EUROPEAN UNION
European Structural and Investment Funds
Operational Programme Research,
Development and Education

MINISTRY OF EDUCATION,
YOUTH AND SPORTS

VSB TECHNICAL UNIVERSITY OF OSTRAVA
FACULTY OF MECHANICAL ENGINEERING
DEPARTMENT OF HYDROMECHANICS AND HYDRAULIC EQUIPMENT

# Fluid Mechanisms

## Practical Tasks and Basics of Pneumatics

Kamil Fojtášek
Lumír Hružík
Lukáš Dvořák
Adam Bureček
Erik Stonawski
Martin Vašina

Ostrava 2022

The study material was created with the support of the project from the Operational Programme Research, Development and Education entitled "Technology for the Future 2.0" with registration number CZ.02.2.69/0.0/0.0/18_058/0010212.

Acknowledgment to Bosch Rexroth company for the provision of visual materials and product photos for the purpose of processing this study materials.

# Contents

# List of Symbols

| ***Symbol*** | ***Unit*** | ***Term*** |
|---|---|---|
| $A_1$ | [$m^2$] | piston area of the hydraulic motor |
| $A_{1t}$ | [$m^2$] | theoretical piston area |
| $A_2$ | [$m^2$] | annulus area on the piston rod side of the hydraulic motor |
| $A_{2t}$ | [$m^2$] | theoretical (required) inter-circular area on the piston rod side |
| $A_C$ | [$m^2$] | heat transfer surface of the cooler |
| $A_p$ | [$m^2$] | pipe cross-sectional area |
| $A_T$ | [$m^2$] | heat transfer area of the tank |
| $A_{TV}$ | [$m^2$], [$mm^2$] | throttling point of the throttle valve |
| $C$ | [$dm^3 \cdot s^{-1} \cdot bar^{-1}$] | sonic conductance |
| $Cv$ | [gal (USwet) $\cdot$ $min^{-1}$] | Cv-value (flow coefficient) |
| $D$ | [m], [mm] | piston diameter |
| $D_D$ | [m] | diameter of winding drum |
| $D_t$ | [m], [mm] | theoretical piston diameter of the hydraulic cylinder |
| $F$ | [N] | force |
| $F_d$ | [N] | dynamic force |
| $F_{fr}$ | [N] | frictional force |
| $F_l$ | [N] | load force |
| $F_M$ | [N] | force on the hydraulic motor |
| $G$ | [N] | gravitational force |
| $H$ | [m], [mm] | cylinder stroke |
| $J$ | [$kg \cdot m^2$] | moment of inertia |
| $J_D$ | [$kg \cdot m^2$] | moment of inertia of drum |
| $J_M$ | [$kg \cdot m^2$] | moment of inertia of the hydraulic motor |
| $J_{red}$ | [$kg \cdot m^2$] | reduced moment of inertia |
| $J_{Tred}$ | [$kg \cdot m^2$] | total reduced moment of inertia |
| $Kv$ | [$m^3 \cdot h^{-1}$] | Kv-value (flow coefficient) |
| $L$ | [m] | pipe length |
| $M_D$ | [$N \cdot m$] | torque on the drum |
| $M_d$ | [$N \cdot m$] | dynamic torque |
| $M_G$ | [$N \cdot m$] | torque on the shaft of the hydraulic pump |
| $M_l$ | [$N \cdot m$] | load torque |

| | | |
|---|---|---|
| $M_M$ | [N · m] | torque of the hydraulic motor |
| $N$ | [-] | number of cycles |
| $P_1$ | [W] | input power of the hydraulic system |
| $P_2$ | [W] | output power of the hydraulic system |
| $P_e$ | [W] | equivalent power |
| $P_G$ | [W] | required power of the electric motor |
| $P_h$ | [W] | hydraulic power |
| $P_l$ | [W] | loss power of the hydraulic system |
| $P_{lRV}$ | [W] | loss power at the relief valve |
| $P_{lTV}$ | [W] | loss power at the throttle valve |
| $P_S$ | [W] | specific power of cooler |
| $P_t$ | [W] | theoretical power of the hydraulic pump |
| $P_u$ | [W] | useful power |
| $Q$ | [$m^3 \cdot s^{-1}$], [$dm^3 \cdot min^{-1}$] | volumetric flow |
| $Q_{ekv}$ | [$dm^3 \cdot min^{-1}$(ANR)] | equivalent flow rate |
| $Q_G$ | [$m^3 \cdot s^{-1}$], [$dm^3 \cdot min^{-1}$] | volumetric flow of the hydraulic pump |
| $Q_{Gr}$ | [$m^3 \cdot s^{-1}$], [$dm^3 \cdot min^{-1}$] | real volumetric flow of the hydraulic pump |
| $Q_{Gt}$ | [$m^3 \cdot s^{-1}$], [$dm^3 \cdot min^{-1}$] | required (theoretical) volumetric flow of the hydraulic pump |
| $Q_L$ | [$m^3 \cdot s^{-1}$], [$dm^3 \cdot min^{-1}$] | leakage volumetric flow |
| $Q_L$ | [$m^3 \cdot s^{-1}$], [$dm^3 \cdot min^{-1}$] | required volumetric flow of the working liquid of the hydraulic system through the cooler |
| $Q_M$ | [$m^3 \cdot s^{-1}$], [$dm^3 \cdot min^{-1}$] | volumetric flow of the hydraulic motor |
| $Q_m$ | [$m^3 \cdot s^{-1}$], [$dm^3 \cdot min^{-1}$] | mean volumetric flow |
| $Q_{Mc}$ | [$dm^3 \cdot min^{-1}$(ANR)] | corrected flow rate |
| $Q_N$ | [$dm^3 \cdot min^{-1}$ (ANR)] | normal nominal flow |
| $Q_n$ | [$dm^3 \cdot min^{-1}$(ANR)] | immediate flow rate |
| $Q_{nM}$ | [$dm^3 \cdot min^{-1}$(ANR)] | required flow to the motor |
| $Q_{ns}$ | [$dm^3 \cdot min^{-1}$(ANR)] | average air consumption |
| $Q_{RV}$ | [$m^3 \cdot s^{-1}$], [$dm^3 \cdot min^{-1}$] | volumetric flow through the relief valve |
| $Q_{TV}$ | [$m^3 \cdot s^{-1}$], [$dm^3 \cdot min^{-1}$] | volumetric flow through the throttle valve |
| $Q_V$ | [$m^3 \cdot s^{-1}$], [$dm^3 \cdot min^{-1}$] | volumetric flow of the cooling medium |
| $R_D$ | [m] | radius of winding drum |
| $Re$ | [-] | Reynolds number |
| $Re_{crit}$ | [-] | critical Reynolds number |

| | | |
|---|---|---|
| $T$ | [s] | heating time constant |
| $T$ | [K] | thermodynamic (absolute) temperature |
| $V$ | [$m^3$] | volume |
| $V_A$ | [$m^3$], [$dm^3$] | 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 |
| $V_{gGt}$ | [$m^3$], [$cm^3$] | theoretical geometric stroke volume of the hydraulic pump |
| $V_{gM}$ | [$m^3$], [$cm^3$] | geometric stroke volume of the hydraulic motor |
| $V_{Lp}$ | [$m^3$] | liquid volume in the pipe |
| $V_n$ | [$dm^3$(ANR)] | volume of air required to extend and retract the piston rod |
| $V_R$ | [$m^3$], [$dm^3$] | reserve volume of the accumulator |
| | | |
| $a$ | [$m \cdot s^{-2}$] | acceleration |
| $b$ | [-] | critical pressure ratio |
| $c$ | [$J \cdot kg^{-1} \cdot K^{-1}$] | specific heat capacity |
| $c_1$ | [$J \cdot kg^{-1} \cdot K^{-1}$] | specific heat capacity of working liquid |
| $c_2$ | [$J \cdot kg^{-1} \cdot K^{-1}$] | specific heat capacity of steel parts in the circuit |
| $c_p$ | [$J \cdot kg^{-1} \cdot K^{-1}$] | specific heat capacity of the gas at constant pressure |
| $c_v$ | [$J \cdot kg^{-1} \cdot K^{-1}$] | specific heat capacity of the gas at constant volume |
| $d$ | [m], [mm] | piston rod diameter |
| $d_p$ | [m] | diameter of the pipe |
| $d_t$ | [m], [mm] | required (theoretical) area on the piston rod side of the hydraulic cylinder |
| $f$ | [gal (Imp) $\cdot$ $min^{-1}$] | f-value (flow coefficient) |
| $g$ | [$m \cdot s^{-2}$] | acceleration due to gravity |
| $h$ | [m] | length |
| $i$ | [-] | gear ratio |
| $i_{GB}$ | [-] | gear ratio of the gearbox |
| $k_C$ | [$W \cdot m^{-2} \cdot K^{-1}$] | heat transfer coefficient of the cooler |
| $k_T$ | [$W \cdot m^{-2} \cdot K^{-1}$] | total heat transfer coefficient of the tank |
| $kv$ | [$dm^3 \cdot h^{-1}$] | kv-value (flow coefficient) |
| $l_p$ | [m] | length of the pipe |
| $m$ | [kg] | weight , mass load |
| $m_1$ | [kg] | mass of working liquid in the system |

| | | |
|---|---|---|
| $m_2$ | [kg] | mass of steel parts in the circuit |
| $m_{Lp}$ | [kg] | liquid mass in the pipe |
| $m_{pred}$ | [kg] | reduced liquid mass in the pipe |
| $m_{Tred}$ | [kg] | total reduced mass on the piston rod of the hydraulic motor |
| $n$ | [$s^{-1}$], [$min^{-1}$] | speed |
| $n_0$ | [$s^{-1}$], [$min^{-1}$] | initial speed |
| $n_D$ | [$s^{-1}$], [$min^{-1}$] | speed of the drum |
| $n_G$ | [$s^{-1}$], [$min^{-1}$] | speed of the hydraulic pump |
| $n_M$ | [$s^{-1}$], [$min^{-1}$] | speed of the hydraulic motor |
| $n_s$ | [$s^{-1}$], [$min^{-1}$] | steady speed |
| $p$ | [Pa], [MPa], [bar] | pressure |
| $p_1$ | [Pa], [MPa], [bar] | input pressure; pressure on the piston side; |
| $p_2$ | [Pa], [MPa], [bar] | output pressure; pressure on the piston rod side; |
| $p_{abs}$ | [Pa], [MPa], [bar] | absolute pressure |
| $p_n$ | [Pa], [MPa], [bar] | normal pressure |
| $p_{RV}$ | [Pa], [MPa], [bar] | pressure adjusted on the relief valve |
| $r$ | [$J \cdot kg^{-1} \cdot K^{-1}$] | gas constant |
| $t$ | [°C] | system (liquid) temperature |
| $t_0$ | [s] | initial time |
| $t_0$ | [°C] | initial temperature, ambient temperature |
| $t_1$ | [°C] | input temperature of the cooling medium |
| $t_2$ | [°C] | output temperature of the cooling medium |
| $t_L$ | [°C] | temperature of the working liquid of the hydraulic system entering the cooler |
| $t_{L1}$ | [°C] | temperature of the working liquid at the cooler input |
| $t_{L2}$ | [°C] | temperature of the working liquid at the cooler output |
| $t_s$ | [s] | start-up time of the hydraulic motor |
| $t_s$ | [°C] | steady (maximum) system temperature |
| $v$ | [$m \cdot s^{-1}$] | velocity |
| $v_0$ | [$m \cdot s^{-1}$] | initial velocity |
| $v_1$ | [$m \cdot s^{-1}$] | extension velocity of the piston rod of the hydraulic motor |
| $v_2$ | [$m \cdot s^{-1}$] | retraction velocity of the piston rod of the hydraulic motor |

| | | |
|---|---|---|
| $v_{max}$ | [m · s$^{-1}$] | maximum velocity |
| $v_p$ | [m · s$^{-1}$] | liquid flow velocity in pipe |
| $v_s$ | [m · s$^{-1}$] | steady velocity |
| $z$ | [-] | number of teeth on gears |
| | | |
| $\Delta p$ | [Pa], [MPa], [bar] | pressure gradient |
| $\Delta p_{CV}$ | [Pa], [MPa], [bar] | pressure gradient on the check valve |
| $\Delta p_G$ | [Pa], [MPa], [bar] | pressure gradient on the hydraulic pump |
| $\Delta p_l$ | [Pa], [MPa], [bar] | pressure loss |
| $\Delta p_M$ | [Pa], [MPa], [bar] | pressure gradient on the hydraulic motor |
| $\Delta p_{TV}$ | [Pa], [MPa], [bar] | pressure gradient on the throttle valve |
| $\Delta t_{CM}$ | [°C] | temperature difference of the cooling medium |
| $\Delta t_m$ | [°C] | mean logarithmic temperature difference |
| $\Delta t_T$ | [°C] | temperature gradient on the tank |
| $\varepsilon$ | [s$^{-2}$] | angular acceleration |
| $\eta_{GB}$ | [-] | gearbox efficiency |
| $\eta_{mh}$ | [-] | mechanical-hydraulic efficiency |
| $\eta_{mhG}$ | [-] | mechanical-hydraulic efficiency of the hydraulic pump |
| $\eta_{mhM}$ | [-] | mechanical-hydraulic efficiency of the hydraulic motor |
| $\eta_Q$ | [-] | volumetric efficiency |
| $\eta_{QG}$ | [-] | volumetric efficiency of the hydraulic pump |
| $\eta_{QM}$ | [-] | volumetric efficiency of the hydraulic motor |
| $\eta_T$ | [-] | total efficiency |
| $\eta_{TM}$ | [-] | total efficiency of the hydraulic motor |
| $\lambda$ | [-] | friction coefficient |
| $\mu_{TV}$ | [-] | flow coefficient of the throttle valve |
| $\nu$ | [mm$^2$ · s$^{-1}$] | kinematic viscosity |
| $\rho$ | [kg · m$^{-3}$] | density |
| $\rho_L$ | [kg · m$^{-3}$] | density of the working liquid |
| $\tau$ | [s] | time |
| $\phi$ | [W] | heat flow (heat power) |
| $\phi_C$ | [W] | required cooler power |
| $\phi_T$ | [W] | amount of heat dissipated by the tank surface |

| | | |
|---|---|---|
| $\omega_D$ | [rad · s$^{-1}$] | angular velocity of the drum |
| $\omega_G$ | [rad · s$^{-1}$] | angular velocity of the input shaft of the hydraulic pump |
| $\omega_M$ | [rad · s$^{-1}$] | angular velocity of the output shaft of the hydraulic motor |

***Abbreviations***

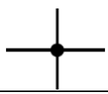
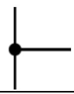
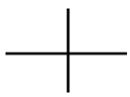
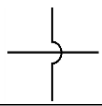
| | |
|---|---|
| ANR | atmospheric normal references |
| CV | check valve |
| ČSN | Czech technical standard |
| DV | directional valve |
| EM | electric motor |
| FCV | 2-way flow control valve |
| GB | gearbox |
| HM | hydraulic motor |
| HP | hydraulic pump |
| ISO | International Organization for Standardization |
| PI | pressure intensifier |
| RV | relief valve |
| TV | throttle valve |

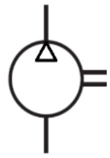
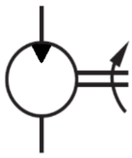
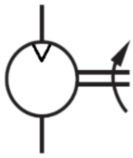
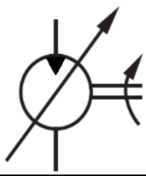
# 1. Symbols of hydraulic and pneumatic elements

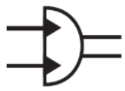
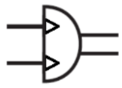
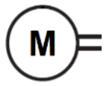
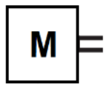
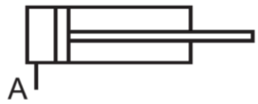
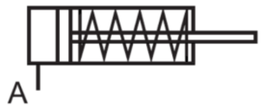
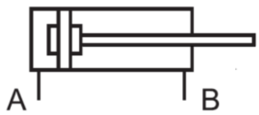
Fluid systems are characterized by a perimeter arrangement. Simplified graphical symbols are used in hydraulic and pneumatic drawings in order to illustrate the circuit connection and function. Each circuit element is assigned its own unique symbol. To understand the correct function of the circuit and to read the technical drawings, it is necessary to become familiar with these symbols.

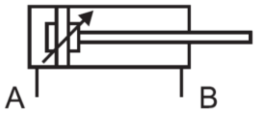
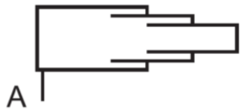
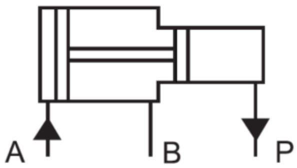
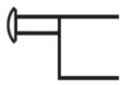
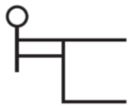
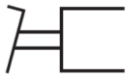
In the Czech Republic, the standard ČSN 01 3624 *Symbols for drawing hydraulic and pneumatic schemes* [1] was valid in the past, but its validity expired in 2002. This standard was followed by ČSN ISO 1219-1 *Hydraulics and pneumatics - Graphical symbols and circuit schemes* [2], which was issued in 1999 and was valid until 2009 when it was cancelled without replacement. In 2012, the international standard ISO 1219-1:2012 *Fluid power systems and components - Graphical symbols and circuit diagrams - Part 1: Graphical symbols for conventional use and data-processing applications* [3] was published and is valid in European Union countries. This standard is still supplemented in some countries, e.g., in Germany there are also valid standards with the designation DIN ISO 1219. It should be noted that it is often possible in technical practice to find element designations and drawings of an older date whose symbols correspond to one of the older standards. However, the differences between standards are usually not very significant. This study text contains a basic overview of the most used symbols for hydraulic and pneumatic circuits, as is shown in Tab 1.1. Because the principle of function of hydraulic and pneumatic components is, in many cases, very similar or the same, some symbols are universal and the same for both hydraulic and pneumatic circuits.

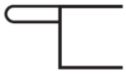
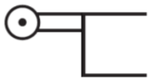
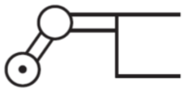
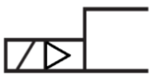
Tab 1.1 Selected graphical symbols of hydraulic and pneumatic elements

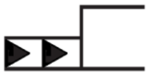
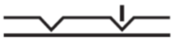
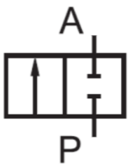
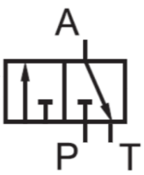
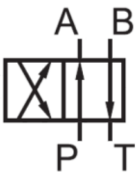
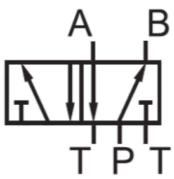
| Significance | Symbol | |
|---|---|---|
| *Continuous line*<br>line, pressure line (main, secondary, return) |  | |
| *Dashed line*<br>control line, drain line (leakages) |  | |
| *Chain dotted line*<br>to group two or more components in a sub-assembly |  | |
| Connected lines |  |  |
| Unconnected lines (crossing) |  |  |
| Hose |  | |

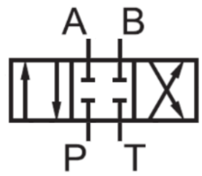
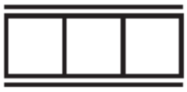
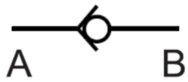
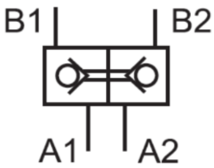
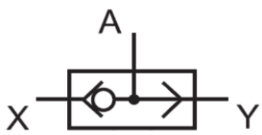
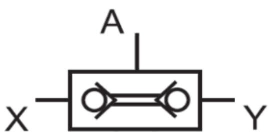
| Significance | Symbol | |
|---|---|---|
| | hydraulics | pneumatics |
| Energy source, flow direction | ▶ | ▷ |
| **Generators**<br>Fixed displacement pump, 1 direction of flow, 1 direction of rotation (left), compressor (right) |  |  |
| Fixed displacement pump, 2 directions of flow, 1 direction of rotation |  | |
| Variable displacement pump, 1 direction of flow, 1 direction of rotation |  | |
| Variable displacement pump, 2 directions of flow, 2 directions of rotation |  | |
| Pressure source (simplified) |  | |
| **Motors**<br>Fixed displacement motor, 1 direction of flow, 1 direction of rotation |  |  |
| Fixed displacement motor, 2 directions of flow, 2 directions of rotation |  |  |
| Variable displacement motor, 1 direction of flow, 1 direction of rotation |  | |
| Variable displacement motor, 2 directions of flow, 2 directions of rotation |  | |

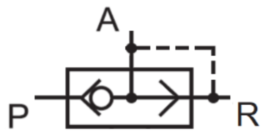
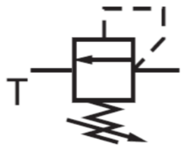
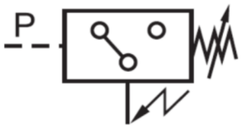
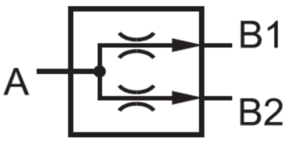
<table>
<tr><th rowspan="2">Significance</th><th colspan="2">Symbol</th></tr>
<tr><th>hydraulics</th><th>pneumatics</th></tr>
<tr><td>Rotary actuator</td><td></td><td></td></tr>
<tr><td>Electrical motor</td><td colspan="2"></td></tr>
<tr><td>Drive unit, except for electrical motor (combustion engine)</td><td colspan="2"></td></tr>
<tr><td>Single acting single-rod hydraulic cylinder (return stroke is given by external force)</td><td colspan="2"></td></tr>
<tr><td>Single acting single-rod hydraulic cylinder with internal spring (return stroke is given by spring)</td><td colspan="2"></td></tr>
<tr><td>Double-acting single-rod hydraulic cylinder</td><td colspan="2"></td></tr>
<tr><td>Double-acting double-rod hydraulic cylinder</td><td colspan="2"></td></tr>
<tr><td>Double-acting single-rod hydraulic cylinder with constant cushioning at both end positions</td><td colspan="2"></td></tr>
</table>

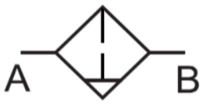
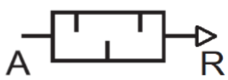
| Significance | Symbol | |
|---|---|---|
| | hydraulics | pneumatics |
| Double-acting single-rod hydraulic cylinder with adjustable cushioning at both end positions |  | |
| Single acting hydraulic telescopic cylinder, (return stroke is given by external force) |  | |
| Double-acting hydraulic telescopic cylinder |  | |
| Pressure intensifier |  | |
| **Operational modes**<br>General symbol |  | |
| Push button |  | |
| Lever |  | |
| Pedal, 1 direction of operation |  | |
| Pedal, 2 directions of operation |  | |

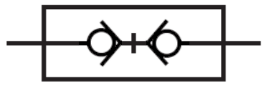
| Significance | Symbol | |
|---|---|---|
| | hydraulics | pneumatics |
| Push rod |  | |
| Roller shaft |  | |
| Roller lever |  | |
| Spring |  | |
| Electrical (by electromagnet) |  | |
| Hydraulic (by liquid pressure) |  | |
| Pneumatic (by air pressure) |  | |
| 2 parallel acting operators (push button or electrical) |  | |
| 2-stage serial electro-hydraulic operation |  | |
| 2-stage serial electro-pneumatic operation |  | |

| Significance | Symbol | |
|---|---|---|
| | hydraulics | pneumatics |
| 2-stage serial hydraulic operation |  | |
| Serial-parallel operation, 2 stage serial electro-hydraulic operation, spring centering of mid-position |  | |
| Detent, maintains specified position |  | |
| **Control elements**<br>Straight and oblique arrows inside elements indicate the direction of fluid flow, connection of ways |  | |
| Way closing | $\perp$ | |
| 2/2-way directional valve |  | |
| 3/2-way directional valve |  | |
| 4/2-way directional valve |  | |
| 5/2-way directional valve |  | |

| Significance | Symbol | |
|---|---|---|
| | hydraulics | pneumatics |
| 4/3-way directional valve |  | |
| Proportional distributor |  | |
| Check valve, without spring loading |  | |
| Check valve, spring loaded |  | |
| Pilot operated check valve |  | |
| Double pilot operated check valve |  | |
| Shuttle valve (logical function „or“) |  | |
| Shuttle valve (logical function „and“) | |  |

| Significance | Symbol | |
|---|---|---|
| | hydraulics | pneumatics |
| Air bleed valve | |  |
| Pressure relief valve |  | |
| Pressure reducing valve |  | |
| Pressure switch |  | |
| Shut-off valve |  | |
| Throttle valve |  | |
| Throttle check valve |  | |
| 2-way flow control valve |  | |
| Flow divider |  | |

| Significance | Symbol | |
| --- | --- | --- |
| | hydraulics | pneumatics |
| **Other elements**<br>Tank |  | |
| Accumulator |  | |
| Air reservoir | |  |
| Filter |  | |
| Separator | |  |
| Filter with separator | |  |
| Lubricator | |  |
| Preparation unit air comprising | |  |
| Noise damper | |  |

| Significance | Symbol | |
|---|---|---|
| | hydraulics | pneumatics |
| Cooler |  | |
| Heater |  | |
| Quick Coupler |  | |
| Pressure gauge (manometer) |  | |
| Flowmeter |  | |
| Thermometer |  | |
| Tachometer |  | |
| Torque meter |  | |

# 2. Basic hydraulic circuits

## 2.1 Circuits for generating translation (linear) motion

An example of a circuit for generating translational motion is the circuit shown in Fig. 2.1. It is an open hydraulic circuit. The pressure energy source is a fixed displacement hydraulic pump (1). The consumer is a double-rod hydraulic cylinder (2). If the diameters of the piston rods of the hydraulic motor are the same, then the same velocity of movement of the piston rods is ensured in both directions, i.e., $v_1 = v_2$. The control of the movement direction of the hydraulic motor is performed by means of the directional valve (3), which is operated by a lever. By adjusting the directional valve to one of the extreme positions, the movement of the piston rod is deduced in the direction of the velocity $v_1$ or $v_2$. The middle (neutral) position of the directional valve is defined by means of springs. This directional valve is in open centre design. When the hydraulic motor is inactive (the directional valve is in the middle position), the liquid from the hydraulic pump is drained back into the tank through the directional valve and the hydraulic pump is unloaded. The maximum pressure in the circuit is limited by the relief valve (4). The check valve (5) protects the hydraulic pump from the adverse effects of pressure shocks generated in the circuit. The low-pressure filter (6) is located in the discharge branch [4].

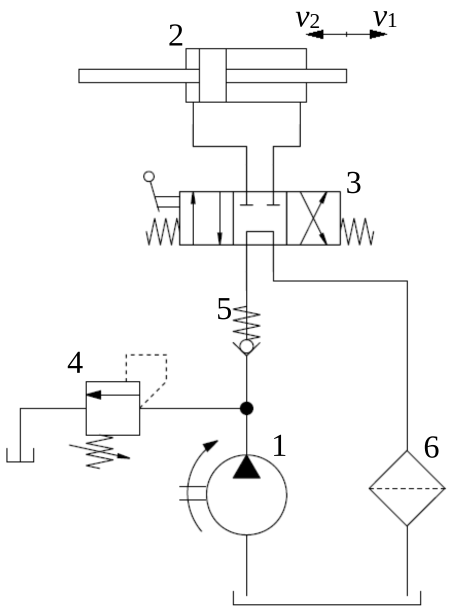



Fig. 2.1 Circuit for generating linear motion

*1 – hydraulic pump, 2 – double-rod hydraulic cylinder, 3 – directional valve, 4 – relief valve, 5 – check valve, 6 - filter*

The single-rod hydraulic cylinder (2) (with differential piston) is used in the circuit shown in Fig. 2.2. In this case, the surfaces on the piston and piston rod sides of the hydraulic motor are different, which ensures different velocities of the piston rod movement in both directions. The volumetric flow of the hydraulic pump $Q$ is given by the equation:

$$Q = A_1 \cdot v_1 = A_2 \cdot v_2 \ , \tag{2.1}$$

where $Q$ [$m^3 \cdot s^{-1}$] is the volumetric flow of the hydraulic pump, $A_1$ [$m^2$] is the piston area of the hydraulic motor, $A_2$ [$m^2$] is the annulus area on the piston rod side of the hydraulic motor, $v_1$ [$m \cdot s^{-1}$] is the extension velocity of the piston rod, and $v_2$ [$m \cdot s^{-1}$] is the retraction velocity of the piston rod.

To control the movement direction of the hydraulic motor, the electrically operated directional valve (3) is located in the circuit. This directional valve has a closed centre; in the neutral position the input from the hydraulic pump is closed. In order to prevent the energy on the relief valve (4) from being dissipated when the hydraulic motor is inactive, the unloading valve (5) is placed in the circuit, which transfers the liquid from the hydraulic pump back into the tank. The filter (6) with the parallel connected check valve (7) is located in the return branch. The check valve allows the liquid to flow into the tank when the pressure gradient on the filter increases (clogging of the filter insert) and protects the circuit from rupture of the filter insert [4], [5].

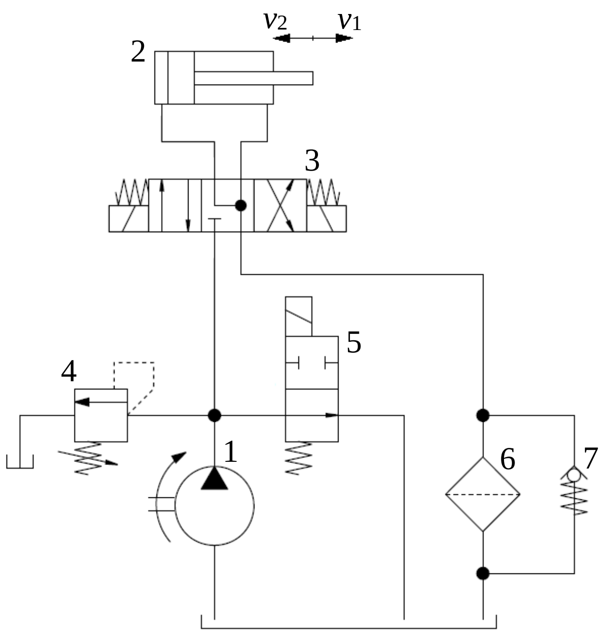



Fig. 2.2 Circuit for generating linear motion with a single-rod hydraulic cylinder

*1 – hydraulic pump, 2 – single-rod hydraulic cylinder, 3 – main directional valve, 4 – relief valve, 5 – unloading directional valve, 6 – filter, 7 – check valve*

Serial or parallel connections of directional valves to the pressure source can be used to control more hydraulic motors. An example of the control of two linear hydraulic motors (2 and 3) by means of two serially connected directional valves (4 and 5) is shown in Fig. 2.3. In this case, it is possible to use both directional valves with an open centre, i.e., when the hydraulic motors are inactive, the hydraulic pump (1) is unloaded. The liquid is filtered by means of the filter (7). When the directional valves are connected in series, the simultaneous operation of both hydraulic motors is not usually assumed [6].

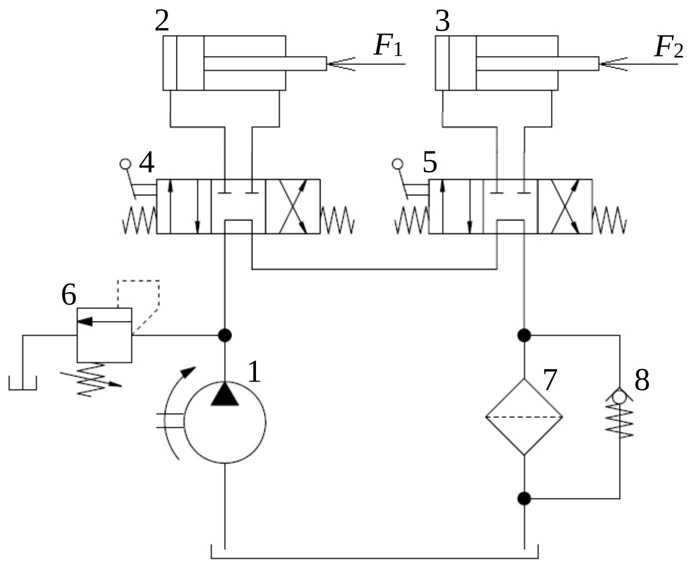



Fig. 2.3 Control of two hydraulic motors by serially connected directional valves

*1 – hydraulic pump, 2, 3 – hydraulic motors, 4, 5 – directional valves, 6 – relief valve, 7 – filter, 8 – check valve*

In Fig. 2.4, the directional valves (4 and 5) are connected in parallel to the pressure source, i.e., to the hydraulic pump (1). In this case, the closed-centre directional valves are used. Unloading of the hydraulic pump when the hydraulic motors (2 and 3) are inactive is solved by means of the unloading directional valve (7). In the case of the parallel connection, both hydraulic motors (2 and 3) can be operated simultaneously.

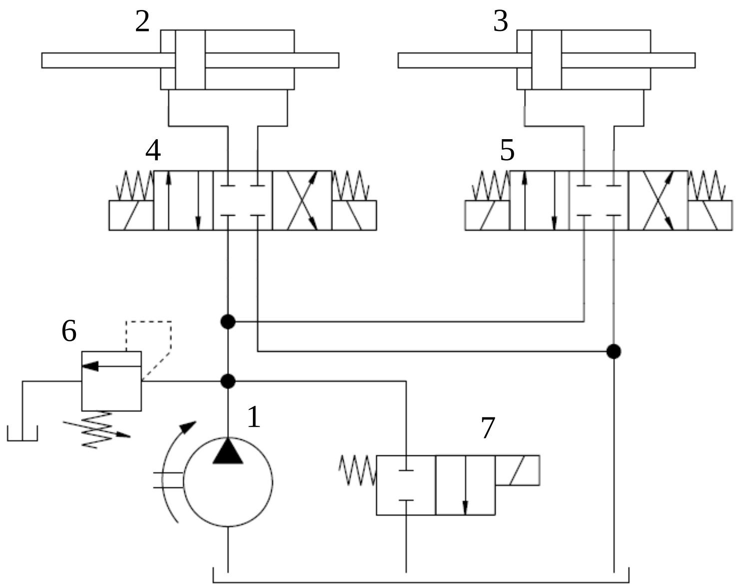



Fig. 2.4 Control of two hydraulic motors by directional valves connected in parallel to the pressure source

*1 – hydraulic pump, 2, 3 – hydraulic motors, 4, 5 – directional valves, 6 – relief valve, 7 – unloading directional valve*

Another typical example of a hydraulic circuit to generate a linear motion can be a load lifting and lowering application. When the load is lifted, it is necessary to overcome the load resistance. In addition, the potential energy of the load is increased. It is then necessary to brake the motor during lowering. In the circuit shown in Fig. 2.5, the single-rod (2) hydraulic motor is used to lift the load. In this case, it is the single acting hydraulic motor because the lowering of the piston rod of the hydraulic motor is caused by an external force from the load. The lifting and lowering are controlled by means of the directional valve (3). The lowering velocity of the load is controlled by the throttle valve (5). The check valve (6) ensures that the load does not fall spontaneously when the hydraulic pump (1) is stopped [7], [8].

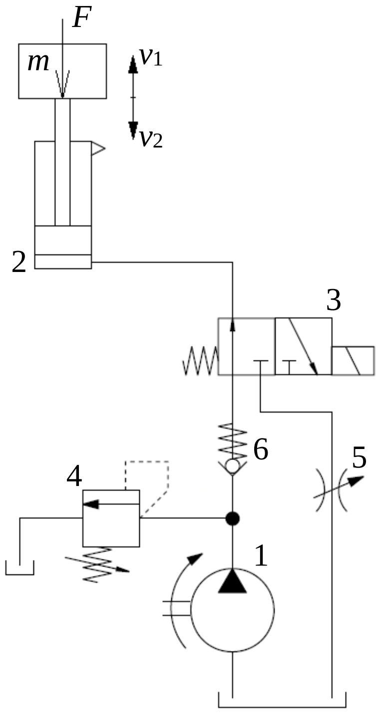



Fig. 2.5 Circuit for load lifting and lowering

*1 – hydraulic pump, 2 – hydraulic motor, 3 – directional valve, 4 – relief valve, 5 – throttle valve, 6 – check valve*

A simple circuit of a manual hydraulic hoist is shown in Fig. 2.6. The shut-off valve (5) must be closed to lift the load. When the lever of the manual hydraulic pump (1) is lifted, the liquid from the tank is supplied to the space under the piston of the hydraulic pump through the check valve (3). When the lever of the hydraulic pump is pushed, the pressure liquid flows through the check valve (4) into the space under the piston of the hydraulic motor (2). The pressure size in the space under the piston of the hydraulic motor (2) depends on the magnitude of the loading force. At the same time, the liquid volume increases. Therefore, the force generated by the piston rod of the hydraulic motor increases and the piston rod moves upwards. The load lowering is possible after the opening of the shut-off valve. In this case, the size of the valve opening leads to a change in the lowering velocity [9].

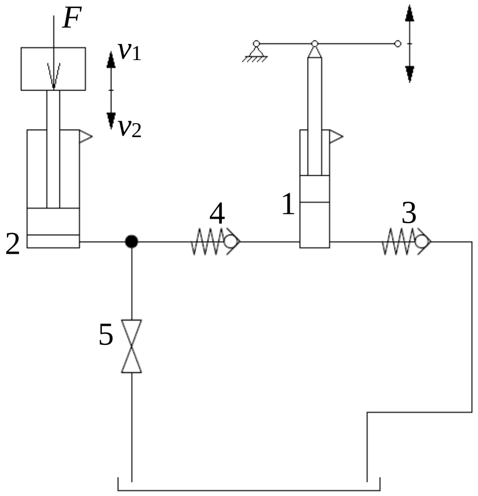



Fig. 2.6 Schematic diagram of manual hydraulic hoist

*1 – hydraulic pump, 2 – hydraulic motor, 3 – check valve, 4 – check valve, 5 – shut-off valve*

## 2.2 Circuits for generating rotary motion

Rotary hydraulic motors are used in hydraulic systems in order to generate rotary motion. The circuit for load lifting and lowering by means of the rotary hydraulic motor (2) with the flow in two directions is shown in Fig. 2.7. The winding drum (7) is connected to the output shaft of the hydraulic motor. The control of the rotation direction of the hydraulic motor is realized by the directional valve (3). When the load is lifted, the liquid from the hydraulic pump (1) flows through the check valve (6). The throttle valve (5) can be used to control the lowering velocity of the load [10].

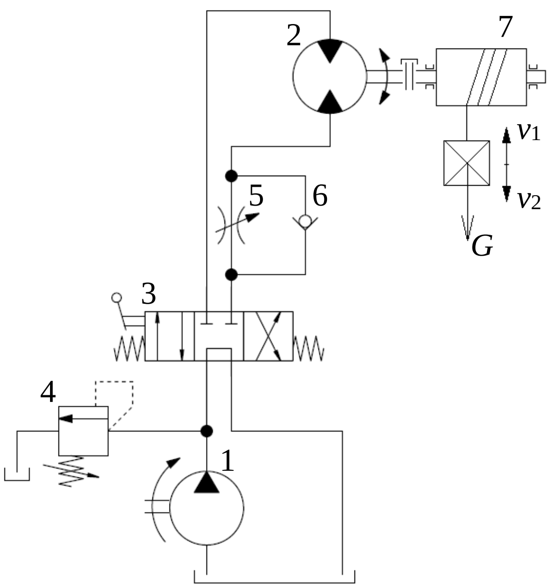



Fig. 2.7 Circuit for load lifting and lowering

*1 – hydraulic pump, 2 – hydraulic motor, 3 – directional valve, 4 – relief valve, 5 – throttle valve, 6 – check valve, 7 – winding drum*

All of the above examples characterized open hydraulic circuits. A simplified closed hydraulic circuit with a rotary hydraulic motor is shown in Fig. 2.8. In the closed circuit, the liquid tank (5) is connected parallel to the line between the hydraulic pump (1) and the hydraulic motor (2). The liquid from the hydraulic motor is not returned to the tank but is fed to the input of the hydraulic pump. In this case, the variable displacement hydraulic pump is used to control the rotational speed of the hydraulic motor. Liquid loss in the main circuit due to leakages on both converters is supplemented by means of the check valves (3). Pressure relief valves 4 are used to limit the maximum pressure in the circuit and protect the system from overload [4], [11].

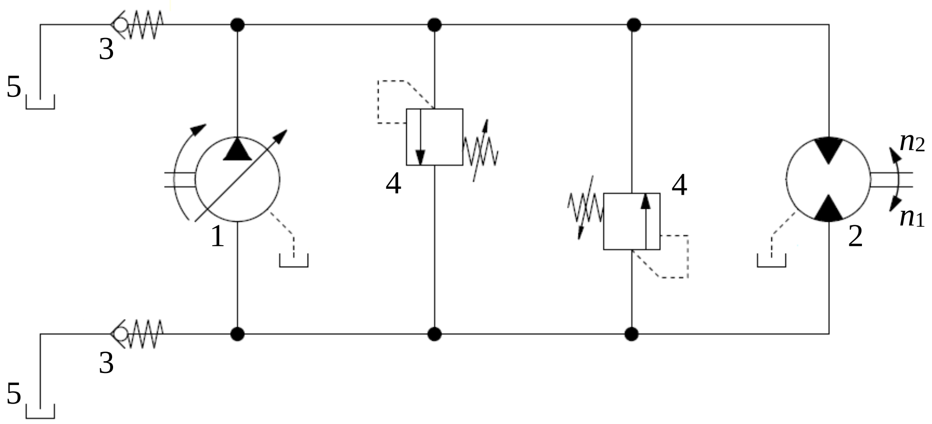



Fig. 2.8 Simplified closed hydraulic circuit

*1 – variable displacement hydraulic pump, 2 – rotary hydraulic rotor, 3 – check valves, 4 – relief valves, 5 – tank*

## 2.3 Control of velocity or speed of the hydraulic motor

Control of the velocity or speed of a hydraulic motor can be **continuous** or **stepped**. For example, the stepped control is characterized by the gradual connection of more hydraulic pumps. An example of the stepped velocity control of the hydraulic motor is shown in Fig. 2.9. Two hydraulic pumps (1 and 2) with different geometric stroke volumes are used in the circuit, allowing to achieve up to three different velocities of the hydraulic motor (3). The movement direction of the hydraulic motor is controlled by means of the electrically operated directional valve (5).

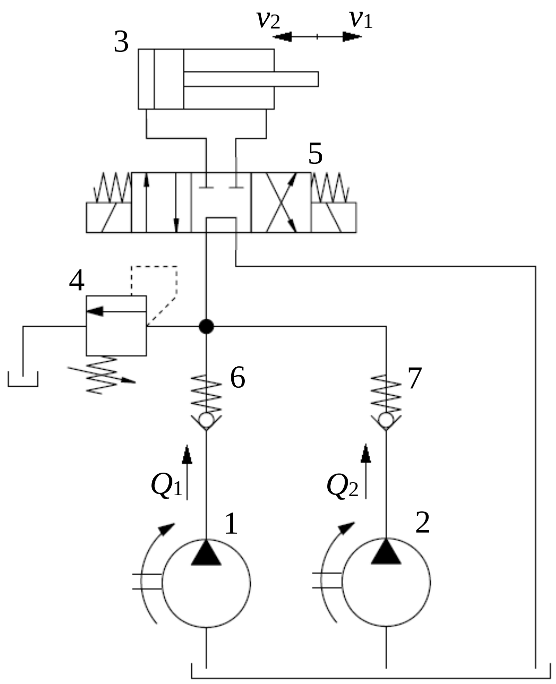



Fig. 2.9 Stepped velocity control of hydraulic motor

*1, 2 – hydraulic pumps, 3 – hydraulic motors, 4 – relief valve, 5 – directional valve, 6, 7 – check valves*

The continuous control of the velocity or speed of the hydraulic motor is more common and can be realized, for example, by adding a variable resistance to the circuit or by changing the geometric stroke volume of the hydraulic pump (or by changing the geometric stroke volume of the rotary hydraulic motor).

The continuous control **by changing the geometric stroke volume** of a hydrostatic converter is characterized by suitable velocity and speed characteristics and a very good total efficiency of the hydraulic system. The control range is usually from 1:10 to 1:100. An example of the continuous velocity control of a hydraulic motor by changing the geometric stroke volume of the hydraulic pump is shown in Fig. 2.10. The variable flow source is the variable displacement hydraulic pump (1) with the continuously variable geometric stroke volume. The volumetric flow $Q$ of the hydraulic pump is given by the equation [12]:

$$Q = V_g \cdot n \ , \tag{2.2}$$

where $Q$ [$m^3 \cdot s^{-1}$] is the volumetric flow of the hydraulic pump, $V_g$ [$m^3$] is the geometric stroke volume of the hydraulic pump, and $n$ [$s^{-1}$] is the speed of the hydraulic pump.

Continuous control by changing the geometric stroke volume is often used, for example, in hydrostatic transmissions of mobile hydraulic machines, in whose closed circuits, variable displacement hydraulic pumps and fixed or variable displacement hydraulic motors are used.

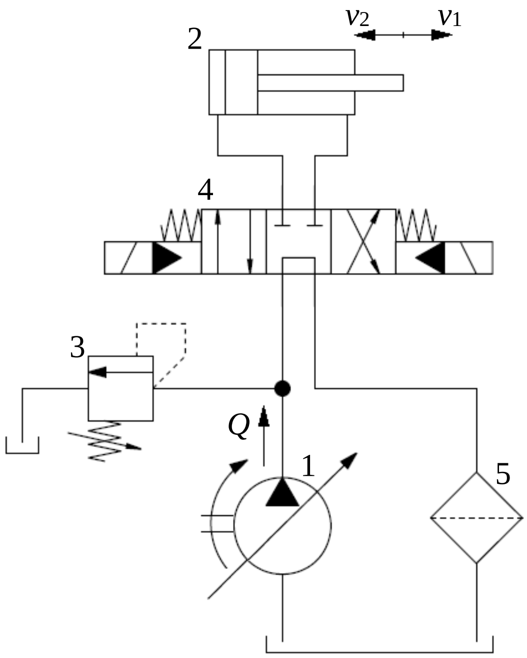



Fig. 2.10 Continuous control by changing the geometric stroke volume of the hydraulic pump

*1 – hydraulic pump, 2 – hydraulic motor, 3 – relief valve, 4 – directional valve, 5 – filter*

In the case of the continuous velocity or speed control of the hydraulic motor by **variable resistance**, an element is placed in the hydraulic circuit whose flow area can be continuously changed. Variable resistances for hydraulic systems can be throttle valves, throttle valves with pressure gradient stabilization (flow controllers), proportional distributors and servo valves. Compared to the geometric stroke volume change control, variable resistance control is characterized by a higher control range from 1:100 to 1:1000 and higher control speed, accuracy, and sensitivity. However, with the variable resistance control, part of the energy of the hydraulic generator is dissipated on the relief valve (when it is opened), and the total efficiency of the hydraulic system is significantly lower compared to the geometric volume change control.

An example of a circuit with continuous velocity control of a hydraulic motor by means of variable resistance is shown in Fig. 2.11. The variable resistances in the circuit are the throttle valves (5) and (6). The piston rod velocity of the hydraulic motor (2) in the direction $v_1$ is controlled by adjusting the flow area of the throttle valve (6). The piston rod velocity of the hydraulic motor in direction $v_2$ is controlled by the throttle valve (5). The check valves (7) and (8), which are connected in parallel to the throttle valves, ensure the bypass of the throttle valves in the opposite direction of liquid flow. This circuit can also be referred to as the piston rod velocity control of the hydraulic motor by throttling at the motor output.

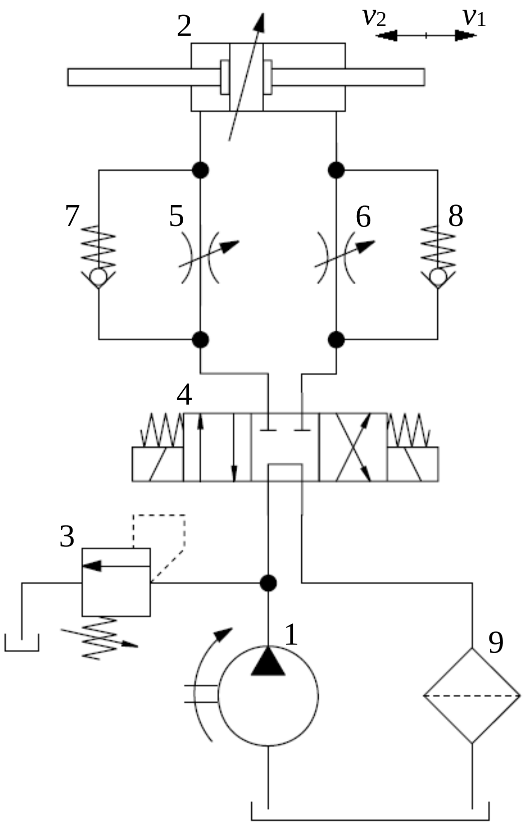



Fig. 2.11 Continuous velocity control by variable resistance using throttle valves

*1 – hydraulic pump, 2 – hydraulic motor, 3 – relief valve, 4 – directional valve, 5, 6 – throttle valves, 7, 8 – check valves, 9 - filter*

In modern hydraulic systems, continuous control of the velocity or speed of the hydraulic motor by changing the speed of the hydraulic pump (using a frequency converter in an electric motor) is also used more often [13], [14].

## 2.4 Control of force or stroke of the hydraulic motor

The force or torque of the hydraulic motor can be controlled by changing the pressure in a circuit with a hydraulic motor or by changing the geometric volume of a rotary hydraulic motor.

The pressure change in the hydraulic circuit can be achieved by means of pressure valves. The connection of the three relief valves (3), (4) and (5), by means of which three different magnitudes of force effect on the hydraulic motor (2) can be achieved, is shown in Fig. 2.12. The maximum pressure must be set on the valve (3). Lower pressures of different values are set on valves (4) and (5). Switching of different working pressures is ensured by means of the directional valve (6).

