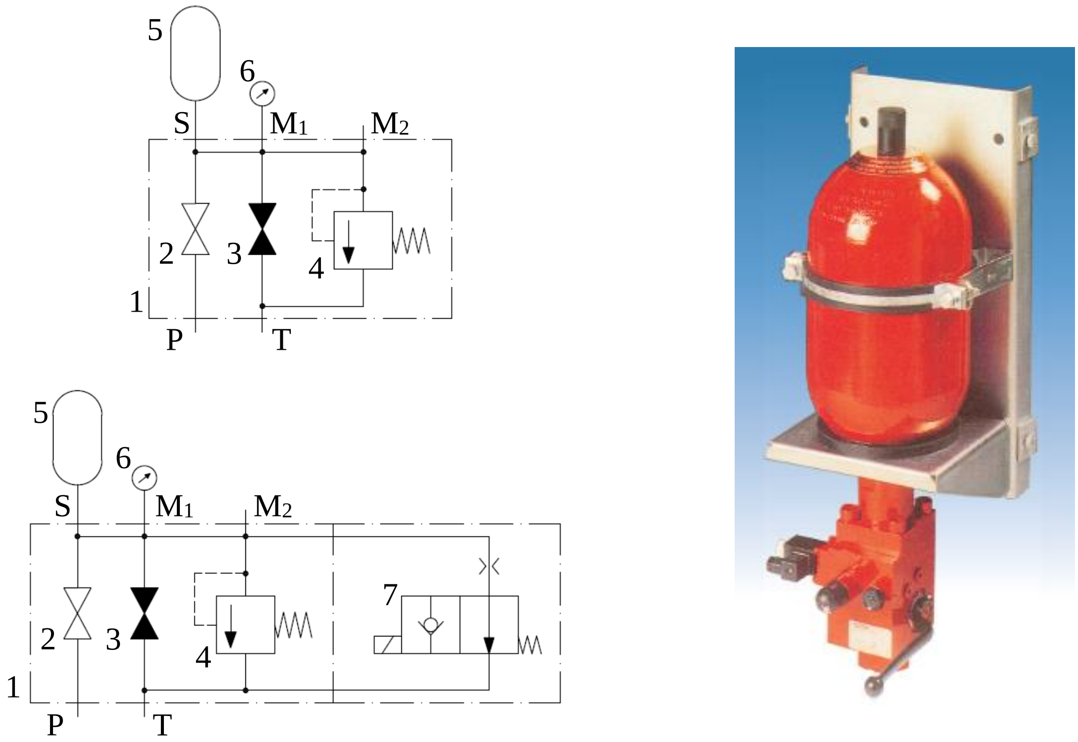


Fig. 10.12 Safety and isolating control block, basic version with manual operation (top left), safety block with additional electrical control (bottom left), an example of connecting the safety block to the accumulator (right)

1 – valve block, 2 - main shut-off valve, 3 – drain valve, 4 – relief valve, 5 – accumulator, 6 – manometer, 7 - electrically operated valve

From the point of view of operational safety, in some cases, it is also suitable to use protection in the gas part of the accumulator. In operations with higher ambient temperatures or in the event of a fire hazard, safety fuses are used to ensure that the gas is discharged from the accumulator. Safety membranes or gas relief valves then monitor the maximum allowable increase in the gas pressure.

The weight of the accumulator and the action of inertial forces during operation must not be transferred to the connected pipes of the hydraulic system. Accumulators must always be fixed to avoid stress on pipes. Calipers or brackets are used, between which rubber pads are inserted. A possible accumulator mounting is shown in Fig. 10.12 (right).

10.4.2 Filling equipment

The filling equipment is used to fill empty accumulators with nitrogen or to revise the gas pressure and its addition to the required value. The filling pressure is defined for a temperature of 20 °C, it differs for individual types of accumulators, and its value is based on the expected working pressure of the hydraulic system. The filling pressure should be checked at regular intervals. Examples of filling equipment from different manufacturers are shown in Fig. 10.13.

Fig. 10.13 Filling and testing equipment, Parker VGU equipment (left), N2S-V mobile filling equipment from Hydac company (right) [23]

10.5 Operating and safety regulations

Accumulators are pressure vessels, and their installation, operation and maintenance must be in accordance with safety regulations that are specified in the relevant standard. The following basic information must be clearly marked on each accumulator: manufacturer (supplier) of the accumulator, production number, year of manufacture, maximum allowable pressure, volume of the accumulator, and range of allowable ambient temperatures. The mounting of the accumulator is also prescribed and must include the safety and isolating control block which were discussed above.

In addition, each pressure vessel must have a technical certificate (according to the applicable standard ČSN 690010, pressure vessels must be documented with a valid passport issued by the manufacturer, importer of pressure vessels or a person with the appropriate authorisation), in which checks, revisions and repairs are recorded. The repairs must only be performed by experts from the accumulator manufacturer. Periodic checks and revisions are also prescribed by the standard and may only be performed by an inspection technician with a certificate for pressure vessels.

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