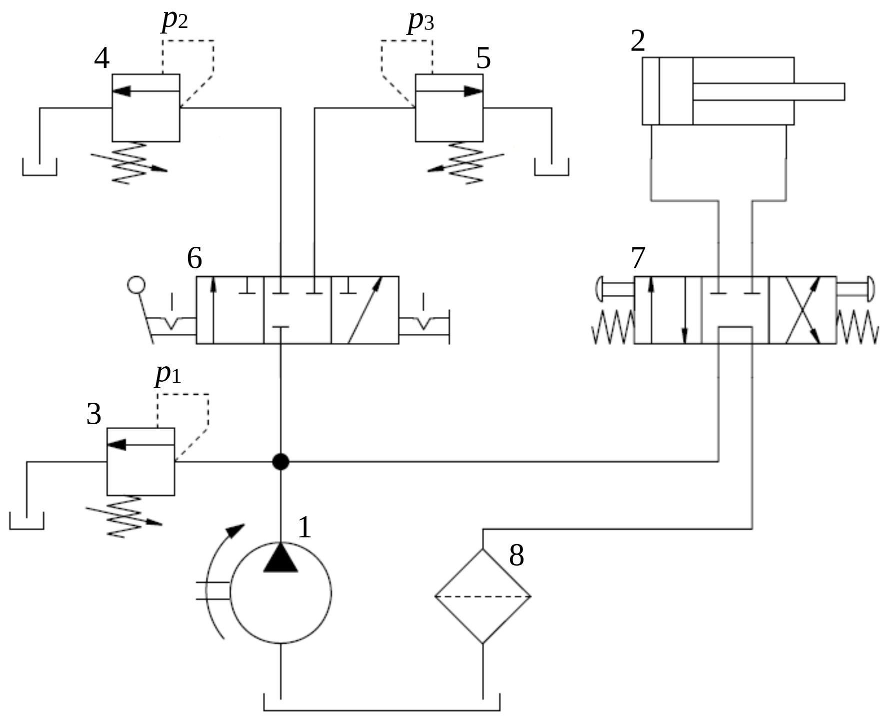



Fig. 2.12 Circuit for adjusting three different working pressures

*1 – hydraulic pump, 2 – hydraulic motor, 3, 4, 5 – relief (overflow) valves, 6, 7 – directional valves, 8 – filter*

The torque control of a rotary hydraulic motor by changing its geometric stroke volume is shown in Fig. 2.13. The dependence of the output torque $M_M$ of the rotary hydraulic motor on its geometric stroke volume $V_{gM}$ can be expressed by the equation:

$$M_M = \frac{V_{gM} \cdot \Delta p_M}{2\pi} \quad , \tag{2.3}$$

where $M_M$ [N · m] is the torque of the hydraulic motor, $V_{gM}$ [m$^3$] is the geometric stroke volume of the hydraulic motor, and $\Delta p_M$ [Pa] is the pressure gradient on the hydraulic motor.

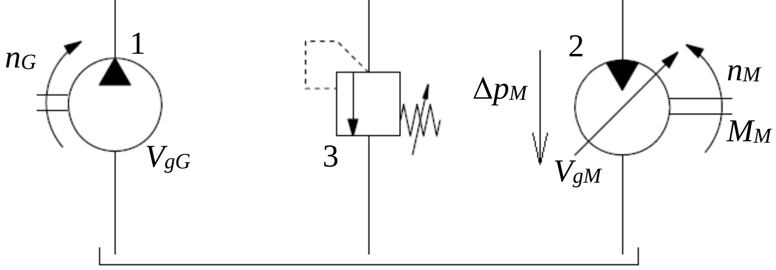



Fig. 2.13 Torque control by changing geometric stroke volume of rotary hydraulic motor

*1 – hydraulic pump, 2 – variable displacement hydraulic motor, 3 – relief valve*

# 3. Calculation of parameters of an ideal (lossless) hydrostatic circuit in steady state

An ideal (lossless) hydrostatic converter is considered in the calculation and hydraulic system where is:

- total efficiency $\eta_T$ = 100 %,
- ideal fluid (frictionless and incompressible),
- hydraulic elements without pressure, flow, and mechanical losses.

However, the back pressure on the check (non-return) and pressure valves obtained by overcoming the spring force is considered. The effect of a load on hydraulic motors is considered (dynamic effects of the load are not considered).

The examples are considerably simplified.

**Example 3.1**

Two identical series-connected hydraulic cylinders HM1 and HM2 are loaded by the external forces $F_1$ a $F_2$. The hydraulic pump HP supplies a constant volumetric flow $Q_G$, and the maximum system pressure $p_{RV}$ is set on the relief valve RV. The check valve CV is located in the return line and generates the back pressure $\Delta p_{CV}$. Calculate the pressure $p$ required to extend the piston rods of both hydraulic cylinders and the piston rod extension velocities $v_1$ a $v_2$.

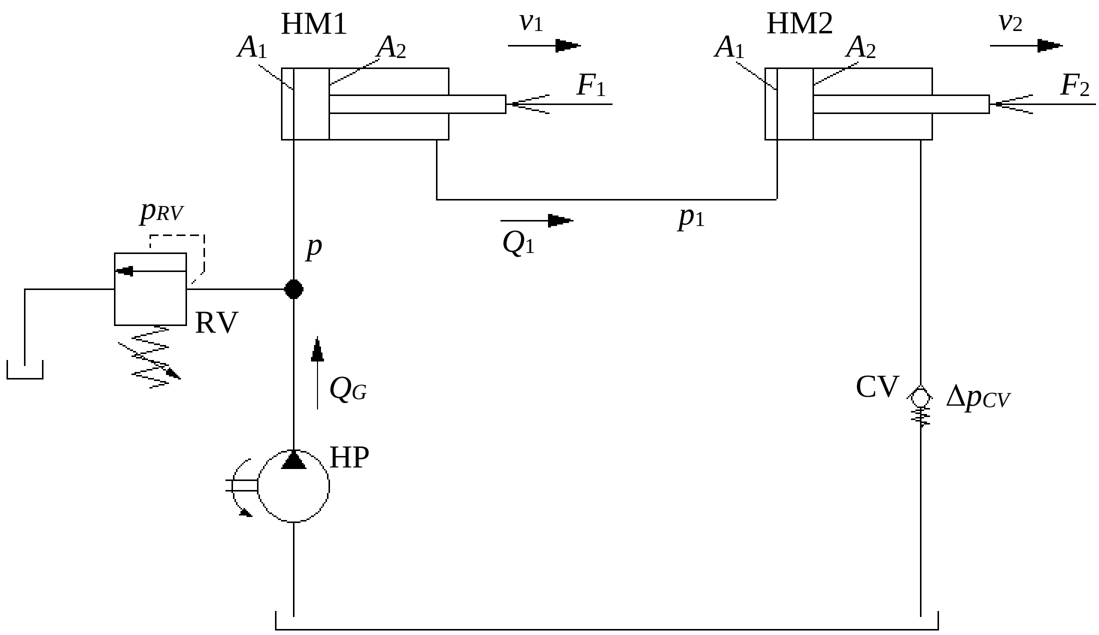



Fig. 3.1 Hydraulic scheme

*HP – hydraulic pump, HM1 and HM2 – hydraulic cylinders, RV – relief valve, CV – check valve*

Entered:

| | |
|---|---|
| piston area | $A_1 = 0.0031\ m^2$ |
| inter-circular area on the piston rod side | $A_2 = 0.0023\ m^2$ |
| force acting on the hydraulic cylinder HM1 | $F_1 = 30\ 000\ N$ |
| force acting on the hydraulic cylinder HM2 | $F_2 = 15\ 000\ N$ |
| pressure adjusted on the relief valve | $p_{RV} = 16\ MPa$ |
| back pressure on the check valve | $\Delta p_{CV} = 2\ MPa$ |
| volumetric flow of the hydraulic pump | $Q_G = 6.2 \cdot 10^{-4}\ m^3 \cdot s^{-1}$ |

Calculate: $p = ?$, $v_1 = ?$, $v_2 = ?$

Calculation:

Firstly, from the equation of motion, it is necessary to calculate the pressure $p_1$ needed to extend the piston rod of the hydraulic cylinder HM2:

$$p_1 \cdot A_1 = F_2 + \Delta p_{CV} \cdot A_2 \Rightarrow p_1\ ,$$
$$p_1 = \frac{F_2 + \Delta p_{CV} \cdot A_2}{A_1} = \frac{15\ 000 + (2 \cdot 10^6 \cdot 0.0023)}{0.0031} = 6.32 \cdot 10^6\ Pa\ , \tag{3.1}$$
$$p_1 = 6.32\ MPa\ .$$

Subsequently, it is necessary to determine the pressure $p$ required to extend the piston rod of the hydraulic cylinder HM1, taking into account the back pressure $p_1$ on the piston rod side:

$$p \cdot A_1 = F_1 + p_1 \cdot A_2 \Rightarrow p\ ,$$
$$p = \frac{F_1 + p_1 \cdot A_2}{A_1} = \frac{30\ 000 + (6.32 \cdot 10^6 \cdot 0.0023)}{0.0031} = 14.37 \cdot 10^6 Pa\ , \tag{3.2}$$
$$p = 14.37\ MPa\ .$$

It was found that the pressure $p < p_{RV}$. This means that the relief valve remains closed. The extension velocities of both hydraulic cylinders are determined using the continuity equation:

$$Q_G = A_1 \cdot v_1 \Rightarrow v_1\ ,$$
$$v_1 = \frac{Q_G}{A_1} = \frac{6.2 \cdot 10^{-4}}{0.0031} = 0.2\ m \cdot s^{-1}\ . \tag{3.3}$$

The extension velocity of the piston rod of the hydraulic cylinder HM1 is $v_1 = 0.2\ m \cdot s^{-1}$.

$$Q_1 = A_1 \cdot v_2 = A_2 \cdot v_1 \Rightarrow v_2\ ,$$
$$v_2 = \frac{A_2 \cdot v_1}{A_1} = \frac{0.0023 \cdot 0,2}{0.0031} = 0.148\ m \cdot s^{-1}\ . \tag{3.4}$$

The extension velocity of the piston rod of the hydraulic cylinder HM2 is $v_2 = 0.148\ m \cdot s^{-1}$.

**Example 3.2**

Two identical hydraulic cylinders HM1 and HM2 are connected in parallel to the flow source and are loaded by the external forces $F_1$ and $F_2$. The hydraulic pump HP supplies a constant volumetric flow $Q_G$, and the maximum system pressure $p_{RV}$ is set on the relief valve RV. The check valve CV is located in the return line and generates the back pressure $\Delta p_{CV}$. Calculate the pressures $p_{11}$ and $p_{12}$ required to extend the piston rods of both hydraulic cylinders, the order of extension of the hydraulic cylinders and the extension velocities $v_1$ a $v_2$.

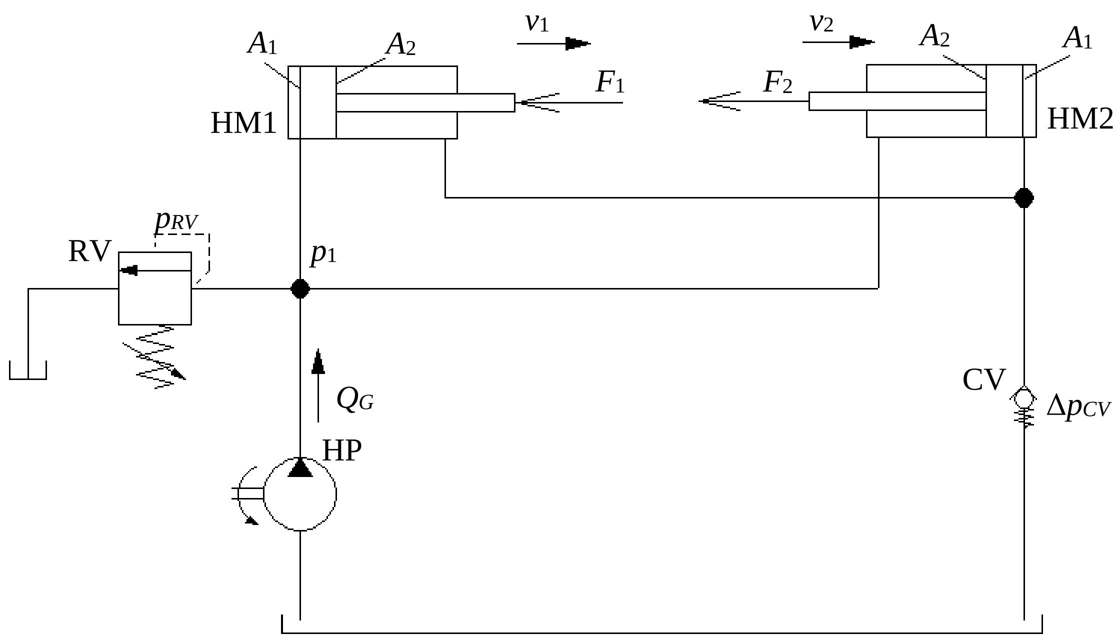



Fig. 3.2 Hydraulic scheme

*HP – hydraulic pump, HM1 and HM2 – hydraulic cylinders, RV – relief valve, CV – check valve*

<u>Entered:</u>

| | |
|---|---|
| piston area | $A_1 = 0.0031\ m^2$ |
| inter-circular area on the piston rod side | $A_2 = 0.0023\ m^2$ |
| force acting on the hydraulic cylinder HM1 | $F_1 = 20\ 000\ N$ |
| force acting on the hydraulic cylinder HM2 | $F_2 = 20\ 000\ N$ |
| pressure adjusted on the relief valve | $p_{RV} = 16\ MPa$ |
| back pressure on the check valve | $\Delta p_{CV} = 2\ MPa$ |
| volumetric flow of the hydraulic pump | $Q_G = 6.2 \cdot 10^{-4}\ m^3 \cdot s^{-1}$ |

<u>Calculate:</u> $p_{11} = ?,\ p_{12} = ?,\ v_1 = ?,\ v_2 = ?$

Calculation:

The pressure $p_{11}$ needed to extend the piston rod of the hydraulic cylinder HM1:

$$p_{11} \cdot A_1 = F_1 + \Delta p_{CV} \cdot A_2 \Rightarrow p_{11} \,,$$

$$p_{11} = \frac{F_1 + \Delta p_{CV} \cdot A_2}{A_1} = \frac{20\,000 + (2 \cdot 10^6 \cdot 0.0023)}{0.0031} = 7.94 \cdot 10^6 \, Pa \,, \tag{3.5}$$

$$p_{11} = 7.94 \, MPa \,.$$

The pressure $p_{12}$ needed to move the piston rod of the hydraulic cylinder HM2:

$$p_{12} \cdot A_2 = F_2 + \Delta p_{CV} \cdot A_1 \Rightarrow p_{12} \,,$$

$$p_{12} = \frac{F_2 + \Delta p_{CV} \cdot A_1}{A_2} = \frac{20\,000 + (2 \cdot 10^6 \cdot 0.0031)}{0.0023} = 11.39 \cdot 10^6 \, Pa \,, \tag{3.6}$$

$$p_{12} = 11.39 \, MPa \,.$$

Because the pressure $p_{12} > p_{11}$, the piston rod of the hydraulic cylinder HM1 will first move with the velocity $v_1$ (while the velocity $v_2 = 0$):

$$v_1 = \frac{Q_G}{A_1} = \frac{6.2 \cdot 10^{-4}}{0.0031} = 0.2 \, m \cdot s^{-1} \,. \tag{3.7}$$

When the piston rod of the hydraulic cylinder HM1 reaches the end position, the pressure in the system increases to $p_{12}$ = 11.39 MPa and the piston rod of the hydraulic motor HM2 starts to move with the velocity $v_2$:

$$v_2 = \frac{Q_G}{A_2} = \frac{6.2 \cdot 10^{-4}}{0.0023} = 0.27 \, m \cdot s^{-1} \,. \tag{3.8}$$

**Example 3.3**

The hydraulic cylinder HM of the piston diameter $D$ and the piston rod diameter $d$ has the piston and piston rod sides connected to increase the extension velocity. The volumetric flow $Q_G$ from the hydraulic pump HP and the pressure $p$ are available. Calculate the extension velocity $v$ of the piston rod of the hydraulic cylinder and the maximum force $F$ that the hydraulic motor is able to develop in this connection.

Entered:

| | |
|---|---|
| piston diameter | $D$ = 120 mm = 0.12 m |
| piston rod diameter | $d$ = 100 mm = 0.1 m |
| system pressure | $p$ = 20 MPa |
| volumetric flow of the hydraulic pump | $Q_G$ = 0.001 $m^3 \cdot s^{-1}$ = 60 $dm^3 \cdot min^{-1}$ |

Calculate: $v$ = ?, $F$ = ?

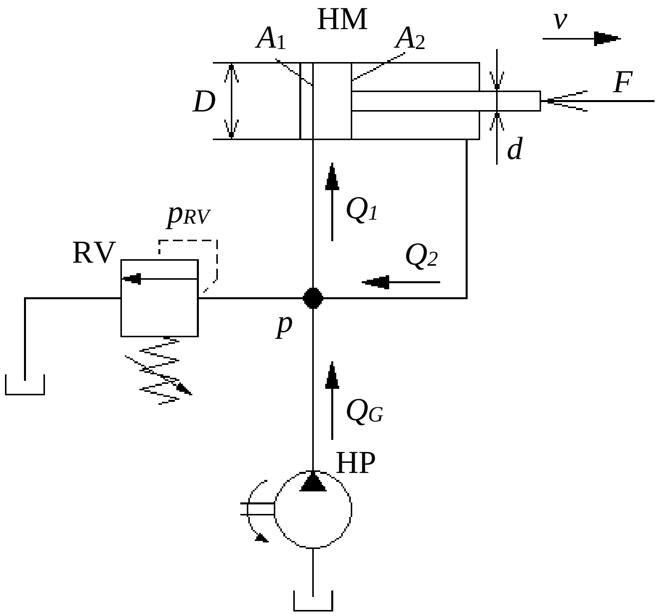



Fig. 3.3 Hydraulic scheme

*HP – hydraulic pump, HM – hydraulic cylinder, RV – relief valve*

Calculation:

The piston area $A_1$:

$$A_1 = \frac{\pi \cdot D^2}{4} = \frac{\pi \cdot 0.12^2}{4} = 0.0113\,m^2\,. \tag{3.9}$$

The inter-circular area on the piston rod side $A_2$:

$$A_2 = \frac{\pi \cdot (D^2 - d^2)}{4} = \frac{\pi \cdot (0.12^2 - 0{,}1^2)}{4} = 0.0034\,m^2\,. \tag{3.10}$$

In order to determine the extension velocity of the piston rod, it is necessary to consider the sum of volumetric flows:

$$Q_1 = Q_G + Q_2\,. \tag{3.11}$$

Velocity $v$ can be calculate using the equation of continuity to (3.11):

$$A_1 \cdot v = Q_G + A_2 \cdot v$$
$$Q_G = (A_1 - A_2) \cdot v \Rightarrow v \tag{3.12}$$
$$v = \frac{Q_G}{(A_1 - A_2)} = \frac{0.001}{(0.0113 - 0.0034)} = 0.126\,m \cdot s^{-1}.$$

The determination of the force using the equation of motion:

$$p_1 \cdot A_1 = F + p_2 \cdot A_2\,. \tag{3.13}$$

With regard to the connection, the pressures on both sides of the hydraulic cylinder are the same and the force $F$ is calculated as follows:

$$p_1 = p_2 = p\,,$$
$$F = p \cdot A_1 - p \cdot A_2 = p \cdot (A_1 - A_2) = 20 \cdot 10^6 \cdot (0.0113 - 0.0034)\,, \tag{3.14}$$
$$F = 158\,000\ N$$

**Example 3.4**

The hydraulic cylinder HM is loaded by force $F$. The pressure on the relief valve RV behind the hydraulic pump HP is $p_{RV}$, and the volumetric flow of the hydraulic pump is $Q_G$. The check valve CV is located in the return line and generates the back pressure $\Delta p_{CV}$. In order to achieve the required pressure on the hydraulic cylinder, a pressure intensifier PI with a constant movement velocity $v_p$ is used in the system. Calculate the pressure $p_3$ needed to extend the piston rod of the hydraulic cylinder, the pressure $p_1$ required at the inlet of the pressure intensifier, the extension velocity $v$ of the piston rod of the hydraulic cylinder, the leakage volumetric flow $Q_L$ of the pressure intensifier and the theoretical power $P_t$ of the hydraulic pump.

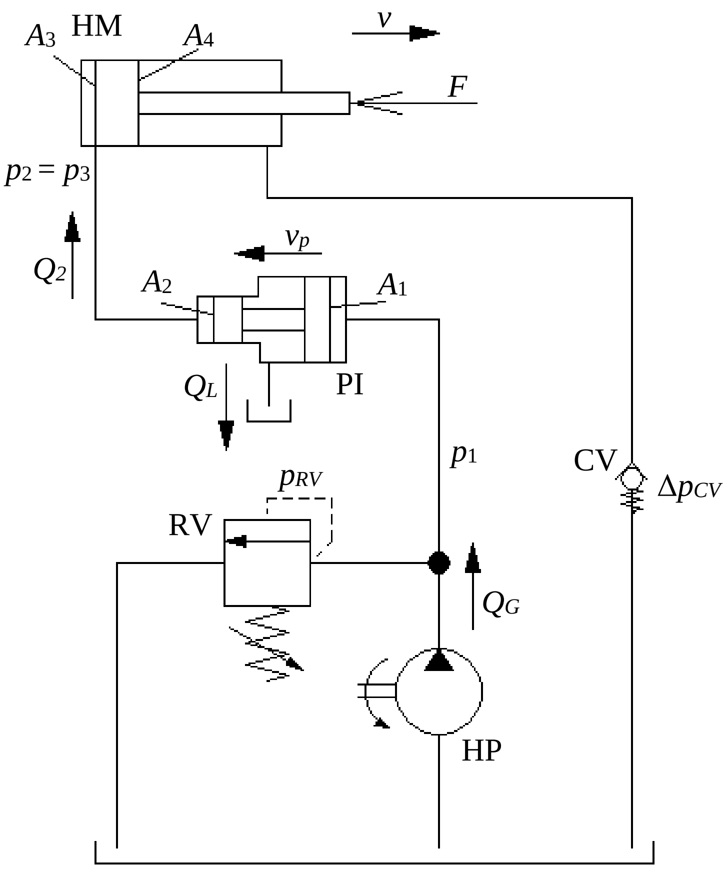



Fig. 3.4 Hydraulic scheme

*HP – hydraulic pump, HM – hydraulic cylinder, RV – relief valve, CV – check valve, PI - pressure intensifier*

Entered:

| | |
|---|---|
| load force on the hydraulic cylinder | $F = 300$ kN |
| input area of the pressure intensifier | $A_1 = 0.02$ m$^2$ |
| output area of the pressure intensifier | $A_2 = 0.005$ m$^2$ |
| piston area of the hydraulic cylinder | $A_3 = 0.00785$ m$^2$ |
| area on the piston rod side of hydraulic cylinder | $A_4 = 0.0028$ m$^2$ |
| back pressure on the check valve | $\Delta p_{CV} = 2$ MPa |
| pressure adjusted on the relief valve | $p_{RV} = 10$ MPa |
| volumetric flow of the hydraulic pump | $Q_G = 0.001$ m$^3 \cdot$ s$^{-1}$ = 60 dm$^3 \cdot$ min$^{-1}$ |

Calculate: $p_3 = ?, p_1 = ?, v = ?, Q_L = ?, P_t = ?$

Calculation:

The pressure $p_3$ needed to extend the loading force $F$, and the piston rod is determined from the equation of motion:

$$p_3 \cdot A_3 = F + \Delta p_{CV} \cdot A_4 \Rightarrow p_3 \,,$$
$$p_3 = \frac{F + \Delta p_{CV} \cdot A_4}{A_3} = \frac{300\,000 + (2 \cdot 10^6 \cdot 0.0028)}{0.00785} = 38.93 \cdot 10^6 Pa \,, \tag{3.15}$$
$$p_3 = 38.93\ MPa \,.$$

The pressure $p_1$ required at the input of the pressure intensifier ($p_2 = p_3$):

$$p_1 \cdot A_1 = p_3 \cdot A_2 \Rightarrow p_1 \,,$$
$$p_1 = \frac{p_3 \cdot A_2}{A_1} = \frac{38.93 \cdot 10^6 \cdot 0{,}005}{0.02} = 9.73 \cdot 10^6 Pa = 9.73\ MPa \,. \tag{3.16}$$

The pressure on the input of the pressure intensifier is lower compared to the pressure on the relief valve, i.e., $p_1 < p_{RV}$. This means that the pressure intensifier can be used in the system and the piston rod of the hydraulic cylinder overcomes the loading force $F$ and will be extended.

The velocity $v_p$ of the pressure intensifier is constant. It follows that:

$$v_p = \frac{Q_G}{A_1} = \frac{Q_2}{A_2} \Rightarrow Q_2 \,,$$
$$Q_2 = Q_G \cdot \frac{A_2}{A_1} = 0.001 \cdot \frac{0.005}{0.02} = 2.5 \cdot 10^{-4}\ m^3 \cdot s^{-1} \,. \tag{3.17}$$

And therefore, the extension velocity $v$ of the piston rod of the hydraulic cylinder:

$$v = \frac{Q_2}{A_3} = \frac{2.5 \cdot 10^{-4}}{0.00785} = 0.032\ m \cdot s^{-1} \,. \tag{3.18}$$

The leakage volumetric flow $Q_L$ of the pressure intensifier:

$$Q_L = Q_G - Q_2 = 0.001 - 2.5 \cdot 10^{-4} = 7.5 \cdot 10^{-4}\ m^3 \cdot s^{-1}\ . \tag{3.19}$$

The theoretical power $P_t$ of the hydraulic pump:

$$P_t = p_1 \cdot Q_G = 9.732 \cdot 10^6 \cdot 0.001 = 9\ 732\ W = 9.732\ kW\ . \tag{3.20}$$

**Example 3.5**

The two-stage telescopic cylinder HM for lifting the mass load *m* is located in a circuit. The pressure on the relief valve RV is $p_{RV}$, and the volumetric flow of the hydraulic pump HP is $Q_G$. Determine the pressure $p_1$ required for the extension of the first stage of the cylinder and the pressure $p_2$ required for the extension of the second stage of the cylinder. Determine in which order and at what velocity $v_1$ and $v_2$ the individual stages of the telescopic cylinder will be extended.

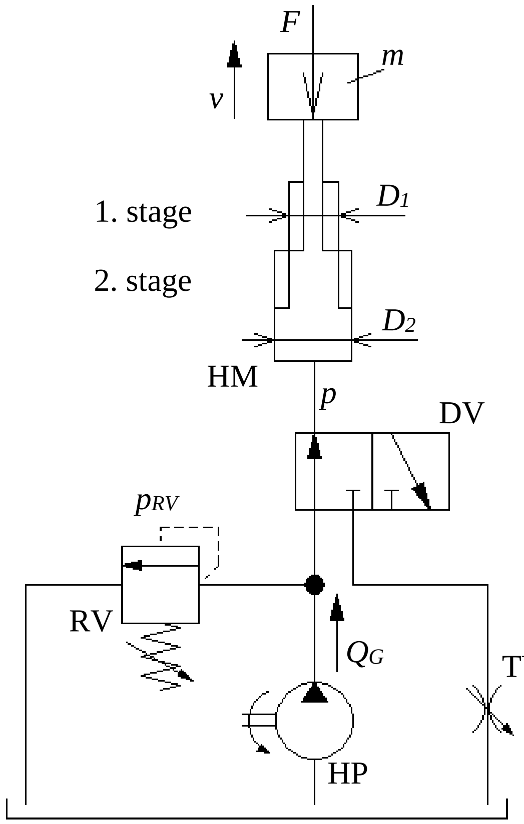



Fig. 3.5 Hydraulic scheme

*HP – hydraulic pump, HM – hydraulic cylinder, RV – relief valve, DV – directional valve, TV – throttle valve*

Entered:

| | |
|---|---|
| mass load | $m$ = 4 000 kg |
| diameter of the first stage of the cylinder | $D_1$ = 63 mm = 0.063 m |
| diameter of the second stage of the cylinder | $D_2$ = 80 mm = 0.08 m |
| pressure adjusted on the relief valve | $p_{PV}$ = 16 MPa |
| volumetric flow of the hydraulic pump | $Q_G$ = 0.001 m$^3 \cdot$ s$^{-1}$ = 60 dm$^3 \cdot$ min$^{-1}$ |

Calculate: $p_1$ = ?, $p_2$ = ?, $v_1$ = ?, $v_2$ = ?

Calculation:

The area $A_1$ of the first stage of the telescopic cylinder:

$$A_1 = \frac{\pi \cdot D_1^2}{4} = \frac{\pi \cdot 0.063^2}{4} = 0.0031\ m^2\ . \tag{3.21}$$

The area $A_2$ of the second stage of the telescopic cylinder:

$$A_2 = \frac{\pi \cdot D_2^2}{4} = \frac{\pi \cdot 0.08^2}{4} = 0.005\ m^2\ . \tag{3.22}$$

The loading force $F$ from the mass load:

$$F = m \cdot g = 4\ 000 \cdot 9.81 = 39\ 240\ N\ . \tag{3.23}$$

The pressure $p_1$ needed to move the first stage of the telescopic cylinder:

$$p_1 = \frac{F}{A_1} = \frac{39\ 240}{0.0031} = 12.66 \cdot 10^6\ Pa = 12.66\ MPa\ . \tag{3.24}$$

The pressure $p_2$ needed to move the second stage of the telescopic cylinder:

$$p_2 = \frac{F}{A_2} = \frac{39\ 240}{0.005} = 7.85 \cdot 10^6\ Pa = 7.85\ MPa\ . \tag{3.25}$$

The extension velocity $v_1$ of the first stage:

$$v_1 = \frac{Q_G}{A_1} = \frac{0.001}{0.0031} = 0.32\ m \cdot s^{-1}\ . \tag{3.26}$$

The extension velocity $v_2$ of the second stage:

$$v_2 = \frac{Q_G}{A_2} = \frac{0.001}{0.005} = 0.2\ m \cdot s^{-1}\ . \tag{3.27}$$

First, at the working pressure $p_2$ = 7.85 MPa, the second stage of the telescopic cylinder is pushed to the stop at the velocity $v_2$ = 0.2 m $\cdot$ s$^{-1}$. Then, after a pressure increase to $p_1$ = 12.66 MPa, the first stage is extended at the velocity $v_1$ = 0.32 m $\cdot$ s$^{-1}$. When the first stage reaches the stop, the pressure in the system rises to the pressure $p_{RV}$ set on the relief valve.

# 4. Calculation of circuits with throttle valves

Throttle valves can be used to control the velocity or speed of hydraulic motors in hydraulic systems. The throttle valve is a variable resistance, and the volumetric flow through this valve is given by the equation:

$$Q_{TV} = \mu_{TV} \cdot A_{TV} \sqrt{\frac{2 \cdot \Delta p_{TV}}{\rho}} \tag{4.1}$$

where $Q_{TV}$ [$m^3 \cdot s^{-1}$] is the volumetric flow, $A_{TV}$ [$m^2$] throttling point, $\Delta p_{TV}$ [Pa] pressure loss (pressure difference in front of and behind a throttle element), $\rho$ [$kg \cdot m^{-3}$] is the fluid density, and $\mu_{TV}$ [-] is the flow coefficient.

Obviously, in addition to the flow area $A_{TV}$, the flow $Q_{TV}$ through the valve also depends on the pressure gradient $\Delta p_{TV}$. So, what is the function of the throttle valve shown in Fig. 4.1.

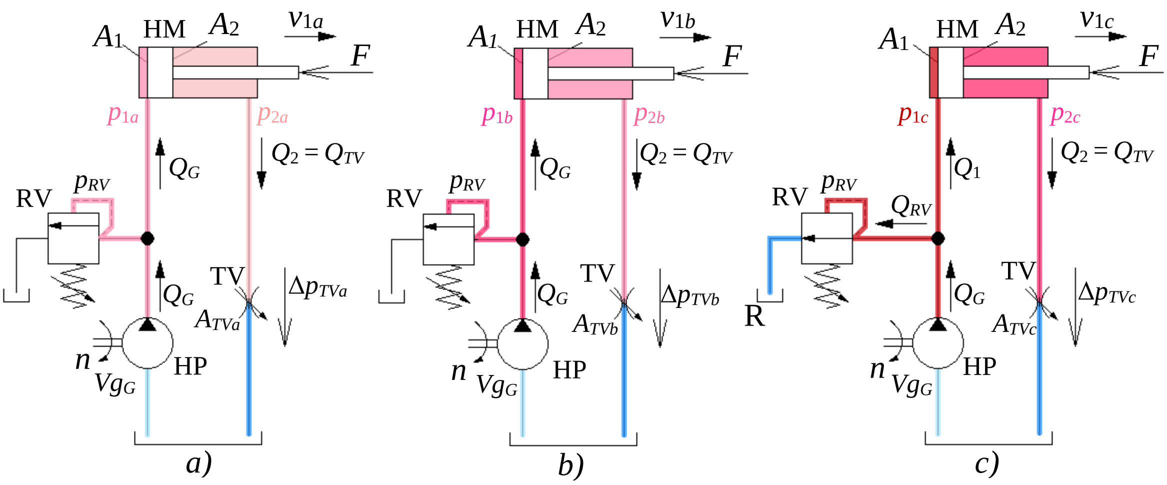



Fig. 4.1 Function of the throttle valve

*HP – hydraulic pump, HM – hydraulic cylinder, RV – relief valve, TV – throttle valve*

The hydraulic cylinder HM is loaded with a constant external force $F$. The hydraulic pump HP has a constant geometric volume $Vg_G$, and the input speed $n$ is constant. The maximum pressure $p_{RV}$ in the system is set using the relief valve RV. The throttle valve TV at the output of the hydraulic cylinder is used to control the velocity $v_1$ of the cylinder movement. The movement of the piston and piston rod is based on the equation of force balance:

$$\begin{gathered} F_1 = F + F_2 \, , \\ p_1 \cdot A_1 = F + p_2 \cdot A_2 \, , \\ p_2 = \Delta p_{TV} \, . \end{gathered} \tag{4.2}$$

In order to extend the piston rod, it is necessary to induce a liquid pressure force $F_1$ on the piston side that is greater compared to the external force $F$ and the liquid pressure force $F_2$ on the annulus side (in the real case, it is necessary to consider also the influence of the friction force expressing the passive resistances in the hydraulic cylinder).

As shown in Fig. 4.1a, the throttle valve TV is maximally open (i.e., $A_{TVa} = A_{max}$). In this case, the throttle valve represents only a minimum resistance to liquid flow, which is represented by the pressure $p_{2a}$ on the annulus side. After starting the hydraulic pump, the pressure in the system is increased to the value $p_{1a}$, which is required for the movement of the piston rod. The piston rod of the hydraulic motor will perform the stroke at the maximum velocity $v_{1a} = v_{max}$. For the stroke time, the hydraulic pump delivers a constant flow $Q_G$ to the hydraulic cylinder, and the movement velocity of the cylinder can be expressed from the continuity equation $Q_G = A_1 \cdot v_{1a}$. When the piston rod is completely extended, the pressure in the system increases to the value set on the relief valve RV, and after its opening, the liquid will flow through it into the tank. In this case, the cylinder velocity is not controlled by the throttle valve. This is a constant flow source, and the safety valve is closed.

In the connection, according to Fig. 4.1b, the throttle valve is slightly opened (i.e., $A_{TVb} < A_{TVa}$). This causes a change in the flow resistance through the throttle valve and increases the pressure on the annulus side (i.e., $p_{2b} > p_{2a}$). In order to set the piston rod in motion in this case, it is necessary to increase the pressure in the system (i.e., $p_{1b} > p_{1a}$). If the necessary pressure to move the piston rod is lower in comparison to the pressure set on the relief valve (i.e., $p_{1b} < p_{RV}$ and the relief valve is closed), then the entire flow $Q_G$ from the hydraulic pump is supplied to the hydraulic motor. Therefore, the velocity of the piston rod movement is always the same (i.e., $v_{1b} = v_{1a} = v_{max}$). Although the throttle valve was throttled, there was no change in the movement velocity of the piston rod. The hydraulic pump operates as a constant flow source in a similar way as in the previous case.

Now, according to Fig. 4.1c, the throttle valve is throttled considerably more (i.e., $A_{TVc} << A_{TVb}$). The valve now presents a significantly greater resistance to liquid flow, which will increase the pressure on the annulus side (i.e., $p_{2c} >> p_{2b}$). This results in an increase in the pressure $p_{1c}$ up to the pressure $p_{RV}$ that is set on the relief valve. The relief valve will be opened, one part of the flow from the hydraulic pump flows back into the tank through the relief valve (i.e., $Q_{RV}$), and the other part (i.e., $Q_1$) flows towards the hydraulic cylinder; therefore, $Q_G = Q_{RV} + Q_1$. The velocity of the piston rod movement is decreased (i.e., $v_{1c} < v_{1b}$), because the volumetric flow into the hydraulic cylinder is $Q_1 = A_1 \cdot v_{1c}$ and $Q_1 < Q_G$. In this case, the hydraulic pump works as a constant pressure source because the relief valve is open.

**Example 4.1**

The hydraulic cylinder HM is loaded by force $F$. The pressure on the relief valve RV behind the hydraulic pump HP is $p_{RV}$, and the volumetric flow of the hydraulic pump is $Q_G$. The throttle valve TV is located in the return line.

Determine:

a) the velocity $v$ of the piston rod extension for the case when the throttle valve is open,
b) adjust the throttle valve so that the relief valve has not yet opened,
c) adjust the throttle valve so that the piston rod extension speed is $v = 0{,}1 \text{ m} \cdot \text{s}^{-1}$,
d) the useful power, the loss power, and the total efficiency of the drive for throttle valve settings (b) and (c)

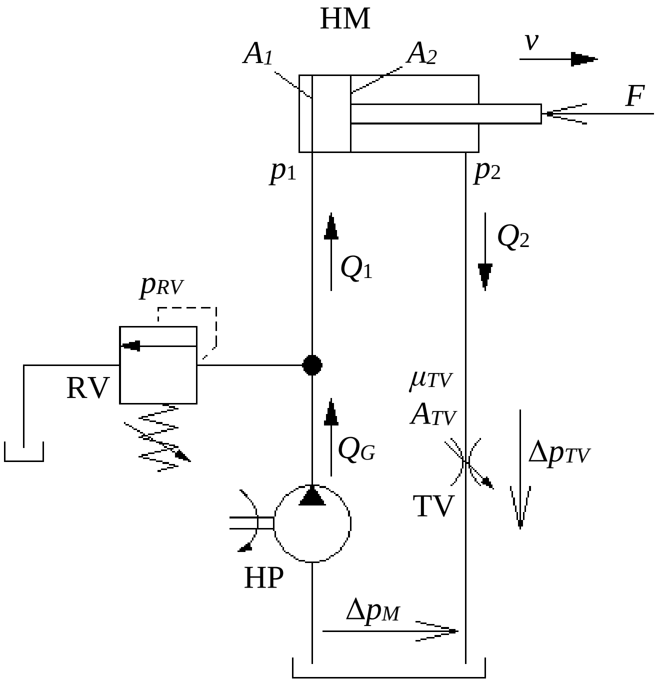



Fig. 4.2 Hydraulic scheme

*HP – hydraulic pump, HM – hydraulic cylinder, RV – relief valve, TV – throttle valve*

<u>Entered:</u>

| | |
|---|---|
| piston area | $A_1 = 0.0031 \text{ m}^2$ |
| inter-circular area on the piston rod side | $A_2 = 0.0023 \text{ m}^2$ |
| force acting on the hydraulic cylinder HM | $F = 15\,000 \text{ N}$ |
| pressure adjusted on the relief valve | $p_{RV} = 16 \text{ MPa}$ |
| volumetric flow of the hydraulic pump | $Q_G = 36 \text{ dm}^3 \cdot \text{min}^{-1} = 0.0006 \text{ m}^3 \cdot \text{s}^{-1}$ |
| oil density | $\rho = 890 \text{ kg} \cdot \text{m}^{-3}$ |
| flow coefficient of the throttle valve | $\mu_{TV} = 0.73$ |

Calculation:

a) Determine the velocity $v$ for the case when the throttle valve is open:

$$v = v_{max} = \frac{Q_G}{A_1} = \frac{0.0006}{0.0031} = 0.1935\ m \cdot s^{-1} \tag{4.3}$$

b) Adjust the throttle valve so that the relief valve has not yet opened.

The system can work with a so-called constant flow source, i.e., the relief valve is closed, and all flow is directed into the cylinder. The gradual closing of the throttle valve results in a greater pressure loss, and at the same time, the total pressure in the system increases. When the pressure set on the relief valve is reached, the valve opens. If the throttle valve continues to close, the pressure will no longer increase, but the velocity of the hydraulic cylinder will be regulated. The flow is divided; one part of the liquid is directed to the cylinder, and the other part of the liquid flows through the relief valve back into the tank. In this case, we are talking about a so-called constant pressure source.

In the case under consideration, the maximum pressure is reached so that $p_1 = p_{RV} = 16$ MPa. The pressure $p_2$ on the piston rod side of the hydraulic cylinder HM is determined from the equation of motion:

$$p_2 = \frac{p_1 \cdot A_1 - F}{A_2} = \frac{16 \cdot 10^6 \cdot 0.0031 - 15\,000}{0.0023} = 15.04 \cdot\ 10^6 Pa\,. \tag{4.4}$$

Volumetric flow $Q_2$ on the output of the hydraulic cylinder for the piston rod extension speed $v_{max}$:

$$Q_2 = A_2 \cdot v_{max} = 0.0023 \cdot 0.1935 = 0.00045\ m^3 \cdot s^{-1}\,. \tag{4.5}$$

By substituting into the equation (4.1) where volumetric flow is $Q_2 = Q_{TV}$ and pressure is $p_2 = \Delta p_{TV}$ can be calculated required throttling point $A_{TV}$:

$$Q_2 = \mu_{TV} \cdot A_{TV} \cdot \sqrt{\frac{2 \cdot p_2}{\rho}} \Rightarrow A_{TV}\,,$$

$$A_{TV} = \frac{Q_2}{\mu_{TV} \cdot \sqrt{\frac{2 \cdot p_2}{\rho}}} = \frac{0.00045}{0.73 \cdot \sqrt{\frac{2 \cdot 15.04 \cdot 10^6}{890}}} = 3.35 \cdot 10^{-6} m^2 = 3.35\ mm^2\,. \tag{4.6}$$

c) Adjust the throttle valve so that the piston rod extension speed is $v = 0.1$ m · s$^{-1}$.

As already mentioned, further closing of the throttle valve does not change the pressure, i.e., $p_1 = p_{RV} = 16$ MPa and $p_2 = 15.04$ MPa. Volumetric flow $Q_2$ on the output of the hydraulic cylinder for the piston rod extension speed $v = 0.1$ m · s$^{-1}$ can be calculated:

$$Q_2 = A_2 \cdot v = 0.0023 \cdot 0.1 = 0.00023\ m^3 \cdot s^{-1}\,. \tag{4.7}$$

Required throttling point $A_{TV}$ for piston rod extension speed $v$:

$$A_{TV} = \frac{Q_2}{\mu_{TV} \cdot \sqrt{\frac{2 \cdot p_2}{\rho}}} = \frac{0.00023}{0.73 \cdot \sqrt{\frac{2 \cdot 15.04 \cdot 10^6}{890}}} = 1.71 \cdot 10^{-6} m^2 = 1.71\ mm^2\ . \tag{4.8}$$

d) Calculate the useful power, the loss power, and the total efficiency of the drive for throttle valve settings (b) and (c).

For the setting according to point b) - the useful power $P_u$ is determined based on the force and the movement velocity of the piston rod:

$$P_u = F \cdot v = 15\ 000 \cdot 0.1935 = 2\ 902.5\ W = 2.9\ kW\ . \tag{4.9}$$

If the other elements are ignored, the greatest loss will occur at the throttle valve. The loss power $P_{lTV}$ can be calculated:

$$P_{lTV} = Q_2 \cdot p_2 = 0.00045 \cdot 15.04 \cdot 10^6 = 6\ 768\ W = 6.77\ kW\ . \tag{4.10}$$

The total efficiency $\eta_T$ of the drive:

$$\eta_T = \frac{P_u}{P_u + P_{lTV}} = \frac{2.9}{2.9 + 6.77} = 0.3\ . \tag{4.11}$$

For the setting according to point c) - in this case, some of the liquid will flow back into the tank through the relief valve, and therefore, loss will also occur on this valve. The useful power $P_u$ can be calculated:

$$P_u = F \cdot v = 15\ 000 \cdot 0.1 = 1\ 500\ W = 1.5\ kW\ . \tag{4.12}$$

The loss power $P_{lTV}$ at the throttle valve:

$$P_{lTV} = Q_2 \cdot p_2 = 0.00023 \cdot 15.04 \cdot 10^6 = 3\ 459\ W = 3.46\ kW\ . \tag{4.13}$$

The loss power $P_{lRV}$ at the relief valve:

$$\begin{gathered} P_{zRV} = p_{RV} \cdot Q_{RV} = p_{RV} \cdot (Q_G - Q_1) = p_{RV} \cdot (Q_G - A_1 \cdot v)\ , \\ P_{zRV} = 16 \cdot 10^6 \cdot (0.0006 - 0.0031 \cdot 0{,}1) = 4\ 640\ W = 4.64\ kW \end{gathered} \tag{4.14}$$

The total efficiency $\eta_T$ of the drive:

$$\eta_T = \frac{P_u}{P_u + P_{lTV} + P_{lRV}} = \frac{1.5}{1.5 + 3.46 + 4.64} = 0.156\ . \tag{4.15}$$

The flow control to the hydraulic motor by means of the throttle valve is one of the most energy-inefficient control methods. It can be used at lower powers or when the hydraulic system operates for a shorter time.

**Example 4.2**

Based on the entered values, calculate the speed $n_M$ of a rotary hydraulic motor HM.

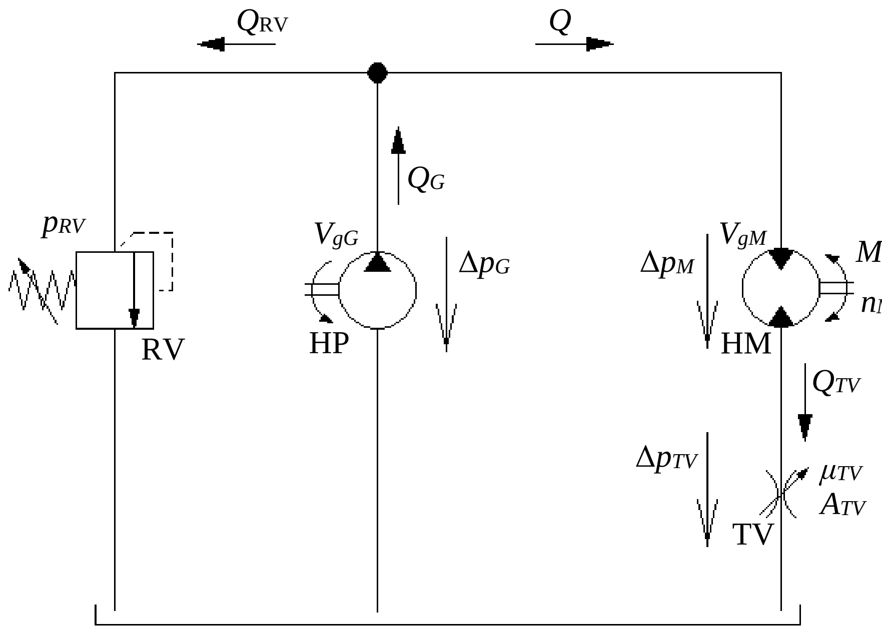



Fig. 4.3 Hydraulic scheme

*HP – hydraulic pump, HM – hydraulic cylinder, RV – relief valve, TV – throttle valve*

Entered:

| | |
|---|---|
| pressure adjusted on the relief valve | $p_{RV}$ = 16 MPa |
| volumetric flow of the hydraulic pump | $Q_G = 300\ dm^3 \cdot min^{-1} = 5 \cdot 10^{-3}\ m^3 \cdot s^{-1}$ |
| motor load torque | $M_M = 100\ N \cdot m$ |
| geometric stroke volume of motor | $V_{gM} = 157\ cm^3 = 1.57 \cdot 10^{-4}\ m^3$ |
| flow coefficient of the throttle valve | $\mu_{TV} = 0.68$ |
| oil density | $\rho = 880\ kg \cdot m^{-3}$ |
| throttle opening area of the throttle valve | $A_{TV} = 10\ mm^2 = 10 \cdot 10^{-6}\ m^2$ |

Calculation:

The calculation is performed by first selecting whether the system is a constant pressure or flow source. So first, let us assume that the relief valve is closed, and all volumetric flow from hydraulic pump HP enters the hydraulic motor HM (i.e., a constant flow source).

$$Q_G = Q\ . \tag{4.16}$$

The speed $n$ of a rotary hydraulic motor:

$$n_M = \frac{Q}{V_{gM}} = \frac{5 \cdot 10^{-3}}{1.57 \cdot 10^{-4}} = 31.85\ s^{-1}\,. \tag{4.17}$$

The resulting speed value looks realistic, but it is necessary to check the original assumption. It is necessary to determine the pressure gradient $\Delta p_M$ on the hydraulic motor:

$$\Delta p_M = \frac{2 \cdot \pi \cdot M_M}{V_{gM}} = \frac{2 \cdot \pi \cdot 100}{1.57 \cdot 10^{-4}} = 4\ MPa\,. \tag{4.18}$$

This is the pressure gradient required to overcome the load on the hydraulic motor. The pressure gradient $\Delta p_{TV}$ across the throttle valve must also be added to this pressure.

$$Q = \mu_{TV} \cdot A_{TV} \cdot \sqrt{\frac{2 \cdot \Delta p_{TV}}{\rho}} \Rightarrow\ \Delta p_{TV}\,,$$

$$\Delta p_{TV} = \frac{Q^2 \cdot \rho}{\mu_{TV}^2 \cdot A_{TV}^2 \cdot 2} = \frac{(5 \cdot 10^{-3})^2 \cdot 880}{0.68^2 \cdot (10 \cdot 10^{-6})^2 \cdot 2} = 237\ MPa \tag{4.19}$$

It is obvious that the above high-pressure value in the hydraulic system is not realistic:

$$\Delta p_M + \Delta p_{TV} > p_{RV}\,. \tag{4.20}$$

The maximum system pressure value will be 16 MPa, which is the value set on the relief valve. Therefore, the check shows that the relief valve will be open, part of the liquid will flow through the relief valve back into the tank and therefore the system will work with a constant pressure source:

$$\Delta p_G = p_{RV} = \Delta p_M + \Delta p_{TV}\,. \tag{4.21}$$

The pressure gradient $\Delta p_{TV}$ across the throttle valve:

$$\Delta p_{TV} = \Delta p_G - \Delta p_M = \Delta p_G - \frac{2 \cdot \pi \cdot M_M}{V_{gM}} = 16 \cdot 10^6 - \frac{2 \cdot \pi \cdot 100}{1.57 \cdot 10^{-4}} = 12\ MPa\,. \tag{4.22}$$

If flow losses in the hydraulic motor are neglected, then the volumetric flow, which flows through the hydraulic motor, also flows through the throttle valve. From the throttle valve setting, the volumetric flow can be determined as follows:

$$Q_{TV} = Q = \mu_{TV} \cdot A_{TV} \cdot \sqrt{\frac{2 \cdot \Delta p_{TV}}{\rho}}\,,$$

$$Q_{TV} = 0.68 \cdot 10 \cdot 10^{-6} \cdot \sqrt{\frac{2 \cdot 12 \cdot 10^6}{880}} = 1.12 \cdot 10^{-3}\ m^3 \cdot s^{-1}\,. \tag{4.23}$$

Then, the speed $n_M$ of the hydraulic motor is:

$$n_M = \frac{Q}{V_{gM}} = \frac{1.12 \cdot 10^{-3}}{1.57 \cdot 10^{-4}} = 7.15\ s^{-1}\,. \tag{4.24}$$

We will check again just to be sure. In this case, assuming that this is a constant pressure source, we will double-check that the volumetric flow through the hydraulic motor is less than the flow supplied by the hydraulic pump, which is OK, i.e. $Q < Q_G$.

We will continue with the example. How will the motor speed be changed when the load (i.e., torque) on the hydraulic motor is increased to $M_M = 150$ N · m?

We will assume a constant pressure source. If the load on the hydraulic motor increases, a greater pressure gradient $\Delta p_M$ will be required to overcome it:

$$\Delta p_M = \frac{2 \cdot \pi \cdot M_M}{V_{gM}} = \frac{2 \cdot \pi \cdot 150}{1.57 \cdot 10^{-4}} = 6\ MPa\ . \tag{4.25}$$

Therefore, the pressure gradient $\Delta p_{ŠV}$ on the throttle valve is changed:

$$\Delta p_{TV} = \Delta p_G - \Delta p_M = \Delta p_G - \frac{2 \cdot \pi \cdot M_M}{V_{gM}}\,,$$
$$\Delta p_{TV} = 16 \cdot 10^6 - \frac{2 \cdot \pi \cdot 150}{1.57 \cdot 10^{-4}} = 10\ MPa\ . \tag{4.26}$$

The equation for the volumetric flow through the throttle valve shows that the flow is dependent on the pressure gradient; therefore, the flow through the entire branch of the circuit will be changed:

$$Q_{TV} = Q = \mu_{TV} \cdot A_{TV} \cdot \sqrt{\frac{2 \cdot \Delta p_{\mathrm{TV}}}{\rho}}\,,$$
$$Q = 0.68 \cdot 10 \cdot 10^{-6} \cdot \sqrt{\frac{2 \cdot 10 \cdot 10^6}{880}} = 1.025 \cdot 10^{-3}\ m^3 \cdot s^{-1}\ . \tag{4.27}$$

A reduction in flow naturally leads to a reduction in speed $n_M$ of the motor:

$$n_M = \frac{Q}{V_{gM}} = \frac{1.025 \cdot 10^{-3}}{1.57 \cdot 10^{-4}} = 6.53\ s^{-1}\ . \tag{4.28}$$

**Example 4.3**

Calculate the throttle valve setting. The condition is to reach speed 1 200 min$^{-1}$ for the hydraulic motor HM2.

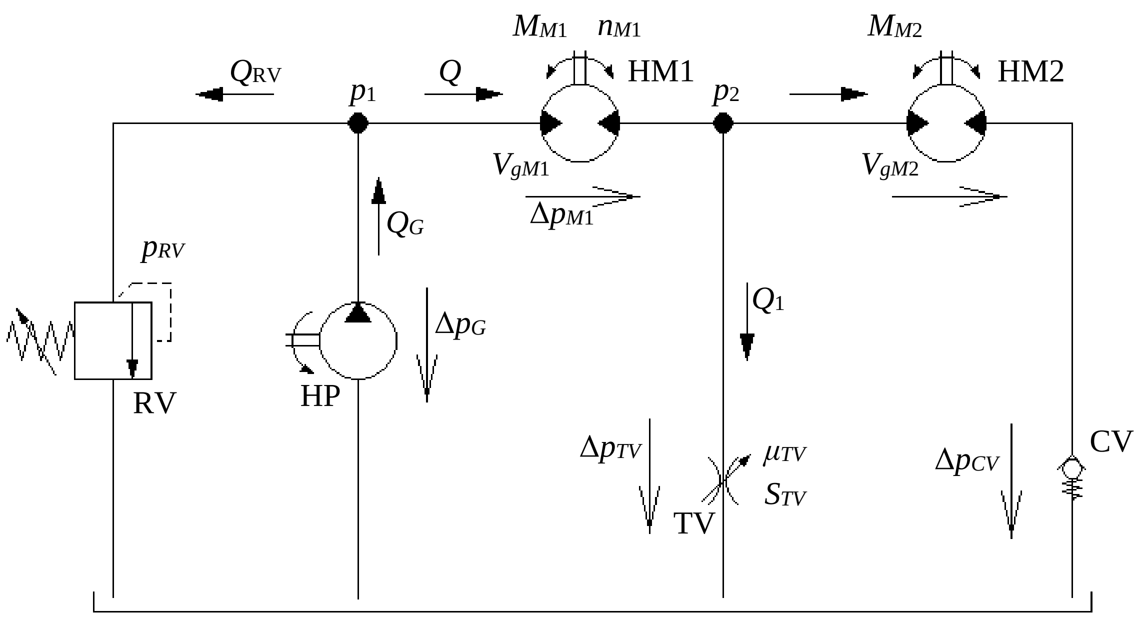



Fig. 4.4 Hydraulic scheme

*HP – hydraulic pump, HM1, HM2 – hydraulic motors, RV – relief valve, TV – throttle valve, CV – check valve*

Entered:

| | |
|---|---|
| pressure adjusted on the relief valve | $p_{RV} = 16$ MPa |
| pressure drop on the check valve | $\Delta p_{CV} = 1$ MPa |
| volumetric flow of the hydraulic pump | $Q_G = 120\ \mathrm{dm^3 \cdot min^{-1}} = 2 \cdot 10^{-3}\ \mathrm{m^3 \cdot s^{-1}}$ |
| load torque on the motor HM1 | $M_{M1} = 20\ \mathrm{N \cdot m}$ |
| load torque on the motor HM2 | $M_{M2} = 100\ \mathrm{N \cdot m}$ |
| geometric stroke volume of the motor HM1 | $V_{gM1} = 35\ \mathrm{cm^3} = 35 \cdot 10^{-6}\ \mathrm{m^3}$ |
| geometric stroke volume of the motor HM2 | $V_{gM2} = 70\ \mathrm{cm^3} = 70 \cdot 10^{-6}\ \mathrm{m^3}$ |
| flow coefficient of the throttle valve | $\mu_{TV} = 0.73$ |
| oil density | $\rho = 890\ \mathrm{kg \cdot m^{-3}}$ |
| required speed of the motor HM2 | $n_{M2} = 1\ 200\ \mathrm{min^{-1}}$ |

Calculate:

opening of throttle valve $A_{TV} = ?$

Calculation:

We will assume a constant flow source, i.e., $p_1 < p_{RV}$ and $Q = Q_G$. First, we will check the pressure. The total pressure $p_1$ is given by the sum of the pressure gradient on the motor HM1, the pressure gradient on the motor HM2 and the pressure drop on the check valve CV.

First, the pressure $p_2$ is calculated:

$$p_2 = \Delta p_{M2} + \Delta p_{CV} = \frac{2 \cdot \pi \cdot M_{M2}}{V_{gM2}} + \Delta p_{CV} \,, \qquad (4.29)$$

$$p_2 = \frac{2 \cdot \pi \cdot 100}{70 \cdot 10^{-6}} + 1 \cdot 10^6 = 9.97 \, MPa \,.$$

And the pressure $p_1$:

$$p_1 = \Delta p_{M1} + p_2 = \frac{2 \cdot \pi \cdot M_{M1}}{V_{gM1}} + p_2 \,, \qquad (4.30)$$

$$p_1 = \frac{2 \cdot \pi \cdot 20}{35 \cdot 10^{-6}} + 9.97 \cdot 10^6 = 13.56 \, MPa \,.$$

The condition $p_1 < p_{RV}$ is fulfilled. Therefore, the relief valve will be closed. All the liquid flows through the first motor HM1, and then it is divided into flow $Q_1$ through the throttle valve and flow $Q_2$ through the second motor HM2. In order to determine the throttle valve setting $A_{TV}$, it is necessary to know the pressure gradient across the throttle valve $\Delta p_{TV}$ (which is equal to the pressure $p_2$) and the flow through the throttle valve $Q_1$. This flow is found from the difference between the flow $Q$ and the flow $Q_2$ required to turn motor HM2 to the desired speed $n_{M2}$:

$$Q_1 = Q - Q_2 = Q - V_{gM2} \cdot n_{M2} \,, \qquad (4.31)$$

$$Q_1 = 2 \cdot 10^{-3} - 70 \cdot 10^{-6} \cdot \frac{1\,200}{60} = 0.0006 \, m^3 \cdot s^{-1} \,.$$

The opening of throttle valve $A_{TV}$:

$$Q_1 = \mu_{TV} \cdot A_{TV} \cdot \sqrt{\frac{2 \cdot \Delta p_{TV}}{\rho}} \Rightarrow A_{TV} \,, \qquad (4.32)$$

$$A_{TV} = \frac{Q_1}{\mu_{TV} \cdot \sqrt{\frac{2 \cdot \Delta p_{TV}}{\rho}}} = \frac{0.0006}{0.73 \cdot \sqrt{\frac{2 \cdot 9.97 \cdot 10^6}{890}}} = 5.49 \cdot 10^{-6} \, m^2 = 5.49 \, mm^2$$

# 5. Measurement of $Q$ - $\Delta p$ characteristic of 2-way flow control valve and characteristic of the conventional throttle valve

**Assignment**

1) Measure $Q$ - $\Delta p$ characteristic of 2-way flow control valve type GFG2PK18-10 produced by Parker company [15]. Perform the measurements for three valve settings (max. 6 divisions).
2) Replace the 2-way flow control valve with a conventional throttle valve and perform the characteristic measurement for the valve set to one turn from the closed position.
3) Compare the measurement characteristics of both valves.

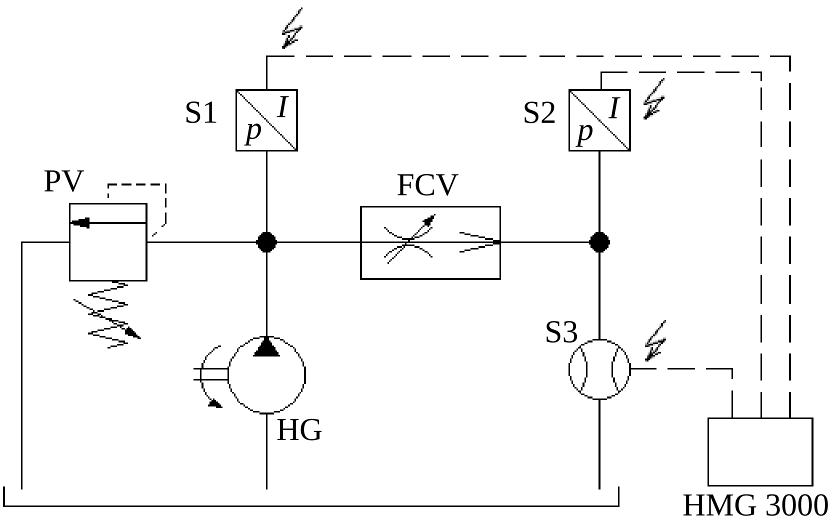



Fig. 5.1 Measurement scheme of $Q$ - $\Delta p$ characteristic of 2-way flow control valve

*HG –hydraulic pump, PV –relief valve, S1 and S2 – pressure sensors, FCV – 2-way flow control valve, S3 – flowmeter, HMG 3000 – measuring system Hydac*

**Measurement procedure**

1) We connect the circuit according to the measurement scheme, connect the electrical outputs of the pressure sensors and the flowmeter to the Hydac HMG 3000 measuring system [16] and set the appropriate channels (the setting of the measuring system is described in the manual "Pressure and flow measurements with the Hydac measuring system", which is available in the laboratory).
2) We set the relief valve RV to a minimum. We set, e.g., 2 divisions on the 2-way flow control valve and turn on the hydraulic pump.
3) We will read the pressures $p_1$ and $p_2$ (in front of and after the valve) and the oil volumetric flow $Q$ through the valve from the display of the HMG 3000 measuring system. We will record the values in the table.
4) We use the valve RV to set a higher pressure p1 (we choose a step of 2 to 3 bar) and for each pressure value we write the values of $p_1$, $p_2$ and $Q$ in the table.
5) We repeat the measurements for two other settings of the 2-way flow control valve (e.g., 4 and 6 divisions, according to the teacher's instructions).

6) Similarly, we will perform the measurements for the conventional throttle valve at one revolution of the control screw from the closed position.

**Used equipment**

- Parker teaching panel with elements (2-way flow control valve, throttle valve), the liquid supply from the hydraulic pump and return lines (to tank) are realized by means of a distribution board in the lower left part.
- Hoses and T-pieces are used to connect pressure sensors.
- Measuring system Hydac HMG 3000, pressure sensors, flowmeter.


Fig. 5.2 Used equipment

*S1 and S2 – pressure sensors, FCV – 2-way flow control valve, S3 – flowmeter, HMG 3000 – measuring system Hydac*

**Parameters of sensors**

- Pressure sensors – Hydac HDA 4446-A-250-000, measuring range (0 ÷ 250) bar, output signal (4 ÷ 20) mA [17].
- Flowmeter – Hydac EVS 3108-H-0020-000, measuring range (1.2 ÷ 20) $dm^3 \cdot min^{-1}$, output signal (0.5 ÷ 4.5) V [18]. There is no need to set the flow meter, the HMG 3000 measuring system detects it automatically.

**Processing of results**

To construct the $Q$ - $\Delta p$ characteristic, it is necessary to know the pressure loss on the throttle valve:

$$\Delta p = p_1 - p_2 \,, \tag{5.1}$$

An example of measured results and their graphical processing are shown below.

Tab 5.1 2-way flow control valve – 4 divisions

| $p_1$ [bar] | $p_2$ [bar] | $\Delta p$ [bar] | $Q$ [$dm^3 \cdot min^{-1}$] |
|---|---|---|---|
| 0 | 0 | 0 | 0 |
| 6.14 | 2 | 4.14 | 2.73 |
| 8.02 | 2.81 | 5.21 | 3.36 |
| … | … | … | … |
| 18.03 | 3.01 | 15.02 | 3.86 |
| 20 | 3.01 | 16.99 | 3.86 |
| 22.02 | 3.02 | 19 | 3.87 |

Tab 5.2 Conventional throttle valve – one revolution of the control screw

| $p_1$ [bar] | $p_2$ [bar] | $\Delta p$ [bar] | $Q$ [$dm^3 \cdot min^{-1}$] |
|---|---|---|---|
| 0 | 0 | 0 | 0 |
| 8.01 | 3.21 | 4.8 | 4.16 |
| 10.02 | 3.86 | 6.16 | 5.12 |
| … | … | … | … |
| 20.05 | 6.84 | 13.21 | 8.97 |
| 24.01 | 7.91 | 16.1 | 10.39 |
| 26.03 | 8.27 | 17.76 | 11.14 |

Record the results of all measurements in one graph.

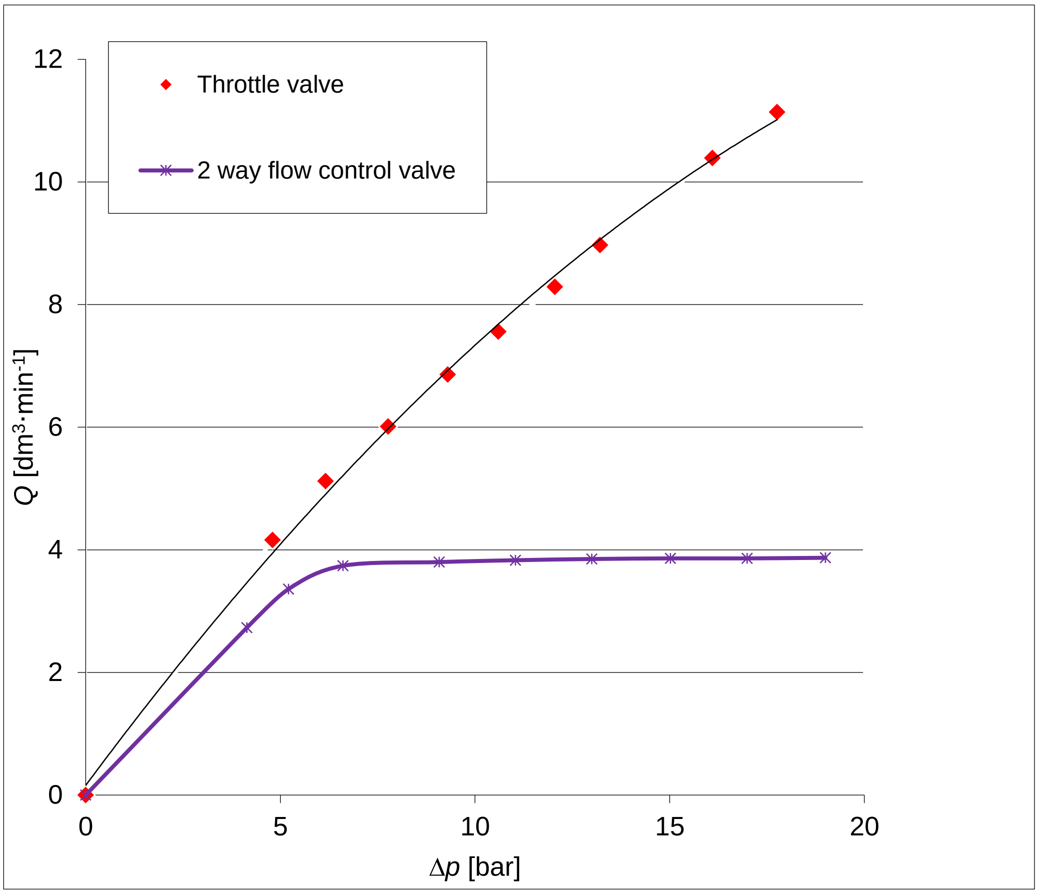



Fig. 5.3 A graphical example of measured results

# 6. Calculation of parameters of the real circuit

**Example 6.1**

The closed hydrostatic drive is used for lifting the mass load *m*. The maximum system pressure $p_{RV}$ is set on the relief valve RV. The winding drum with the radius $R_D$ is connected via a gearbox to the hydraulic motor HM. Find out the required electric motor power $P_G$ for the hydraulic pump HP drive. Calculate the lifting speed *v* of the mass load. Neglect pressure losses in pipes.

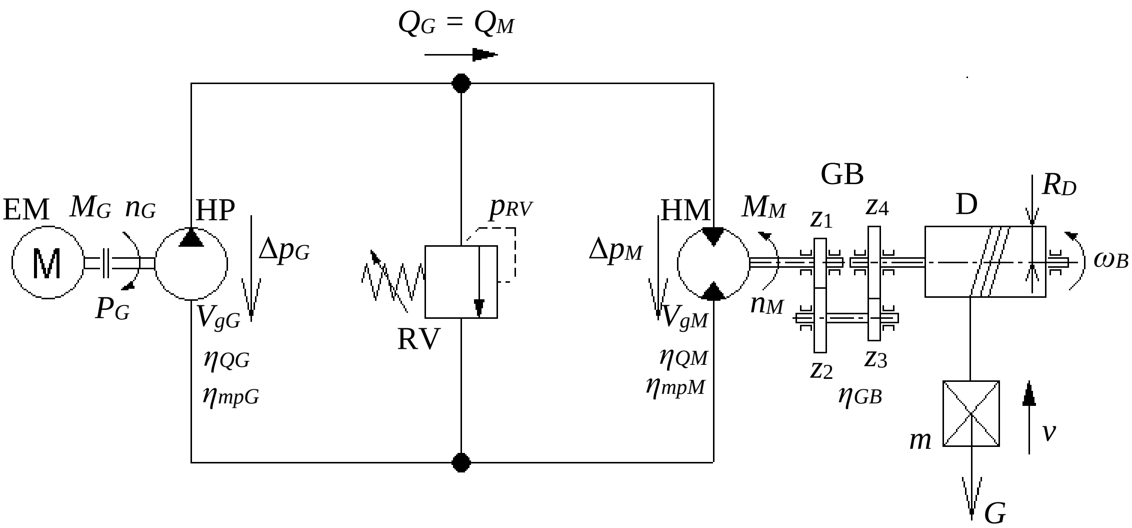



Fig. 6.1 Hydraulic scheme

*EM – electric motor, HP –hydraulic pump, RV –relief valve, HM – hydraulic motor, GB - gearbox, D – winding drum*

<u>Entered:</u>

| | |
|---|---|
| geometric stroke volume of the hydraulic pump | $V_{gG} = 20 \cdot 10^{-6}$ m$^3$ = 20 cm$^3$ |
| speed of the hydraulic pump | $n_G = 24$ s$^{-1}$ |
| geometric stroke volume of the hydraulic motor | $V_{gM} = 100 \cdot 10^{-6}$ m$^3$ = 100 cm$^3$ |
| volumetric efficiency of the hydraulic pump | $\eta_{QG} = 0.96$ |
| mechanical-hydraulic efficiency of the hydraulic pump | $\eta_{mhG} = 0.95$ |
| volumetric efficiency of the hydraulic motor | $\eta_{QM} = 0.92$ |
| mechanical-hydraulic efficiency of the hydraulic motor | $\eta_{mhM} = 0.95$ |
| pressure adjusted on the relief valve | $p_{RV} = 32$ MPa |
| number of teeth on gears | $z_1 = 12$, $z_2 = 24$, $z_3 = 12$, $z_4 = 36$ |
| gearbox efficiency | $\eta_{GB} = 0.96$ |
| radius of winding drum | $R_D = 0.25$ m |
| load weight | $m = 600$ kg |

<u>Calculate:</u> $v$ = ?, $P_G$ = ?

<u>Calculation:</u>

The gravitational force $G$ from the mass load:

$$G = m \cdot g = 600 \cdot 9.81 = 5\,886\; N\,. \tag{6.1}$$

The torque $M_D$ on the drum:

$$M_D = G \cdot R_D = 5\,886 \cdot 0.25 = 1\,471.5\; N \cdot m\,. \tag{6.2}$$

The gear ratio $i_{GB}$ of the gearbox:

$$i_{GB} = \frac{z_4}{z_3} \cdot \frac{z_2}{z_1} = \frac{36}{12} \cdot \frac{24}{12} = 6\,. \tag{6.3}$$

The torque $M_M$ on the hydraulic motor:

$$M_M = M_D \cdot \frac{1}{i_{GB}} \cdot \frac{1}{\eta_{GB}} = 1\,471.5 \cdot \frac{1}{6} \cdot \frac{1}{0.96} = 255.5\; N \cdot m\,. \tag{6.4}$$

The pressure gradient $\Delta p_M$ on the hydraulic motor:

$$\Delta p_M = \frac{2 \cdot \pi \cdot M_M}{V_{gM}} \cdot \frac{1}{\eta_{mhM}} = \frac{2 \cdot \pi \cdot 255.5}{100 \cdot 10^{-6}} \cdot \frac{1}{0.95} = 16.9 \cdot 10^6\; Pa = 16.9\; MPa\,. \tag{6.5}$$

If pressure losses in pipes are neglected, then $\Delta p_M = \Delta p_G$ =16.9 MPa, which is less compared to the pressure set on the relief valve. Therefore, the relief valve RV remains closed.

The torque $M_G$ on the shaft of the hydraulic pump:

$$M_G = \frac{V_{gG} \cdot \Delta p_G}{2 \cdot \pi} \cdot \frac{1}{\eta_{mhG}} = \frac{20 \cdot 10^{-6} \cdot 16.9 \cdot 10^6}{2 \cdot \pi} \cdot \frac{1}{0.95} = 56.6\; N \cdot m\,. \tag{6.6}$$

The required power $P_G$ of the electric motor:

$$P_G = M_G \cdot \omega_G = M_G \cdot 2 \cdot \pi \cdot n_G = 56.6 \cdot 2 \cdot \pi \cdot 24 = 8\,535\; W = 8.53\; kW\,. \tag{6.7}$$

The real volumetric flow $Q_G$ of the hydraulic pump:

$$Q_G = V_{gG} \cdot n_G \cdot \eta_{QG} = 20 \cdot 10^{-6} \cdot 24 \cdot 0.96 = 4.61 \cdot 10^{-4}\; m^3 \cdot s^{-1}\,,$$
$$Q_G = 27.66\; dm^3 \cdot min^{-1}. \tag{6.8}$$

Since the relief valve is closed, this volumetric flow enters the hydraulic motor (i.e., $Q_G = Q_M$ ), and the speed $n_M$ of the hydraulic motor is:

$$n_M = \frac{Q_M}{V_{gM}} \cdot \eta_{QM} = \frac{4.61 \cdot 10^{-4}}{100 \cdot 10^{-6}} \cdot 0.92 = 4.24\; s^{-1}\,. \tag{6.9}$$

The speed $n_D$ of the drum:

$$n_D = n_M \cdot \frac{1}{i_{GB}} = 4.24 \cdot \frac{1}{6} = 0.707\ s^{-1}\ . \tag{6.10}$$

The lifting speed $v$ of the mass load:

$$v = \omega_D \cdot R_D = 2 \cdot \pi \cdot n_D \cdot R_D = 2 \cdot \pi \cdot 0.707 \cdot 0.25 = 1.11\ m \cdot s^{-1}\ . \tag{6.11}$$

The calculation of the real circuit, including a description of the selection of individual elements of a hydraulic system, is given in Chapter 10.

# 7. Calculation of the start-up time of the hydraulic motor

### Example 7.1

Determine the start-up time $t_s$ of the hydraulic motor, which is connected to a pressure source $p_1$. At the initial time ($t_0 = 0$), the velocity of the hydraulic motor is zero ($v_0 = 0$). Pressure losses in the pipes are neglected. Neglect the weight of the piston, piston rod and liquid in the hydraulic motor.

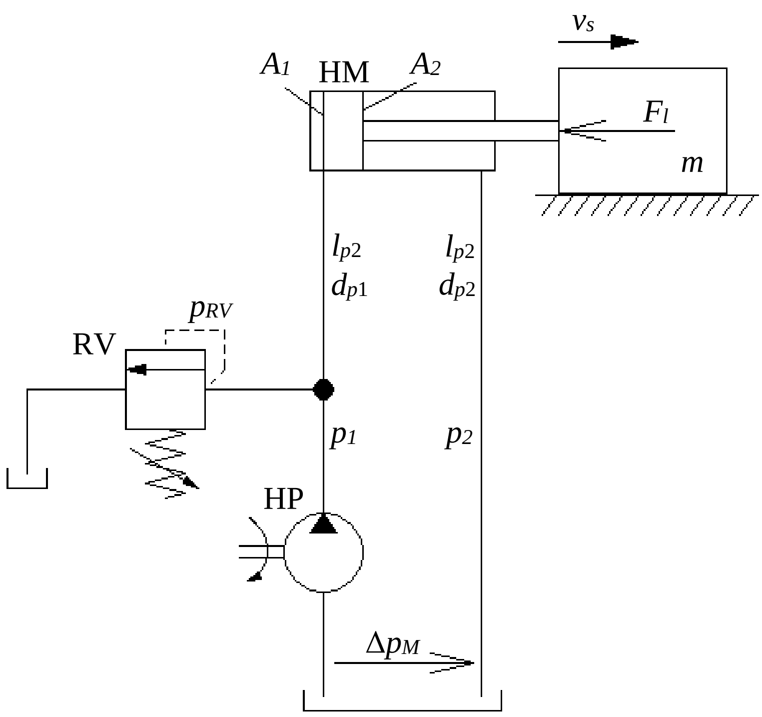



Fig. 7.1 Hydraulic scheme

*HP –hydraulic pump, RV –relief valve, HM – hydraulic cylinder*

<u>Entered:</u>

| | |
|---|---|
| piston area | $A_1 = 0.0031\ m^2$ |
| intercircular area on the piston rod side | $A_2 = 0.0023\ m^2$ |
| force acting on the hydraulic motor | $F_l = 15\ 000\ N$ |
| steady velocity of the piston rod | $v_s = 0.1\ m \cdot s^{-1}$ |
| lengths of the pipes | $l_{p1} = l_{p2} = 10\ m$ |
| diameters of the pipes | $d_{p1} = d_{p2} = 10\ mm = 0.01\ m$ |
| efficiency of the hydraulic motor | $\eta_{TM} = 1$ |
| load weight | $m = 3\ 000\ kg$ |
| oil density | $\rho = 900\ kg \cdot m^{-3}$ |

<u>Calculate:</u> $t_s = ?$

Calculation:

Equation of motion for the piston rod extension:

$$F_M - F_l = F_d \,. \tag{7.1}$$

where $F_M$ [N] is the force on the hydraulic motor, $F_l$ [N] the load force, and $F_d$ [N] is the dynamic force.

If the mass of the load $m$ = const., then the dynamic force can be expressed by the formula:

$$\sum F_{di} = m \cdot a \,,$$
$$\sum F_{di} = m_{Tred} \cdot \frac{dv}{dt} \,. \tag{7.2}$$

where $m$ [kg] is the mass, $a$ [m · s$^{-2}$] is the acceleration of the piston rod of the hydraulic motor, $m_{Tred}$ [kg] is the total reduced mass on the piston rod of the hydraulic motor, $dv$ [m · s$^{-1}$] is the velocity change, and $dt$ [s] is the time change.

If at the initial time $t_0 = 0$, the velocity of the hydraulic motor is zero ($v_0 = 0$), the following relation can be considered for the linear start-up of the hydraulic motor:

$$\frac{dv}{dt} \cong \frac{\Delta v}{\Delta t} = \frac{v_s - v_0}{t_s - t_0} = \frac{v_s}{t_s} \,. \tag{7.3}$$

where $v_0$ [m · s$^{-1}$] is the initial velocity of the piston rod of the hydraulic motor, $v_s$ [m · s$^{-1}$] is the steady velocity of the piston rod of the hydraulic motor, $t_0$ [s] is the initial time, which corresponds to the initial velocity of the piston rod, and $t_s$ [s] is the start-up time of the hydraulic motor.

To determine the force $F_M$ from the liquid on the hydraulic motor, the relationship applies:

$$F_M = p_1 \cdot A_1 - p_2 \cdot A_2 \,, \tag{7.4}$$

where $p_1$ [Pa] is the pressure on the piston side of the hydraulic motor, $p_2$ [Pa] is the pressure on the piston rod side of the hydraulic motor, $A_1$ [m$^2$] is the area on the piston side of the hydraulic motor, and $A_2$ [m$^2$] is the area on piston rod side of hydraulic motor.

By substituting into equation (7.1) it is possible to write the equation of motion for the extension of the piston rod of the hydraulic motor in the form:

$$p_1 \cdot A_1 - p_2 \cdot A_2 - F_l = m_{Tred} \cdot \frac{v_s}{t_s} \,. \tag{7.5}$$

The pressure in the waste (return) branch can be considered zero ($p_2 = 0$), then it is possible to express the starting time $t_s$ of the hydraulic motor:

$$p_1 \cdot A_1 - F_l = m_{Tred} \cdot \frac{v_s}{t_s} \Rightarrow t_s = \frac{m_{Tred} \cdot v_s}{p_1 \cdot A_1 - F_l} \,. \tag{7.6}$$

In order to express the total reduced mass, the reduced mass of the liquid in the pressure pipe at the input to the hydraulic motor must be considered (for short pipes this can be neglected).The reduced mass of the liquid in the pressure pipe can be determined from the equality of kinetic energies:

$$\frac{1}{2} \cdot m_{Lp1} \cdot v_{p1}^2 = \frac{1}{2} \cdot m_{p1red} \cdot v_s^2 \Rightarrow m_{p1red} = m_{Lp1} \cdot \left(\frac{v_{p1}}{v_s}\right)^2 . \tag{7.7}$$

where $m_{Lp1}$ [kg] is the liquid mass in the pipe at the input to the hydraulic motor, $v_{p1}$ [$m \cdot s^{-1}$] is the liquid flow velocity in the pipe at the input to the hydraulic motor, and $m_{p1red}$ [kg] is the reduced liquid mass in the pipe at the input to the hydraulic motor.

Although we do not know the volumetric flow, it is evident that the volumetric flow in the pipe is equal to the volumetric flow on the hydraulic motor, i.e., $Q_{p1} = Q_{M1}$. Using the continuity equation:

$$Q_{p1} = Q_{M1} = v_{p1} \cdot A_{p1} = v_s \cdot A_1 \Rightarrow \frac{v_{p1}}{v_s} = \frac{A_1}{A_{p1}} . \tag{7.8}$$

where $Q_{p1}$ [$m^3 \cdot s^{-1}$] is the volumetric flow in the pipe at the input to the hydraulic motor, $Q_{M1}$ [$m^3 \cdot s^{-1}$] is the volumetric flow on the hydraulic motor, and $A_{p1}$ [$m^2$] is the pipe cross-sectional area at the input to the hydraulic motor.

The liquid weight in the pressure pipe at the inlet to the hydraulic motor $m_{Lp1}$:

$$\rho = \frac{m_{Lp1}}{V_{Lp1}} \Rightarrow m_{Lp1} = \rho \cdot V_{Lp1} = \rho \cdot A_{p1} \cdot l_{p1} . \tag{7.9}$$

where $\rho$ [$kg \cdot m^{-3}$] is the liquid density, $V_{Lp1}$ [$m^3$] is the liquid volume in the pipe at the input to the hydraulic motor, and $l_{p1}$ [m] is the pipe length at the input to the hydraulic motor.

By substituting the previous two relations (7.8) and (7.9) into the equation (7.7) for the reduced mass of the liquid in the pressure pipe:

$$m_{p1red} = \rho \cdot A_{p1} \cdot l_{p1} \cdot \left(\frac{A_1}{A_{p1}}\right)^2 . \tag{7.10}$$

The same procedure would be used in the case of determining the reduced mass of the liquid in the return pipe $m_{p2red}$:

$$m_{p2red} = \rho \cdot A_{p2} \cdot l_{p2} \cdot \left(\frac{A_2}{A_{p2}}\right)^2 . \tag{7.11}$$

where $m_{p2red}$ [kg] is the volumetric flow in the pipe at the output of the hydraulic motor, $A_{p2}$ [m2] is the pipe cross-sectional area at the output of the hydraulic motor, and $l_{p2}$ [m] is the pipe length at the output of the hydraulic motor.

The pipe area $A_{p1} = A_{p2}$ can be calculated:

$$A_{p1} = \frac{\pi \cdot d_{p1}^2}{4} = \frac{\pi \cdot 0.01^2}{4} = 7.85 \cdot 10^{-4}\ m^2 . \tag{7.12}$$

The total reduced weight $m_{Tred}$:

$$m_{Tred} = m + m_{p1red} + m_{p2red} \,,$$

$$m_{Tred} = m + \rho \cdot A_{p1} \cdot l_{p1} \cdot \left(\frac{A_1}{A_{p1}}\right)^2 + \rho \cdot A_{p2} \cdot l_{p2} \cdot \left(\frac{A_2}{A_{p2}}\right)^2 \,, \tag{7.13}$$

$$m_{Cred} = 3\,000 + 900 \cdot 7.85 \cdot 10^{-4} \cdot 10 \cdot \left(\frac{0.0031}{7.85 \cdot 10^{-4}}\right)^2 + 900 \cdot 7.85 \cdot 10^{-4} \cdot 10 \cdot \left(\frac{0.0023}{7.85 \cdot 10^{-4}}\right)^2 = 4\,708.5\ kg$$

And the start-up time $t_s$ of the hydraulic motor to the steady velocity:

$$t_s = \frac{m_{Tred} \cdot v_s}{p_1 \cdot A_1 - F_l} = \frac{4\,708.5 \cdot 0.1}{16 \cdot 10^6 \cdot 0.00031 - 15\,000} = 1.36 \cdot 10^{-2}\ s\,. \tag{7.14}$$

**Example 7.2**

Determine the load lifting velocity $v$ and start-up time $t_s$ of the hydraulic motor, which is connected to a pressure source. At the initial time $t_0 = 0$, the speed of the hydraulic motor is zero ($n_0 = 0$). Pressure losses in pipes and gear box efficiency are neglected. Also, the weight of the liquid in the pipe will be neglected (it is a short pipe).

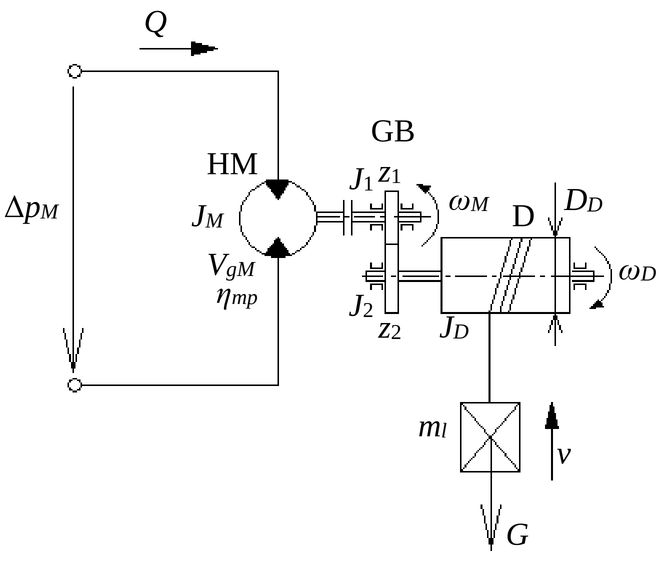



Fig. 7.2 Hydraulic scheme

*HM – hydraulic motor, GB - gearbox, D – winding drum*

<u>Entered:</u>

| | |
|---|---|
| volumetric flow | $Q = 1 \cdot 10^{-3}\ m^3 \cdot s^{-1} = 60\ dm^3 \cdot min^{-1}$ |
| pressure gradient on the hydraulic motor | $\Delta p_M = 16$ MPa |
| geometric stroke volume of the hydraulic motor | $V_{gM} = 1 \cdot 10^{-3}\ m^3 = 1\ cm^3$ |
| mechanical-hydraulic efficiency of hyd. motor | $\eta_{mhM} = 0.95$ |

| | |
|---|---|
| moment of inertia of the hydraulic motor | $J_M$ = 1.2 kg · m$^2$ |
| number of teeth of the small gear | $z_1$ = 20 |
| moment of inertia of the small gear | $J_1$ = 0.1 kg · m$^2$ |
| number of teeth of the large gear | $z_2$ = 40 |
| moment of inertia of the large gear | $J_2$ = 0.4 kg · m$^2$ |
| drum diameter | $D_D$ = 0.5 m |
| moment of inertia of drum | $J_D$ = 15 kg · m$^2$ |
| load weight | $m_l$ = 1 500 kg |

Calculate: $t_s$ = ?, $v$ = ?

Calculation:

The torque equation on hydraulic motor HM:

$$M_M - M_l = M_d \,. \tag{7.15}$$

where $M_M$ [N · m] is the torque on the hydraulic motor, $M_l$ [N · m] is the load torque, and $M_d$ [N · m] is the dynamic torque.

$$\sum M_{di} = J \cdot \varepsilon \,,$$
$$\sum M_{di} = J_{Tred} \cdot \frac{d\omega}{dt} \,. \tag{7.16}$$

where $J$ [kg · m$^2$] is the moment of inertia, $\varepsilon$ [s$^{-2}$] is the angular acceleration, $J_{Tred}$ [kg · m$^2$] is the total reduced moment of inertia, $d\omega$ [s$^{-1}$] is the change in angular velocity, and $dt$ [s] is the time change.

If at the initial time ($t_0$ = 0), the speed of the hydraulic motor is zero ($n_0$ = 0), then assuming a linear start-up:

$$\frac{d\omega}{dt} \cong \frac{\Delta\omega}{\Delta t} = 2 \cdot \pi \cdot \frac{n_s - n_0}{t_s - t_0} = 2 \cdot \pi \cdot \frac{n_s}{t_s} \,. \tag{7.17}$$

where $t_0$ [s] is the initial time, $t_s$ [s] is the start-up time of the hydraulic motor, $n_0$ [s$^{-1}$] is the initial speed of the hydraulic motor, and $n_s$ [s$^{-1}$] is the steady speed of the hydraulic motor.

The start-up time $t_s$ of the hydraulic motor (without considering the effect of gearbox efficiency):

$$M_M - M_l = J_{Tred} \cdot 2 \cdot \pi \cdot \frac{n_s}{t_s} \Rightarrow t_s = \frac{J_{Tred} \cdot 2 \cdot \pi \cdot n_s}{M_M - M_l} \,. \tag{7.18}$$

The gear ratio $i$:

$$i = \frac{\omega_M}{\omega_D} = \frac{z_2}{z_1} = \frac{40}{20} = 2\,. \tag{7.19}$$

where $i$ [-] is the gear ratio of the gearbox, $\omega_M$ [$s^{-1}$] is the angular velocity on the output shaft of the hydraulic motor, $\omega_D$ [$s^{-1}$] is the angular velocity on the winding drum, $z_1$ [-] is the number of teeth of the small gear wheel, and $z_2$ [-] is the number of teeth of the large gear wheel.

The load torque $M_l$:

$$M_l = M_D \cdot \frac{1}{i} = G \cdot \frac{D_D}{2} \cdot \frac{1}{i} = m \cdot g \cdot \frac{D_D}{2} \cdot \frac{1}{i} = 1\,500 \cdot 9.81 \cdot 0.25 \cdot \frac{1}{2}, \tag{7.20}$$
$$M_z = 1\,839.4\ N \cdot m\,.$$

where $M_B$ [N · m] is the torque on the winding drum, $G$ [N] is the gravitational force from the mass load, and $D_B$ [m] is the diameter of the winding drum.

The torque of the hydraulic motor $M_M$:

$$M_M = \frac{\Delta p_M \cdot V_{gM}}{2 \cdot \pi} \cdot \eta_{mhM} = \frac{16 \cdot 10^6 \cdot 1 \cdot 10^{-3}}{2 \cdot \pi} \cdot 0{,}95 = 2\,419.2\ N \cdot m\,. \tag{7.21}$$

where $\eta_{mhM}$ [-] is mechanical-pressure efficiency of the hydraulic motor.

Reduction of moments of inertia $J_{red1}$ large gear and drum on the output shaft of the hydraulic motor:

$$\frac{1}{2} \cdot J_{red1} \cdot \omega_M^2 = \frac{1}{2} \cdot J_D \cdot \omega_D^2 + \frac{1}{2} \cdot J_2 \cdot \omega_D^2 \Rightarrow J_{red1}\,,$$
$$J_{red1} = (J_D + J_2) \cdot \left(\frac{\omega_D}{\omega_M}\right)^2 = (J_D + J_2) \cdot \frac{1}{i^2}\,. \tag{7.22}$$

where $J_{red1}$ [$kg \cdot m^2$] is the reduced moment of inertia for rotary motion.

Reduction of linear motion of mass load $J_{red2}$ on the output shaft of the hydraulic motor:

$$\frac{1}{2} \cdot J_{red2} \cdot \omega_M^2 = \frac{1}{2} \cdot m_l \cdot v^2 \Rightarrow J_{red2}\,,$$
$$J_{red2} = m_l \cdot \left(\frac{v}{\omega_M}\right)^2 = m_l \cdot \left(\frac{R_D \cdot \omega_D}{\omega_M}\right)^2 = m_l \cdot R_D^2 \cdot \frac{1}{i^2}\,. \tag{7.23}$$

where $J_{red2}$ [$kg \cdot m^2$] is the reduced moment of inertia for linear motion, and $R_B$ [m] is the radius of the winding drum.

The total reduced moment of inertia:

$$J_{Tred} = J_M + J_1 + J_{red1} + J_{red2} = J_M + J_1 + (J_2 + J_D) \cdot \frac{1}{i^2} + m_l \cdot R_D^2 \cdot \frac{1}{i^2}, \tag{7.24}$$
$$J_{Tred} = 1.2 + 0.1 + (0.4 + 15) \cdot \frac{1}{2^2} + 1\,500 \cdot 0.25^2 \cdot \frac{1}{2^2} = 28.59\ kg \cdot m^2\,.$$

The steady speed $n_s$ of the hydraulic motor:

$$Q = V_{gM} \cdot n_s \Rightarrow n_s = \frac{Q}{V_{gM}} = \frac{1 \cdot 10^{-3}}{1 \cdot 10^{-3}} = 1\ s^{-1}\ . \tag{7.25}$$

The start-up time $t_s$ of the hydraulic motor:

$$t_s = \frac{J_{Tred} \cdot 2 \cdot \pi \cdot n_s}{M_M - M_l} = \frac{28.59 \cdot 2 \cdot \pi \cdot 1}{2\,419.2 - 1\,839.4} = 0.31\ s\ . \tag{7.26}$$

The movement velocity $v$ of the mass:

$$v = \omega_D \cdot R_D = \omega_M \cdot \frac{1}{i} \cdot R_D = 2 \cdot \pi \cdot n_s \cdot \frac{1}{i} \cdot R_D = 2 \cdot \pi \cdot 1 \cdot \frac{1}{2} \cdot 0.25\ , \tag{7.27}$$

$$v = 0.785\ m \cdot s^{-1}\ .$$

# 8. Hydraulic accumulators

## 8.1 Application of accumulators in circuits

Accumulators find many applications in hydraulic systems. They can be used as a source of pressure, flow or as a damping element.

**Damping of pulsations**

Hydraulic pumps are the source of flow and pressure pulsations in a system. The occurrence of pulsations is based on their construction principles and can result in the uneven operation of a hydraulic motor. In addition, the pulsations can also cause other undesirable phenomena, such as increased noise, pipe vibrations, or valve oscillations and lead to their damage. To reduce pulsations, the accumulator is located in the pressure branch immediately behind the hydraulic pump Fig. 8.1. In this case, membrane or bladder accumulators are used [7], [20].

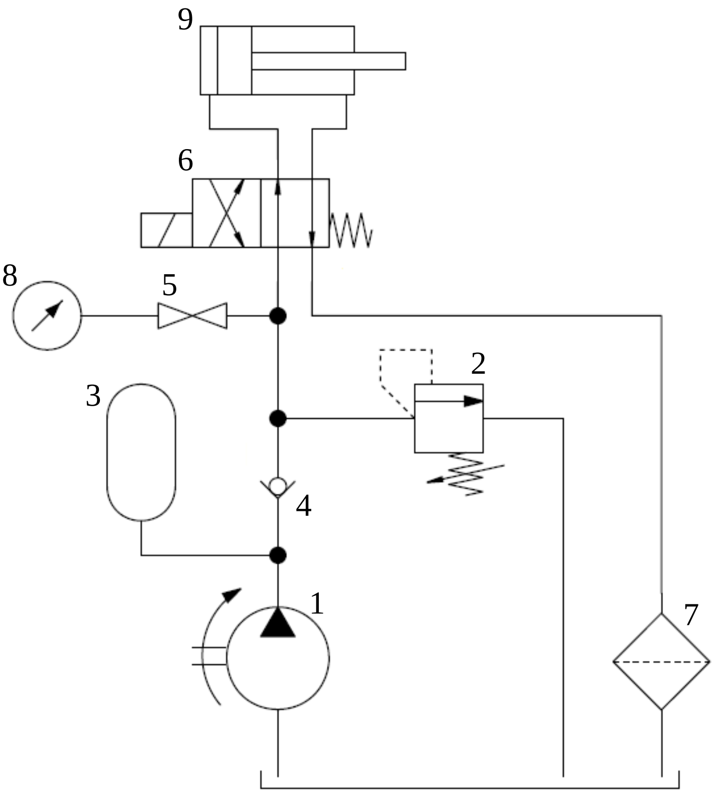



Fig. 8.1 Use of an accumulator for damping of pulsations

*1 – hydraulic pump, 2 – relief valve, 3 – accumulator, 4 – check valve, 5 – shut-off valve, 6 – directional valve, 7 – filter, 8 – manometer, 9 – hydraulic cylinder*

**Damping of pressure peaks and shocks**

Shocks in a hydraulic system can be caused by sudden interruption or flow reversion, e.g. when the directional valve is repositioned or due to changing loads on a hydraulic motor. They can cause dangerous pressure peaks. Accumulators with small volumes and fast reaction velocities are used to dampen them. An example of a possible connection is shown in Fig. 8.2 (left). The excess energy during the sudden liquid pressure increase is absorbed by gas compression in the accumulator. The most effective way is to place the accumulator as close as possible to the place where the pressure peak occurs. A possible solution for damping the vehicle chassis using an accumulator is presented in Fig. 8.2 (right).

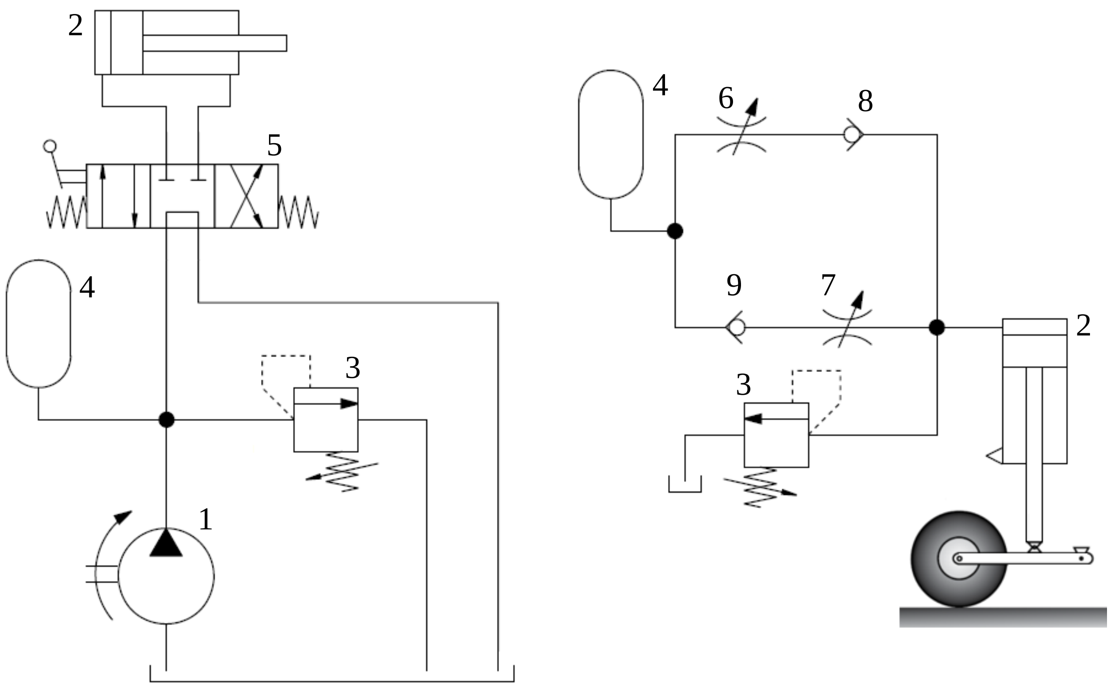



Fig. 8.2 Use of an accumulator for damping of shocks

*1 – hydraulic pump, 2 – hydraulic cylinder, 3 – relief valve, 4 – accumulator 5 – directional valve, 6, 7 – throttle valves, 8, 9 – check valves*

In addition to common accumulators, hydraulic shock absorbers of special construction are also used for the above-mentioned applications [19].

**Compensation of flow losses and derivation of constant force**

According to Fig. 8.3, the connection of the accumulator allows for keeping working pressure in the circuit and compensates for possible flow losses on the directional valve (11) or fluctuations of the liquid volume due to temperature changes. This can be used, for example, to derive a constant clamping force for the hydraulic motor (2). After starting the hydraulic pump (1), the piston rod of the hydraulic motor starts to extend. When the clamping jaw comes into contact with the material, the pressure in the circuit starts to increase and at the same time, the accumulator (7) starts to be filled through the check valve (5). After reaching the working clamping pressure $p_2$, the hydraulic pump can be disconnected. Now, the accumulator covers volume losses and monitors the necessary clamping pressure in the space of the hydraulic motor through the pressure switch (9). If the clamping pressure drops to the $p_1$ value, the hydraulic pump can be started again in order to pressurize the system to the pressure $p_2$. With such a connection, energy savings are obtained when the hydraulic pump is not in operation. If an accumulator were not placed in the system, the hydraulic pump would have to continuously supply pressure liquid to create the pressure on the hydraulic motor (the energy would be dissipated at the relief valve (3)) and cover possible flow losses.

After the control signal is applied to the directional valve and its adjustment to the left position, the reverse motion of the hydraulic motor can be realized by the liquid from the accumulator or after starting the hydraulic pump by the sum of the flow from the hydraulic pump and accumulator, which would achieve a faster movement of the motor [21].

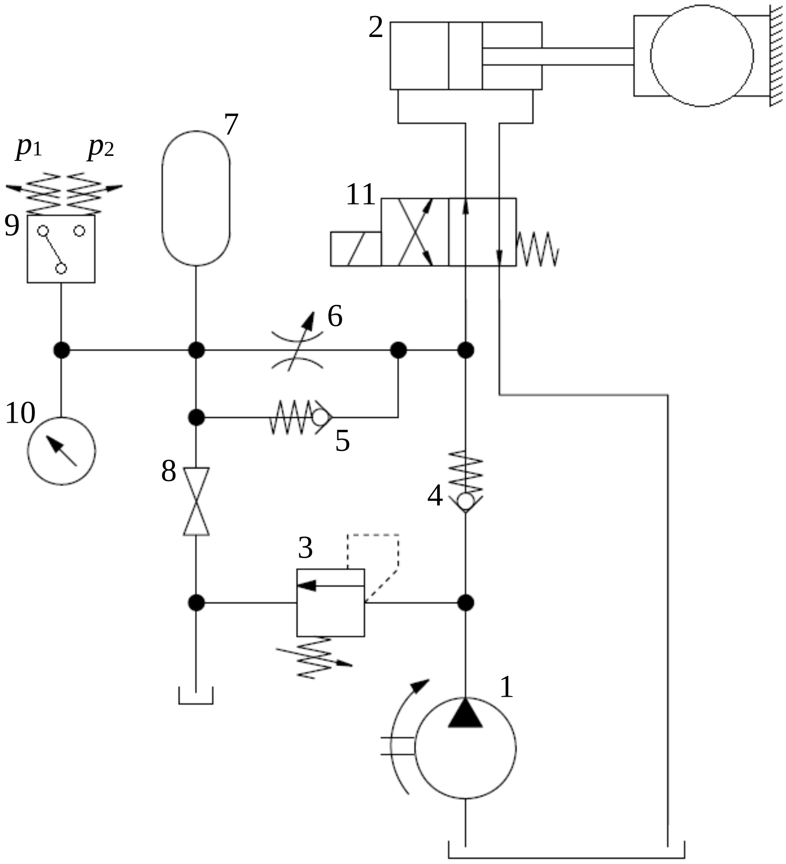



Fig. 8.3 Derivation of constant force and compensation of flow losses

*1 – hydraulic pump, 2 – hydraulic cylinder, 3 – relief valve, 4, 5 – check valves, 6 – throttle valve, 7 – accumulator, 8 – shut-off valve, 9 – pressure switch, 10 – manometer, 11 – directional valve*

A similar connection is shown in Fig. 8.4. In this case, the three-position directional valve (5) is used in the system. The accumulator (6) performs a similar function as in the previous system. The open middle position allows for unloading hydraulic pump 1 or using it for other work operations [22].

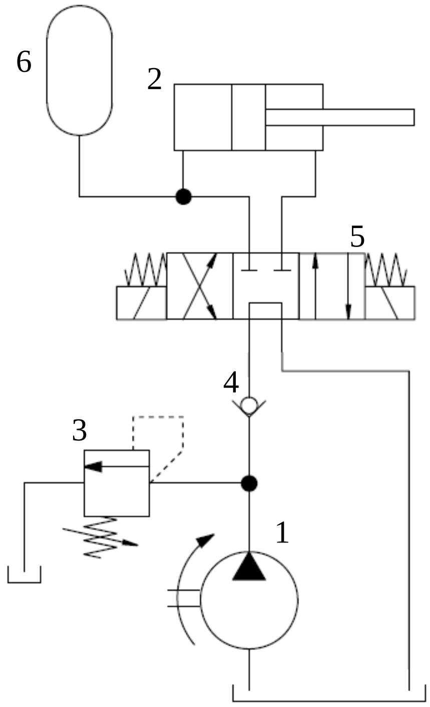



Fig. 8.4 Compensation of flow losses

*1 – hydraulic pump, 2 – hydraulic cylinder, 3 – relief valve, 4 – check valve, 5 – directional valve, 6 - accumulator*

**Pneumohydraulic spring**

As is shown in Fig. 8.5 (left), the extension of the piston rod of the hydraulic motor (2) is achieved by means of a hydraulic pump (1) using the directional valve (4). At the same time, the liquid from the piston rod area fills the accumulator (5) through the check valve (6). The reverse movement of the hydraulic motor is possible after repositioning the directional valve to its left position with liquid from the accumulator.

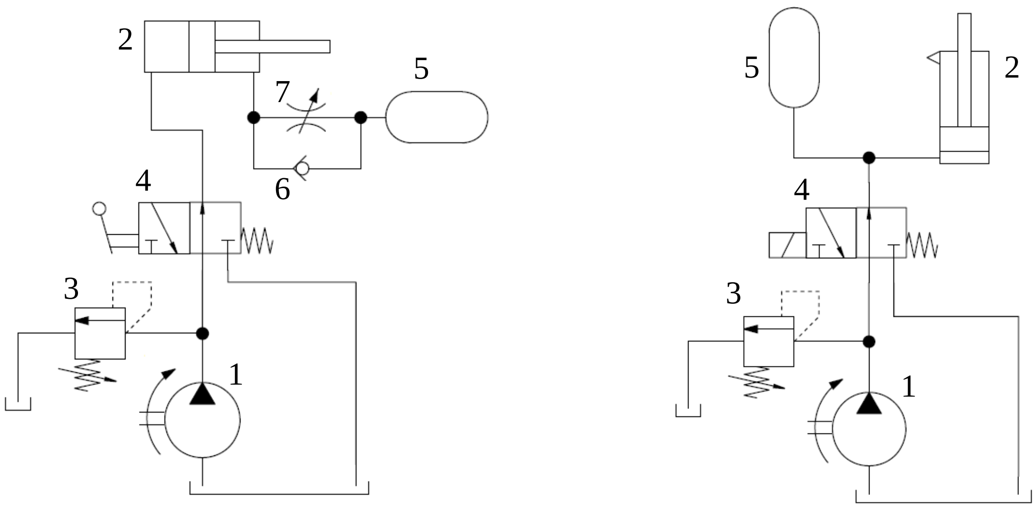



Fig. 8.5 Accumulator as pneumohydraulic spring

*1 – hydraulic pump, 2 – hydraulic cylinder, 3 – relief valve, 4 – directional valve, 5 – accumulator, 6 – check valve, 7 – throttle valve*

Another possible application of accumulators is the so-called load balancing; see Fig. 8.5 (right). This can be used for manipulation with heavy loads. The accumulator (5) keeps the pressure on the piston side, which creates the compression force acting against the gravitational force of the load and, therefore, easier manipulation of the hydraulic cylinder (2) with the load. It is used, for example, in the machinery industry, for the movement of a machine head of great weight and dimensions [9], [*14*].

**Uneven withdrawal of liquid**

In applications where, for example, more hydraulic motors of different sizes and liquid consumptions are required, an accumulator can be placed in the system (Fig. 8.6). Without the accumulator, the hydraulic pump would have to be dimensioned to cover the largest consumption of the given consumer (hydraulic motor). If it is a cyclically repeating working process, it is possible to design a hydraulic pump with a smaller geometric stroke volume. The mean liquid flow rate is determined for the working cycle, and subsequently, the required useful volume of the accumulator is subtracted from the consumption diagram. It is a combined pressure source, where the accumulator is filled during times of lower consumption and, in the case of peak consumption, the flow to the consumer is supplemented by the accumulator [23].

The use of the accumulator is energy-saving, reduces operating costs and enables the hydraulic aggregate to be dimensioned for lower power.

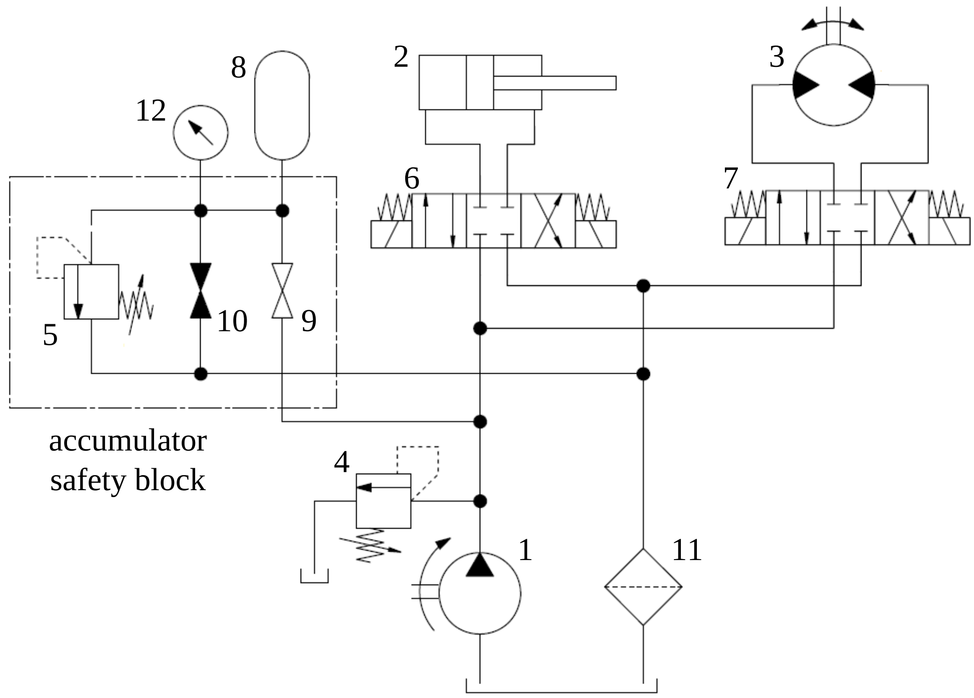



Fig. 8.6 Coverage of more consumers with different consumption

*1 – hydraulic pump, 2 – hydraulic cylinder, 3 – rotary hydraulic motor, 4, 5 – relief valves, 6, 7 – directional valves, 8 – accumulators, 9, 10 – shut-off valve, 11 – filter, 12 - manometer*

**Ensuring short-term high consumption**

In equipment where a large amount of pressure liquid is needed for a short time, it is advantageous to use an accumulator. As in the previous case, the hydraulic pump is dimensioned for moderate power only. The hydraulic pump fills the accumulator for a particular time. Subsequently, the accumulated energy is used to perform the necessary work operation. These are applications for construction, forming and machining machines, metallurgical equipment and conveyors. For power plants, these are fast shutdown turbine safety systems, where there is only minimal liquid consumption during regular operation. In the case of injection moulding machines, maximum power is required but only for a short period of time. A simplified circuit of a hydraulic shear for cutting a continuously cast steel casting is shown in Fig. 8.7. The delay between individual cuts is long, on the order of several minutes. A cheaper hydraulic pump (1) with a lower flow rate can be used in the system, which continuously fills the accumulator (4). After the signal is applied to the directional valve (3), the energy from the accumulator (4) is used to cut the material quickly. The accumulator (5) is used as a hydropneumatic spring for the reverse movement of the hydraulic motor (2) [23].

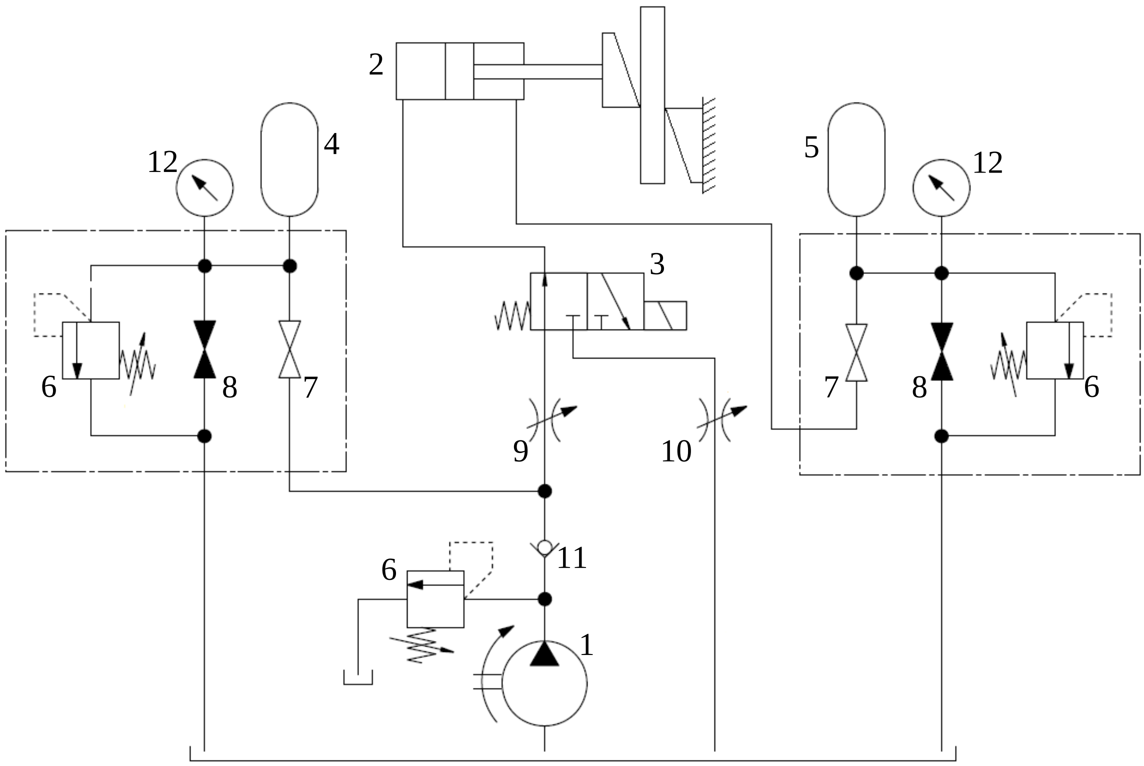



Fig. 8.7 Short-term high liquid withdrawal

*1 – hydraulic pump, 2 – hydraulic cylinder, 3 – directional valve, 4, 5 – accumulators, 6 – relief valves, 7, 8 – shut-off valves, 9, 10 – throttle valves, 11 – check valve, 12 - manometers*

### Reduction of stroke time

This can be machining or pressing applications, where a rapid feed can be used before the actual work operation. The low-pressure hydraulic pump (1) (see Fig. 8.8) is used for the accumulator (6) filling. When the hydraulic cylinders (3) move without load, the liquid from both hydraulic pumps and the accumulator is used. At the end of the stroke of the hydraulic cylinders, there is an increase in the system pressure when the check valve is closed (8). The pressing process is ensured by the high-pressure pump (2) at a low movement velocity. The accumulator is filled during the pressing process [19].

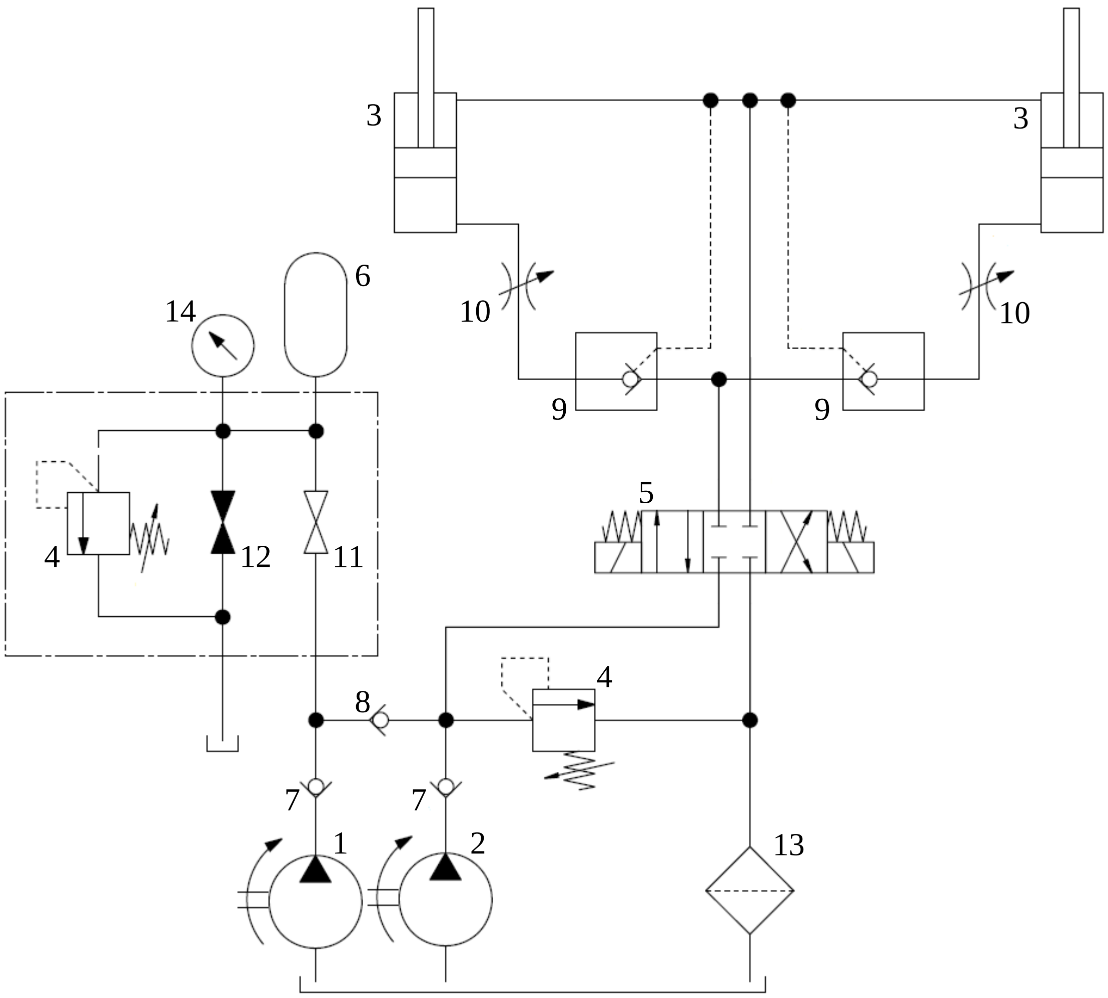



Fig. 8.8 Reduction of stroke time – rapid feed

*1, 2 – hydraulic pump, 3 – hydraulic cylinders, 4 – relief valves, 5 – directional valve, 6 – accumulator, 7, 8 – check valves, 9 – pilot operated check valve, 10 – throttle valves, 11, 12 – shut-off valves, 13 – filter, 14 - manometer*

### Emergency control

Hydraulic accumulators are often used as backup devices, for example, in the case of a power cut. This is applied, e.g. in bearings lubrication of large turbines and pumps, in machine tools to protect tools from damage, for safety control of flaps and valves in the energy industry, or for emergency control of brakes and doors of various mobile equipment.

In Fig. 8.9, during a power failure, the accumulator (6) energy is used through the directional valve (5) to retract the hydraulic motor (2) [23].

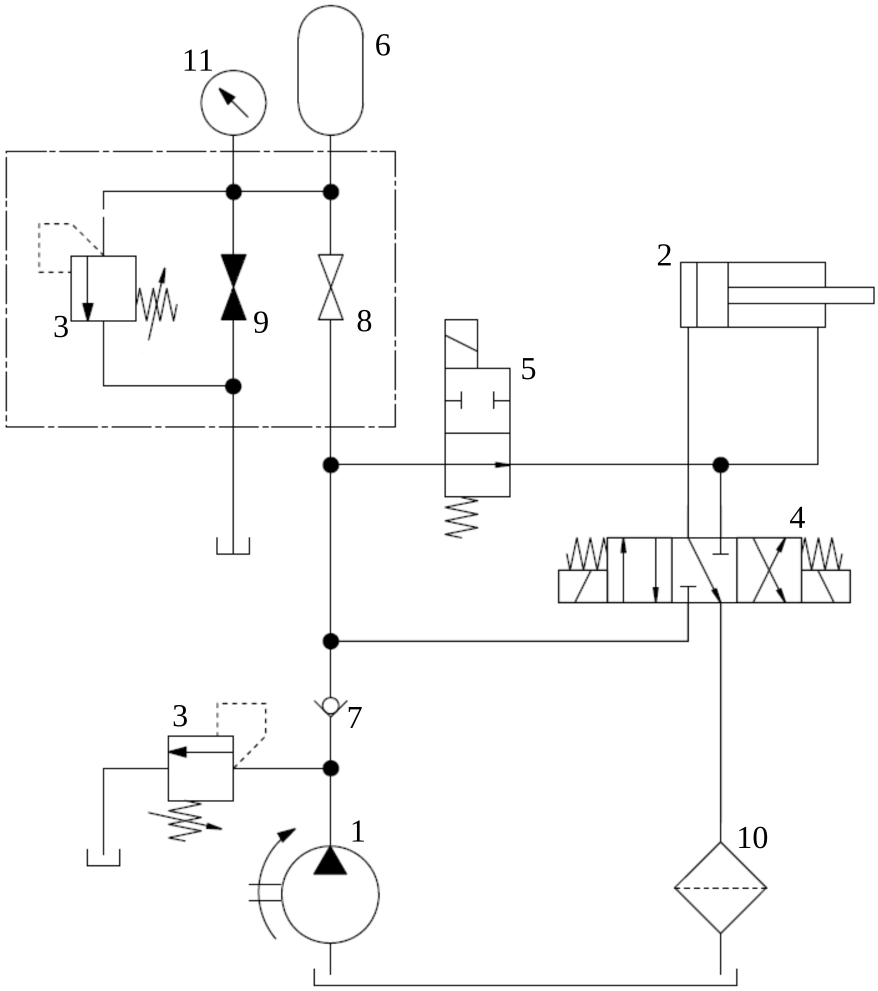



Fig. 8.9 Emergency control

*1 – hydraulic pump, 2 – hydraulic cylinder, 3 – relief valve, 4, 5 – directional valves, 6 – accumulator, 7 – check valve, 8, 9 – shut-off valves, 10 – filter, 11 - manometer*

**Pressure source for circuits with fast-switching control valves**

Proportional valves and servo valves are characterized by a high repositioning velocity. In order to fully utilize their potential, the accumulator (5) is connected in parallel to the hydraulic pump (1) in the circuit Fig. 8.10. The accumulator is used here as a temporary pressure source until the variable displacement hydraulic pump is adjusted to the required volumetric flow.

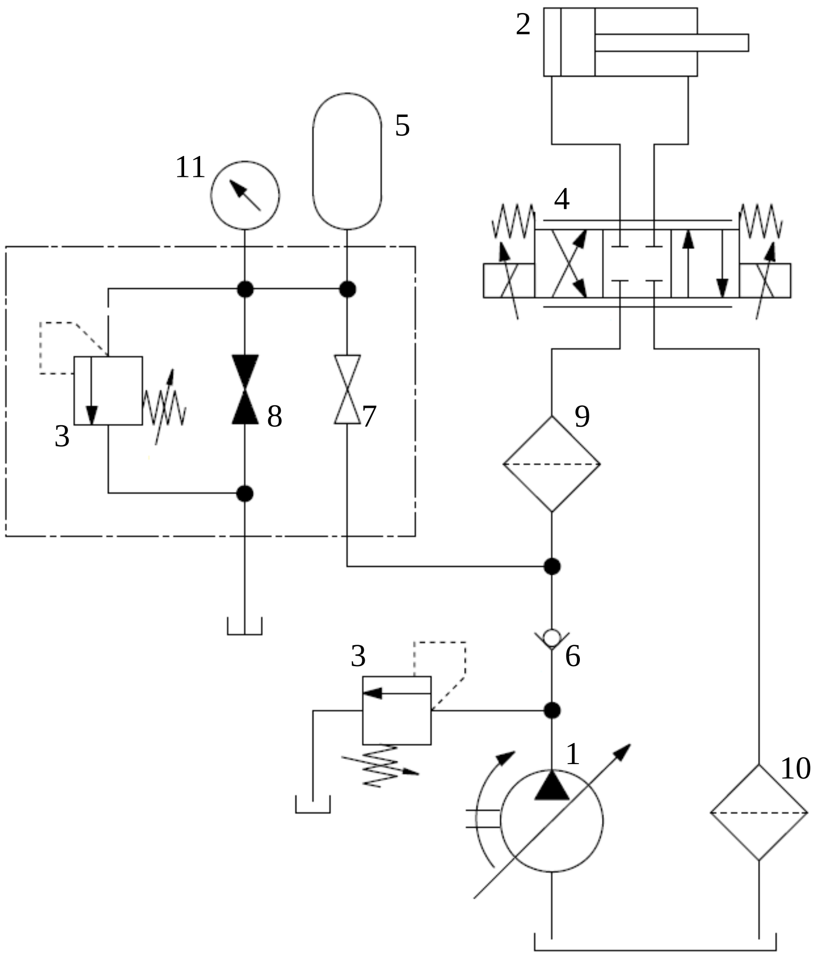



Fig. 8.10 Pressure source for circuits with fast-switching control valves

*1 – hydraulic pump, 2 – hydraulic motor, 3 – relief valve, 4 – proportional valve, 5 – accumulator, 6 – check valve, 7, 8 – shut-off valves, 9 – pressure filter, 10 – low pressure (return line) filter, 11 - manometer*

## 8.2 Calculation and design of the accumulator

The calculation of accumulators is based on the equation of the state of gases. The closed gas in the accumulator behaves in accordance with the laws of thermodynamics, then for the polytropic change of state of the gas:

$$p \cdot V^n = const., \tag{8.1}$$

where $p$ [Pa] is the absolute pressure of the gas, $V$ [$m^3$] is the gas volume at the pressure $p$, and $n$ [-] is the polytropic exponent depending on the type of state change.

There are always changes in the volume and pressure of the gas during the operation of the accumulator. Therefore, isochoric and isobaric changes are not considered. For an ideal gas, three state changes can occur:

**Isothermal change of state *n* = 1** – there is no change in temperature (in the accumulator); a complete heat exchange between the gas and the outside occurs. The isothermal change can be considered for slow processes when the time of filling or emptying of the accumulator is longer than 3 minutes.

**Adiabatic change of state *κ* = 1.4** – is for rapid changes and with perfect thermal insulation of the gas from the environment. The filling or emptying of the accumulator would be so fast that heat exchange could not occur. The adiabatic change can be considered for rapid processes where the filling or emptying time is less than 1 minute.

The exponent $\kappa$ gives the ratio of the specific heat capacities at constant pressure and volume:

$$\kappa = \frac{c_p}{c_v}, \tag{8.2}$$

where $c_p$ [$J \cdot kg^{-1} \cdot K^{-1}$] is the specific heat capacity of the gas at constant pressure, and $c_v$ [$J \cdot kg^{-1} \cdot K^{-1}$] is the specific heat capacity of the gas at constant volume.

**Polytropic change of state *n* = (1 ÷ 1.4)** - it is evident that perfect thermal insulation is not really possible. The isothermal and adiabatic changes are limit states. Real processes in the accumulator operation will be according to the polytropic change, somewhere between the limit states, as described in the state diagram, as shown in Fig. 8.11.

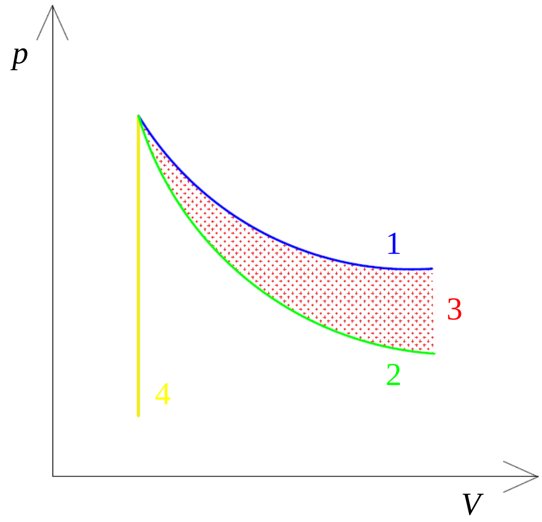



Fig. 8.11 *p* - *V* state diagram of gases

*1 – isothermal change, 2 – adiabatic change, 3 – polytropic change, 4 – isochoric change*

Thus, the equation (8.3) applies for the state changes of the gas. The individual states are graphically shown in Fig. 8.12.

$$p_0 \cdot V_0^n = p_1 \cdot V_1^n = p_2 \cdot V_2^n, \tag{8.3}$$

where $p_0$ [Pa] is the gas pre-fill pressure, $V_0$ [m$^3$] is the effective gas volume at the pressure $p_0$, which also corresponds to the maximum accumulator volume, $p_2$ [Pa] is the maximum operating pressure corresponding to the maximum working pressure in the hydraulic system, $V_2$ [m$^3$] is the gas volume at the pressure $p_2$, it is the minimum gas volume, $p_1$ [Pa] is the minimum operating pressure corresponding to the minimum working pressure in the hydraulic system, and $V_1$ [m$^3$] is the gas volume at the pressure $p_1$.

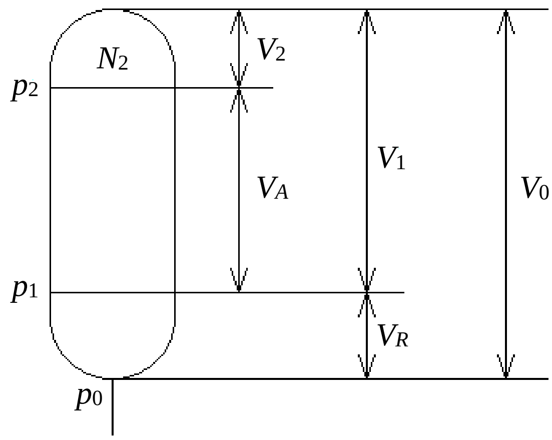



Fig. 8.12 Working cycles of the accumulator

Four states occur during the accumulator operation. These are gas filling, liquid filling to operating pressure, liquid withdrawal (pressure decrease in the accumulator), and accumulator refilling to operating pressure.

Firstly, the accumulator is filled with gas to the so-called filling pressure $p_0$, which is less than the minimum operating pressure of the hydraulic system (it is shown for each accumulator type in Tab 8.1), most often:

$$p_0 = 0.9 \cdot p_1 \,. \tag{8.4}$$

In the next step, when the accumulator is slowly filled with liquid, an isothermal change can be considered. The allowable ratio of the maximum and filling gas pressure is given in Tab 8.1. If the volume $V_0$ is known, it is possible to calculate the gas volume $V_2$:

$$\begin{gathered} p_0 \cdot V_0 = p_2 \cdot V_2 \,, \\ V_2 = \frac{p_0}{p_2} \cdot V_0 \,. \end{gathered} \tag{8.5}$$

In the third phase, the liquid is withdrawn from the accumulator, and the useful volume $V_A$ of the liquid can be calculated as:

$$V_A = V_0 - V_2 \,, \tag{8.6}$$

where $V_A$ [m$^3$] is useful volume of the accumulator.

This equation would be valid if the accumulator were made exactly to measure with the required volume. Usually, the accumulator is selected according to the commonly supplied sizes, and the so-called reserve volume $V_R$ is also left:

$$V_A = V_0 - V_2 - V_R \,, \tag{8.7}$$

where $V_R$ [m$^3$] is reserve volume of the accumulator.

After emptying the liquid from the accumulator, there is a phase of refilling the liquid to its original state [7].

**Calculation procedure**

Most often, when designing the accumulator, the operating pressures of the hydraulic system $p_1$ and $p_2$ are known, as well as the required useful volume of the accumulator $V_A$, then the calculation procedure is as follows:

$$\begin{gathered} p_1 \cdot V_1^n = p_2 \cdot V_2^n \,, \\ p_1 \cdot (V_2 + V_A)^n = p_2 \cdot V_2^n \,, \\ V_2 + V_A = \left(\frac{p_2}{p_1}\right)^{\frac{1}{n}} \cdot V_2 \,, \\ V_A = V_2 \left[\left(\frac{p_2}{p_1}\right)^{\frac{1}{n}} - 1\right]. \end{gathered} \tag{8.8}$$

The gas volume in the accumulator at maximum (working) pressure in the hydraulic system is given:

$$V_2 = \frac{V_A}{\left(\frac{p_2}{p_1}\right)^{\frac{1}{n}} - 1} \, . \tag{8.9}$$

The exponent $n$ is selected depending on the desired function of the accumulator in the system. The volume $V_1$ can be calculated as follows:

$$V_1 = V_2 + V_A \, . \tag{8.10}$$

The accumulator with the next higher volume is selected from the manufacturer's catalogue. The reserve volume $V_R$ of the accumulator is then determined, whereby:

$$V_0 = V_1 + V_R \, . \tag{8.11}$$

When calculating and selecting the battery, we usually follow the manufacturer's data. It is possible to use their calculation software or nomograms, as is shown in Fig. 8.13. With the known ratio of pressures $p_2$ a $p_1$, the useful volume of the liquid (marked $\Delta V$ in Fig. 8.13) and the filling and emptying velocities of the accumulator, it is possible to determine the total volume of the accumulator.

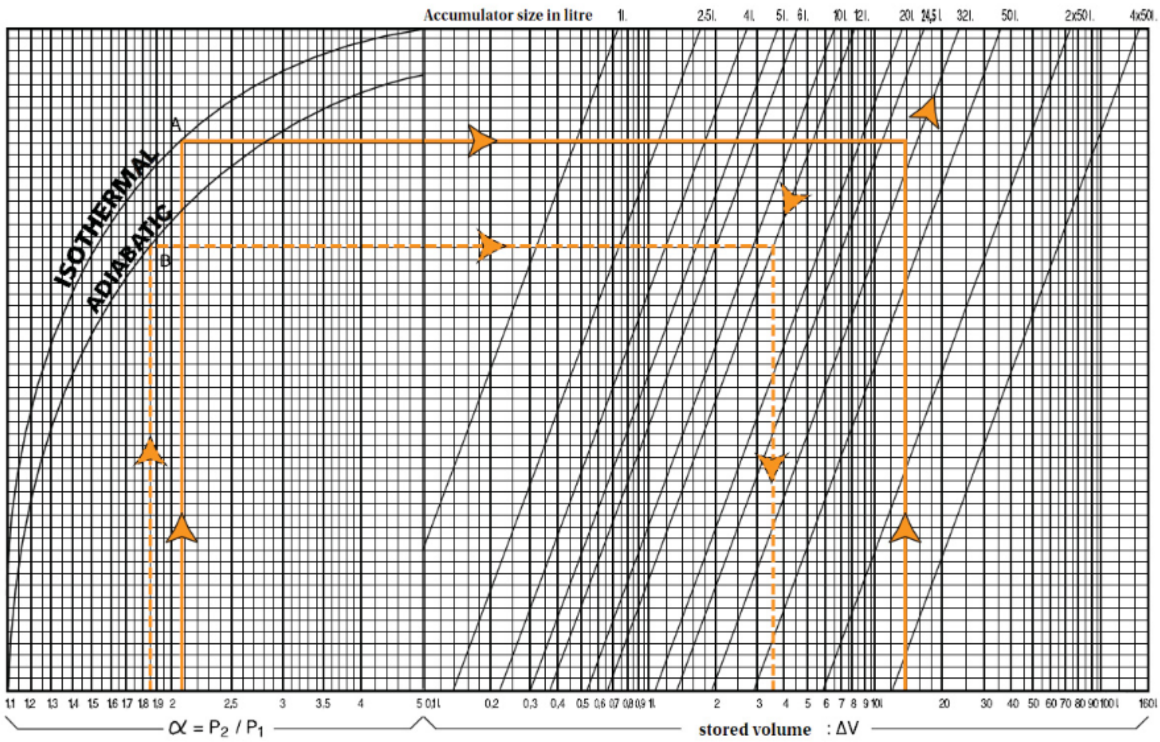



Fig. 8.13 Nomogram of PARKER company [24]

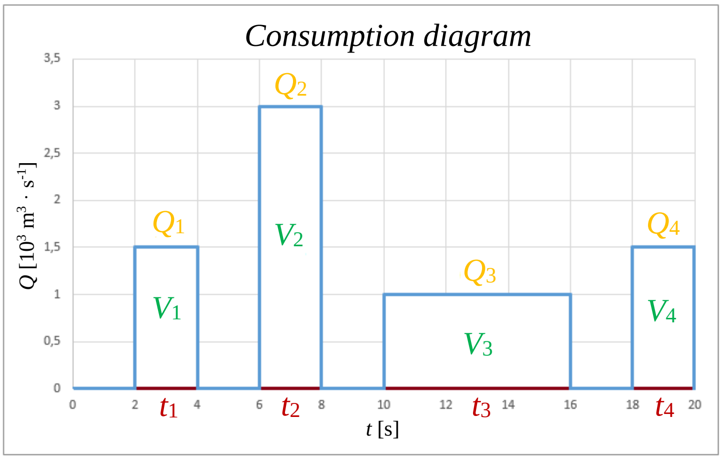
An overview of the basic parameters of gas accumulators is shown in Tab 8.1.

Tab 8.1 An overview of the basic parameters of gas accumulators [7], [14], [19], [20], [24], [25]

| Parameters | Piston accumulator | Bladder accumulator | Membrane accumulator |
|---|---|---|---|
| **liquid** | oils, anhydrous synthetic liquids | all types of hydraulic liquids (depending on bladder material) | all types of hydraulic liquids (depending on membrane material) |
| **maximum working pressures** | 35 MPa (high-pressure to 55 MPa) | up to 55 MPa (low-pressure to 3.5 MPa) | 35 MPa (welded)<br>up to 75 MPa (screwed) |
| **maximum volumes** | 400 $dm^3$ | 200 $dm^3$ (low-pressure up to 450 $dm^3$) | $(0.2 \div 4)\ dm^3$ |
| **filling pressure $p_0$** | $p_0 \leq p_1 - 0.5$ MPa | for common applications $p_0 \leq 0.9 \cdot p_1$ for damping of shocks and pulsations $p_0 = (0.6 \div 0.9) \cdot p_1$ | $p_0 \leq 0.9 \cdot p_1$ |
| **maximum allowable pressure ratio $p_2/p_0$** | without limitations | $p_2/p_0 \leq 4$ | $p_2/p_0 = 6 \div 8$ (welded)<br>$p_2/p_0 \leq 10$ (screwed) |
| **maximum volumetric flow** | is given by the piston velocity, when using a standard seal $0.5\ m \cdot s^{-1}$, maximum $5\ m \cdot s^{-1}$ | $(4 \div 30)\ dm^3 \cdot s^{-1}$<br>in "High-flow" design up to $140\ dm^3 \cdot s^{-1}$ | up to $6\ dm^3 \cdot s^{-1}$ |
| **applications** | slow processes, short-term withdrawals of large liquid volumes | universal use | damping of shocks and pulsations |

**Example 8.1**

Design an accumulator for a combined drive with a hydraulic pump. The volumetric flow from the source is irregular; consider the load according to the consumption diagram Fig. 8.14. It is a periodically repeating cycle with the period $T$ = 20 s. For a given cycle, calculate the mean volumetric flow $Q_m$ delivered by the hydraulic pump and the useful volume $V_A$ of the accumulator.



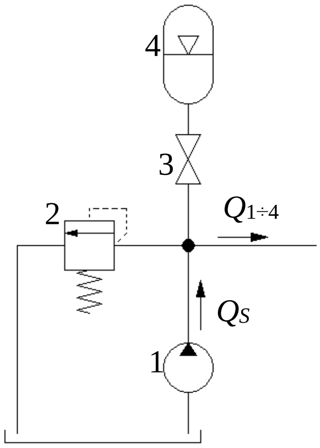




Fig. 8.14 Consumption diagram (left), hydraulic scheme (right)

*1 – hydraulic pump, 2 – relief valve, 3 – shut-off valve, 4 - accumulator*

<u>Entered:</u>

| | |
|---|---|
| sampling of volumetric flows in individual sections | $Q_1 = 1.5 \cdot 10^{-3}\ m^3 \cdot s^{-1}$ |
| | $Q_2 = 3 \cdot 10^{-3}\ m^3 \cdot s^{-1}$ |
| | $Q_3 = 1 \cdot 10^{-3}\ m^3 \cdot s^{-1}$ |
| | $Q_4 = 1.5 \cdot 10^{-3}\ m^3 \cdot s^{-1}$ |
| time of individual withdrawals | $t_1 = t_2 = t_4 = 2$ s |
| | $t_3 = 6$ s |
| period time | $T = 20$ s |

<u>Calculate:</u> $Q_m = ?$, $V_A = ?$

<u>Calculation:</u>

If there was only the hydraulic pump as a flow source in the system, it would have to be sized to cover the maximum volumetric flow $Q_2$. When an accumulator is used, the mean volumetric flow $Q_m$ (i.e., the volumetric flow of the hydraulic pump) is calculated, and the rest of the required volumetric flow to the system is supplied by the accumulator:

$$Q_m = Q_G = \frac{\sum_{i=1}^{n} \Delta V_i}{T} = \frac{\sum_{i=1}^{n} Q_i \cdot t_i}{T} ,$$
$$Q_m = Q_G = \frac{1.5 \cdot 10^{-3} \cdot 2 + 3 \cdot 10^{-3} \cdot 2 + 1 \cdot 10^{-3} \cdot 6 + 1.5 \cdot 10^{-3} \cdot 2}{20} , \tag{8.12}$$
$$Q_m = Q_G = 0.9 \cdot 10^{-3}\ m^3 \cdot s^{-1} .$$

From the mean volumetric flow $Q_m$ and the speed of an electric motor, it is possible to calculate the geometric stroke volume of the hydraulic pump and, subsequently, to design the hydraulic pump. The liquid volume $V_T$ delivered by the hydraulic pump per cycle time is:

$$V_T = Q_G \cdot T = 0.9 \cdot 10^{-3} \cdot 20 = 18 \cdot 10^{-3}\ m^3 = 18\ dm^3 . \tag{8.13}$$

Volumes $V_i$ withdrawn by the equipment in individual time periods $t_i$:

$$\begin{aligned} V_1 &= Q_1 \cdot t_1 = 1.5 \cdot 10^{-3} \cdot 2 = 3 \cdot 10^{-3}\ m^3 = 3\ dm^3 , \\ V_2 &= Q_2 \cdot t_2 = 3 \cdot 10^{-3} \cdot 2 = 6 \cdot 10^{-3}\ m^3 = 6\ dm^3 , \\ V_3 &= Q_3 \cdot t_3 = 1 \cdot 10^{-3} \cdot 6 = 6 \cdot 10^{-3}\ m^3 = 6\ dm^3 , \\ V_4 &= Q_4 \cdot t_4 = 1.5 \cdot 10^{-3} \cdot 2 = 3 \cdot 10^{-3}\ m^3 = 3\ dm^3 . \end{aligned} \tag{8.14}$$

The volume withdrawn by the equipment $\sum_{i=1}^{n} V_i$ must be equal to the volume $V_T$ supplied by the hydraulic pump.

The accumulator will be designed to cover the difference between the instantaneous consumption and the mean volumetric flow of the hydraulic pump by determining the useful volume of the accumulator $V_A$, as the difference between the maximum positive and negative deviation $\Delta V$, see Fig. 8.16:

$$V_A = (2.4 + 1.8) \cdot 10^{-3} = 4.2 \cdot 10^{-3}\ m^3 = 4.2\ dm^3 . \tag{8.15}$$

For the equipment with the given consumption diagram, design an accumulator for a combined drive with the hydraulic pump, the working pressure of the mechanism $p_2$ = 16 MPa, the minimum allowable pressure $p_1$ = 10 MPa and the polytropic exponent $n$ = (1.1 ÷ 1.4).

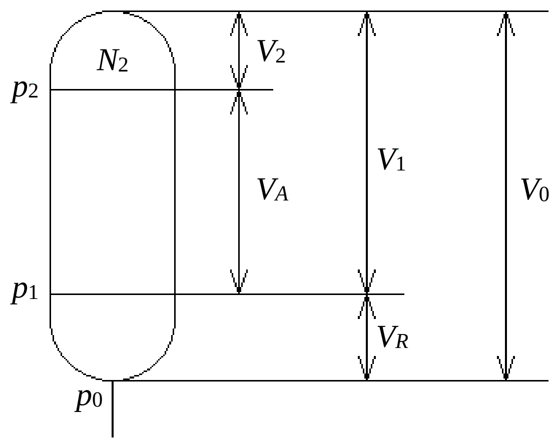



Fig. 8.15 Working cycles of the accumulator

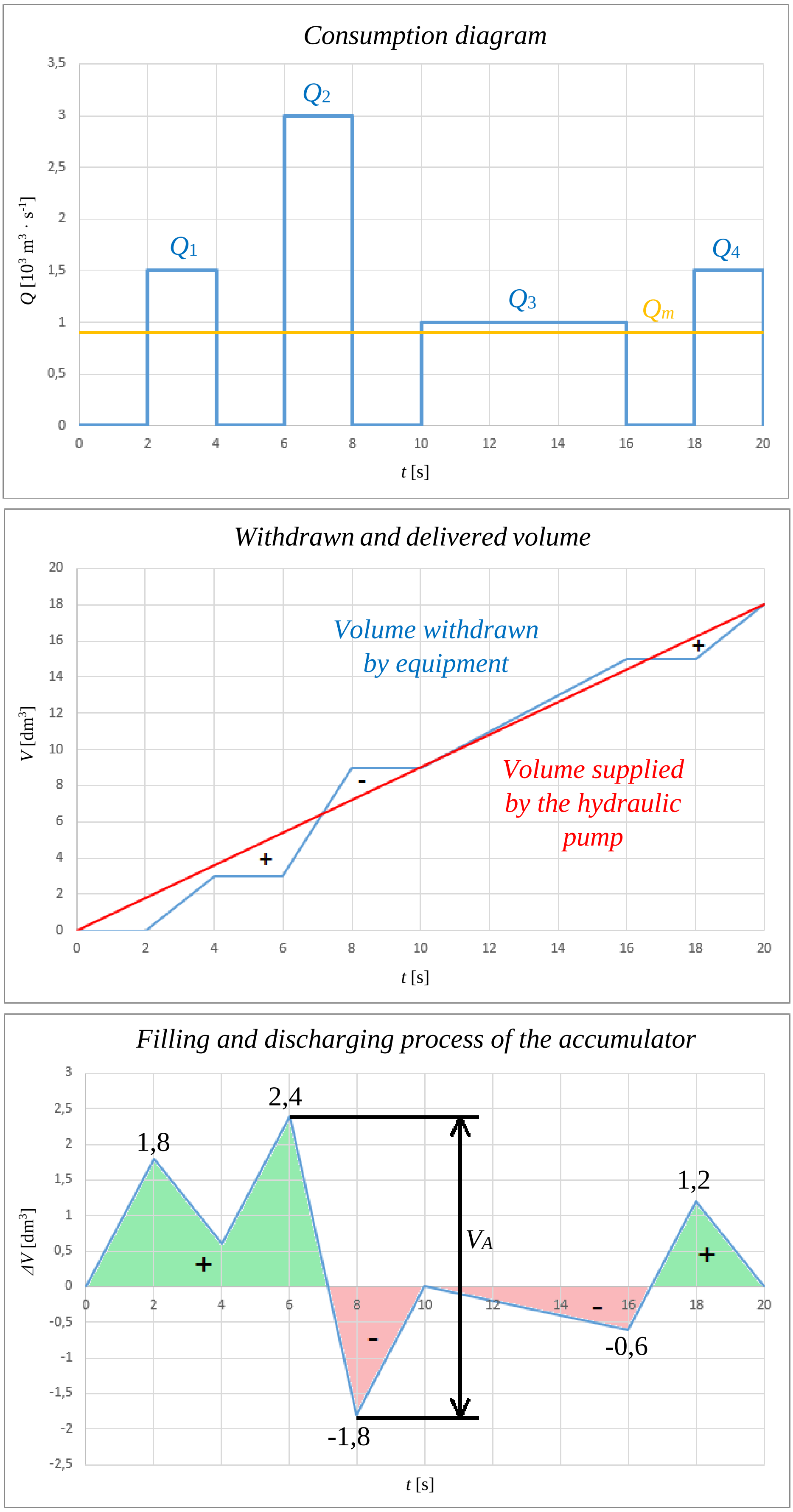



Fig. 8.16 Graphical determination of the useful volume $V_A$ of the accumulator

Assuming the polytropic change holds $p \cdot V^n$ = const., the polytropic exponent is $n$ = 1,2. From equitation (8.8) can be calculated the gas volume $V_2$ in the accumulator at the maximum (working) pressure in the hydraulic system:

$$V_2 = \frac{V_A}{\left(\frac{p_2}{p_1}\right)^{\frac{1}{n}} - 1} = \frac{4.2 \cdot 10^{-3}}{\left(\frac{16.1 \cdot 10^6}{10.1 \cdot 10^6}\right)^{\frac{1}{1.2}} - 1} = 8.84 \cdot 10^{-3}\, m^3 = 8.84\, dm^3\,. \tag{8.16}$$

The gas volume $V_1$ at the minimum pressure in the hydraulic system can be calculated from Fig. 8.15:

$$V_1 = V_2 + V_A = 8.84 \cdot 10^{-3} + 4.2 \cdot 10^{-3} = 13.04 \cdot 10^{-3}\, m^3 = 13.04\, dm^3\,. \tag{8.17}$$

Choosing the accumulator from the manufacturer's catalogue; see Fig. 8.17. We choose a bladder accumulator type SB40-20 with a nominal volume of 20 dm$^3$ from Hydac company.

**SB40-2.5 ... 50**

Permitted operating pressure 40 bar (PED)

| Nominal volume [l] | Eff. gas volume [l] | Weight [kg] | A [mm] | B [mm] | C [mm] | Ø D [mm] | J thread ISO DIN 13 | K thread ISO 228 | SW [mm] | Q 1) [l/s] |
|---|---|---|---|---|---|---|---|---|---|---|
| 2.5 | 2.5 | 9 | 541 | 122 | 68 | 108 | M100x2 | G 2 | 36 | 7 |
| 5 | 5 | 13 | 891 | | | | | | | |
| 10 | 9.3 | 14 | 533 | 106 | | 219 | | | | |
| 20 | 18 | 23 | 843 | | | | | | | |
| 32 | 33.5 | 38 | 1363 | | | | | | | |
| 50 | 48.6 | 52 | 1875 | | 78 | | | | 68 2) | |

1) Q = max. flow rate of operating fluid (at approx. 0.5 bar pressure drop via connection)
2) use C-spanner

Fig. 8.17 Choosing the bladder accumulator from Hydac company catalogue [26]

Calculation of the reserve volume $V_R$ of the accumulator:

$$V_0 = V_1 + V_R \Rightarrow V_R = V_0 - V_1 = 20 \cdot 10^{-3} - 13.04 \cdot 10^{-3} = 6.96 \cdot 10^{-3}\, m^3\,, \tag{8.18}$$
$$V_R = 6.96\, dm^3$$

# 9. Thermal calculation of the hydraulic system

During the operation of a machine or mechanism, part of the input energy is used (consumed) to overcome the resistances in the system. These resistances can be referred to as resistances to motion, to deformation, and to acceleration. When these resistances are overcome, energy changes occur, which can be reversible or irreversible. In the case of the resistance to deformation and the resistance to acceleration, this is a reversible energy change since the resulting deformation or kinetic energy can be accumulated. In the case of the resistance to motion, this is an irreversible change and is converted to thermal energy [27].

Suppose this formulation is applied to the liquid flow through a hydraulic circuit. In that case, a part of the liquid's pressure energy is converted into thermal energy at the resistance to motion, and thus, the energy is dissipated. Resistances to motion are mainly local resistances such as orifices, throttle valves, proportional valves, and pressure valves. Heat generation also occurs in hydraulic generators and hydraulic motors and can be defined by their volumetric and mechanical-hydraulic efficiency. Frictional losses in pipes and hoses are also resistance to motion.

If a steady state is considered, then the change and transfer of energy in a hydraulic system can be expressed by the total efficiency $\eta_T$ of the system. The total system efficiency can be determined as the ratio of the output power $P_2$ to the input power $P_1$:

$$\eta_T = \frac{P_2}{P_1}, \tag{9.1}$$

where $\eta_T$ [-] is the total efficiency of the hydraulic system, $P_2$ [W] is the output power of the hydraulic system, and $P_1$ [W] is the input power of the hydraulic system.

In general, the total efficiency can also be expressed as the product of the partial efficiencies $\eta_i$ of individual system components:

$$\eta_T = \prod_{i=1}^{n} \eta_i \,. \tag{9.2}$$

To determine the thermal balance of a system, it is necessary to determine the total power loss $P_l$, which is given by the difference between the input and output powers. When the total efficiency of the hydraulic system is known, the total power loss can be calculated from the input power $P_1$:

$$P_l = P_1 \cdot (1 - \eta_T)\,, \tag{9.3}$$

where $P_l$ [W] is the loss power of the hydraulic system.

Alternatively, the power loss can be calculated from the output power $P_2$:

$$P_l = P_2 \cdot \left(\frac{1}{\eta_T} - 1\right). \tag{9.4}$$

It can now be determined with some simplification that the power loss of the hydraulic system is converted into heat. And therefore, the power loss of the $P_l$ will be proportional to the heat flow (heat power) $\phi$:

$$P_l = \phi \,, \tag{9.5}$$

where $\phi$ [W] is the heat flow or heat power.

The generated heat leads to the heating of the elements of the hydraulic circuit as well as the working liquid. Part of the heat is removed through the surface of the tank, hydraulic elements, and pipes to the surroundings. To determine the heating course in the hydraulic circuit it is necessary to follow the differential equation of thermal equilibrium (9.6). The calculation can be simplified by assuming a homogeneous system where the temperature is the same at all points. Furthermore, only two different system materials with different specific heat capacities are considered in the calculation. As a rule, these are the materials with the highest mass (usually steel and working liquid). Only the tank's heat transfer surface is considered for heat exchange with the surroundings [28].

$$\phi \cdot d\tau = (m_1 \cdot c_1 + m_2 \cdot c_2) \cdot dt + k_T \cdot A_T \cdot \Delta t_T \cdot d\tau \,, \tag{9.6}$$

where $\tau$ [s] is the time, $m_1$ [kg] is the mass of working liquid in the system, $m_2$ [kg] is the mass of steel parts in the circuit, $c_1$ [J · kg · K$^{-1}$] is the specific heat capacity of working liquid, $c_2$ [J · kg · K$^{-1}$] is the specific heat capacity of steel parts in the circuit, d$t$ [°C] is the temperature change, $k_T$ [W · m$^{-2}$ · K$^{-1}$] is the total heat transfer coefficient of the tank, $A_T$ [m$^2$] is the heat transfer area of the tank, and $\Delta t_T$ [°C] is the temperature gradient on the tank.

In the given equation (1.6), the term on the left side represents the amount of heat that enters the system as a result of the conversion of pressure energy into thermal energy. The first term on the right side of the equation represents the amount of heat that is accumulated in the system and causes it to heat up by the temperature d$t$. The second term on the right side of the equation represents the heat dissipated to the surroundings by the heat transfer surface area of the tank $A_N$.

The temperature gradient on the tank represents the difference between the temperature $t$ of the system (liquid) and the ambient temperature $t_a$:

$$\Delta t_T = t - t_0 \,, \tag{9.7}$$

where $t$ [°C] is the system (liquid) temperature, and $t_0$ [°C] is the ambient temperature.

Considering the previously mentioned assumptions, the solution of equation (9.6) can be used to determine the relationship for the system heating course:

$$t = t_0 + \frac{\phi}{k_T \cdot A_T} \cdot \left(1 - e^{-\frac{\tau}{T}}\right), \tag{9.8}$$

where $T$ [s] is the heating time constant and $t_\mathrm{a}$ [°C] is the initial system temperature, which is equal to the ambient temperature.

It is possible to express the heating time constant $T$ by the equation:

$$T = \frac{m_1 \cdot c_1 + m_2 \cdot c_2}{k_T \cdot A_T} \,. \tag{9.9}$$

Assuming that the maximum or steady temperature is reached in time $\tau$ = ∞, then by substituting into equation (9.8), it is possible to obtain the equation for the calculation of the steady temperature $t_S$:

$$t_s = t_0 + \frac{\phi}{k_T \cdot A_T}, \tag{9.10}$$

where $t_s$ [°C] is the steady (maximum) system temperature.

The heating course can be expressed graphically, as shown in Fig. 9.1. The heating time constant $T$ represents the time it takes for the temperature increase to reach 63.2 % of the difference between the steady state and initial temperature $t_s - t_0$.

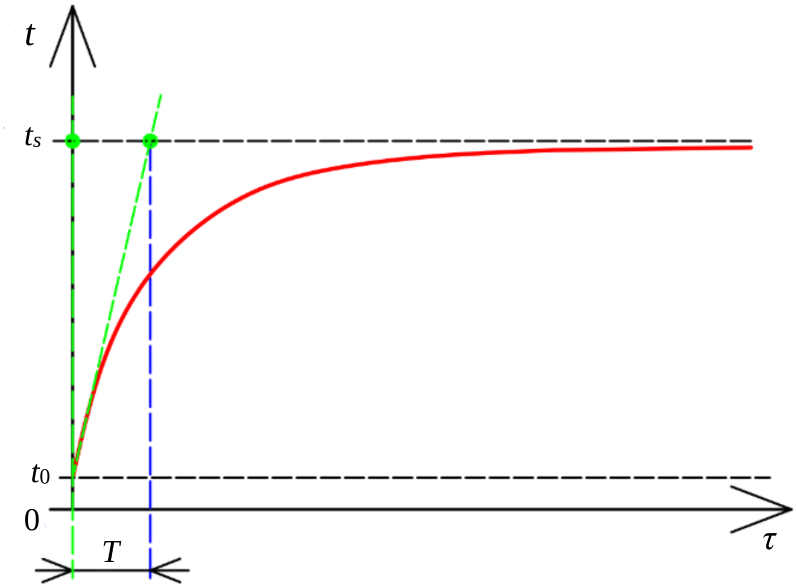



Fig. 9.1 Graphical dependence of heating process [29]

The time $\tau$ it takes to heat the circuit to temperature $t$ can be expressed from equation (1.8) as:

$$\tau = T \cdot \ln \frac{t_s - t_0}{t_s - t}. \tag{9.11}$$

If the heat transfer through the tank surface and individual components is not sufficient and the maximum operating temperature could be exceeded, the working liquid must be cooled, and a cooler must be installed in the system The required cooler power $\phi_C$ can be defined as the difference between the heat that enters the system and the heat removed by the heat exchange surface of the tank to the surroundings from the equation:

$$\phi_C = \phi - \phi_T, \tag{9.12}$$

where $\phi_C$ [W] is the required cooler power.

The amount of heat dissipated by the tank surface $\phi_T$ is determined from equation (1.6) as:

$$\phi_T = k_T \cdot A_T \cdot \Delta t_T \,. \tag{9.13}$$

where $\Delta t_T$ [°C] is the difference between the temperature $t_L$ of the working liquid in the tank and the air temperature $t_0$ around the tank.

In order to determine the heat transfer coefficient of the tank $k_T$, it is necessary to know the wall thickness of the tank, the coefficient of thermal conductivity of the tank wall material, the heat transfer coefficient between the liquid and the tank wall and the heat transfer coefficient between the tank wall and the surroundings. Its exact determination is relatively complicated and for practical purposes the values of the heat transfer coefficient of the tank were determined experimentally, see Tab 9.1.

Tab 9.1 Recommended values for selecting the heat transfer coefficient $k_T$ [7], [27]

| Sector | Application | $k_T$ [W · m$^{-2}$ · K$^{-1}$] |
|---|---|---|
| Stationary hydraulics | limited air flow around the tank | 7 ÷ 10 |
| | free air flow around the tank | 10 ÷ 15 |
| | intensive air flow around the tank | 17 ÷ 22 |
| Mobil hydraulics | tractors and loaders | 18 ÷ 20 |
| | excavators and cranes | 13 ÷ 15 |
| | agricultural machinery | 12 ÷ 18 |

Only the area in contact with the liquid on one side and the air on the other side is considered to be the heat transfer surface area $A_T$ of the tank. The tank cover and the tank bottom when standing on the floor are not considered.

From the cooling power of the cooler, it is possible to determine the required heat transfer surface area of the cooler:

$$A_C = \frac{\phi_C}{k_C \cdot \Delta t_m}\,, \tag{9.14}$$

where $A_C$ [m$^2$] is the necessary heat transfer surface of the cooler, $k_C$ [W · m$^{-2}$ · K$^{-1}$] is the heat transfer coefficient of the cooler, and $\Delta t_m$ [°C] is the mean logarithmic temperature difference.

The heat transfer surface $k_C$ of the cooler depends on the used cooler design. Approximate values of this coefficient during oil cooling by air are in the range of $k_C$ = (30 ÷ 200) W · m$^{-2}$ · K$^{-1}$.

The mean logarithmic temperature difference is given by the formula:

$$\Delta t_m = \frac{t_2 - t_1}{\ln \dfrac{t_L - t_1}{t_L - t_2}}\,, \tag{9.15}$$

where $t_L$ [°C] is the temperature of the working liquid of the hydraulic system entering the cooler, $t_1$ [°C] is the input temperature of the cooling medium, and $t_2$ [°C] is the output temperature of the cooling medium.

In hydraulic systems, in most cases, coolers with a forced flow of the cooling medium are used. This is usually water in the case of water coolers or air in the case of air coolers. For the design of a water pump or an air fan, it is necessary to determine the required volumetric flow of the cooling medium, which is determined by the equation:

$$Q_V = \frac{\phi_C}{c \cdot \rho \cdot \Delta t_{CM}}, \tag{9.16}$$

where $Q_V$ [$m^3 \cdot s^{-1}$] is the volumetric flow of the cooling medium (water or air) through the cooler, $c$ [$J \cdot kg^{-1} \cdot K^{-1}$] is the specific heat capacity of the cooling medium, $\rho$ [$kg \cdot m^{-3}$] is the density of the cooling medium, and $\Delta t_{CM}$ [°C] is the temperature difference of the cooling medium.

Similarly, it is possible to determine the required volumetric flow of the working (cooled) liquid of the hydraulic system through the cooler:

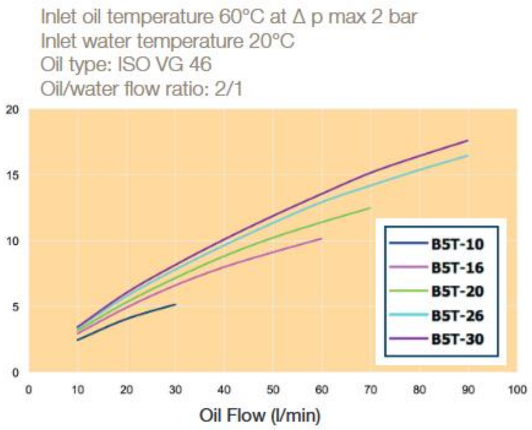
$$Q_L = \frac{\phi_C}{c_1 \cdot \rho_L \cdot (t_{L1} - t_{L2})}, \tag{9.17}$$

where $Q_L$ [$m^3 \cdot s^{-1}$] is the required volumetric flow of the working liquid of the hydraulic system through the cooler, $c_1$ [$J \cdot kg^{-1} \cdot K^{-1}$] is the specific heat capacity of the working liquid, $\rho_L$ [$kg \cdot m^{-3}$] is the density of the working liquid, $t_{L1}$ [°C] is the temperature of the working liquid at the cooler input, and $t_{L2}$ [°C] is the temperature of the working liquid at the cooler output.

The liquid flow rate $Q_L$ through the cooler depends on the type of hydraulic circuit and its function. In order to select a cooler, it is possible to use online software programs of cooling technology manufacturers or to select a cooler based on calculated values directly from the manufacturer's catalogue. The cooler selection is usually on the basis of the power characteristic in the case of water coolers, where the x-axis shows the volumetric flow $Q_L$ [$dm^3 \cdot min^{-1}$] of the working liquid through the cooler and on the y-axis the required cooling power $\phi_C$ [kW], as shown in Fig. 9.2 (left). For air coolers, the specific (specific cooling) power of the cooler $P_S$ is usually given; see Fig. 9.2 (right). This power is defined by the formula:

$$P_S = \frac{\phi_C}{(t_{L1} - t_1)}, \tag{9.18}$$

where $P_S$ [$W \cdot °C^{-1}$] is the specific power of cooler, $t_{L1}$ [°C] is maximal temperature of the working liquid (usually 60 °C), and $t_1$ [°C] is an ambient air temperature (usually 20 °C).



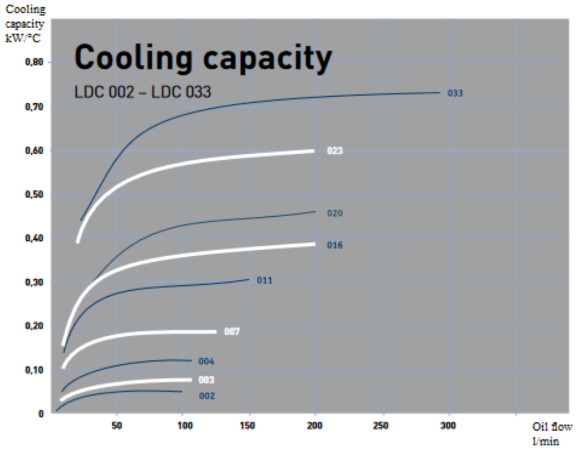




Fig. 9.2 Power characteristics for cooler selection, water plate cooler SWO by Parker company (left) [30], air cooler LAC by Parker company (right) [31]

## 9.1 Coolers

The basic classification of coolers used in hydraulic systems according to the coolant type is as follows:

- water coolers,
- air coolers.

Water coolers use water as the cooling medium. They are characterized by high dissipated heat power with relatively small cooler dimensions. They are mainly used in stationary hydraulic applications where the supply and discharge of the cooling medium (water) are ensured. In terms of construction, the most common are pipe coolers (see Fig. 9.3). Pipe coolers consist of a set of pipes that are coiled in the cooler body. From the point of view of the flow of two fluids, they can be either co-current or counter-current coolers. Pipe coolers are commonly manufactured for cooling powers up to 600 kW and coolant flow rates up to 1200 $dm^3 \cdot min^{-1}$.

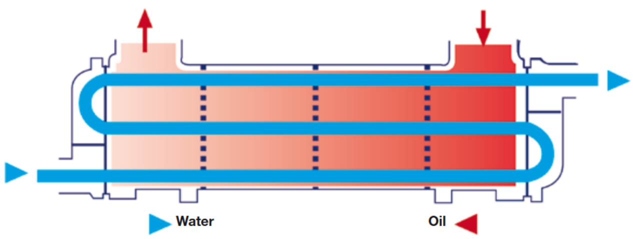



Fig. 9.3 Water pipe cooler SWO from Parker company (left), working principle of water pipe cooler (right) [32]

Another possible construction design of water coolers is the so-called plate coolers, as is shown in Fig. 9.4. These coolers are characterized by their high compactness. They consist of corrugated channel plates that separate the coolant and the cooled liquid and, with their small dimensions, ensure a large heat exchange surface area between the two liquids. Standard reported cooling powers of plate coolers are up to 600 kW for coolant flow rates up to 1800 $dm^3 \cdot min^{-1}$.

A certain disadvantage when using water coolers is the possibility of water leakage into the hydraulic system, which can occur especially when the cooler is damaged.


Fig. 9.4 Water plate cooler PWO from Parker company (left), operating principle of the plate water cooler (right) [30]

Air coolers use air as the cooling medium. The main parts of an air cooler are the body, in which the pipe for the flow of the coolant is guided, and the fan, which ensures the flow of cooling air. The fan can be driven by an electric motor or by a hydraulic motor. These coolers are especially suitable for cooling the hydraulic system of mobile hydraulic machines. Air coolers are manufactured for cooling powers up to 300 kW and coolant flow rates up to 400 $dm^3 \cdot min^{-1}$.

Compared to water coolers, air coolers have larger installation dimensions and are characterized by higher noise levels. An example of an LAC air cooler with an AC electric motor is shown in Fig. 9.5.


Fig. 9.5 Air cooler LAC with AC electric motor by Parker company [31]

**Example 9.1**

Determine for a stationary hydraulic device what heat output $\phi_T$ is dissipated through the tank surface to the surroundings. The tank length is $a$, the tank height is $b$, and the tank width is $c$ (see Fig. 9.6). The tank is filled with oil up to the height h, with the temperature $t_L$. The temperature around the tank is $t_0$. The tank is placed on the floor, and the bottom of the tank cannot be considered a heat transfer surface area. The heat transfer coefficient of the tank is $k_T$.

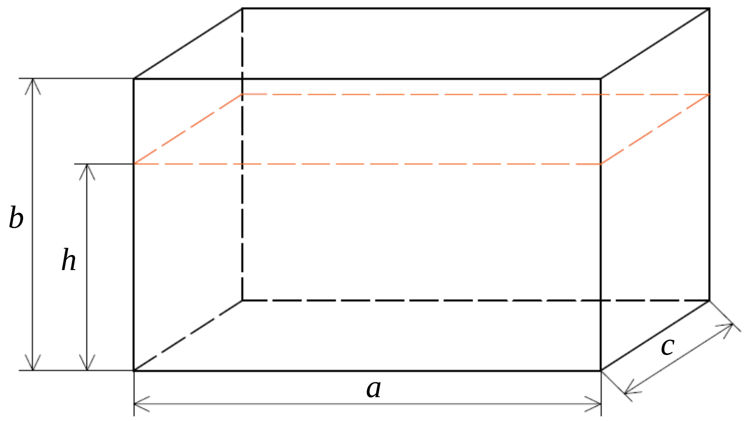



Fig. 9.6 Tank dimensions

Entered:

| | |
|---|---|
| tank length | $a = 1$ m |
| tank height | $b = 0.8$ m |
| tank width | $c = 0.4$ m |
| liquid height in tank | $h = 0.6$ m |
| liquid temperature in tank | $t_L = 55$ °C |
| ambient temperature around tank | $t_0 = 20$ °C |
| heat transfer coefficient of tank | $k_T = 10$ W · m$^{-2}$ · K$^{-1}$ |

Calculate:

| | |
|---|---|
| heat power dissipated through tank | $\phi_T = ?$ |

Calculation:

The tank heat transfer area $A_T$ without considering the tank bottom can be calculated as follows:

$$A_T = 2 \cdot a \cdot h + 2 \cdot c \cdot h = 2 \cdot 1 \cdot 0.6 + 2 \cdot 0.4 \cdot 0.6 = 1.68\, m^2 \,. \tag{9.19}$$

The heat power $\phi_T$ dissipated through the tank is calculated from the equation (9.13):

$$\begin{gathered} \phi_T = k_T \cdot A_T \cdot \Delta t_T \,, \\ \phi_T = k_T \cdot A_T \cdot (t_L - t_0) \,, \\ \phi_T = 10 \cdot 1.68 \cdot (55 - 20) = 588\, W \,. \end{gathered} \tag{9.20}$$

How will be changed the heat power $\phi_{T2}$ dissipate through the tank if the tank is placed on a pedestal (when the bottom surface of the tank is in contact with air)? In this case, heat is also exchanged between the tank bottom and the surrounding air. The tank heat transfer area $A_{T2}$ is determined as follows:

$$\begin{gathered} A_{T2} = 2 \cdot a \cdot h + 2 \cdot c \cdot h + a \cdot c \,, \\ A_{T2} = 2 \cdot 1 \cdot 0.6 + 2 \cdot 0.4 \cdot 0.6 + 0.6 \cdot 0.4 = 2.08\, m^2 \,. \end{gathered} \tag{9.21}$$

The heat power $\phi_{T2}$ dissipated through the tank is increased as follows:

$$\begin{gathered} \phi_{T2} = k_T \cdot A_{T2} \cdot \Delta t_T \,, \\ \phi_{T2} = k_T \cdot A_{T2} \cdot (t_L - t_0) \,, \\ \phi_{T2} = 10 \cdot 2.08 \cdot (55 - 20) = 728\, W \,. \end{gathered} \tag{9.22}$$

It is also possible to determine how the heat power dissipated through this tank would change if the tank was part of a mobile hydraulic system. The heat transfer coefficient of the tank is chosen from Tab 9.1; for mobile hydraulics, it is $k_{Tm}$ = 18 W · m$^{-2}$ · K$^{-1}$. The amount of heat $\phi_{Tm}$ dissipated through the tank in the mobile equipment is calculated:

$$\phi_{Tm} = k_{Tm} \cdot A_{T2} \cdot \Delta t_T \,,$$
$$\phi_{Tm} = k_{Tm} \cdot A_{T2} \cdot (t_L - t_0) \,, \tag{9.23}$$
$$\phi_{Tm} = 18 \cdot 2.08 \cdot (55 - 20) = 1310.4 \, W \,.$$

**Example 9.2**

Calculate the power loss in a closed hydraulic circuit. The circuit uses Sauer SPV22 hydraulic pump and SMF 22 hydraulic motor. The hydraulic circuit operates with constant input speed $n_1$ of the hydraulic pump at its constant geometric volume $V_{gG}$ and at different load values graphically shown in Fig. 9.7 (the equipment operates in a periodic production cycle). The values of the working pressures ($p_1 \div p_4$) correspond to the individual working periods ($\tau_1 \div \tau_4$). The pressure in the return line behind the hydraulic motor is constant, maintained by the relief valve at the value $p_0$.

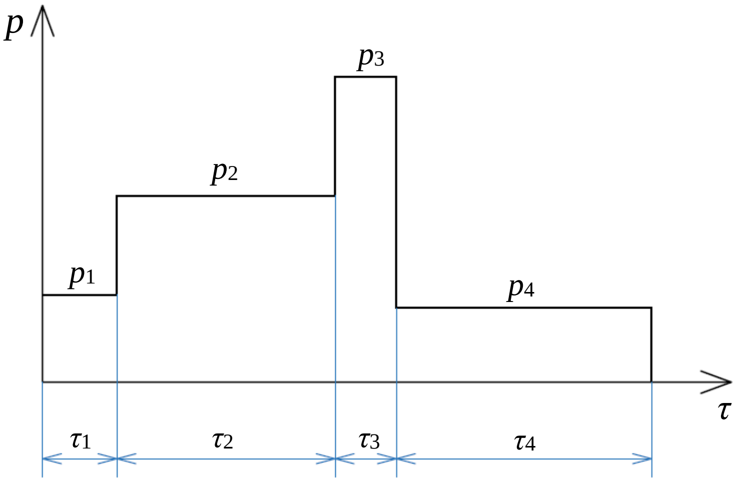



Fig. 9.7 Load course in a closed hydraulic circuit

<u>Entered:</u>

| | |
|---|---|
| speed of the hydraulic pump | $n_1$ = 2500 min$^{-1}$ |
| geometric stroke volume of the hydraulic pump | $V_{gG}$ = 80 cm$^3$ |
| working pressures in time intervals | $p_1$ = 10 MPa |
| | $p_2$ = 22 MPa |
| | $p_3$ = 35 MPa |
| | $p_4$ = 8 MPa |
| time working intervals | $\tau_1$ = 30 s |
| | $\tau_2$ = 120 s |
| | $\tau_3$ = 20 s |

$\tau_4 = 160$ s

pressure in the return line behind hydraulic motor $p_0 = 1.4$ MPa

<u>Calculate:</u>

power loss in a closed hydraulic circuit $P_l = ?$

<u>Calculation:</u>

The power of the hydraulic system is given by the equation:

$$P_i = (p_i - p_0) \cdot n_1 \cdot V_{gG} \,. \tag{9.24}$$

For the relevant load, the powers ($P_1 \div P_4$) are determined according to the equation (9.24):

$$\begin{aligned}
P_1 &= (p_1 - p_0) \cdot n_1 \cdot V_{gHP} = (10 - 1.4) \cdot 10^6 \cdot \frac{2500}{60} \cdot 80 \cdot 10^{-6} = 28.66\ kW\,, \\
P_2 &= (p_2 - p_0) \cdot n_1 \cdot V_{gHP} = (22 - 1.4) \cdot 10^6 \cdot \frac{2500}{60} \cdot 80 \cdot 10^{-6} = 68.66\ kW\,, \\
P_3 &= (p_3 - p_0) \cdot n_1 \cdot V_{gHP} = (35 - 1.4) \cdot 10^6 \cdot \frac{2500}{60} \cdot 80 \cdot 10^{-6} = 112\ kW\,, \\
P_4 &= (p_4 - p_0) \cdot n_1 \cdot V_{gHP} = (8 - 1.4) \cdot 10^6 \cdot \frac{2500}{60} \cdot 80 \cdot 10^{-6} = 22\ kW\,.
\end{aligned} \tag{9.25}$$

The equivalent power $P_e$ for the whole working cycle of the equipment is calculated from the formula:

$$\begin{aligned}
P_e &= \sqrt{\frac{\sum P_i^2 \cdot \tau_i}{\sum \tau_i}}\,, \\
P_e &= \sqrt{\frac{P_1^2 \cdot \tau_1 + P_2^2 \cdot \tau_2 + P_3^2 \cdot \tau_3 + P_4^2 \cdot \tau_4}{\tau_1 + \tau_2 + \tau_3 + \tau_4}}\,, \\
P_e &= \sqrt{\frac{28.66^2 \cdot 30 + 68.66^2 \cdot 120 + 112^2 \cdot 20 + 22^2 \cdot 160}{30 + 120 + 20 + 160}} = 52.76\ kW\,.
\end{aligned} \tag{9.26}$$

The flow and mechanical-pressure efficiency values correspond to the operating pressures at given speeds. The total efficiency of a hydraulic converter is given by the product of the volumetric and mechanical-hydraulic efficiencies. Determine the total efficiency for the values given in Tab 9.2.

Tab 9.2 Efficiency values of hydraulic converters for individual working loads

| Working pressure | Hydraulic pump | | | Hydraulic motor | | |
|---|---|---|---|---|---|---|
| | $\eta_{mhG}$ | $\eta_{volG}$ | $\boldsymbol{\eta_{TG}}$ | $\eta_{mhM}$ | $\eta_{volM}$ | $\boldsymbol{\eta_{TM}}$ |
| $p_1$ = 10 MPa | 0,880 | 0,983 | **0,865** | 0,901 | 0,987 | **0,889** |
| $p_2$ = 22 MPa | 0,933 | 0,962 | **0,897** | 0,928 | 0,977 | **0,906** |
| $p_3$ = 35 MPa | 0,944 | 0,945 | **0,892** | 0,948 | 0,953 | **0,903** |
| $p_4$ = 8 MPa | 0,850 | 0,988 | **0,839** | 0,824 | 0,994 | **0,819** |

To calculate the power loss, it is necessary to know the total efficiency of the two hydraulic converters, which corresponds to the whole working cycle of the equipment. The total efficiency can be simplified as the mean value of the individual efficiencies, or as the equivalent value.

The mean value of the total efficiency of the hydraulic pump $\eta_{TGm}$ is calculated from the equation:

$$\eta_{TGm} = \frac{\eta_{TG1} + \eta_{TG2} + \eta_{TG3} + \eta_{TG4}}{4}, \tag{9.27}$$

$$\eta_{TGm} = \frac{0.865 + 0.897 + 0.892 + 0.839}{4} = 0.873\,.$$

The mean value of the total efficiency of the hydraulic motor $\eta_{TMm}$ is calculated from the equation:

$$\eta_{TMm} = \frac{\eta_{TM1} + \eta_{TM2} + \eta_{TM3} + \eta_{TM4}}{4}, \tag{9.28}$$

$$\eta_{TMm} = \frac{0.889 + 0.906 + 0.903 + 0.819}{4} = 0.879.$$

The power loss for the mean efficiency values $P_{lm}$ can be calculated using equation (9.3) as:

$$P_{lm} = P_e \cdot (1 - \eta_{TGm} \cdot \eta_{TMm}) = 52.76 \cdot (1 - 0.873 \cdot 0.879) = 12.29\ kW\,. \tag{9.29}$$

The equivalent value of the total efficiency of the hydraulic pump $\eta_{TGe}$ is calculated from the equation:

$$\eta_{TGe} = \sqrt{\frac{\sum \eta_i^2 \cdot \tau_i}{\sum \tau_i}},$$

$$\eta_{tGe} = \sqrt{\frac{\eta_{TG1}^2 \cdot \tau_1 + \eta_{TG2}^2 \cdot \tau_2 + \eta_{TG3}^2 \cdot \tau_3 + \eta_{TG4}^2 \cdot \tau_4}{\tau_1 + \tau_2 + \tau_3 + \tau_4}}, \tag{9.30}$$

$$\eta_{tGe} = \sqrt{\frac{0.865^2 \cdot 30 + 0.897^2 \cdot 120 + 0.892^2 \cdot 20 + 0.839^2 \cdot 160}{30 + 120 + 20 + 160}} = 0.865\,.$$

The equivalent value of the total efficiency of the hydraulic motor $\eta_{TMe}$ is calculated from the equation:

$$\eta_{TMe} = \sqrt{\frac{\sum \eta_i^2 \cdot \tau_i}{\sum \tau_i}},$$

$$\eta_{TMe} = \sqrt{\frac{\eta_{TM1}^2 \cdot \tau_1 + \eta_{TM2}^2 \cdot \tau_2 + \eta_{TM3}^2 \cdot \tau_3 + \eta_{TM4}^2 \cdot \tau_4}{\tau_1 + \tau_2 + \tau_3 + \tau_4}}, \tag{9.31}$$

$$\eta_{TMe} = \sqrt{\frac{0.889^2 \cdot 30 + 0.906^2 \cdot 120 + 0.903^2 \cdot 20 + 0.819^2 \cdot 160}{30 + 120 + 20 + 160}} = 0.863 .$$

And the power loss for the equivalent efficiency values $P_{le}$ is given by the formula:

$$P_{le} = P_e \cdot (1 - \eta_{TGe} \cdot \eta_{TMe}) = 52.76 \cdot (1 - 0.865 \cdot 0.863) = 13.4\ kW . \tag{9.32}$$

**Example 9.3**

Determine the heating course of oil in a tank whose heat transfer surface is $A_T$ nd which contains oil of volume $V_T$. The input power of the hydraulic mechanism is $P_1$. The total efficiency of the hydraulic pump is $\eta_{TG}$, and the total efficiency of the hydraulic motor is $\eta_{TM}$. The output pressure from the hydraulic pump is $p_1$. The pressure loss of lines and hydraulic elements between the hydraulic pump and the hydraulic motor is $\Delta p_l$. The weight $m$ of the hydraulic pump, hydraulic motor, hydraulic elements, tank, and lines is estimated. The initial oil temperature in the tank is $t_0$, and the heat transfer coefficient of the tank is $k_T$.

<u>Entered:</u>

| | |
|---|---|
| heat transfer area of tank | $A_T = 3.2\ m^2$ |
| oil volume in the tank | $V_T = 0.42\ m^3$ |
| input power of the hydraulic pump | $P_1 = 12$ kW |
| total efficiency of the hydraulic pump | $\eta_{TG} = 0.92$ |
| total efficiency of the hydraulic motor | $\eta_{TM} = 0.94$ |
| output pressure behind the hydraulic pump | $p_1 = 16$ MPa |
| pressure loss of lines and hydraulic elements | $\Delta p_l = 0.62$ MPa |
| weight of tank and all elements of the hydraulic circuit | $m$ =145 kg |
| initial oil temperature in tank (ambient temperature) | $t_0 = 18$ °C |
| oil density | $\rho = 900\ kg \cdot m^{-3}$ |
| heat transfer coefficient | $k_T = 15\ W \cdot m^{-2} \cdot K^{-1}$ |

Calculate:

Calculate and graphically display the heating course of the hydraulic system.

Calculation:

The efficiency of lines and hydraulic elements of the circuit $\eta_l$ is given by the formula:

$$\eta_l = 1 - \frac{\Delta p_l}{p_1} = 1 - \frac{0.62}{16} = 0.961 \,. \tag{9.33}$$

The total efficiency $\eta_T$ of the hydraulic system is given:

$$\eta_T = \eta_{TG} \cdot \eta_{TM} \cdot \eta_l = 0.92 \cdot 0.94 \cdot 0.961 = 0.831 \,. \tag{9.34}$$

The power loss $P_l$ of the hydraulic system is calculated according to the equation (9.3):

$$P_l = P_1 \cdot (1 - \eta_T) = 12\,000 \cdot (1 - 0.831) = 2028\,W \,. \tag{9.35}$$

The heating time constant $T$ of the hydraulic system is determined according to the equation (9.9). The predominant materials of the hydraulic system are included in the calculation. Among the solid parts, it is steel. The substantial part of the hydraulic system consists of the working fluid (oil). The oil weight $m_1$ can be calculated from its volume and density:

$$m_1 = V_T \cdot \rho = 0.42 \cdot 900 = 378\,kg \,. \tag{9.36}$$

The specific heat capacity of oil is $c_1$ = 1850 J · kg$^{-1}$ · K$^{-1}$, and the specific heat capacity of steel is $c_2$ = 460 J · kg$^{-1}$ · K$^{-1}$. To simplify the calculation, the heat dissipation to the surroundings is considered only through the heat transfer surface of the tank. Now, the heating time constant $T$ of the hydraulic system is given by the formula:

$$T = \frac{m_1 \cdot c_1 + m_2 \cdot c_2}{k_T \cdot A_T} = \frac{378 \cdot 1850 + 145 \cdot 460}{15 \cdot 3.2} = 15\,958\,s \,. \tag{9.37}$$

The heating process can be calculated according to the equation (9.8); for the graphical expression of the heating course it is necessary to choose different times $\tau$. The heat power $\phi$ is equal to the loss power $P_l$. As a calculation example, the calculation of the oil temperature $t$ over the time $\tau$ = 3600 s = 1 hour is:

$$t = t_0 + \frac{\phi}{k_T \cdot A_T} \cdot \left(1 - e^{-\frac{\tau}{T}}\right) = 18 + \frac{2028}{15 \cdot 3.2} \cdot \left(1 - e^{-\frac{3600}{15958}}\right) = 26.53\,°C \,, \tag{9.38}$$

Calculation of the oil temperature $t$ for different time intervals $\tau$ is shown in Tab 9.3.

Tab 9.3 Calculation of the oil temperature $t$ for different time intervals $\tau$

| $\tau$ [s] | 0 | 3600 | 7200 | 16000 | 32000 | 64000 | 120000 |
|---|---|---|---|---|---|---|---|
| $t$ [°C] | $t_0$ | 26,53 | 33,34 | 44,74 | 54,56 | 59,48 | 60,23 |

According to the values calculated in Tab 9.3 it is possible to draw the oil heating course as is shown in Fig. 9.8.

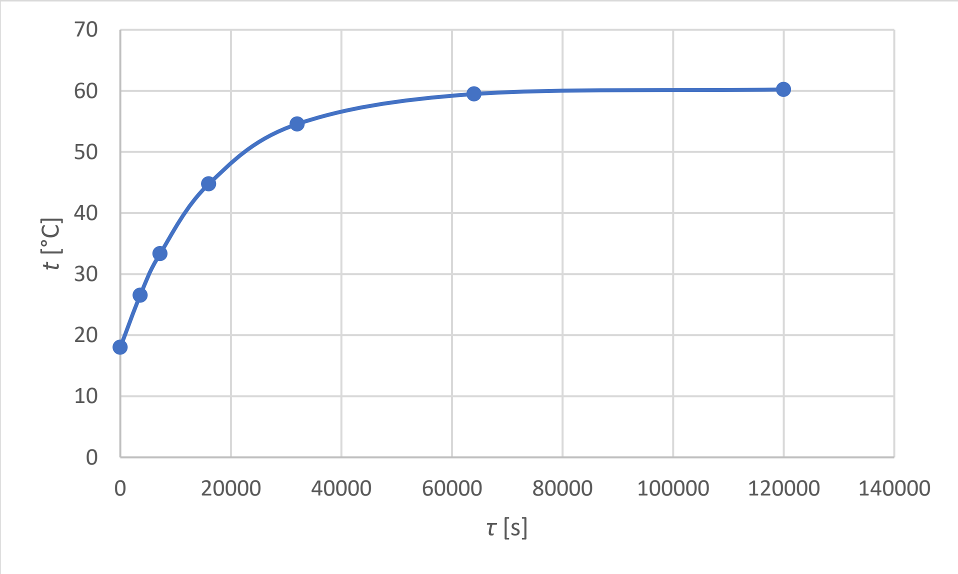



Fig. 9.8 Oil heating course in the tank of the hydraulic system

**Example 9.4**

Determine the time required to adjust the automatic oil temperature control timer in the steel tank that controls the valve in the cooling water supply to the cooler. When the oil temperature in the tank reaches $t_i$, this temperature must be reduced by cooling to $t_e$. The air temperature around the tank is $t_0$. The tank dimensions are the tank length $a$, the tank height $b$ and the tank width $c$. The tank is placed on the floor and is filled with oil from 80% of its volume $V_T$. The tank weight (without oil) is $m_2$.

<u>Entered:</u>

| | |
|---|---|
| tank length | $a = 0.8$ m |
| tank height | $b = 0.5$ m |
| tank width | $c = 0.6$ m |
| oil temperature at which the cooling is switched on | $t_i = 60$ °C |
| temperature to which the oil in the tank should be cooled | $t_e = 40$ °C |
| air temperature around the tank | $t_0 = 20$ °C |
| heat transfer coefficient of the tank | $k_T = 12\ \mathrm{W \cdot m^{-2} \cdot K^{-1}}$ |
| weight of the tank (steel) | $m_2 = 55$ kg |
| oil density | $\rho = 900\ \mathrm{kg \cdot m^{-3}}$ |

<u>Calculate:</u>

The time for which the timer will be adjusted $\tau = ?$

Calculation:

The tank volume $V_T$ can be calculated from the equation:

$$V_T = a \cdot b \cdot c = 0.8 \cdot 0.5 \cdot 0.6 = 0.24\ m^3\ . \tag{9.39}$$

The tank is filled with oil from 80% of its volume. Therefore, the liquid volume $V_L$ in the tank is:

$$V_L = V_T \cdot 0.8 = 0.192\ m^3\ . \tag{9.40}$$

The height of the oil level $h$ in the tank is determined from the tank dimensions and the liquid volume in the tank according to the formula:

$$V_L = a \cdot c \cdot h \rightarrow h = \frac{V_L}{a \cdot c} = \frac{0.192}{0.8 \cdot 0.6} = 0.4\ m\ . \tag{9.41}$$

The oil weight $m_1$ in the tank is calculated as follows:

$$m_1 = V_L \cdot \rho = 0.192 \cdot 900 = 172.8\ kg\ . \tag{9.42}$$

The heat transfer area $A_T$ of the tank is:

$$A_T = 2 \cdot a \cdot h + 2 \cdot c \cdot h = 2 \cdot 0.8 \cdot 0.4 + 2 \cdot 0.6 \cdot 0.8 = 1.12\ m^2\ . \tag{9.43}$$

The specific heat capacity of oil is $c_1 = 1850$ J · kg$^{-1}$ · K$^{-1}$, and the specific heat capacity of steel is $c_2 = 460$ J · kg$^{-1}$ · K$^{-1}$. It is possible to determine the cooling time constant $T$:

$$T = \frac{m_1 \cdot c_1 + m_2 \cdot c_2}{k_T \cdot A_T} = \frac{172.8 \cdot 1850 + 55 \cdot 460}{12 \cdot 1.12} = 25\ 668\ s\ . \tag{9.44}$$

It is now possible to determine the time $\tau$ for setting the time switch. The equation (9.8) is applied, where the maximum oil temperature is $t_i$, and the required oil temperature is $t_e$. For setting the time switch, it is given:

$$\tau = T \cdot \ln \frac{t_i - t_0}{t_e - t_0} = 25\ 668 \cdot\ \ln \frac{60 - 20}{40 - 20} = 17\ 791.7\ s. \tag{9.45}$$

The timer will be set to the time 4.94 hours ≈ 4 hours and 56 minutes.

**Example 9.5**

Determine the heating course in a system with throttle valve control if the volumetric flow through the throttle valve is $Q_{TV}$ and the pressure gradient on the throttle valve is $\Delta p_{TV}$. Calculate the maximum steady-state oil temperature $t_s$, determine the heating time constant $T$, the oil heating time $\tau$ to a temperature of 55 °C, the required cooling power $\phi_C$ of a cooler, and design a water cooler.

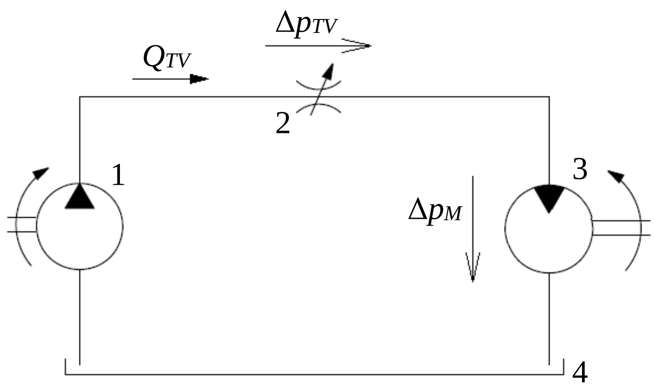



Fig. 9.9 Hydraulic scheme

*1 – hydraulic pump, 2 – throttle valve, 3 – hydraulic motor, 4 - tank*

Entered:

| | |
|---|---|
| volumetric flow through the throttle valve | $Q_{TV} = 7.5 \cdot 10^{-4}\ m^3 \cdot s^{-1}$ |
| pressure gradient on the throttle valve | $\Delta p_{TV} = 6$ MPa |
| tank volume | $V_T = 400\ dm^3$ |
| heat exchange surface of the tank | $A_T = 2\ m^2$ |
| heat transfer coefficient of the tank | $k_T = 14\ W^2 \cdot K^{-1} \cdot m^{-2}$ |
| initial oil temperature | $t_0 = 18$ °C |
| oil density | $\rho_F = 880\ kg \cdot m^{-3}$ |
| weight of steel parts | $m_2 = 130$ kg |
| specific heat capacity of oil | $c_1 = 1.8 \cdot 10^3\ J \cdot kg^{-1} \cdot K^{-1}$ |
| specific heat capacity of steel | $c_2 = 0.45 \cdot 10^3\ J \cdot kg^{-1} \cdot K^{-1}$ |

Calculate: $t_s = ?$, $T = ?$, $\phi_C = ?$

Calculation:

The power loss $P_L$ is equal to the heat flow $\phi$. For the given case, it is considered with the simplification that the energy is converted into heat during liquid flow through the throttle valve:

$$\phi = P_L = Q_{TV} \cdot \Delta p_{TV} = 7.5 \cdot 10^{-4} \cdot 6 \cdot 10^6 = 4\,500\ W = 4.5\ kW\,. \tag{9.46}$$

The steady-state temperature $t_s$ can be calculated from the equation:

$$t = t_0 + \frac{\phi}{k_T \cdot A_T} \cdot \left(1 - e^{-\frac{\tau}{T}}\right). \tag{9.47}$$

If we theoretically assume that the maximum steady-state temperature $t_s$ can be continuously increased, i.e., it will be reached in time $\tau = \infty$, by substituting into the previous equation:

$$t = t_0 + \frac{\phi}{k_T \cdot A_T} \cdot \left(1 - e^{-\frac{\infty}{T}}\right),$$
$$e^{-\infty} = 0\,, \tag{9.48}$$
$$t_s = t_0 + \frac{\phi}{k_T \cdot A_T} \cdot (1 - 0) = t_0 + \frac{\phi}{k_T \cdot A_T} = 18 + \frac{4\,500}{14 \cdot 2} = 178.7\,°C\,.$$

The oil weight $m_1$ can be calculated using the oil density $\rho_L$ and the useful tank volume $V_T$:

$$\rho_L = \frac{m_1}{V_T} \Rightarrow m_1 = \rho_F \cdot V_T = 880 \cdot 0.4 = 352\,kg\,. \tag{9.49}$$

The heating time constant $T$ can be calculated:

$$T = \frac{m_1 \cdot c_1 + m_2 \cdot c_2}{k_T \cdot A_T} = \frac{352 \cdot 1.8 \cdot 10^3 + 130 \cdot 0.45 \cdot 10^3}{14 \cdot 2} = 24\,718\,s\,, \tag{9.50}$$
$$T = 6.86\,hrs$$

The heating time constant $T$ can be determined graphically from the heating course Fig. 9.10:

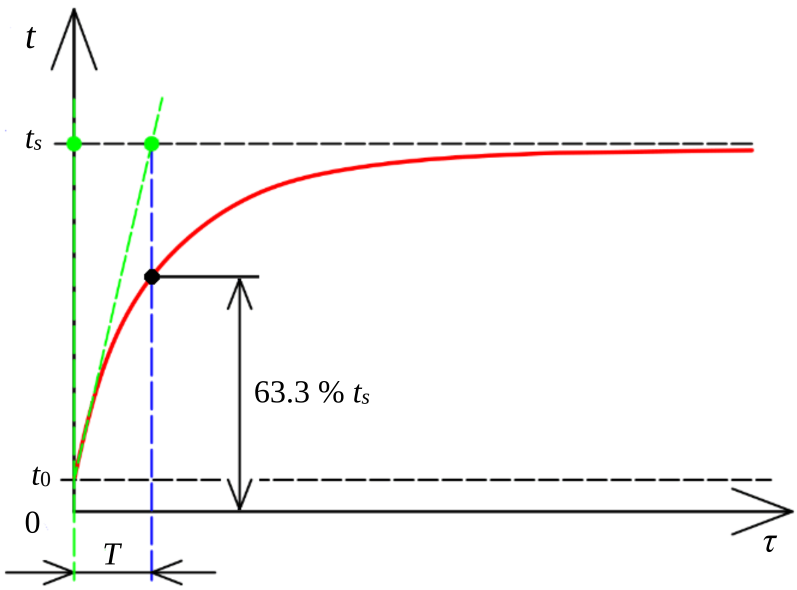



Fig. 9.10 Heating course

Determine the oil heating time $\tau$ to a temperature of $55°C$:

$$\tau = T \cdot ln\frac{t_s - t_0}{t_s - t} = 24\,718 \cdot ln\frac{178.7 - 18}{178.7 - 55} = 6\,468\,s = 1.79\,hrs\,. \tag{9.51}$$

Design a separate cooling system so that the temperature in the tank is stabilized at $t_T = 55$ °C.

The required cooler power $\phi_C$ is calculated from the loss power $P_L$, from which the thermal power removed by the tank $\phi_T$ is subtracted:

$$\phi_C = \phi - \phi_T = \phi - k_T \cdot A_T \cdot (t_T - t_0) = 4\,500 - 14 \cdot 2 \cdot (55 - 18)\,, \tag{9.52}$$

$$\phi_C = 3\,464\,W\,.$$

For the oil flow $Q_L$ through the cooler, we know that the temperature $t_T = t_{L1} = 55$ °C and we choose the oil temperature coming out of the cooler $t_{L2} = 40$ °C:

$$Q_L = \frac{\phi_C}{\rho_L \cdot c_1 \cdot (t_{L1} - t_{L2})} = \frac{3\,464}{880 \cdot 1.8 \cdot 10^3 \cdot (55-40)}, \tag{9.53}$$

$$Q_L = 1.46 \cdot 10^{-4}\, m^3 \cdot s^{-1} = 8.7\, l \cdot min^{-1}$$

We can choose a suitable type of a water cooler from a catalogue, e.g., from the Parker company, as shown in the figure below:

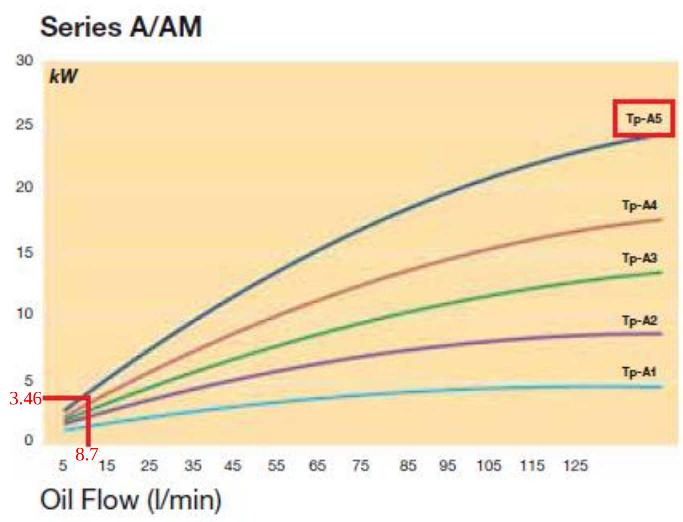
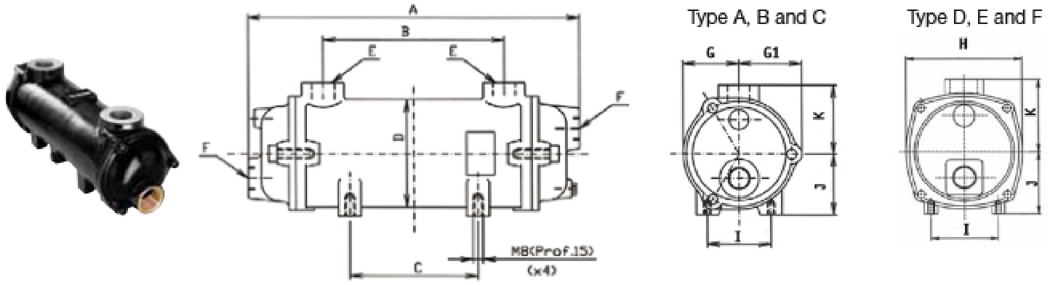



Fig. 9.11 Choosing a water cooler from the Parker catalogue [32]

**Main dimensions and typical performance**




| Model | Dimensions mm | | | | | | | | | | | Dis-sipated Power | Oil Flow Rate | Water Flow Rate | Oil Pres-sure Drop | Water Pres-sure Drop | Sur-face | Weight |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | A | B | C | D | E F | G | G1 | H | I | J | K | kW | l/min | l/min | bar | bar | m² | kg |
| Tp-A 1 | 195 | 72 | 38 | 86 | ¾" | 50 | 55 | - | 54 | 55 | 60 | 3 | 30 | 15 | 0.10 | 0.02 | 0.13 | 3 |
| Tp-A 2 | 263 | 142 | 106 | 86 | ¾" | 50 | 55 | - | 54 | 55 | 60 | 6 | 46 | 23 | 0.19 | 0.05 | 0.22 | 3.5 |
| Tp-A 3 | 349 | 228 | 192 | 86 | ¾" | 50 | 55 | - | 54 | 55 | 60 | 9 | 56 | 28 | 0.36 | 0.09 | 0.32 | 4 |
| Tp-A 4 | 448 | 326 | 290 | 86 | ¾" | 50 | 55 | - | 54 | 55 | 60 | 13 | 64 | 32 | 0.60 | 0.13 | 0.46 | 4.7 |
| Tp-A 5 | 576 | 454 | 418 | 86 | ¾" | 50 | 55 | - | 54 | 55 | 60 | 16 | 56 | 28 | 0.56 | 0.12 | 0.68 | 5.5 |
| Tp-B 1 | 273 | 123 | 109 | 108 | 1" | 60 | 65 | - | 77 | 65 | 70 | 8 | 66 | 33 | 0.16 | 0.02 | 0.33 | 5 |
| Tp-B 2 | 355 | 205 | 190 | 108 | 1" | 60 | 65 | - | 77 | 65 | 70 | 12 | 80 | 40 | 0.32 | 0.03 | 0.48 | 6 |
| Tp-B 3 | 452 | 302 | 289 | 108 | 1" | 60 | 65 | - | 77 | 65 | 70 | 18 | 104 | 52 | 0.96 | 0.07 | 0.66 | 7 |
| Tp-B 4 | 587 | 437 | 422 | 108 | 1" | 60 | 65 | - | 77 | 65 | 70 | 25 | 106 | 53 | 1 | 0.11 | 0.90 | 8.2 |
| Tp-B 5 | 730 | 580 | 566 | 108 | 1" | 60 | 65 | - | 77 | 65 | 70 | 29 | 98 | 49 | 1.04 | 0.14 | 1.16 | 10 |
| Tp-C 1 | 372 | 187 | 93 | 130 | 1¼" | 70 | 80 | - | 77 | 75 | 80 | 16 | 100 | 50 | 0.28 | 0.04 | 0.64 | 9 |
| Tp-C 2 | 472 | 287 | 193 | 130 | 1¼" | 70 | 80 | - | 77 | 75 | 80 | 26 | 120 | 60 | 0.55 | 0.07 | 0.90 | 10 |
| Tp-C 3 | 600 | 416 | 322 | 130 | 1¼" | 70 | 80 | - | 77 | 75 | 80 | 36 | 140 | 70 | 0.74 | 0.13 | 1.23 | 12.5 |
| Tp-C 4 | 744 | 559 | 465 | 130 | 1¼" | 70 | 80 | - | 77 | 75 | 80 | 48 | 160 | 80 | 1.06 | 0.17 | 1.60 | 14.5 |
| Tp-C 5 | 922 | 737 | 643 | 130 | 1¼" | 70 | 80 | - | 77 | 75 | 80 | 56 | 140 | 70 | 0.95 | 0.16 | 2.07 | 17.5 |

Fig. 9.12 Technical parameters of the cooler given in the Parker catalogue [32]

In this case, the oil/water tubular cooler type Tp-A5 was selected.

# 10. Design of hydraulic circuit for the derivation of pushing force

**Example 10.1**

Design a hydraulic circuit of a pushing device to derive the pushing force $F$, the extension velocity of the piston rod $v_1$ the retraction velocity of the piston rod $v_2$. Choose the mechanical-hydraulic efficiency for the first design $\eta_{mhM}$. The pressure on the relief valve is $p_{RV}$. Choose other necessary parameters from standards and manufacturer's documents.

| | |
|---|---|
| required pushing force | $F = 85\ 000$ N |
| extension velocity of the piston rod | $v_1 = 0.1\ \mathrm{m \cdot s^{-1}}$ |
| retraction velocity of the piston rod | $v_2 = 0.2\ \mathrm{m \cdot s^{-1}}$ |
| mechanical-hydraulic efficiency of the hydraulic cylinder | $\eta_{mhM} = 0.95$ |
| pressure on the relief valve | $p_{RV} = 16$ MPa |
| pipe length | $L_1 = L_2 = L_3 = L_4 = 2$ m |

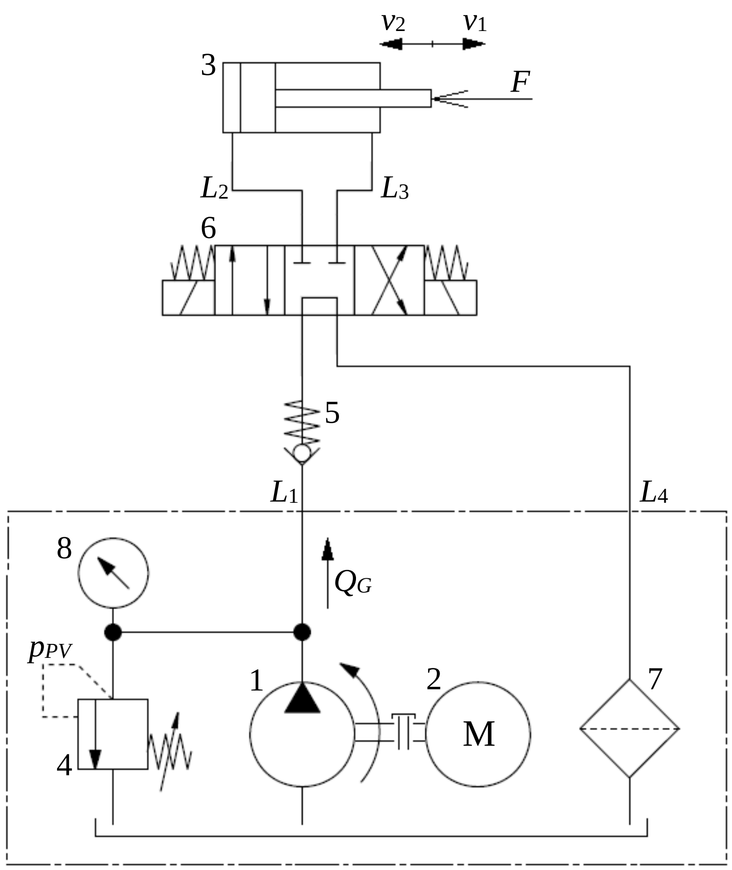



Fig. 10.1 Diagram of the hydraulic circuit

*1 – hydraulic pump, 2 – electric motor, 3 – hydraulic cylinder, 4 – relief valve, 5 – check valve, 6 – directional valve, 7 – filter, 8 - manometer*

**Design of the hydraulic cylinder**

According to the specification, the pressure on the relief valve is set to $p_{RV}$ = 16 MPa. For initial calculations, however, we will assume that the pressure $p_1$ is only 12 MPa. In this way, we keep a reserve for pressure losses that occur due to the liquid flow through the lines and control elements. The reserve is also used to start the hydraulic motor (dynamic pressure to accelerate the parts connected to the piston rod).

We start with the balance equation of the forces acting on the hydraulic cylinder:

$$p_1 \cdot A_1 - p_2 \cdot A_2 - \frac{F}{\eta_{mhM}} = 0\,, \tag{10.1}$$

where $p_1$ [Pa] is pressure acting on the piston side of the hydraulic cylinder, $p_2$ [Pa] pressure acting on the piston rod side of the hydraulic cylinder, $A_1$ [m$^2$] the piston area, and $A_2$ [m$^2$] is the piston rod side area of the hydraulic cylinder.

We can consider the pressure $p_2$ = 0 MPa. From the above equation, we will determine the theoretical value of the area $A_{t1}$:

$$p_1 \cdot A_{1t} - F = 0 \;\rightarrow\; A_{1t} = \frac{F}{p_1 \cdot \eta_{mhM}} = \frac{85\,000}{12 \cdot 10^6 \cdot 0.95} = 7.465 \cdot 10^{-3}\, m^2\,. \tag{10.2}$$

where $A_{1t}$ [m$^2$] is the theoretical piston area.

Selection of the piston diameter $D$ of the hydraulic cylinder:

$$A_{1t} = \frac{\pi \cdot D_t^2}{4} \;\rightarrow\; D_t = \sqrt{\frac{4 \cdot A_{1t}}{\pi}} = \sqrt{\frac{4 \cdot 7.465 \cdot 10^{-3}}{\pi}} = 0.097\, m\,. \tag{10.3}$$

where $D_t$ [m] is the theoretical piston diameter of the hydraulic cylinder.

According to ČSN 119101 [33] standard, basic and additional standard piston diameters are specified.

Tab 10.1 Standard piston diameters of hydraulic cylinders according to ČSN 119101 standard

| Standard piston diameters [mm] | |
|---|---|
| basic | 25; 32; 40; 50; 63; 80; 100; 125; 160; 200; 250; 320 |
| additional | 90; 110; 140; 180; 220; 280 |

Basically, all suppliers offer cylinders in the basic dimension range. According to the Bosch Rexroth catalogue, we can choose the piston diameter $D$ = 100 mm (see the catalogue preview below in Fig. 10.2).

Recalculation of the actual piston area $A_1$:

$$A_1 = \frac{\pi \cdot D^2}{4} = \frac{\pi \cdot 0.1^2}{4} = 0.007854\,m^2\,. \tag{10.4}$$

The actual required pressure $p_{1r}$ by modifying the equation (10.2):

$$p_{1r} = \frac{F}{A_1 \cdot \eta_{mhM}} = \frac{85\,000}{0.007854 \cdot 0.95} = 11\,392\,143\,Pa = 11.39\,MPa\,. \tag{10.5}$$

The pressure $p_{1r}$ is lower than the chosen pressure of 12 MPa, i.e., the total reserve to cover pressure losses increased even more.

Next, it is necessary to calculate the diameter of the piston rod. In this case, the required extension velocity $v_1$ and the retraction velocity $v_2$ must be considered. The continuity equation $Q = A \cdot v =$ const. is used to calculate the required (theoretical) inter-circular area $A_{2t}$ on the piston rod side:

$$A_1 \cdot v_1 = A_{2t} \cdot v_2 \Longrightarrow A_{2t} = A_1 \cdot \frac{v_1}{v_2} = 0.007854 \cdot \frac{0.1}{0.2} = 0.003927\,m^2\,, \tag{10.6}$$

where $A_{2t}$ [m$^2$] is the theoretical (required) inter-circular area on the piston rod side.

And the required diameter of the piston rod $d_t$:

$$A_{2t} = \frac{\pi \cdot D^2}{4} - \frac{\pi \cdot d_t^2}{4} \Longrightarrow d_t = \sqrt{D^2 - \frac{4 \cdot A_{2t}}{\pi}} = \sqrt{0.1^2 - \frac{4 \cdot 0.003927}{\pi}}\,, \tag{10.7}$$

$$d_t = 0.0707\,m\,,$$

where $d_t$ [m] is the required (theoretical) area on the piston rod side of the hydraulic cylinder.

Similar to the piston diameter selection, there are basic and additional standard piston rod diameters $d_t$ Tab 10.2:

Tab 10.2 Standard piston rod diameters of hydraulic cylinders according to ČSN 119101 standard

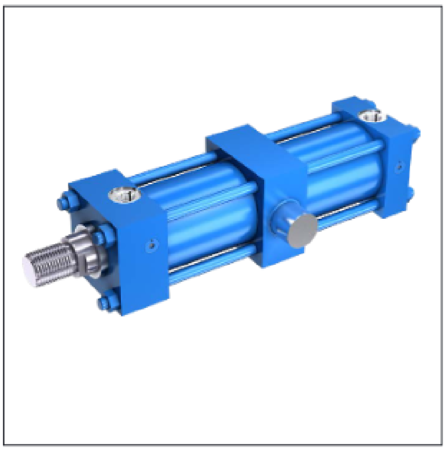
| Standard piston rod diameters [mm] | |
|---|---|
| basic | 25; 32; 40; 50; 63; 80; 100; 125; 160; 200; 250; 320 |
| additional | 90; 110; 140; 180; 220; 280 |

Therefore, a piston rod diameter of 70 mm is chosen from the above possibilities. As can be seen from the equation (10.6), the inter-circular area should be equal to half the retraction velocity, i.e., the area ratio $A_1/A_2 = 2$. Some manufacturers specify this ratio in their catalogues, and therefore, the piston rod can be selected directly from the catalogue, i.e., without prior calculation (in the Bosch Rexroth catalogue in Fig. 10.2, the piston area is marked $A_1$ and the inter-circular area is marked $A_3$).

The inter-circular area on the piston rod side of the hydraulic cylinder $A_2 = 0.00401$ m$^2$.

## Hydraulic cylinder
## Tie rod design

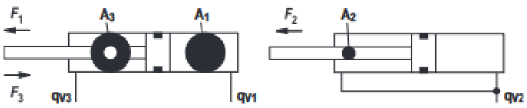
## Series CDT3...Z



- Nominal pressure 160 bar
- Maximum operating pressure up to 210 bar
- Component series 3X

**Areas, forces, flow** (for operating pressure up to 210 bar)

| Piston ØAL in mm | Piston rod ØMM in mm 160 bar | Piston rod ØMM in mm 210 bar | Area ratio $\phi$ A1/A3 | Areas: Piston $A_1$ in cm² | Areas: Rod $A_2$ in cm² | Areas: Ring $A_3$ in cm² | Force [1] Pressure $F_1$ kN 160 bar | Force [1] Pressure $F_1$ kN 210 bar | Force [1] Diff. $F_2$ kN 160 bar | Force [1] Diff. $F_2$ kN 210 bar | Force [1] Pulling $F_3$ kN 160 bar | Force [1] Pulling $F_3$ kN 210 bar | Flow at 0.1 m/s [2] OFF $q_{V1}$ l/min | Flow at 0.1 m/s [2] Diff. $q_{V2}$ l/min | Flow at 0.1 m/s [2] ON $q_{V3}$ l/min | max. available stroke length in mm [4] |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 25 | 12 | - | 1.3 | 4.9 | 1.1 | 3.8 | 7.9 | - | 1.8 | - | 6.1 | - | 2.9 | 0.7 | 2.3 | 600 |
| | 18 | 18 | 2.1 | | 2.5 | 2.4 | | 10.3 | 4.1 | 5.3 | 3.8 | 5.0 | | 1.5 | 1.4 | |
| 32 | 14 | - | 1.3 | 8.0 | 1.5 | 6.5 | 12.9 | - | 2.5 | - | 40.4 | | 4.8 | 0.9 | 3.9 | 800 |
| | 22 | 22 | 1.9 | | 3.8 | 4.2 | | 16.9 | 6.1 | 8.0 | 6.8 | 8.9 | | 2.3 | 2.5 | |
| 40 | 18 | - | 1.3 | 12.6 | 2.5 | 10.0 | 20.1 | - | 4.1 | - | 16.0 | - | 7.5 | 1.5 | 6.0 | 1000 |
| | 22 | 22 | 1.4 | | 3.8 | 8.8 | | 26.4 | 6.1 | 8.0 | 14.0 | 18.4 | | 2.3 | 5.3 | |
| | 28 | 28 | 2.0 | | 6.2 | 6.4 | | 26.4 | 9.9 | 12.9 | 10.2 | 13.4 | | 3.7 | 3.8 | |
| 50 | 22 | - | 1.3 | 19.6 | 3.8 | 15.8 | 31.4 | - | 6.1 | - | 25.3 | - | 11.8 | 2.3 | 9.5 | 1200 |
| | 28 | 28 | 1.5 | | 6.2 | 13.5 | | 41.2 | 9.9 | 12.9 | 21.6 | 28.3 | | 3.7 | 8.1 | |
| | 36 | 36 | 2.1 | | 10.2 | 9.5 | | 41.2 | 16.3 | 21.4 | 15.1 | 19.9 | | 6.1 | 5.7 | |
| 63 | 28 | - | 1.3 | 31.2 | 6.2 | 25.0 | 49.9 | - | 9.9 | 12.9 | 40.0 | - | 18.7 | 3.7 | 15.0 | 1400 |
| | 36 | 36 | 1.5 | | 10.2 | 21.0 | | 65.5 | 16.3 | 21.4 | 33.6 | 44.1 | | 6.1 | 12.6 | |
| | 45 | 45 | 2.1 | | 15.9 | 15.3 | | 65.5 | 25.4 | 33.4 | 24.4 | 32.1 | | 9.5 | 9.2 | |
| 80 | 36 | - | 1.3 | 50.3 | 10.2 | 40.1 | 80.4 | - | 16.3 | 21.4 | 64.1 | - | 30.2 | 6.1 | 24.0 | 1700 |
| | 45 | 45 | 1.5 | | 15.9 | 34.4 | | 105.6 | 25.4 | 33.4 | 55.0 | 72.2 | | 9.5 | 20.6 | |
| | 56 | 56 | 2.0 | | 24.6 | 25.6 | | 105.6 | 39.4 | 51.7 | 41.0 | 53.8 | | 14.8 | 15.4 | |
| 100 | 45 | - | 1.3 | 78.5 | 15.9 | 62.6 | 125.7 | - | 25.4 | 33.4 | 100.2 | - | 47.1 | 9.5 | 37.6 | 2000 |
| | 56 | 56 | 1.5 | | 24.6 | 53.9 | | 164.9 | 39.4 | 51.7 | 86.3 | 113.2 | | 14.8 | 32.3 | |
| | 70 | 70 | 2.0 | | 38.5 | 40.1 | | 164.9 | 61.6 | 80.8 | 64.1 | 84.1 | | 23.1 | 24.0 | |
| 125 | 56 | - | 1.3 | 122.7 | 24.6 | 98.1 | 196.4 | - | 39.4 | 51.7 | 156.9 | - | 73.6 | 14.8 | 58.9 | 2300 |
| | 70 | 70 [3] | 1.5 | | 38.5 | 84.2 | | [3] | 61.6 | 80.8 | 134.8 | [3] | | 23.1 | 50.5 | |
| | 90 | 90 [3] | 2.1 | | 63.6 | 59.1 | | [3] | 101.8 | 133.6 | 94.6 | [3] | | 38.2 | 35.5 | |
| 160 | 70 | - | 1.3 | 201.1 | 38.5 | 162.6 | 321.7 | - | 61.6 | 80.8 | 260.1 | - | 120.6 | 23.1 | 97.5 | 2600 |
| | 90 | 90 | 1.5 | | 63.6 | 137.4 | | 422.2 | 101.8 | 133.6 | 219.9 | 288.6 | | 38.2 | 82.5 | |
| | 110 | 110 [3] | 1.9 | | 95.0 | 106.0 | | [3] | 152.1 | 199.6 | 169.7 | [3] | | 57.0 | 63.6 | |
| 200 | 90 | - | 1.3 | 314.2 | 63.6 | 250.5 | 502.7 | - | 101.8 | 133.6 | 400.9 | - | 188.5 | 38.2 | 150.3 | 3000 |
| | 110 | 110 | 1.4 | | 95.0 | 219.1 | | 659.7 | 152.1 | 199.6 | 350.6 | 460.2 | | 57.0 | 131.5 | |
| | 140 | 140 [3] | 2.0 | | 153.9 | 160.2 | | [3] | 246.3 | 323.3 | 256.4 | [3] | | 92.4 | 96.1 | |




1) Theoretical static cylinder force (without consideration of the efficiency and admissible load for attachment parts such as swivel heads, plates, or valves, etc.)

2) Stroke velocity

3) On request

4) Larger stroke lengths upon request

Fig. 10.2 An example of a catalogue sheet Bosch Rexroth company of the hydraulic cylinders CDT3 [34]

Therefore, the Bosch Rexroth CDT3-100/70/stroke… the hydraulic motor is selected (the type designation of the hydraulic cylinder contains several other characters that specify other specifics, such as the cylinder mounting to the construction, the thread for connecting the liquid line, the end of the piston rod, the damping method, the type and material of the seal, etc. – but we will not specify this; we will focus only on the basic calculations).

**Design of the hydraulic pump**

Next, the selection of the hydraulic pump is performed. To do this, the speed of the electric motor drive must be known, so its selection is made first.

Calculation of required (theoretical) volumetric flow $Q_{Gt}$:

$$Q_{Gt} = v_1 \cdot A_1 = v_1 \cdot \frac{\pi \cdot D^2}{4} = 0.1 \cdot \frac{\pi \cdot 0.1^2}{4} = 7.85 \cdot 10^{-4}\, m^3 \cdot s^{-1}\,, \tag{10.8}$$

$$Q_{Gt} = 47{,}1\, dm^3 \cdot min^{-1}$$

where $Q_{Gt}$ [$m^3 \cdot s^{-1}$] is required (theoretical) volumetric flow of the hydraulic pump.

Hydraulic power $P_h$ can be calculated:

$$P_h = p_{RV} \cdot Q_{Gt} = 16 \cdot 10^6 \cdot 7.85 \cdot 10^{-4} = 12\,560\, W = 12.56\, kW, \tag{10.9}$$

where $P_h$ [W] is hydraulic power.

An electric motor 15 kW 2LC160L04, $n$ = 1 475 $min^{-1}$ from the company VYBO Electric is chosen [35].

Calculation of theoretical geometric stroke volume $V_{Gt}$ of the hydraulic pump:

$$Q_{Gt} = V_{gGt} \cdot n \implies V_{gGt} = \frac{Q_{Gt}}{\frac{n}{60}} = \frac{7.85 \cdot 10^{-4}}{\frac{1475}{60}} = 31.9 \cdot 10^{-6}\, m^3, \tag{10.10}$$

$$V_{gGt} = 31.9\, cm^3\,,$$

where $V_{gGt}$ [$m^3$] is theoretical geometric stroke volume of the hydraulic pump.

The axial piston hydraulic pump in bent axis design from the company Bosch Rexroth type A2FO-32 with the catalogue geometric stroke volume $V_{gG} = 32 \cdot 10^{-6}$ $m^3$ is selected.

# Axial piston fixed pump AA2FO

RA-A 91401/07.2014 1/32
Replaces: 03.08

Data sheet

Series 6
Sizes — Nominal pressure/Maximum pressure
10 to 180 — 5800/6500 psi (400/450 bar)
250 — 5100/5800 psi (350/400 bar)
Open circuits

## Technical data

**Table of values** (theoretical values, without efficiency and tolerances; values rounded)

| Size | | NG | | 10 | 12 | 16 | 23 | 28 | 32 | 45 | 56 |
|---|---|---|---|---|---|---|---|---|---|---|---|
| Displacement geometric, per revolution | | $V_g$ | in³ | 0.63 | 0.73 | 0.98 | 1.40 | 1.71 | 1.95 | 2.78 | 3.42 |
| | | | cm³ | 10.3 | 12 | 16 | 22.9 | 28.1 | 32 | 45.6 | 56.1 |
| Speed maximum[1)] | | $n_{nom}$ | rpm | 3150 | 3150 | 3150 | 2500 | 2500 | 2500 | 2240 | 2000 |
| | | $n_{max}$[2)] | rpm | 6000 | 6000 | 6000 | 4750 | 4750 | 4750 | 4250 | 3750 |
| Flow at $n_{nom}$ | | $q_V$ | gpm | 8.6 | 10.0 | 13.2 | 15.1 | 18.5 | 21.1 | 27.0 | 29.6 |
| | | | L/min | 32 | 38 | 50 | 57 | 70 | 80 | 102 | 112 |
| Power at | $\Delta p = 5100$ psi | P | HP | 25 | 30 | 39 | 44 | 55 | 63 | 80 | 88 |
| | $\Delta p = 350$ bar | P | kW | 19 | 22 | 29 | 33 | 41 | 47 | 60 | 65 |
| | $\Delta p = 5800$ psi | P | HP | 30 | 34 | 45 | 51 | 63 | 71 | 91 | 100 |
| | $\Delta p = 400$ bar | P | kW | 22 | 25 | 34 | 38 | 47 | 53 | 68 | 75 |
| Torque[3)] at $V_g$ and | $\Delta p = 5100$ psi | T | lb-ft | 42 | 50 | 65 | 94 | 116 | 132 | 189 | 232 |
| | $\Delta p = 350$ bar | T | Nm | 57 | 67 | 89 | 128 | 157 | 178 | 254 | 313 |
| | $\Delta p = 5800$ psi | T | lb-ft | 48 | 56 | 75 | 107 | 131 | 150 | 214 | 263 |
| | $\Delta p = 400$ bar | T | Nm | 66 | 76 | 102 | 146 | 179 | 204 | 290 | 357 |
| Rotary stiffness | | c | kNm/rad | 0.92 | 1.25 | 1.59 | 2.56 | 2.93 | 3.12 | 4.18 | 5.94 |
| Moment of inertia for rotary group | | $J_{GR}$ | lbs-ft² | 0.0095 | 0.0095 | 0.0095 | 0.0285 | 0.0285 | 0.0285 | 0.0569 | 0.0997 |
| | | | kgm² | 0.0004 | 0.0004 | 0.0004 | 0.0012 | 0.0012 | 0.0012 | 0.0024 | 0.0042 |
| Maximum angular acceleration | | $\alpha$ | rad/s² | 5000 | 5000 | 5000 | 6500 | 6500 | 6500 | 14600 | 7500 |
| Case volume | | V | gal | 0.045 | 0.045 | 0.045 | 0.053 | 0.053 | 0.053 | 0.087 | 0.119 |
| | | | L | 0.17 | 0.17 | 0.17 | 0.20 | 0.20 | 0.20 | 0.33 | 0.45 |
| Mass (approx.) | | m | lbs | 12 | 12 | 12 | 21 | 21 | 21 | 30 | 40 |
| | | | kg | 6 | 6 | 6 | 9.5 | 9.5 | 9.5 | 13.5 | 18 |

Fig. 10.3 An example of a catalogue sheet for the A2FO hydraulic pump of Bosch Rexroth company [36]

Note: If the geometric stroke volume is, e.g., 30 cm$^3$, the choice of the hydraulic pump depends on the specific situation. If the size 28 cm$^3$ is selected, the cylinder will move slower, if the size 32 cm$^3$ is selected, on the contrary, it will move faster. In this case, however, the system could be supplemented with velocity control, e.g., by a throttle valve. Another option is to choose a variable displacement hydraulic pump and set the desired geometric stroke volume.

However, even in the chosen case, the velocity will be slightly lower because the volumetric efficiency $\eta_{QG}$ of the hydraulic pump must be included in the flow calculation. In the case of the axial piston hydraulic pump, the volumetric efficiency is approximately $\eta_{QG} = 0.95$.

Calculation of real volumetric flow $Q_{Gr}$ of the hydraulic pump:

$$Q_{Gr} = V_{gG} \cdot n \cdot \eta_{QG} = 32 \cdot 10^{-6} \cdot \frac{1\,475}{60} \cdot 0.95 = 7.47 \cdot 10^{-4}\, m^3 \cdot s^{-1}, \tag{10.11}$$

$$Q_{Gr} = 44.84\, dm^3 \cdot min^{-1} ,$$

where $Q_{Gr}$ [$m^3 \cdot s^{-1}$] is the real volumetric flow of the hydraulic pump and $\eta_{QG}$ [-] is volumetric efficiency of the hydraulic pump.

The real extension velocity $v_{1r}$ of the piston rod:

$$v_{1r} = \frac{Q_{Gr}}{A_1} = \frac{Q_{Gr}}{\frac{\pi \cdot D^2}{4}} = \frac{7.47 \cdot 10^{-4}}{\frac{\pi \cdot 0.1^2}{4}} = 0.095\, m \cdot s^{-1} , \tag{10.12}$$

where $v_{1r}$ [$m \cdot s^{-1}$] is the real extension velocity of the piston rod.

The real retraction velocity $v_{2r}$ of the piston rod:

$$v_{2r} = \frac{Q_{Gr}}{A_2} = \frac{Q_{Gr}}{\frac{\pi \cdot (D^2 - d^2)}{4}} = \frac{7.47 \cdot 10^{-4}}{\frac{\pi \cdot (0.1^2 - 0.07^2)}{4}} = 0.186\, m \cdot s^{-1} , \tag{10.13}$$

where $v_{2r}$ [$m \cdot s^{-1}$] is the real retraction velocity of the piston rod.

**Design of pipes**

After choosing both converters (i.e., the hydraulic motor and the hydraulic pump), it is necessary to proceed to the selection of control elements and pipes. Firstly, the diameters of pipes or hoses are chosen and the pressure loss during the flow of the working fluid is calculated.

The choice of the pipe diameter is based on the recommended range of liquid flow velocities in various parts of the hydraulic mechanism Tab 10.3.

Tab 10.3 Recommended values of liquid flow velocity

| Pipe type | Recommended range of liquid flow velocities [$m \cdot s^{-1}$] | Selected velocity [$m \cdot s^{-1}$] |
|---|---|---|
| pressure | $v_p = (4 \div 10)$ | $v_p = 5$ |
| low-pressure (return line) | $v_l = (1{,}5 \div 4)$ | $v_l = 2{,}5$ |
| suction | $v_s = (0{,}5 \div 1)$ | $v_s = 0{,}5$ |

Calculation of the pressure pipe

Pressure pipe is pipe 1 from the hydraulic pump to the directional valve, pipes 2 and 3 between the hydraulic cylinder and the directional valve.

The theoretical cross-section area $A_{pt}$ of pressure pipe:

$$A_{pt} = \frac{Q_{Gr}}{v_p} = \frac{7.47 \cdot 10^{-4}}{5} = 1.494 \cdot 10^{-4}\, m^2\,, \tag{10.14}$$

where $A_{pt}$ [m$^2$] is the theoretical cross-section area of the pressure pipe and $v_p$ [m · s$^{-1}$] is the selected velocity of liquid in the pressure pipe.

Required (theoretical) diameter $d_{pt}$ of the pressure pipe:

$$d_{pt} = \sqrt{\frac{4 \cdot A_{pt}}{\pi}} = \sqrt{\frac{4 \cdot 1.494 \cdot 10^{-4}}{\pi}} = 0.0138\, m = 13.8\, mm\,, \tag{10.15}$$

where $d_{pt}$ [m] is theoretical diameter of the pressure pipe.

Similar to the piston and piston rod diameters, there are prescribed dimensions for pipes too:

Tab 10.4 Nominal size (diameter) of the pipe

| Nominal size [mm] |
|---|
| 4, 6, 8, 10, 13, 16, 20, 25, 32, 40, 50, 63, 80, 100, 125, 160 |

Hose diameters may vary slightly and should be looked up in catalogues. From the above-mentioned diameters, a diameter $d_p$ = 13 mm is chosen for the pressure pipes.

<u>Calculation of the low-pressure pipe</u>

Low-pressure pipe 4 is from the directional valve back to the tank. The theoretical cross-section area $A_{lt}$ of low-pressure pipe:

$$A_{lt} = \frac{Q_{Gr}}{v_l} = \frac{7.47 \cdot 10^{-4}}{2.5} = 0.0003\, m^2\,, \tag{10.16}$$

where $A_{lt}$ [m$^2$] is the theoretical cross-section area of the low-pressure pipe, and $v_p$ [m · s$^{-1}$] is the selected velocity of liquid in the low-pressure pipe.

Required (theoretical) diameter $d_{lt}$ of the low-pressure pipe:

$$d_{lt} = \sqrt{\frac{4 \cdot A_{lt}}{\pi}} = \sqrt{\frac{4 \cdot 3 \cdot 10^{-4}}{\pi}} = 0.0195\, m = 19.5\, mm\,, \tag{10.17}$$

where $d_{lt}$ [m] is the theoretical diameter of the low-pressure pipe.

From Tab 10.4, the low-pressure pipe with a diameter $d_l$ = 20 mm is chosen.

After choosing the pipe diameters, it is possible to calculate the pressure losses. For this purpose, it is necessary to know the properties of the liquid used. The choice of the working liquid depends on many factors, such as working temperature, fire risk, ecological requirements, etc. We will assume that the mechanism works indoors at around 20 °C, and there are no special liquid requirements. From Tab 10.5, mineral oil of viscosity class VG 32 is chosen. The density of oil is $\rho$ = 865 kg · m$^{-3}$, and kinematic viscosity is $\nu$ = 32 mm$^2$ · s$^{-1}$.

Tab 10.5 Basic parameters of mineral oils of HV group

| **Parameters** | **Oil type** | | |
|---|---|---|---|
| | **HV 32** | **HV 46** | **HV 68** |
| Density at 20 °C [kg · m$^{-3}$] | 865 | 855 | 870 |
| Kinematic viscosity at 40 °C [mm$^2$ · s$^{-1}$] | 32 | 46 | 68 |
| Viscosity index [-] | 170 | 170 | 170 |
| Flash point [°C] | 195 | 230 | 230 |
| Fluidity point [°C] | -39 | -36 | -33 |

<u>Pressure losses in lines</u>

All pressure losses are calculated for the working motion, which is the piston rod extension in this case. In our case, pipes 1 and 2 are identical (i.e., the same diameters and lengths), the same amount of oil will flow through these pipes, and therefore, the pressure loss will be the same. Firstly, it is necessary to determine the real flow velocity $v_{p1}$ and $v_{p2}$ in pressure pipes 1 and 2:

$$v_{p1} = v_{p2} = \frac{Q_{Gr}}{A_p} = \frac{Q_{Gr}}{\frac{\pi \cdot d_p^2}{4}} = \frac{7.47 \cdot 10^{-4}}{\frac{\pi \cdot 0.013^2}{4}} = 5.63\ m \cdot s^{-1}\ , \tag{10.18}$$

where $v_{p1}$ [m · s$^{-1}$] is the real flow velocity in pressure pipe 1, $v_{t2}$ [m · s$^{-1}$] is the real flow velocity in pressure pipe 2, and $A_p$ [m$^2$] is the real cross-section area of the pressure pipe.

The real flow velocity in the pressure pipe is in line with a recommended range from Tab 10.3.

Calculation of Reynolds number $Re_1$ and $Re_2$ in pressure pipes 1 and 2:

$$\begin{gathered} Re_1 = Re_2 = \frac{v_{p1} \cdot d_p}{\nu} = \frac{5.63\ \cdot 0.013}{32 \cdot 10^{-6}} = 2287\ , \\ Re_1 < Re_{crit}\ , \\ 2287 < 2320 \rightarrow \text{laminar flow}\ , \end{gathered} \tag{10.19}$$

where $Re_1$ [-] is Reynolds number in pressure pipe 1, $Re_2$ [-] Reynolds number in pressure pipe 2, $Re_{crit}$ [-] critical Reynolds number, and $\nu$ [mm$^2$ · s$^{-1}$] is kinematic viscosity of oil.

Calculation of friction coefficient $\lambda_1$ and $\lambda_2$ for laminar flow in pressure pipes 1 and 2:

$$\lambda_1 = \lambda_2 = \frac{64}{Re} = \frac{64}{2287} = 0.028\,, \tag{10.20}$$

where $\lambda_1$ [-] is friction coefficient in pressure pipe 1, and $\lambda_2$ [-] is friction coefficient in pressure pipe 2.

Calculation of pressure loss $\Delta p_{l1}$ and $\Delta p_{l2}$ in pressure pipes 1 and 2:

$$\Delta p_{l1} = \Delta p_{l2} = \lambda_1 \cdot \frac{L_1}{d_p} \cdot \rho \cdot \frac{v_{p1}^2}{2} = 0.028 \cdot \frac{2}{0.013} \cdot 865 \cdot \frac{5.63^2}{2} = 59053\,Pa\,, \tag{10.21}$$

$$\Delta p_{l1} = \Delta p_{l2} = 0.59\,bar$$

where $\Delta p_{l1}$ [Pa] is pressure loss in pipe 1, $\Delta p_{l2}$ [Pa] pressure loss in pipe 2, $L_1$ [m] is pipe length, and $\rho$ [$kg \cdot m^{-3}$] is the density of oil.

Oil with the volumetric flow $Q_3$ flows through pressure pipe 3 from the hydraulic motor (i.e., from the piston rod side) to the directional valve. This flow is determined from the area ratio on the piston:

$$Q_3 = Q_{Gr} \cdot \frac{A_2}{A_1} = 7.47 \cdot 10^{-4} \cdot \frac{4.01 \cdot 10^{-3}}{7.85 \cdot 10^{-3}} = 3.8 \cdot 10^{-4}\,m^3 \cdot s^{-1}\,, \tag{10.22}$$

$$Q_3 = 22.8\,dm^3 \cdot min^{-1}\,,$$

where $Q_3$ [$m^3 \cdot s^{-1}$] is volumetric flow in pressure pipe 3, and $A_2$ [$m^2$] is the inter-circular area on the piston rod side of the hydraulic cylinder.

The real flow velocity $v_{p3}$ in pressure pipe 3:

$$v_{p3} = \frac{Q_3}{A_p} = \frac{Q_3}{\frac{\pi \cdot d_p^2}{4}} = \frac{3.8 \cdot 10^{-4}}{\frac{\pi \cdot 0.013^2}{4}} = 2.86\,m \cdot s^{-1}\,, \tag{10.23}$$

where $v_{p3}$ [$m \cdot s^{-1}$] is the real flow velocity in pressure pipe 3.

The real flow velocity in pressure pipe 3 is in line with the recommended range from Tab 10.3.

Calculation of the Reynolds number $Re_3$ in pressure pipe 3:

$$Re_3 = \frac{v_{p3} \cdot d_p}{\nu} = \frac{2.86 \cdot 0.013}{32 \cdot 10^{-6}} = 1162\,,$$

$$Re_1 < Re_{crit}\,, \tag{10.24}$$

$$1162 < 2320 \rightarrow \text{laminar flow}\,,$$

where $Re_3$ [-] is the Reynolds number in pressure pipe 3.

Calculation of friction coefficient $\lambda_3$ for laminar flow in pressure pipe 3:

$$\lambda_3 = \frac{64}{Re} = \frac{64}{1162} = 0.055\,, \tag{10.25}$$

where $\lambda_3$ [-] is friction coefficient in pressure pipe 3.

Calculation of pressure loss $\Delta p_{l3}$ in pressure pipe 3:

$$\Delta p_{l3} = \lambda_3 \cdot \frac{L_3}{d_p} \cdot \rho \cdot \frac{v_{p3}^2}{2} = 0.055 \cdot \frac{2}{0.013} \cdot 865 \cdot \frac{2.86^2}{2} = 29\,934\ Pa\,, \tag{10.26}$$

$$\Delta p_{l3} = 0.3\ bar$$

where $\Delta p_{l3}$ [Pa] is pressure loss in pipe 3, and $L_3$ [m] is the pipe length.

The same volumetric flow as through pipe 3 also flows through low-pressure (return) pipe 4, which has a different diameter. The real flow velocity $v_{l4}$ in low-pressure pipe 4:

$$v_{l4} = \frac{Q_3}{A_l} = \frac{Q_3}{\frac{\pi \cdot d_l^2}{4}} = \frac{3.8 \cdot 10^{-4}}{\frac{\pi \cdot 0.02^2}{4}} = 1.21\ m \cdot s^{-1}\,, \tag{10.27}$$

where $v_{l4}$ [m · s$^{-1}$] is the real flow velocity in low-pressure pipe 4, $A_l$ [m$^2$] real cross-section area of low-pressure pipe, and $d_l$ [m] is low-pressure pipe diameter.

Calculation of the Reynolds number $Re_4$ in low-pressure pipe 4:

$$Re_4 = \frac{v_{l4} \cdot d_l}{\nu} = \frac{1.21 \cdot 0.02}{32 \cdot 10^{-6}} = 756\,,$$
$$Re_4 < Re_{crit}\,, \tag{10.28}$$
$$756 < 2320 \rightarrow \text{laminar flow}\,,$$

where $Re_4$ [-] is the Reynolds number in low-pressure pipe 4.

Calculation of the friction coefficient $\lambda_4$ for laminar flow in low-pressure pipe 4:

$$\lambda_4 = \frac{64}{Re} = \frac{64}{756} = 0.085\,, \tag{10.29}$$

where $\lambda_4$ [-] is friction coefficient in low-pressure pipe 4.

Calculation of pressure loss $\Delta p_{l4}$ in low-pressure pipe 4:

$$\Delta p_{l4} = \lambda_4 \cdot \frac{L_4}{d_l} \cdot \rho \cdot \frac{v_{l4}^2}{2} = 0.085 \cdot \frac{2}{0.02} \cdot 865 \cdot \frac{1.21^2}{2} = 5\,382\ Pa\,, \tag{10.30}$$

$$\Delta p_{l4} = 0.06\ bar$$

where $\Delta p_{l4}$ [Pa] is pressure loss in pipe 4 and $L_4$ [m] is pipe length.

**Choice of control elements**

The next step is to select and check the control elements of the circuit, i.e., the directional valve, the check valve, the relief valve and, of course, the filter. Hydraulic elements for industrial use are usually designed in such a way that they do not have a thread in the body for connecting hoses or pipes, and it is therefore necessary to seat and screw these elements to the connection plate, which contains the threads. Then it is possible to connect the individual elements. However, there is another option, which is to assemble a valve block from the elements, seated on one common connection plate, as shown in Fig. 10.4.

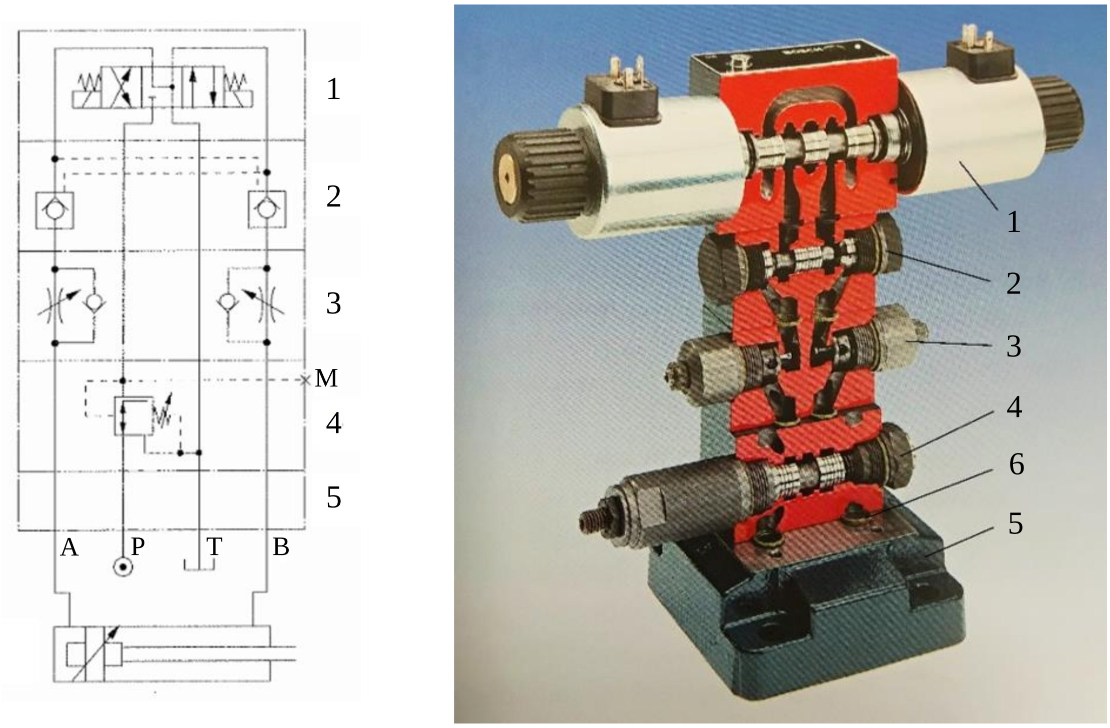



Fig. 10.4 Valve block from the elements, seated on one common connection plate Bosch Rexroth, hydraulic scheme (left), real elements in cut (right) [19]

*1 – directional valve, 2 – double pilot operated check valve, 3 – throttle and check valves, 4 – relief valve, 5 – base plate, 6 – O-ring*

From the figure, it can be seen that between directional valve 1 and base plate 5, it is possible to insert elements that are placed in front of the directional valve (between the hydraulic pump and the directional valve), i.e., the relief valve 4. It is also possible to mount elements in the block that are connected to outputs from the directional valve. In this case, it is double pilot operated check valve 2 and throttle check valves 3.

In our case, it is necessary to choose three elements in the pressure line – the directional, check and return valves. The control elements are chosen, for example, from the Argo-Hytos company.

<u>Directional valve</u>

For the selection of the directional valve and all other elements, two basic data are required, namely the maximum pressure in the circuit (i.e., 16 MPa) and the volumetric flow. The real volumetric flow of the hydraulic pump (in pressure lines 1 and 2) was calculated in the equation (10.11) and is $Q_{Gr}$ = 44,84 dm$^3 \cdot$ min$^{-1}$.

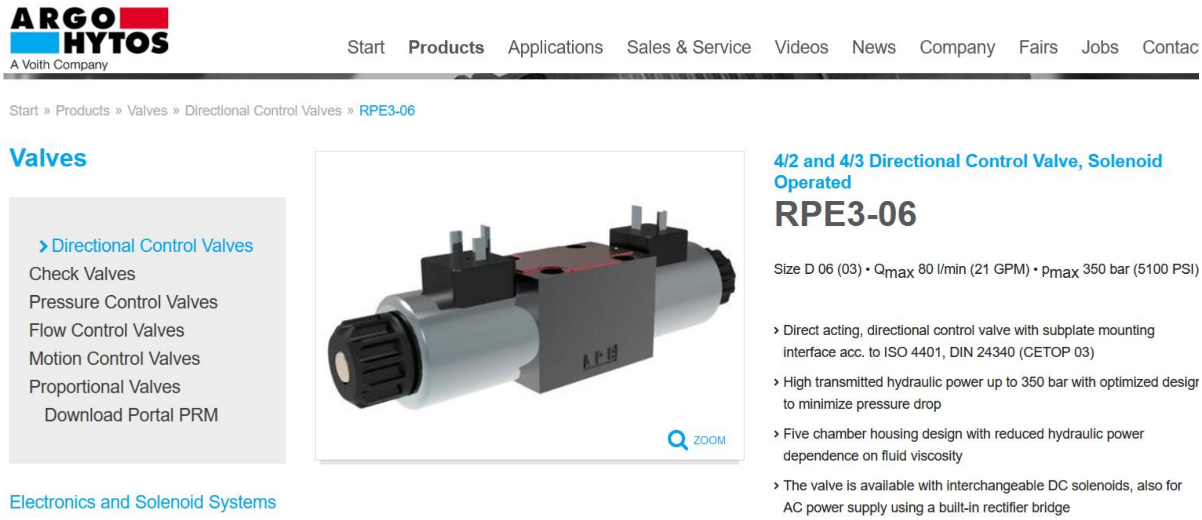



Fig. 10.5 A view of the front page of the RPE3-06 directional valve from Argo-Hytos company [37]

The directional valve type RPE3-06 is preliminarily selected, which is suitable for pressure up to 35 MPa and the maximum volumetric flow of 80 $dm^3 \cdot min^{-1}$. The diameter (i.e., the diameter of connecting channels) is 6 mm. However, this is not enough, and the choice must be checked from the power point of view, and then the pressure loss must be determined.

Firstly, it is necessary to choose the right spool type. According to the hydraulic scheme shown in Fig. 10.1, port P should be connected to port T in the middle position. Ports A and B should be closed. This corresponds to the spool marked C11 in Fig. 10.6.

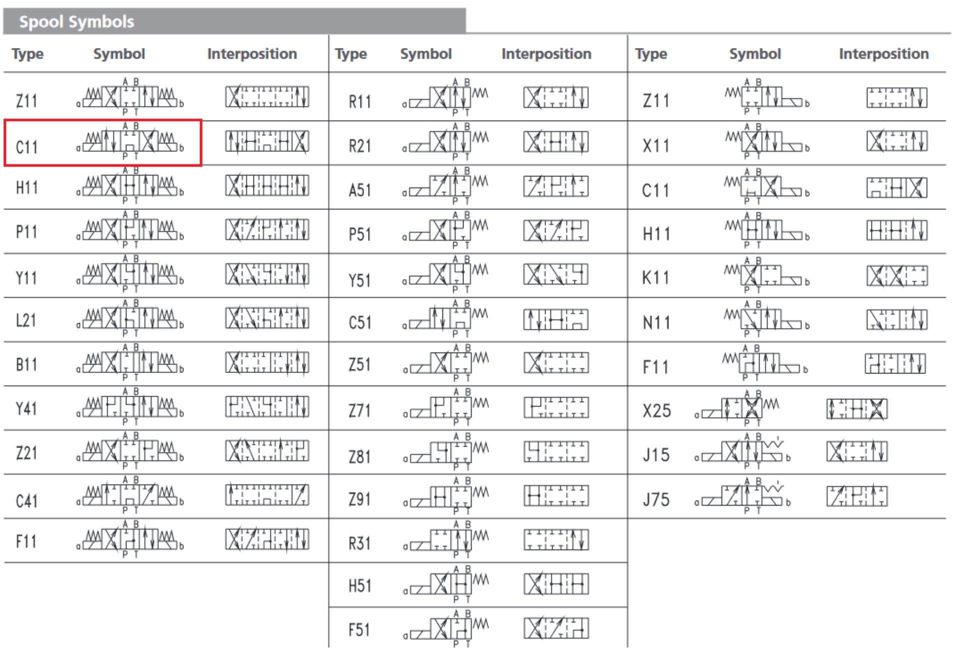
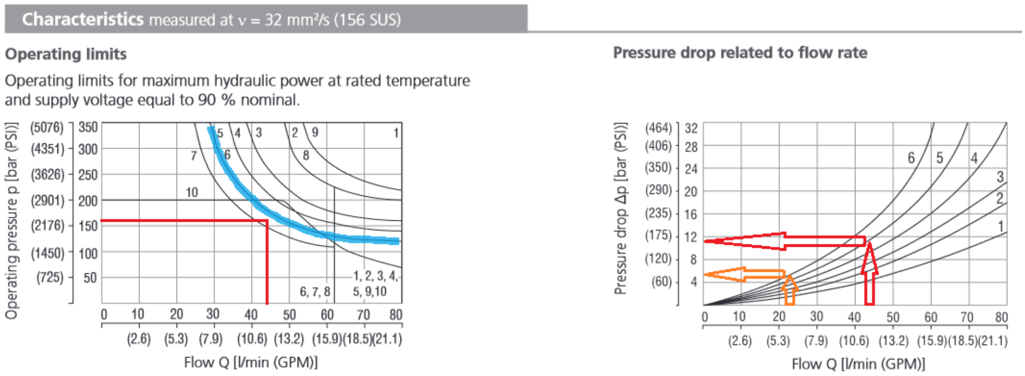



Fig. 10.6 Possible types of spools (function of directional valves) of the RPE3-06 directional valve from Argo-Hytos company [37]

Now it is possible to check whether the directional valve is suitable from the power point of view. The intersection of the maximum pressure and the volumetric flow must lie at the bottom left of the characteristic (in this case, characteristic 6 - marked in blue) of the given spool type,

Fig. 10.7 (left). If the intersection is above this characteristic, it would be necessary to select a directional valve of a larger size (note: the characteristics express the limiting possibilities of the electromagnets for the spool repositioning).




| Spool symbol | | | | | |
|---|---|---|---|---|---|
| 1 | Z11 | 5 | F11 | 7 | Z91 |
| 6 | C11 | 3 | R11 | 5 | R31 |
| 5 | H11 | 4 | R21 | 5 | H51 |
| 1 | P11 | 5 | A51 | 7 | F51 |
| 2 | Y11 | 1 | P51 | 3 | X11 |
| 5 | L21 | 2 | Y51 | 7 | K11 |
| 8 | B11 | 6 | C51 | 7 | N11 |
| 6 | Y41 | 1 | Z51 | 10 | X25 |
| 1 | Z21 | 7 | Z71 | 1 | J15 |
| 5 | C41 | 7 | Z81 | 9 | J75 |

| Spool symbol | P-A | P-B | A-T | B-T | P-T |
|---|---|---|---|---|---|
| Z11,L21,B11,R11 R21,X11,N11,J15 | 2 | 2 | 3 | 3 | |
| C11 | 5 | 5 | 5 | 6 | 3 |
| H11 | 2 | 2 | 2 | 3 | 3 |
| P11 | 1 | 1 | 3 | 3 | |
| Y11 | 2 | 2 | 2 | 2 | |
| Y41 | 3 | 3 | 3 | 3 | |
| Z21,Z51,H51 | | 2 | 3 | | |
| C41 | 4 | 4 | | | 5 |
| F11 | 1 | 2 | | 3 | 3 |
| A51,J75 | 2 | 2 | | | |

| | P-A | P-B | A-T | B-T | P-T |
|---|---|---|---|---|---|
| P51 | | 1 | 3 | | |
| Y51 | | 2 | 2 | | |
| C51 | 2 | | | 3 | 4 |
| Z71 | 3 | 3 | | | |
| Z81 | | | 3 | 3 | |
| Z91 | 3 | | | 3 | 3 |
| R31 | 2 | | | 3 | |
| F51 | | 2 | 3 | | |
| K11 | | 2 | 3 | | |
| X25 | 3 | 3 | 3 | | |

For operating limits under conditions and flow directions other than shown contact our technical support. Admissible operating limits may be considerably lower with only one direction of flow (A or B plugged, or without flow.)

Fig. 10.7 Power characteristics and pressure losses of the RPE3-06 directional valve from Argo-Hytos company [37]

If the directional valve is suitable from the power point of view, pressure losses can be determined in Fig. 10.7 (right). When the piston rod is extended, the ports P and A, B and T are connected in the directional valve. The pressure loss is determined from the relevant characteristics of the C11-type spool valve and the volumetric flow through this valve. When ports P and T are connected and the volumetric flow $Q_{Gr}$ = 44,84 dm$^3$ · min$^{-1}$, a pressure loss of approx. $\Delta p_{dvPA}$ = 1.2 MPa = 12 bar occurs (shown in red). By connecting ports B and T, the volumetric flow $Q_3$ = 22,8 dm$^3$ · min$^{-1}$ flows out of the hydraulic cylinder, and this corresponds to a pressure loss of approx. $\Delta p_{dvBT}$ = 0.5 MPa = 5 bar (shown in orange).

After choosing the directional valve, it is possible to choose other elements. If they will be mounted on top of each other in the valve block, it is necessary to choose the same size, which is determined by the diameter of the directional valve, in this case, 6 mm.

<u>Check valve</u>

The check valve MVJ3-06 from the Argo-Hytos company Fig. 10.8 with a diameter of 6 mm is designed for pressure of 35 MPa and volumetric flow of 50 dm$^3$ · min$^{-1}$, which is satisfactory.

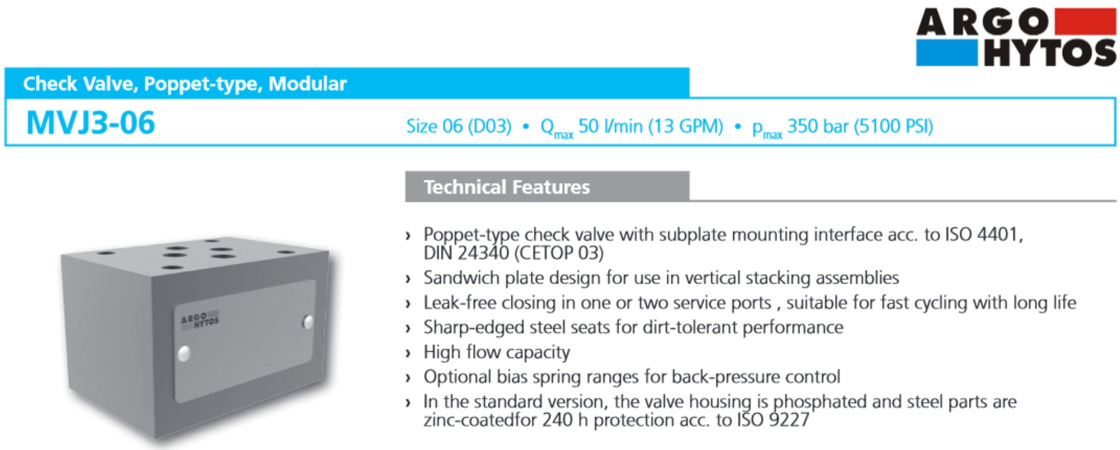



Fig. 10.8 A view of basic information MVJ3-06 check valve from Argo-Hytos company [38]

Next, the correct design (i.e., location and direction) of the check valve must be selected; see Fig. 10.9. In this case, the valve needs to be located in channel P and allow the liquid flow in the direction from channel P2 (connection plate side - inlet from the hydraulic pump) to channel P1 (valve side – directional valve).

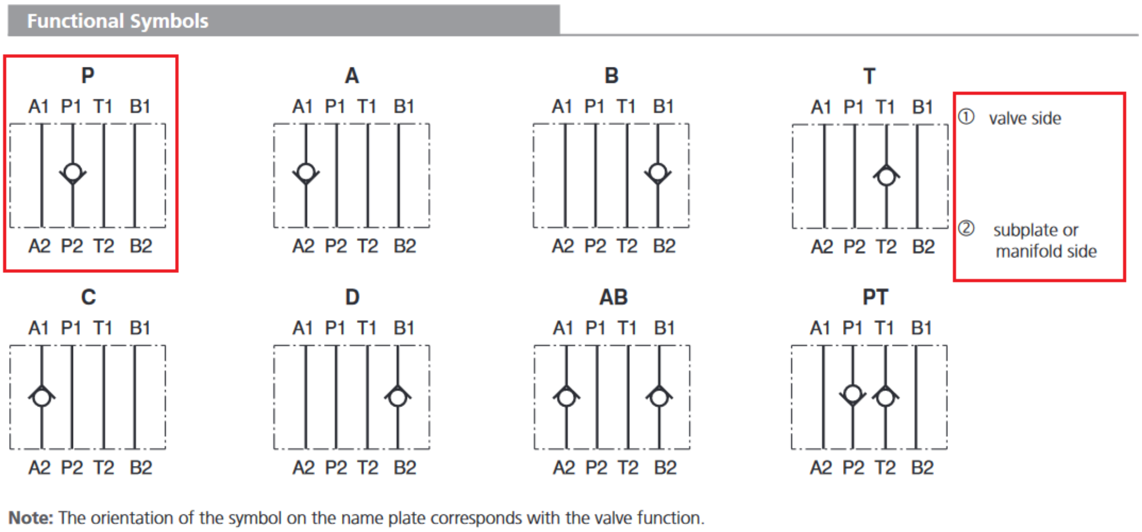



Fig. 10.9 Possible designs of MVJ3-06 check valve from Argo-Hytos company [38]

The check valve can also be ordered in a design with different opening pressures (spring preload). In our case, no preload is required. Therefore, for the variant of the valve with the smallest preload, the pressure loss is subtracted from the characteristic in Fig. 10.10. The pressure loss of the valve is therefore, approximately, $\Delta p_{CV}$ = 0.4 MPa = 4 bar.

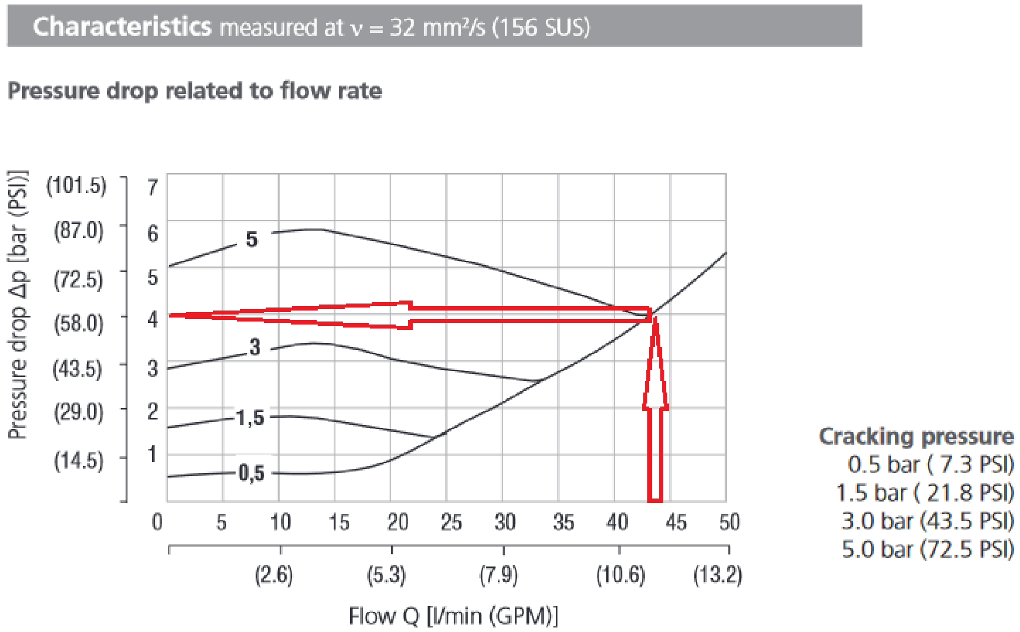



Fig. 10.10 Determination of pressure loss of MVJ3-06 check valve from Argo-Hytos company [38]

Relief valve

For the design of the relief valve, we are primarily guided by the maximum working pressure and maximum flow rate. Again, it is necessary to choose a diameter of 6 mm. The Argo-Hytos company offers, for example, relief valves of type VPP1-06 Fig. 10.11, which can be ordered in a version for mounting in a separate block, for mounting in a pipe, or with a body for mounting on a plate.

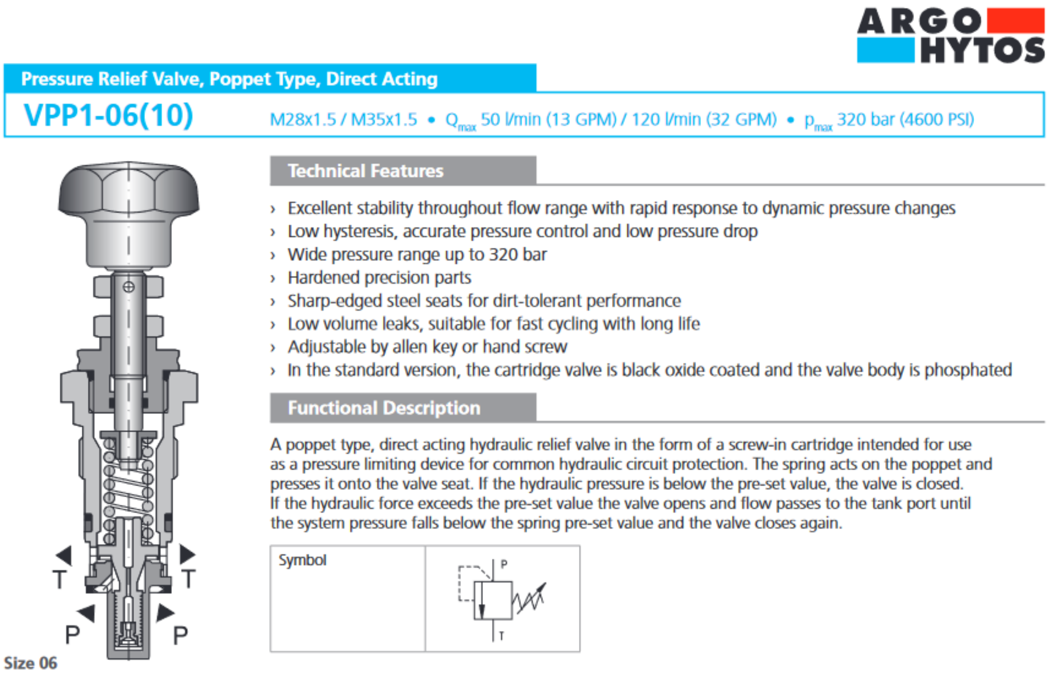



Fig. 10.11 A view of the catalogue of relief valve VPP1-06 from Argo-Hytos company [39]

Even in the case of the relief valve for mounting on a plate, it is possible to perform multiple connections of the relief valve between individual channels. An example of a relief valve catalogue from Bosch Rexroth company is shown below, see Fig. 10.12. When comparing two products of the same diameter (Argo-Hytos a Bosch Rexroth), it is possible to notice that there are differences in the size of the possible volumetric flow.


Fig. 10.12 A view of the catalogue of relief valve ZDBD from Bosch Rexroth company [40]

In our case, option “P” is selected (Fig. 10.13), where the pressure in channel P is monitored, and if the pressure rises to a higher value than the value set on the valve, it will release the liquid into the waste (return) channel T.

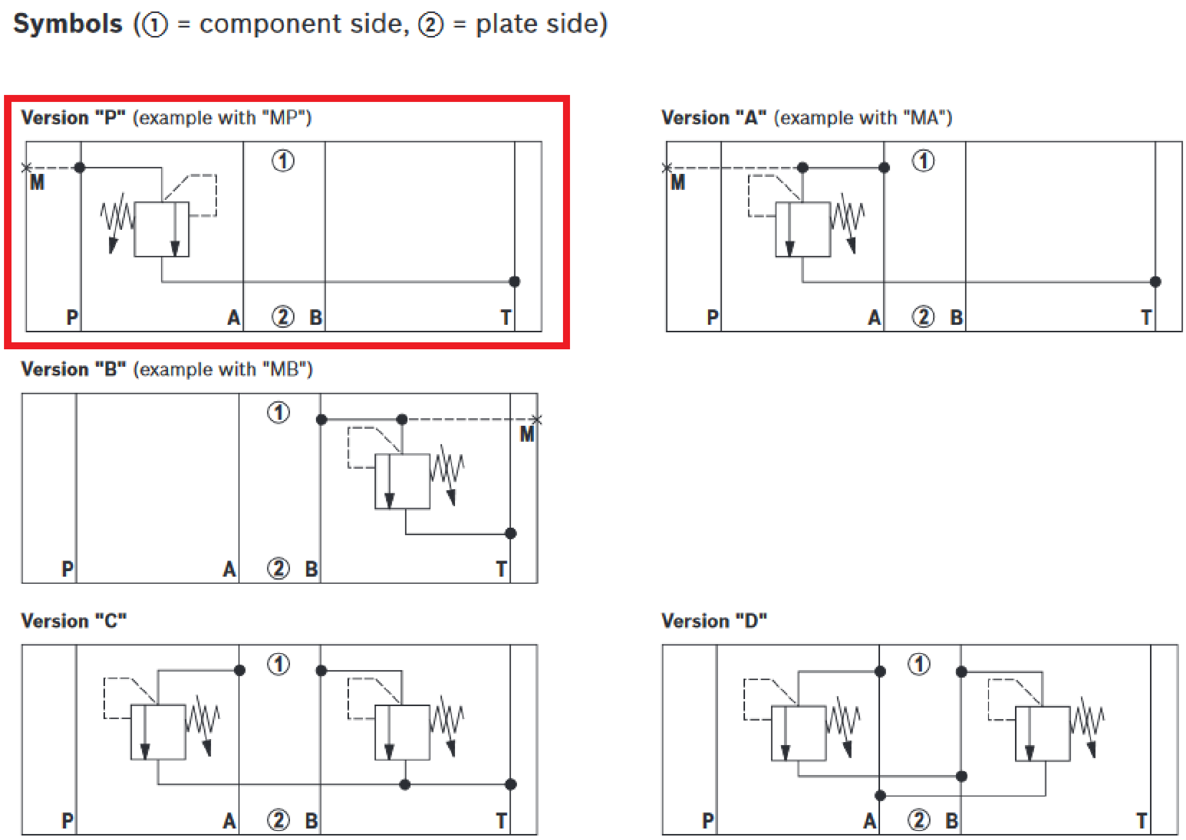



Fig. 10.13 Versions of relief valve ZDBD from Bosch Rexroth company [40]

The pressure loss will not be determined on the relief valve since the relief valve will be closed (no flow) when the piston rod is extended.

Filter

The last element that needs to be selected is the filter. According to the diagram, the filter is placed in the return line, so it is necessary to choose a so-called return or low-pressure filter. These filters can be built into the pipeline or placed (mounted in the tank cover). For example, the following option is selected: Fig. 10.14 shows a preview of the catalogue of return filters from the Argo-Hytos company. The filter must be able to pass the entire volumetric flow $Q_{Gr}$ from the hydraulic pump. Other parameters for the selection of the filter are filtration ability, filter insert material and filter type (with bypass, pressure loss measurement, pollution indication, etc.). The choice of the filter is not the purpose of this calculation, so for example the filter E 072-168 is chosen, which contains the filter insert 16EX2, through which up to 70 $dm^3 \cdot min^{-1}$ flows, and in case of clogging, the liquid flows through the bypass (check valve). The filter can also be equipped with a pressure sensor to indicate clogging. This filter is then characterized by characteristic 3 (Fig. 10.15), from which the pressure loss is approximately $\Delta p_F$ = 0.4 MPa = 4 bar.

ARGO HYTOS

Return Filters

## E 043 · E 072

Tank top mounting · Connection up to G¾ / -12 SAE · Nominal flow rate up to 70 l/min / 18.5 gpm

### Description

**Application**

In the return line circuits of hydraulic systems.

**Performance features**

*Protection against wear:*
By means of filter elements that even in full-flow filtration meet the highest demands regarding cleanliness classes.

*Protection against malfunction:*
By means of full-flow filtration in the system return, the pumps above all are protected from dirt particles remaining in the system after assembly, repairs, or which are generated by wear or enter the system from outside.

**Special features**

› By-pass valve:
The location close to the inlet port prevents dirt particles retained by the filter element from entering into the clean oil side.

› Removable bowl:
In case of maintenance the filter bowl is removed together with the filter element – therefore dirt particles are not flushed back into the tank.

› Extension pipe:
A correct extension pipe length ensures oil outlet below minimum oil level and prevents foaming.

Fig. 10.14 A view of the catalogue of low-pressure filter E 072 from Argo-Hytos company [41]

**Selection Chart**

| Part No. | Nominal flow rate | Pressure drop see diagram **D**/ curve no. | Filter fineness see Diagr. **Dx** | Dirt-holding capacity | Connection A | Cracking pressure of by-pass | Symbol | Replacement element Part No. | Weight |
|---|---|---|---|---|---|---|---|---|---|
| | l/min | | | g | | bar | | | kg |
| 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 | 10 |
| **E 043-156**[1] | 25 | **D1**/1 | 10EX2 | 6.1 | G½ | 2.5 | 2 | V3.0510-56 | 0.6 |
| **E 043-166**[1] | 25 | **D1**/1 | 10EX2 | 6.1 | G½ | 2.5 | 1 | V3.0510-56 | 0.6 |
| **E 043-158**[1] | 35 | **D1**/2 | 16EX2 | 6.1 | G½ | 2.5 | 2 | V3.0510-58 | 0.6 |
| **E 043-168**[1] | 35 | **D1**/2 | 16EX2 | 6.1 | G½ | 2.5 | 1 | V3.0510-58 | 0.6 |
| E 043-151 | 30 | **D1**/3 | 30P | 4.0 | G½ | 1.5 | 2 | P3.0510-51 | 0.6 |
| E 043-161 | 30 | **D1**/3 | 30P | 4.0 | G½ | 1.5 | 1 | P3.0510-51 | 0.6 |
| E 072-153 | 25 | **D2**/1 | 5EX2 | 7.7 | G¾ | 2.5 | 2 | V3.0520-53 | 0.8 |
| E 072-163 | 25 | **D2**/1 | 5EX2 | 7.7 | G¾ | 2.5 | 1 | V3.0520-53 | 0.8 |
| **E 072-156**[1] | 50 | **D2**/2 | 10EX2 | 13 | G¾ | 2.5 | 2 | V3.0520-56 | 0.8 |
| **E 072-166**[1] | 50 | **D2**/2 | 10EX2 | 13 | G¾ | 2.5 | 1 | V3.0520-56 | 0.8 |
| **E 072-158**[1] | 70 | **D2**/3 | 16EX2 | 13 | G¾ | 2.5 | 2 | V3.0520-58 | 0.8 |
| **E 072-168**[1] | 70 | **D2**/3 | 16EX2 | 13 | G¾ | 2.5 | 1 | V3.0520-58 | 0.8 |
| E 072-151 | 50 | **D2**/4 | 30P | 6.6 | G¾ | 1.5 | 2 | P3.0520-51[2] | 0.8 |
| E 072-161 | 50 | **D2**/4 | 30P | 6.6 | G¾ | 1.5 | 1 | P3.0520-51[2] | 0.8 |

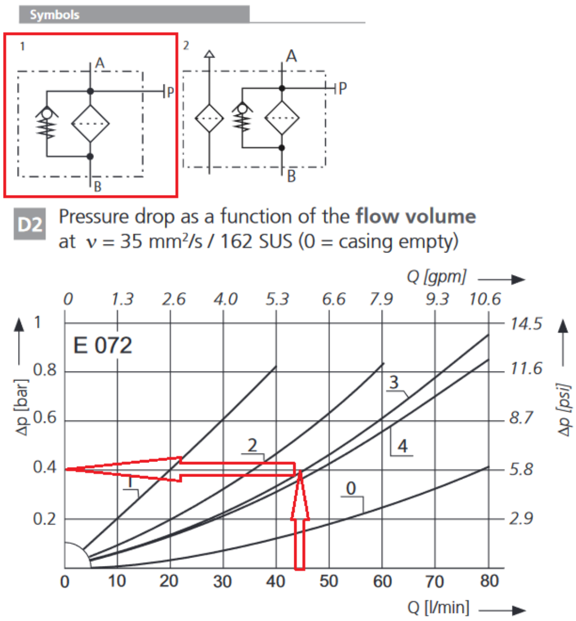



Fig. 10.15 Determination of pressure loss of low-pressure filter E 072 from Argo-Hytos company [41]

**Total pressure loss**

The total pressure loss for extending the hydraulic cylinder piston rod can be determined by adding up all partial losses in the pipe and in hydraulic elements, see Tab 10.6.

Tab 10.6 Determination of the total pressure loss for extending the piston rod

| Element | Marking | Pressure loss | |
|---|---|---|---|
| | | [MPa] | [bar] |
| Pressure pipe 1 | $\Delta p_{p1}$ | 0.059 | 0,59 |
| Pressure pipe 2 | $\Delta p_{p2}$ | 0.059 | 0,59 |
| Pressure pipe 3 | $\Delta p_{p3}$ | 0.03 | 0,3 |
| Low-pressure pipe 4 | $\Delta p_{l4}$ | 0.006 | 0.06 |
| Directional valve RPE3-06 (P → A) | $\Delta p_{dvPA}$ | 1.2 | 12 |
| Directional valve RPE3-06 (B → T) | $\Delta p_{dvBT}$ | 0.5 | 5 |
| Check valve MVJ3-06 | $\Delta p_{CV}$ | 0.4 | 4 |
| Low-pressure filter E 072-168 | $\Delta p_F$ | 0.4 | 4 |
| **Total pressure loss** | $\boldsymbol{\Delta p_T}$ | **2.654** | **26.54** |

The pressure $p_{1r}$ = 11.39 MPa required to overcome the load was calculated in the equation (10.5). After adding the total pressure losses $\Delta p_T$, the total pressure $p_T$ of the system can be calculated:

$$p_T = p_{1r} + \Delta p_T = 11.39 + 2.65 = 14.04\ MPa\,. \tag{10.31}$$

The total pressure $p_T$ must be lower compared to the pressure $p_{RV}$ adjusted on the relief valve:

$$p_T < p_{RV}\,, \qquad 14.04 < 16 \tag{10.32}$$

This condition is fulfilled with a pressure reserve of 1.96 MPa.

# 11. Basics of pneumatic mechanisms

Pneumatic mechanisms belong, together with hydraulic mechanisms, to the group of fluid mechanisms. Medium, in this case, is most often compressed air. According to the type of energy used, pneumatic mechanisms can be divided into pneumostatic mechanisms that use pressure energy and pneumodynamic mechanisms that use the kinetic energy of air. Due to the fact that pneumodynamic machines are at an absolute minimum and therefore, pneumostatic machines and mechanisms predominate, they are simply called pneumatic mechanisms.

At the beginning, it is possible to mention the following advantages and disadvantages of pneumatic mechanisms, as they are listed, e.g., in [42].

The advantages of pneumatic mechanisms are:

- clean operation,
- operation in explosive atmospheres or under water,
- operation in a wide range of temperatures,
- high velocity,
- easy velocity, force and power control by simple elements and the possibility of automation,
- easy installation and maintenance without requiring special qualifications,
- technically simple and therefore inexpensive production of elements,
- single line for air distribution,
- overloading of motors without damage and energy losses.

The disadvantages of pneumatic mechanisms are:

- low efficiency - compressed air is the most expensive energy,
- low stiffness of the mechanism,
- noise caused by air expansion at the air motor outlet,
- low working pressure - small forces and powers.

Pneumatic mechanisms have the same physical basis as hydraulic mechanisms; therefore, the function and design of the individual system elements are very similar. However, it should be noted that due to the air properties, a number of differences can be found between hydraulics and pneumatics. In this part of the study text, the basics of pneumatic element design and system composition will be presented.

As already mentioned, the differences are due to differences between the properties of liquids and gases, especially the compressibility of air. This affects some calculations of the system. The dependence of volume on pressure and temperature can be expressed by the ideal gas state equation:

$$p_a \cdot V = m \cdot r \cdot T \tag{11.1}$$

where $p_a$ [Pa] is the absolute pressure, $V$ [$m^3$] the gas volume, $T$ [K] the thermodynamic (absolute) temperature, $m$ [kg] the mass of gas, and $r$ [$J \cdot kg^{-1} \cdot K^{-1}$] is the gas constant - for air $r = 286.9 \ J \cdot kg^{-1} \cdot K^{-1}$.

Of course, the air is a mixture of real gases, but for simplicity and with sufficient accuracy the above equation for an ideal gas can be used for basic calculations.

In the field of pneumatic mechanisms, it is important to note that air pressure can be defined in either absolute or relative form. It depends on whether the value is related to vacuum (absolute zero) or to atmospheric pressure (relative zero). In thermodynamic calculations (based on the equation of state), absolute pressure is always used, while in parametric calculations of pneumatic mechanisms (such as the calculation of the cylinder piston size), overpressure is most often used. When sizing vacuum suction cups, underpressure is considered.

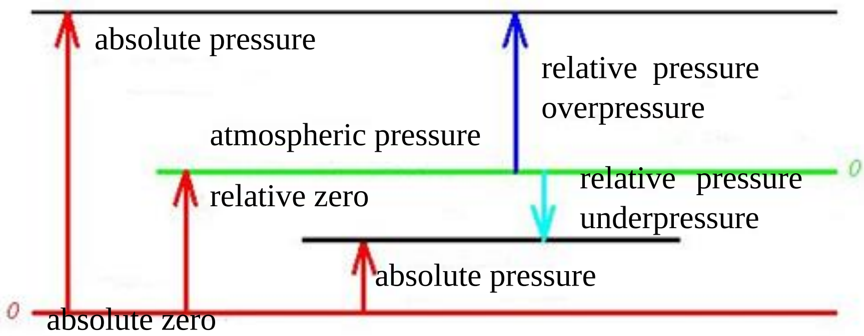



Fig. 11.1 Absolute and relative pressure [43]

**Normal (normalized) air conditions**

Due to the fact that the volume and, therefore, the volumetric flow rate of air depends on other state variables ($p$, $T$), it is necessary to indicate under which conditions the value is valid or the volume (volumetric flow rate) is given under the so-called normal conditions. For engineering calculations, the **normal technical conditions** are most often used, where the normal pressure is $p_n = 1 \cdot 10^{-5}$ Pa and the thermodynamic temperature is $T = 293.15$ K.

There are several ways to write the unit of volume converted to normal conditions. One of the ways is to add the index "n" to the unit of volume, e.g. $m_n^3$ or $dm_n^3 \cdot min^{-1}$. The second way that can be encountered is to add a capital "N" before the unit, e.g. $Nm^3$ or $Ndm^3 \cdot min^{-1}$, which can be misleading.

Currently, ISO/DIS5598 [44] and JISB0142 [45] standards apply, which require the indication of "ANR" after the unit of volume or flow. The abbreviation ANR means "atmospheric normal references". These are defined in the relevant standards as 20 °C, 1 013 mbar, 65 % relative humidity. An example of unit notation is as follows: $m^3$(ANR), $dm^3 \cdot min^{-1}$(ANR) [46].

## 11.1 Production, treatment and distribution of compressed air

At the beginning, it is necessary to mention, at least briefly, the production, distribution, and treatment of compressed air. Compressed air is produced from atmospheric air by means of compressors. Depending on their principle of operation, compressors can be divided into displacement compressors and turbo compressors.

**Displacement compressor**

Suction flow rate ranges (10 ÷ 1 500) $m^3 \cdot h^{-1}$(ANR), pressure (0.6 ÷ 1) MPa for single-stage compressors, up to 2 MPa for two-stage compressors. For example, reciprocating compressors (single and two-stage), diaphragm and screw compressors are commonly used types.

**Turbo-compressors**

These are turbomachines of axial or radial blade design. They are used in central compressor stations of large industrial complexes. Their suction flow is in the range of $(10^4 \div 10^6)$ $m^3 \cdot h^{-1}$(ANR), pressure (1 ÷ 2) MPa.

The compressor is used to produce compressed air, but it needs to be complemented with other elements to form a compressor station (compressor set). The typical composition of a compressor station is as follows Fig. 11.2.

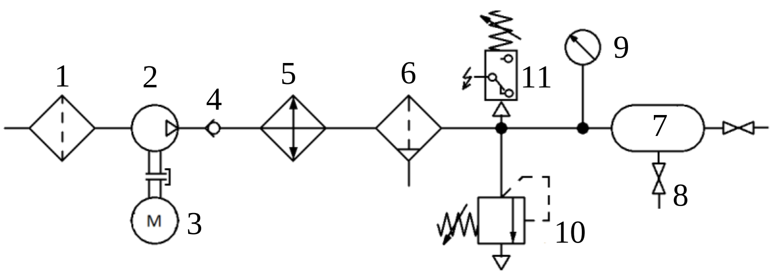



Fig. 11.2 Compressor station

*1 – filter, 2 – compressor, 3 – electric motor, 4 – check valve, 5 – cooler, 6 – condensate separator, 7 – air tank, 8 – shut-off valve, 9 – pressure gauge, 10 – relief valve, 11 – pressure switch*

The filter (1) prevents impurities from entering the compressor (2). The compressor is usually driven by an electric motor (3) or an internal combustion engine in the case of mobile compressors and vehicles. During compression, the air heats up, so cooling (5) is required. Cooling is done by ribbing on the compressor head and cylinder and on the discharge pipe. For larger compressors, water cooling is used. After cooling, the condensate must be separated from the air and drained (6). The check valve (4) prevents backflow when the compressor is switched off. Due to the variable air consumption, it is necessary to include an air tank (7) in the system. This serves as a reservoir of pressurized air and compensates for the above-mentioned fluctuations in consumption. The air is also cooled here, and the steam in the air may condense. To drain the condensate, the air tank must be equipped with a valve (8) for manual or automatic drainage of the condensate. There must be the a pressure gauge (9) on the air tank, and the safety relief valve (10) is an essential safety feature. If the system control is working properly, e.g. by means of pressure switch (11), the safety relief valve is permanently closed and really only serves as a fail-safe in case of failure of the pressure switch or other control element.

Depending on the design and application of the compressor, the **quantity of supplied air is controlled** in the following ways [42]:

**Start-stop control** - after reaching the maximum pressure in the air tank, the compressor drive is switched off or disconnected by means of a coupling. When the pressure drops to the lower pressure limit set on the pressure switch, the drive restarts, and the compressor refills the air tank to maximum pressure.

**Speed control** - the amount of supplied air can be controlled by the continuous or stepwise change in compressor speed.

**Control by air exhaust after the compressor** - it is used where the compressor is permanently connected to the drive and no other control can be primarily used, such as in trucks, buses, and other vehicles. In this case, the compressor supplies air continuously, and the overflow is exhausted back to the atmosphere.

**Air quality requirements**

Air quality is defined by ISO 8573-1 [47]. For common pneumatic mechanisms, class 4 or 5 is sufficient. If the quality of one of the parameters in the "row" is not suitable for a given application, the classes can be combined and defined more precisely, e.g., 2:4:3, class 2 then applies to particulate matter, class 4 to water content, and class 3 to oil content.

Tab 11.1 Air quality according to ISO 8573-1

| Class | Solid particles | | Water content | Oil content |
|---|---|---|---|---|
| | maximum particle diameter [μm] | maximum particle concentration [mg · $m^{-3}$] | temperature of water condensation [°C] | maximum oil concentration [mg · $m^{-3}$] |
| 1 | 0.1 | 0.1 | -70 | 0.01 |
| 2 | 1 | 1 | -40 | 0.1 |
| 3 | 5 | 5 | -20 | 1 |
| 4 | 15 | 8 | +3 | 5 |
| 5 | 40 | 10 | +7 | 25 |
| 6 | - | - | +10 | - |

In order to ensure the necessary air quality, the air must be filtered, firstly by a filter at the compressor inlet, then at the compressor outlet and always at the inlet of the working mechanism. The compressed air must also be de-humidified and, if necessary, lubricated.

**Drying the air** (removing water vapour) or also lowering the dew point can be performed in several ways. One of them is **drying by cooling**. After compression, the compressed air is cooled, thereby precipitating the water vapour. Another method is **adsorption drying**. Adsorption is the binding of water to the surface of a substance. Moisture is trapped by capillary action - the physical principle of drying. Simply put, vapour particles are trapped in the pores of the desiccant. The desiccant must be de-humidified, so-called regenerated, with warm air. This is why adsorption dryers are designed with two chambers - one chamber is used to dry the air, and the other one regenerate. **Absorption drying** can also be mentioned. In this case, the moisture binds chemically to the desiccant. The desiccant must be replaced after a certain period of time.

However, even after drying, the air may contain a small amount of water vapour that can condense in the pipelines. **Condensate separators** are used to prevent condensate entering the system and are placed in front of every pneumatic mechanism. Separators are often combined with a filter in one unit, as is shown in Fig. 11.3. The addition of a pressure regulator (see Fig. 11.4) results in a basic unit for air treatment (Filter-Regulator – FR Unit). It can be represented by a single combined unit, or it can be assembled from individual elements. If the compressed air needs to be lubricated, a lubricator can be included as the last element.

The condensate separator works as follows. At the inlet, the air is forced to rotate, and by centrifugal force, droplets of condensate are ejected from the stream onto the walls of the vessel. The condensate then flows to the bottom of the vessel, which is equipped with a manual or automatic valve for draining the condensate. The condensate-free air then passes through a replaceable (usually plastic) filter insert which removes mechanical impurities.

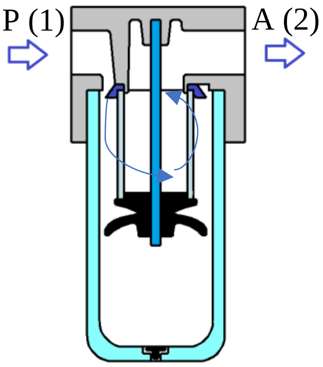



Fig. 11.3 Filter with condensate separator

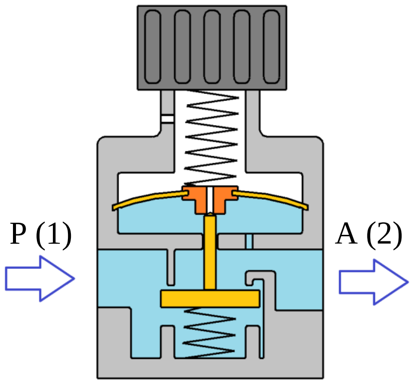



Fig. 11.4 Pressure regulator

It has been mentioned that the air treatment unit may contain a lubricator. A few comments on this. At the present time, lubrication of pneumatic mechanisms with oil mist is being abandoned mainly for health and environmental reasons. This is made possible by the fact that the materials used for the construction of the elements do not require lubrication, or, as is the case with linear and semi-rotary actuators, clamping elements, as well as directional valves, these elements already contain the grease filling. The only elements that need to be lubricated with oil mist are rotary motors. However, there is an exception. Turbine engines (turbomachines) do not require lubrication. Two other important observations are that lubrication is guaranteed within 10 m from the lubricator, and once the mechanism has started to lubricate, lubrication cannot be abandoned.

### Air distribution

The air distribution can be divided into external lines and internal lines in the halls, which are often made of steel pipes, but nowadays, internal pipelines made of aluminium profiles and plastic are also used. The piping's layout should have a slope of (1 ÷ 2) % in the direction of the air flow, and must include water collecting wells at the end of each course of the duct, with the possibility of draining downward and must be easy to discharge the moisture. The brunches to devices should be made from the top of the pipe to prevent condensate from entering the device's pneumatic circuit.

The connection of the individual elements of the pneumatic circuit is most often realized by plastic tubing. Polyurethane or polyamide hoses are often used. Depending on the chemical composition, hoses can have different mechanical, chemical or heat resistance; they can be antistatic. Hoses are also available directly for the food industry, etc. The hoses are usually calibrated to the outer diameter because of the seals in the one-touch fittings, which are widely used to connect the hoses to the individual elements, see Fig. 11.5. After the hose (1) is inserted into the fitting, a collet (2) holds it in place. In case of disassembly, the collet must be opened by pushing on the release ring (4).

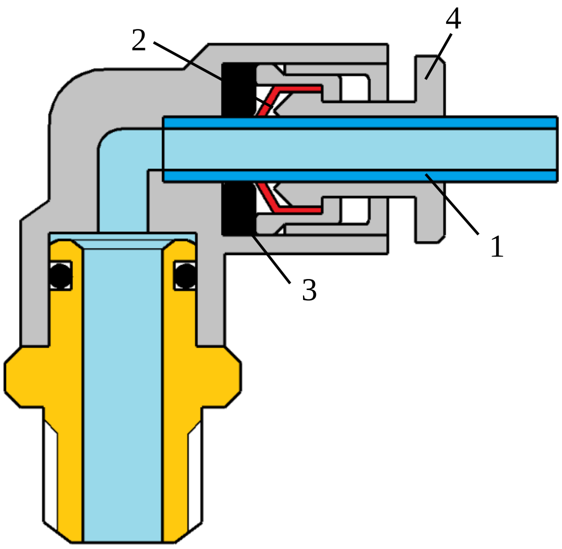



Fig. 11.5 One-touch fitting

*1 – hose, 2 – collet, 3 – seal, 4 – release ring*

## 11.2 Pneumatic actuators

Pneumatic actuators are used to transform air pressure energy into mechanical work. According to the movement of the output element (shaft, flange), actuators can be divided into linear, semi-rotary and rotary motors. The most common are **linear motors (cylinders)**, of which manufacturers offer a large number in various designs and sizes, in more detail, e.g., in. [48], [49].

Fig. 11.6 shows a section of a pneumatic cylinder, which consists of the following main parts: 1 - cylinder tube, 2 - rod cover, 3 - head cover, 4 - piston (these parts are usually made of aluminium alloys, the cylinder tube can be steel), 5 - piston rod, 6 – tie-rod, 7 – tie-rod nut, 8 - piston rod nut (these parts are steel), 9 - cushion ring, 10 - valve for cushion adjustment, 11 -

cushion seal, 12 - piston rod seal, 13 - bushing, 14 - piston seal, 15 - magnetic ring for position sensing, 16 - wear ring, 17 - bumper.

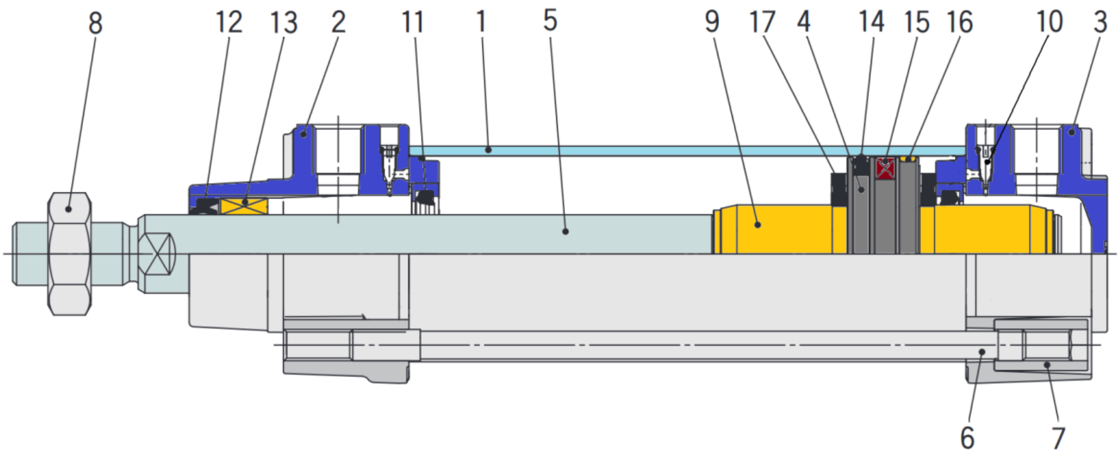



Fig. 11.6 Main parts of the pneumatic cylinder

Depending on the specific type of cylinder, the individual parts may differ or may not be included in the cylinder at all, such as the cushioning or the magnetic ring. There are many types of pneumatic cylinders. First of all, there are the following three types of linear motors according to ISO standards, Fig. 11.7, Fig. 11.8 and Fig. 11.9.

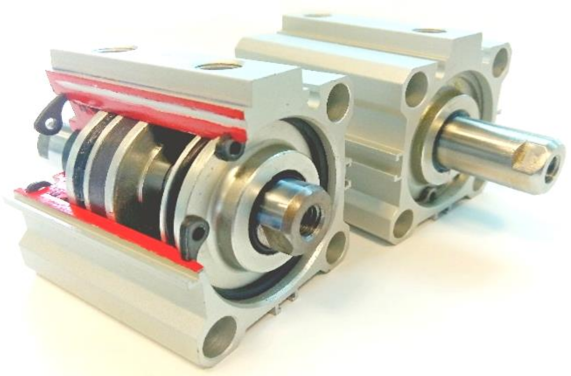


Fig. 11.7 Compact Cylinders according to ISO 21287

Fig. 11.8 Round Body Cylinders according to ISO 6432

Fig. 11.9 Tie rod cylinders according to ISO 15552

The parameters and characteristics of the standardized cylinders can be summarized as follows.

**Round Body cylinders** according to ISO 6432 - piston diameter (8 ÷ 25) mm and standard stroke (10 ÷ 300) mm, with single or double piston rods, up to stroke of 150 mm are also available as single-acting cylinders with spring. They are also available with a non-rotating piston rod. Damping in the end position is realized either by a rubber ring or by pneumatic damping.

**Compact cylinders** according to ISO 21287 – piston diameter (20 ÷ 100) mm, standard stroke (10 ÷ 150) mm. No damping in the end position or the cylinder is equipped with a rubber damping element. In the basic version, the cylinders are supplied with a single or double piston rod. In addition, some companies also offer other designs, such as tandem cylinders for greater force, multi-position cylinders, cylinders with non-rotating rods and others. Selected designs are schematically indicated in Fig. 11.10.

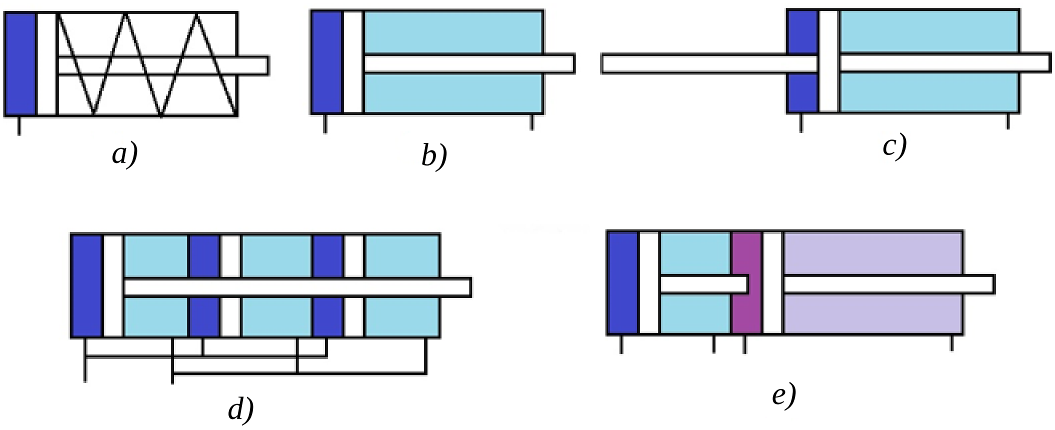



Fig. 11.10 Cylinder types

*a – single-acting cylinder with spring, b – double-acting cylinder with single-rod,*
*c – double-acting cylinder with double-rod, d – tandem cylinder, e – multi-position cylinder*

**Tie rod cylinders** according to ISO 15552 – piston diameter (32 ÷ 125) mm, stroke up to 2 m. They are available with a cylindrical tube with tie rods, or the body consists of an aluminium-drawn profile to which the covers are screwed. As with compact cylinders, a range of designs is available; see Fig. 11.10.

In addition to standardized cylinders, manufacturers offer miniature cylinders with piston diameters from 2.5 mm to cylinders with a diameter of 250 mm, actuators with a non-circular piston, cylinders with a piston rod guide, locks in the end positions, etc. However, these are no more standardized. Fig. 11.11 shows an example of a compact cylinder with a guide that can tolerate relatively high transverse, even impact loads.

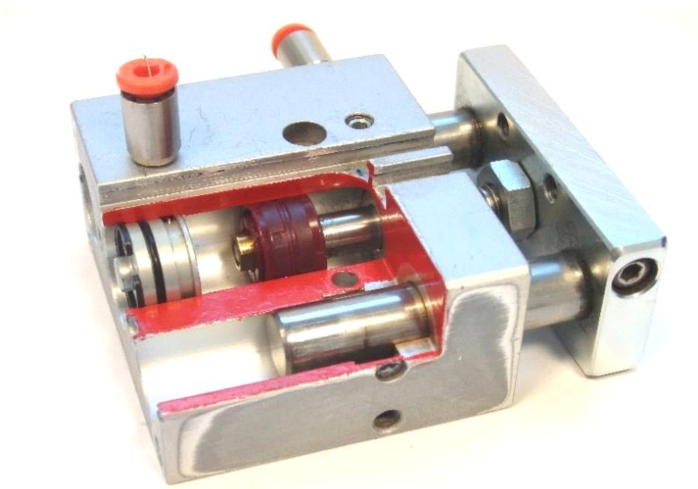


Fig. 11.11 Compact cylinder with a guide

At this point, it is also necessary to mention the **basic calculations**. The scheme of a single-rod pneumatic cylinder is shown in Fig. 11.12.

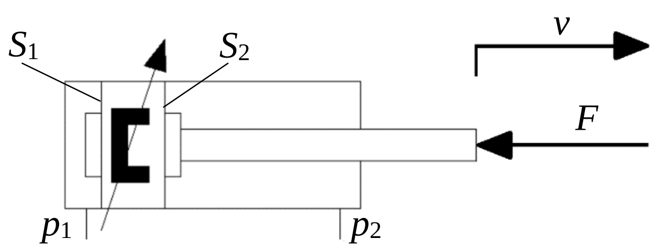



Fig. 11.12 The scheme of single-rod pneumatic cylinder

When calculating the **cylinder size**, we start with the equation of force balance on the piston:

$$p_1 \cdot A_1 = p_2 \cdot A_2 + F + F_{fr} \,, \tag{11.2}$$

where $p_1$ [bar] is pressure on the piston side, $p_2$ [bar] pressure on the piston rod side, $A_1$ [m$^2$] is the piston area, $A_2$ [m$^2$] is the piston rod area, $F$ [N] is the loading force, and $F_{fr}$ [N] is the frictional force.

In equation (11.2), $p_1$ is the pressure on the piston area $A_1$ during movement, i.e., the working pressure set on the pressure regulator reduced by the supposed pressure drop $p_1 = p - \Delta p$. The supposed pressure drop $\Delta p$ is chosen in the range of (0.5 ÷ 1) bar. We will suppose that the pressure on the other side of the piston (in the piston rod chamber) $p_2 = 0$. $F_{fr}$ is the frictional force generated by the friction of the piston seal and piston rod seal. It can be replaced by multiplying the loading force by coefficient $a = (0.1 \div 0.3)$, i.e., $F_{fr} = F \cdot a$. The force balance equation then takes the form:

$$(p - \Delta p) \cdot A_1 = 0 \cdot A_2 + F + F \cdot a \,. \tag{11.3}$$

After adjustment:

$$(p - \Delta p) \cdot A_1 = F \cdot (1 + a) \,. \tag{11.4}$$

The piston area is then expressed from equation (11.4), the piston diameter is calculated, and the nearest greater cylinder diameter is selected from the catalogue based on this value.

**Air consumption** $V_n$, i.e., the volume of air required to extend and retract the piston rod is determined from the volume of the piston chamber and piston rod chamber and by converting to normal conditions, i.e.:

$$V_n = \left( \frac{\pi \cdot D^2}{4} + \frac{\pi \cdot (D^2 - d^2)}{4} \right) \cdot H \cdot \frac{p_{abs}}{p_n} \cdot 1\,000 \,, \tag{11.5}$$

where $V_n$ [dm$^3$(ANR)] is the volume of air required to extend and retract the piston rod, $D$ [m] is the piston diameter, $d$ [m] is the piston rod diameter, $H$ [m] is the cylinder stroke, $p_{abs}$ [bar] is the absolute pressure of compressed air, and $p_n$ [bar] is the normal pressure. The constant 1000 is the conversion from cubic metres to litres.

This equation is often simplified by calculating only the volume of the piston chamber and multiplying it by two. It is true that the piston rod volume reading is neglected in this way, but the error is minimal. In addition, thermodynamic processes occur in the cylinder, and for this reason, the consumption cannot be calculated completely accurately, so it is advisable to slightly overestimate the calculation. The measurements made it is clear that the results of the simplified calculation above are satisfactory.

The **average air consumption** (average flow rate) $Q_{na}$ for piston rod extension and retraction and for $N$ cycles per minute can be found from the following equation:

$$Q_{na} = 2 \cdot \frac{\pi \cdot D^2}{4} \cdot H \cdot N \cdot \frac{p_{abs}}{p_n} \cdot 1\,000 \,, \tag{11.6}$$

where $Q_{ns}$ [dm$^3 \cdot$ min$^{-1}$(ANR)] is average air consumption, and $N$ [-] is the number of cycles.

In the case of a single-acting cylinder, the consumption is half. In order to size the control elements, it is necessary to determine the **immediate flow rate** $Q_n$ that the system must pass to achieve the desired piston rod extension rate.

$$Q_n = \frac{\pi \cdot D^2}{4} \cdot v \cdot \frac{p_{abs}}{p_n} \cdot 60\,000 \,, \tag{11.7}$$

where $Q_n$ [dm$^3$ · min$^{-1}$(ANR)] is immediate flow rate, and $v$ [m · s$^{-1}$] is the required piston rod extension velocity.

Since the piston in the cylinder can move at a relatively high speed (normally up to 1 m · s$^{-1}$, even more in special designs), the cylinder could be damaged when the piston reaches the end position. It is, therefore, necessary to provide **damping in the end position**. This can be performed by a rubber element or the frequently used internal air cushioning, see Fig. 11.13. In the last phase of the movement, part of the air leaving the cylinder is enclosed between the piston and the cover. This enclosed air is discharged through a throttle valve, which can be used to control the damping. When the piston moves in the opposite direction, the air is fed into the enclosed space through a check valve.

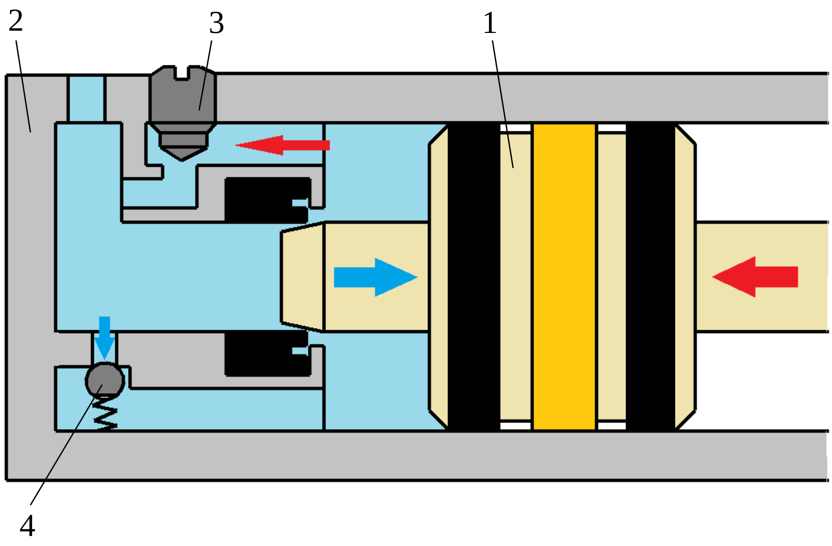



Fig. 11.13 Pneumatic cushioning

*1 – piston, 2 – motor cover, 3 – throttle valve, 4 – check valve*

Various feet and flanges can be used to connect the cylinder to the mechanism frame. These allow the cylinder to be firmly connected to the frame with screws, or the cylinder can be mounted swinging by means of clevis on the head cover or by means of flanges with pins. The end of the piston rod can be connected to the manipulated part of the mechanism with nuts or swingingly with rod clevis or with a ball joint. In addition, the cylinders can be equipped with a brake to secure the position of the piston rod.

Other types of linear piston actuators are **rodless actuators**. In this case, the output element is a flange (slider) that moves along the actuator body. The transmission of force and motion from the piston to the slider is provided, for example, by a steel belt that is routed through two pulleys, magnetically, where the piston and slider are equipped with permanent magnets, or by a mechanical connection, as is shown in Fig. 11.14.

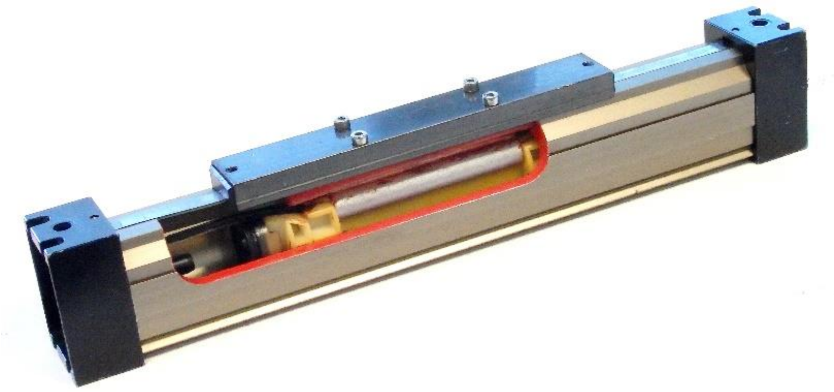


Fig. 11.14 Rodless actuators

Manufacturers supply a wide range of these drives, which differ not only in design but also in transmitted force and bearing stiffness. The undeniable advantages of these motors include the ability to apply forces and torques in all three axes. The load is, of course, limited, and the bearing capacity must be checked by calculation.

### Rotary actuators

In addition to pneumatic actuators with a linear movement of the output component, actuators with a swivel movement of the output shaft are also used. So-called rotary actuators include single **vane actuators** with a swivel angle of up to 290° or double vane motors with a swivel angle of up to 120°. Double vane motors develop twice the torque for the same dimensions. The swivel angle is usually adjustable by means of fixed stops.

Fig. 11.15 Vane rotary actuator

Another way to achieve the swivel motion is by using a geared transmission Fig. 11.16. Here, the piston's linear motion is transformed into rotary motion by means of a **gear rack and pinion**. By selecting a longer rack, it is possible to achieve a swivel of more than 360°, which is an advantage over vane actuators. This type of actuator also achieves higher torques.

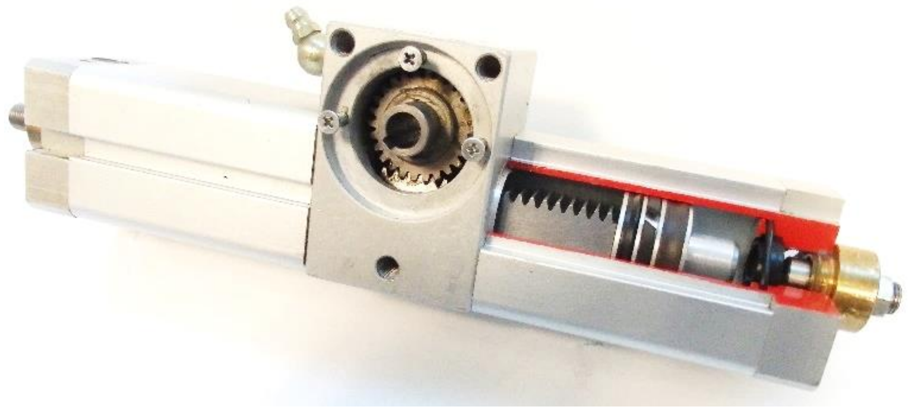


Fig. 11.16 Rack & Pinion rotary actuator

**Air motors**

Air motors are not very common, so only a brief mention is made here. They are mainly used in hand tools such as drills, grinders, tighteners and so on. The simplest and most common designs are vane motors (see Fig. 11.17) and gear motors, but piston and turbine motors cannot be overlooked.

Fig. 11.17 Vane air motor

## 11.3 Valves

Control elements are used to ensure the required movements of pneumatic actuators and the correct operation of pneumatic mechanisms. These can be divided into the following groups:

- valves for control of air flow direction - shut-off and check valves, directional control valves, logic elements,
- valves for control of the flow rate or speed of movement - throttle valves,
- pressure control elements - pressure reducing and relief valves,
- special elements - time and pressure relays.

By selecting the appropriate elements and designing the control part of the mechanism correctly, a purely pneumatic mechanism can be assembled, usually with two or three actuators or even more in more complicated cases. The control part of a mechanism with several pneumatic actuators is quite difficult, and it is nowadays more convenient to use electro-pneumatic control by means of programmable controllers. In this part, we will look at the design and function of the individual control elements and the basic pneumatic circuits that can be implemented using them.

### 11.3.1 Control of air flow direction

This category mainly includes **directional control valves**, which are used to control the direction of movement of the pneumatic actuator. Regardless of whether the mechanism is controlled mechanically, pneumatically or by an electrical signal, if we do not consider the air source and treatment elements, the pneumatic mechanism always consists of at least two elements, namely the pneumatic actuator and the directional control valve.

They must ensure that the piston moves in both directions, as is shown, for example, in Fig. 11.18. In one functional position, they must ensure that the cylinder chamber is filled and the piston rod is extended. In the other position, the valve exhausts the chamber, and the piston rod retracts.

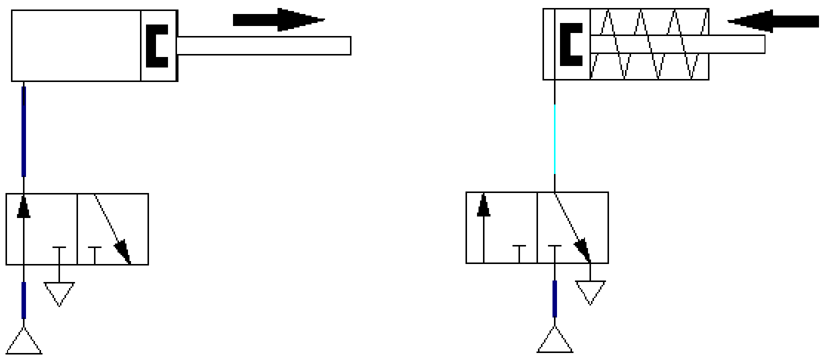


Fig. 11.18 Function of directional control valve

Directional valves can be divided according to the number of ways and the number of positions, each of which provides a different connection of ways. In addition, directional valves can vary in design (valve or spool design) and in control principle (mechanical, pneumatic, electrical, or their combination).

The simplest directional valves are **two-way two-position** (2/2) directional valves. These have one input port, and one output port, with both ports closed in one position and connected in the other.

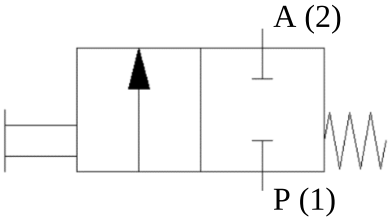



Fig. 11.19 Symbol of 2/2 directional valve

In the symbol of the directional valve, the basic position is usually the right position - inputs and outputs are indicated here. In this case, the basic position is spring-secured, so that if the button is not pressed, the directional valve remains in the closed basic position. This is usually called a normally closed - NC valve. If the positions were drawn in reverse, it would be a normally open - NO valve. In this case, the inlet and outlet would be connected and only when the button is pressed would the supply of compressed air between the inlet and outlet be interrupted.

The directional valves in Fig. 11.19 and Fig. 11.20 can also be designated as monostable. A spring maintains the directional valve in the basic (closed) position. A continuous control signal X is required to keep the directional valve in the open position. When this signal is lost, the directional valve returns to the basic (closed) position.

A frequently used directional valve (see Fig. 11.20) is a three-way two-position (3/2) directional valve in the basic position closed (NC), in this case, operated pneumatically, monostable (basic position is spring-secured).

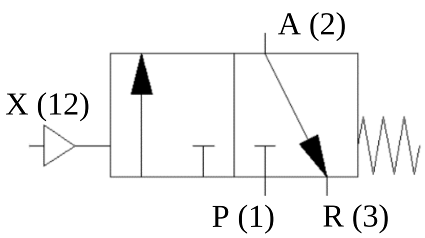



Fig. 11.20 Symbol of 3/2 directional valve

For proper function, the directional valves and other elements must be connected correctly. Therefore, all inputs and outputs of the directional valves are marked either numerically or with letters, which have the following meaning:

P (1) – inlet of the compressed air,

R, S, T (3, 5, 7) – exhaust port of outlet,

A, B – (2, 4) – outlet of the compressed air,

X, Y, Z (12, 14, 16) – control signal input (e.g., signal 12 allows connection of input 1 and output 2).

For some logic elements, which have two inputs and one output, it is possible to find the designation IN (input) and OUT (output).

The 3/2 directional valve is used as the main valve for controlling single-acting cylinders or for obtaining a pneumatic control signal. For example, Fig. 11.21 shows a manually operated 3/2 directional valve that can be used as a start button. If another control is used, e.g., by roller lever, it can serve as a limit switch.

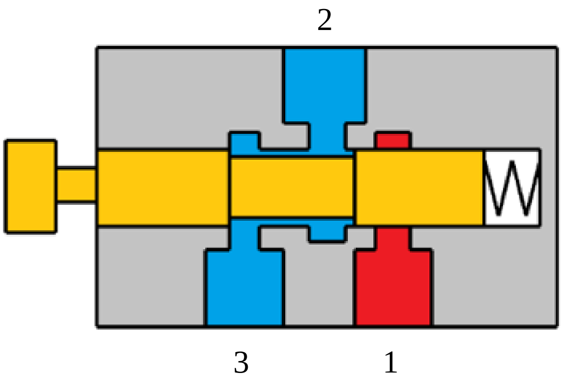



Fig. 11.21 3/2 directional valve

For controlling the movement direction of double-acting cylinders, it is possible to use four-way double-position (4/2) directional valve (see Fig. 11.22) or more often five-way double-position (5/2) directional valve, as is shown in Fig. 11.23. The function is exactly the same, the only difference is in the internal construction. Externally, the difference is that the 5/2 valves have two exhaust ports, while the 4/2 valve has only one port.

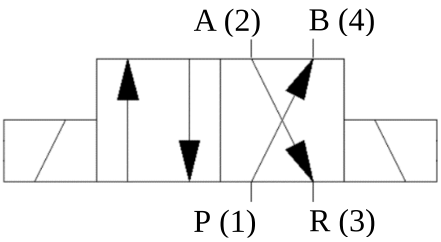



Fig. 11.22 Symbol of 4/2 directional valve

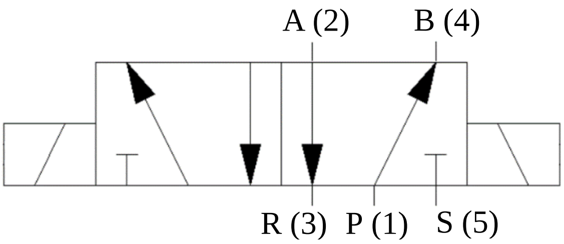



Fig. 11.23 Symbol of 5/2 directional valve

Two-position directional valves controlled from both sides mechanically except springs, pneumatically or electrically, can be called bistable. In these valves, a short pulse of the control signal is sufficient to move the valve spool to the second position, where it remains until a signal is applied from the other side. On the other hand, as already mentioned, in monostable directional valves a permanent signal is required for the changeover. After the signal has been lost, the directional valve is automatically moved to the basic position by a spring.

Manual and mechanical control of the valves has been mentioned above, e.g., by means of a button or roller lever. In these cases, the valve spool is moved by mechanical force. In the following figure (i.e. Fig. 11.24), the control is pneumatic. The spool is moved by supplying compressed air to the control input, and the air then acts on the spool face. The spool movement is therefore ensured by the pressure force.

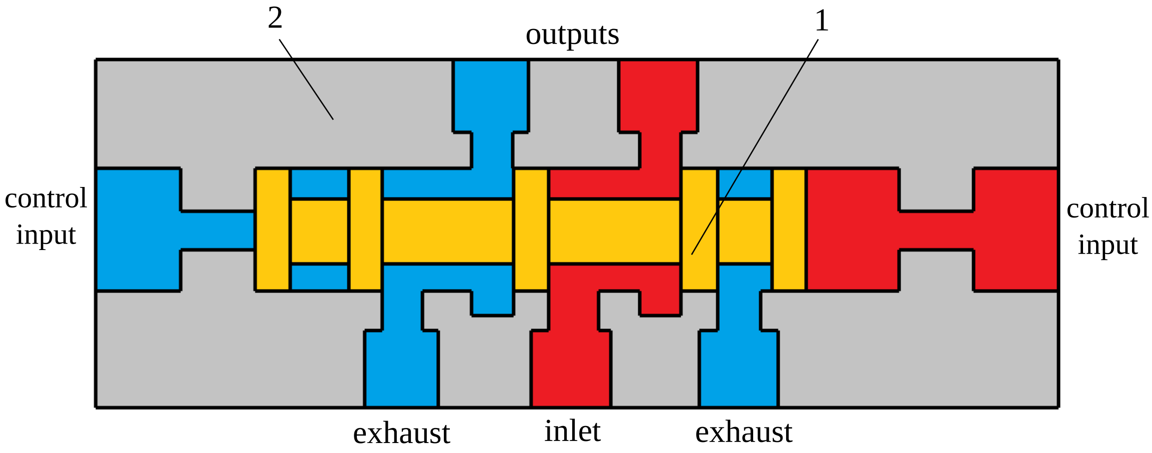



Fig. 11.24 5/2 pneumatically operated directional valve

*1 – spool, 2 – valve body*

If an electrical signal is used for control, the spool can be moved directly by an electromagnet, or more commonly, as is shown in Fig. 11.25, a two-stage electro-pneumatic control can be used. In this case, the solenoid controls a small pilot valve that supplies compressed air to the spool face. In the event of a power failure, the pilot valve can be opened by an emergency manual control.

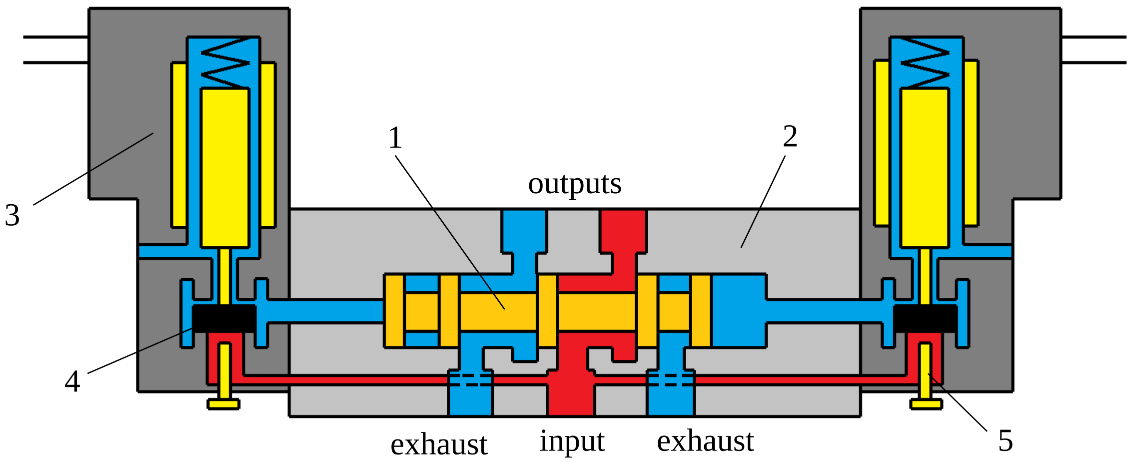



Fig. 11.25 5/2 electro-pneumatically operated directional valve

*1 – spool, 2 – valve body, 3 – electromagnet, 4 – pilot valve, 5 – emergency manual control*

**Check (one-way) valves**

As the name implies, these elements allow air flow in one direction only; in the other direction, they must guarantee absolute tightness. This is ensured, e.g., by using a cone valve, where in one direction the compressed air pushes the cone away and flows further; in the opposite direction, it pushes it against the seat and thus closes the valve Fig. 11.26. They are used, e.g., behind the compressor to prevent air backflow or as a one-way bypass for other elements, as will be shown later.

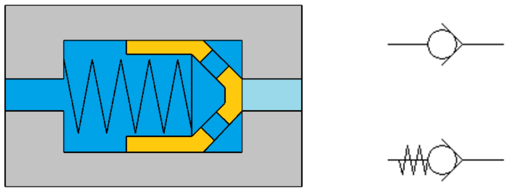


Fig. 11.26 Check valve

If it is necessary to allow flow in the opposite direction, a **pilot check valve** can be used. This allows air to flow from A to P after a control signal is applied to port X (the control piston moves the seal plate away from the seat), see Fig. 11.27. Air flow from P to A is always possible.

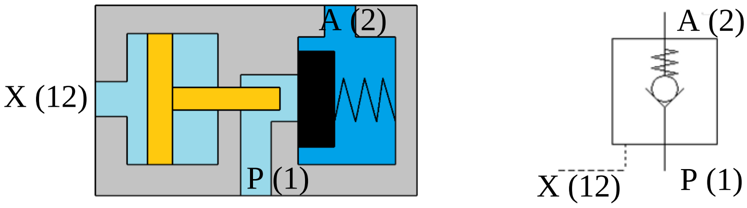



Fig. 11.27 Pilot check valve

By combining two check valves or pilot check valves, so-called logic elements can be realized.

**Shuttle valve**

This provides the function of logical sum, i.e., the signal at the output of an element will be in case there is a signal at one OR the other input or at both. Fig. 11.28 shows the possible internal design and the symbol of the element.

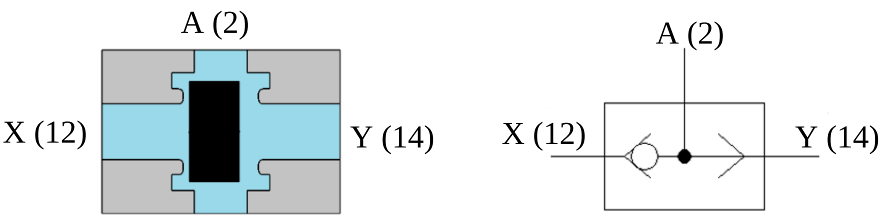



Fig. 11.28 Shuttle valve

**AND valve**

This element shown in Fig. 11.29 provides the logical product function. In order to have a signal on the output, there must be a signal on one AND the other input.

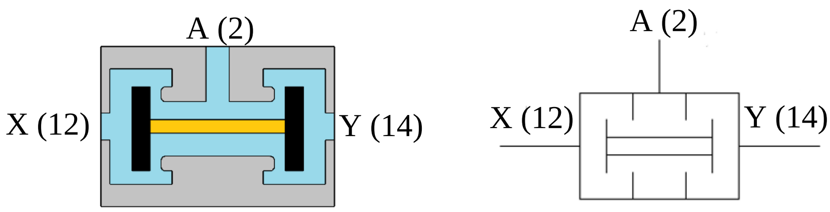



Fig. 11.29 AND valve

**Quick Exhaust Valve**

From a design point of view, it is a combination of a check and a pilot check valve. However, with this element it is possible to influence the velocity of cylinder movement quite significantly. It allows in one direction the air flow from the directional valve to the cylinder chamber (connection of ports 1 and port 2). After switching the directional valve to the position for extending the piston rod, the flow should start in the opposite direction (from port 2 to port 1). However, the air flow will move the sealing element (Fig. 11.30 shown in black), and the connection between ports 2 and 3 will occur. The air then flows into the atmosphere.

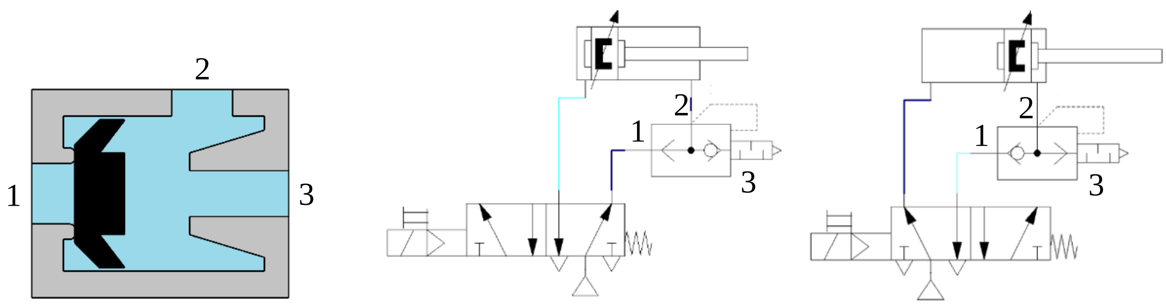



Fig. 11.30 Quick exhaust valve and its connection

The quick exhaust valves are mounted as close to the cylinder as possible or directly into the thread in the cover. Because air does not have to flow through the entire line and directional valve, which can cause quite a lot of resistance, air exhausts through the valve quickly from the cylinder's chamber. The air in the exhausted chamber does not restrict the piston movement, and therefore, a higher movement velocity is achieved. An example of the quick exhaust valve connection is shown in Fig. 11.30. In this case, it allows faster piston rod extension.

### 11.3.2 Flow control valves

By controlling the amount of air that flows into the pneumatic actuator, it is possible to control the piston movement speed or the rotation velocity of the air motor easily. This can be performed in a very simple way by changing the flow cross section, called throttling. The throttling elements can be either fixed, called orifices, or adjustable, called throttle valves. In this case, turning the adjusting screw increases or decreases the flow area between the cone and the seat.

The orifice and throttle valve restrict the air flow in both directions. Often, however, it is necessary to control the amount of flow in one direction only, so a bypass is used with the help of the check valve. Thus, it is obtained a combined throttle and check valve (also speed control valve), see Fig. 11.31.

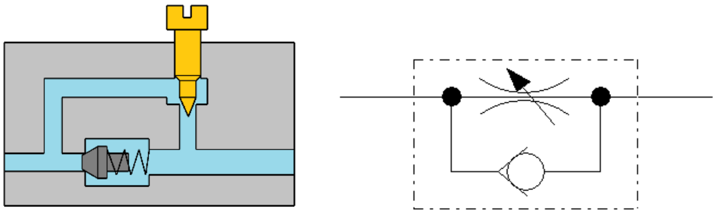


Fig. 11.31 Combined throttle and check valve

Throttle valves are available in two versions - for mounting between the hoses (inline version) and for mounting in to the thread in the cylinder cover, see Fig. 11.32. In the cylinder-mounted version, the speed control valve can be supplemented by a pilot check valve or a pressure-reducing valve, which can be used to adjust the pressure for piston movement in a given direction.

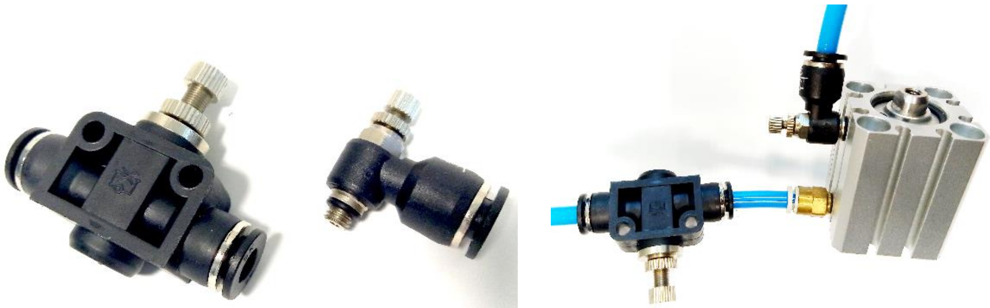


Fig. 11.32 Speed control valve - inline version and cylinder-mounted version

The speed control of pneumatic motor depends on how the speed control valve is connected to the circuit (how the check valve is oriented). According to the orientation of the check valve, the control is divided into meter-in (flow control when the air enters the cylinder chamber) and meter-out (flow control when air flows out of the cylinder chamber).

The meter-out control is preferable for pneumatic mechanisms; see Fig. 11.33. The piston chamber is filled with pressurized air, and the piston rod chamber is gradually exhausted (back pressure remains here). The piston is thus clamped between two columns of pressurized air, the system is relatively rigid, and the movement is more steady compared to the meter-in control, where the piston chamber is rapidly exhausted, and the piston chamber is gradually filled. With the meter-in control, the movement is unsteady and, in extreme cases, can be jerky (stopping the movement).

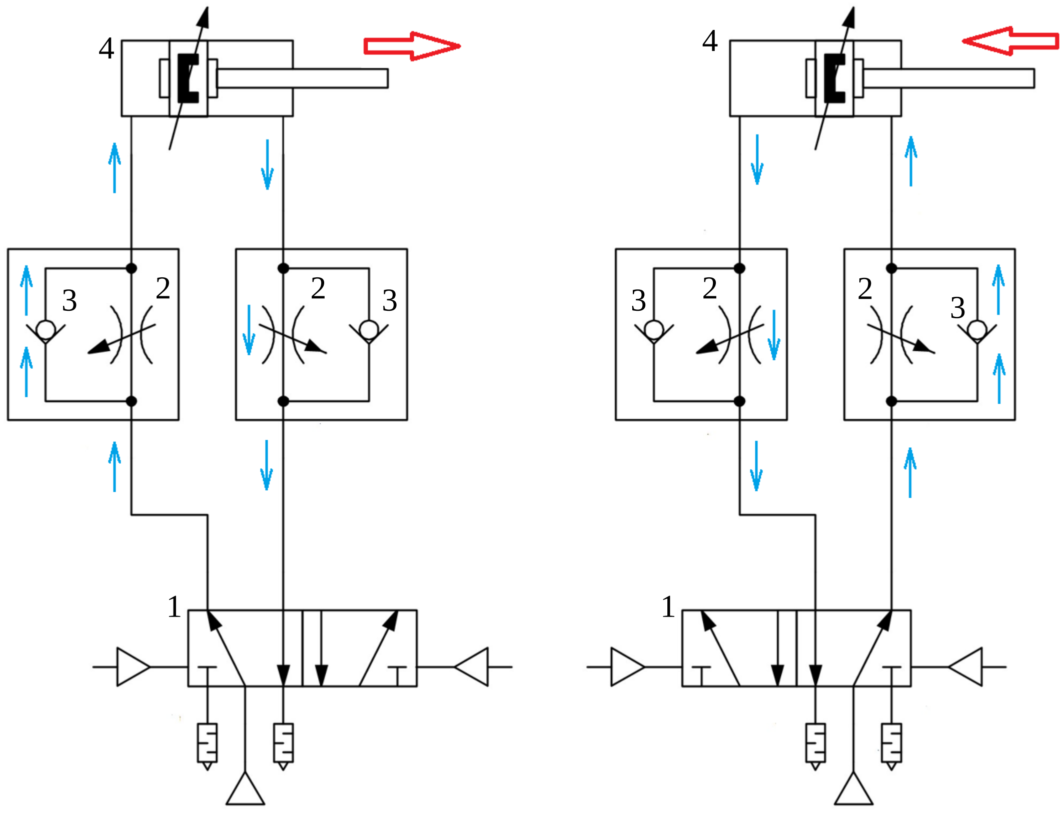



Fig. 11.33 Meter-out control during piston rod extension and retraction

*1 – directional valve, 2 – throttle valve, 3 – check valve, 4 – pneumatic cylinder*

### 11.3.3 Pressure control elements

This group includes relief valves, which must always be located behind the compressor, and pressure-reducing valves (pressure regulators), which will be described in more detail.

**Pressure regulators**

Pressure regulators are used to setting the required pressure in the circuit. They are always part of the air treatment unit. The pressure regulator, see Fig. 11.34, is open under normal conditions and allows compressed air to flow from the inlet (1) to the circuit outlet (2). When the circuit behind the valve is filled to the required pressure value set by the screw and the spring (4), the membrane (5) is lifted, thus closing the valve (6) and not allowing further pressure increase. Thus, the pressure regulator reduces the higher pressure in the distribution system to the lower pressure required in the mechanism. The valve only reopens when there is a consumption on the mechanism side (e.g., filling of the cylinder), which causes a slight pressure drop, and the valve compensates for this drop by reconnecting to the distribution system.

(Note: If the pressure in the distribution line is lower compared to the pressure set on the pressure regulator, the valve is still open and the inlet pressure is equal to the outlet pressure. The valve does not increase the pressure to the required value!)

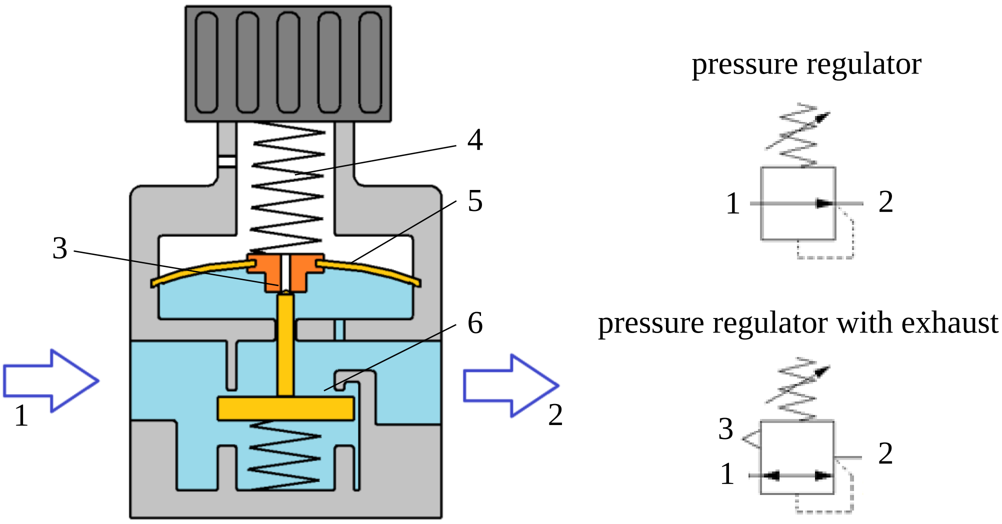



Fig. 11.34 Pressure regulator

*1 – input, 2 – output, 3 – exhaust, 4 – spring, 5 – membrane, 6 - valve*

Some pressure-reducing valves can be designed with a relief function or with an exhaust. In this case, if for some reason the pressure in the circuit is increased, the membrane (2) is deflected more upwardly, thereby releasing the throat of the valve (6) and some of the air exhausts to the atmosphere.

As mentioned, separate **relief valves** are used minimally in pneumatic mechanisms. They are an essential part of the compressor station, where they "control" the maximum allowable pressure in the air tank. If this is exceeded, they open and exhaust air into the atmosphere.

### 11.3.4 Special elements – time and pressure relays

Among a number of other elements that can be created, for example, by combining some basic elements, two relays will be mentioned, i.e., time and pressure relays.

By combining a 3/2 directional valve, check valve, throttle valve and a small air tank, time relays can be realized. Depending on the directional valve used and the orientation of the check valve, a time relay with delayed switching can be built (ON delay), for example, as is shown in Fig. 11.35.

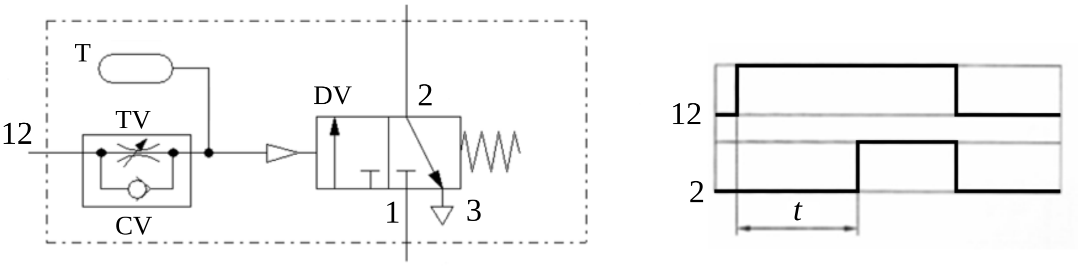



Fig. 11.35 Time relay with ON delay function

*DV – directional valve, TV – throttle valve, CV – check valve, T – air tank*

The function is as follows. A control signal (compressed air) enters port 12 and flows through the throttle valve into the airbox. When it fills after a pre-set time, the 3/2 directional valve is switched, and the compressed air passes from port 1 to port 2. The time delay can be set by adjusting the throttle valve.

Time relays offer even more connection combinations. If the check valve is oriented in the opposite direction, the 3/2 directional valve will be switched almost immediately when control signal 12 is applied. After the loss of signal 12, the air will be slowly discharged from the air tank through the throttle valve, causing a delayed shutdown of the 3/2 directional valve (OFF delay). The other two combinations are obtained by changing the positions of the directional valve to NO (normally open) and combining the orientation of the check valve.

In addition to the time relay, a **pressure relay** Fig. 11.36 can also be used in pneumatic control circuits. Its function is as follows. The pressure air is supplied to inlet 12 of the pressure valve from the place where the desired pressure is to be achieved (e.g., for clamping parts). When the pressure set on the pressure valve is reached, the signal passes through this valve to the 3/2 directional valve. This is switched, and a signal is obtained at output 2.

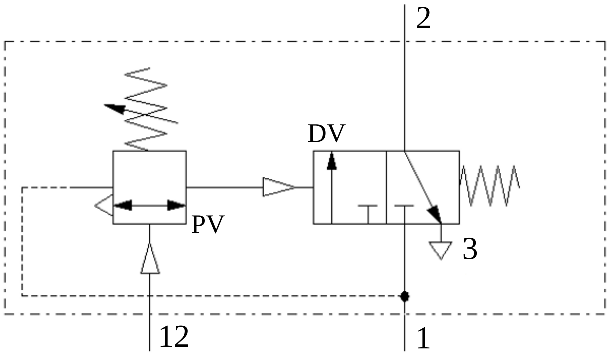



Fig. 11.36 Pressure relay

*PV – pressure valve, DR – directional valve*

## 11.4 An example of the basic pneumatic circuit

Fig. 11.37 shows an example of a simple pneumatic system consisting of one working element 1A, in this case, the double-acting cylinder, and the elements that control its operation. But first, a mention of air treatment. They are grouped into the so-called air treatment unit 0Z1, which consists of the filter with condensate separator, the pressure regulator and the manometer. The required working pressure can be set using the pressure regulator. In front of the treatment unit is located the 3/2 manually controlled directional valve, which has a safety function here. After its switching, the air is supplied further into the system. When the work is finished and the air supply is switched off, the entire system is exhausted at the same time. Only after the exhausting is it possible to intervene in the system, e.g., when replacing damaged elements.

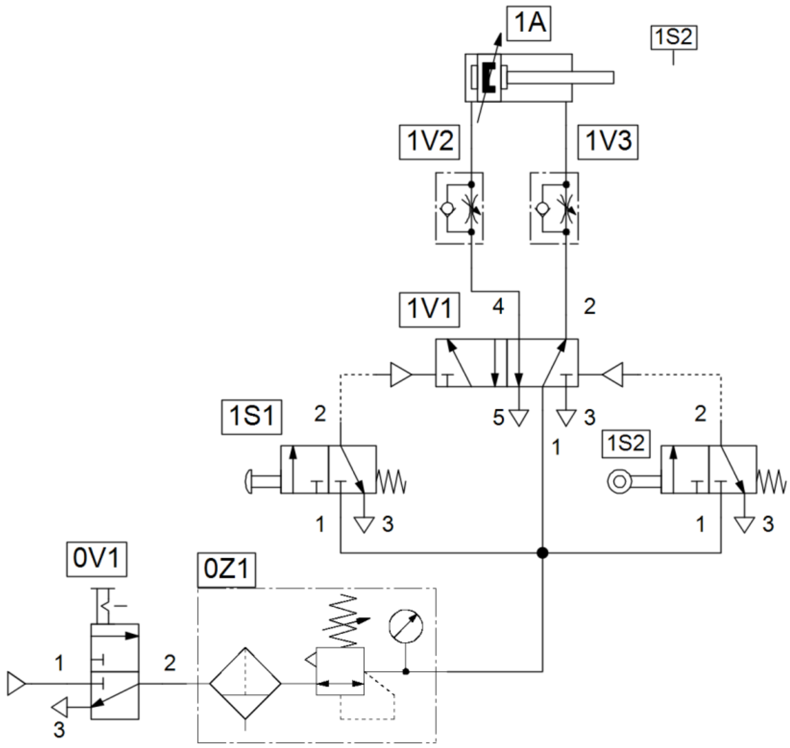



Fig. 11.37 An example of a simple pneumatic system

Now to the system control: After pressing the button of the directional valve 1S1, the signal comes from the left to the main directional valve 1V1. The directional valve spool is moved, ports 1 and 4 are connected, and the air begins to flow through the check valve (contained in the 1V2 combination element) into the cylinder piston chamber. From the second opposite chamber (i.e., the piston rod side), the compressed air begins to flow through the throttle valve 1V3 to the directional valve and through connected ports 2 and 3 out into the atmosphere. The piston rod starts to extend, and its velocity can be controlled by valve 1V3, which is used to throttling at the outlet (meter-out).

As soon as the piston rod extends to the end position, the limit switch (i.e., the mechanically operated directional valve 1S2) is switched on. The limit switch is physically located at the end of the piston rod stroke, it is shown in the diagram by the thick line, and its designation is 1S2. However, for the sake of clarity, all signal elements are drawn in the lower part of the diagram, and then these elements are assigned the appropriate designation, in this case, 1S2. After the limit switch is switched on, the signal comes from the right to the directional valve 1V1, which is adjusted to its original position and the piston rod is retracted. The velocity of the piston rod movement is realized by throttling at the outlet using the valve 1V2.

As the second example, the mechanism for cleaning the washers in the bath is selected, and the circuit design task is as follows. Design a pneumatic circuit for the cleaning bath mechanism. Washers for injection pumps should be cleaned in the bath. A pneumatic motor moves down and up in the bath with a container filled with washers. The cleaning will be started by pressing the Start button, and the cleaning will continue until the Stop button is pressed. It must be ensured that the piston rod returns to the upper (retracted) position when the stop button is pressed so that the washers can be removed.

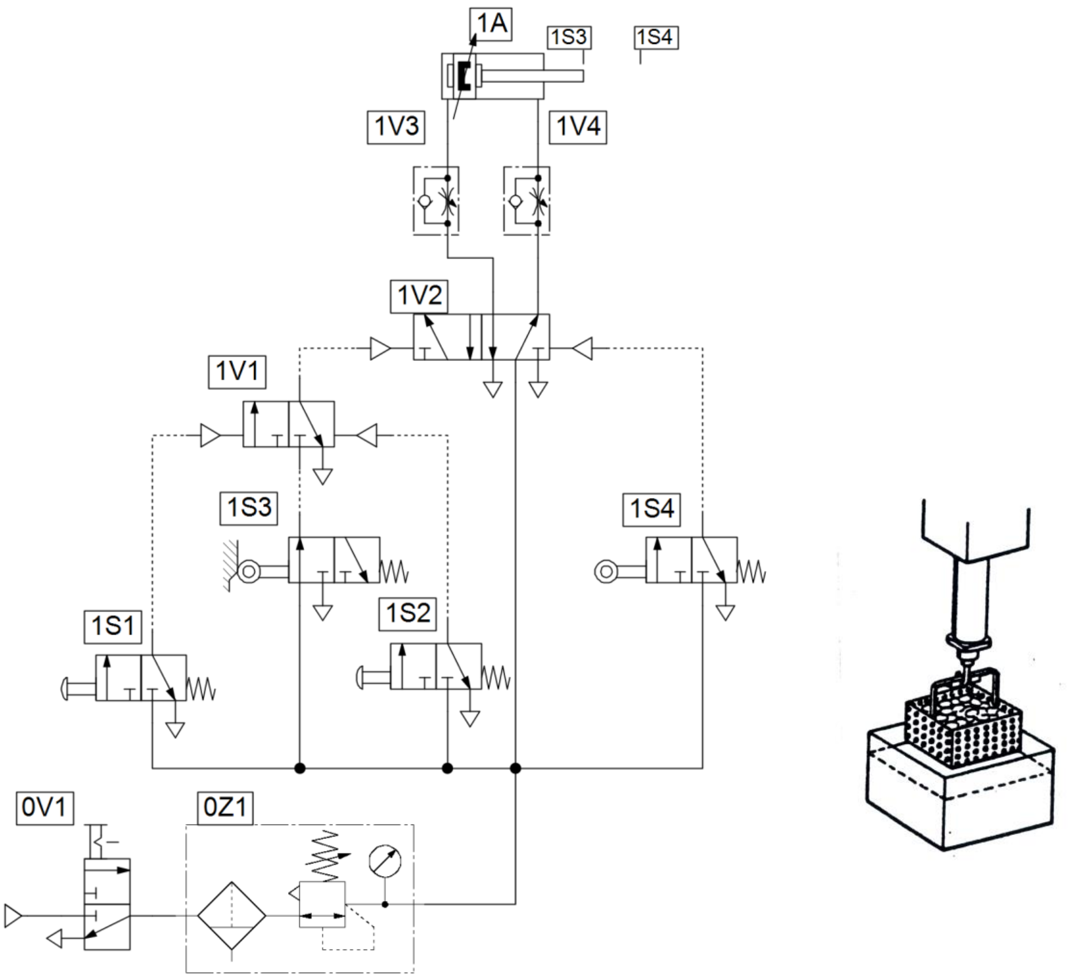



Fig. 11.38 Cleaning the washers - variant 1 and the illustration for the assignment

Description of the circuit function: Automatic up-down operation is provided by the limit switches 1S3 and 1S4, which give signals for one or the other direction of the movement Fig. 11.38. The start of the cycle is realized by pressing the directional valve 1S1 (Start) button, which sends a signal to reposition the directional valve 1V1, which unblocks the signal from the 1S3 limit switches. To stop the cycle, the button 1S2 must be pressed, which gives the command to block the signal from the limit switch 1S3.

If the Stop button is pressed during the downward movement (the piston rod extended) in variant 1, the system completes the cycle (piston rod goes down, comes up). If this is undesirable, the signal from the Stop button can be used not only to block the signal but also to change the position of the main direction valve 1V2 (by pressing Stop, the piston rod is immediately retracted). In variant 2, an alternative location of the blocking valve is also indicated - it blocks the air supply to both limit switches Fig. 11.39. In this case, the circuit must be completed with a shuttle valve (with OR function), that feeds the signal from the Stop button to the 1V2 directional valve. If the OR valve is not there, the condition of returning to the upper position when the Stop button is pressed while moving downwards would not be guaranteed.

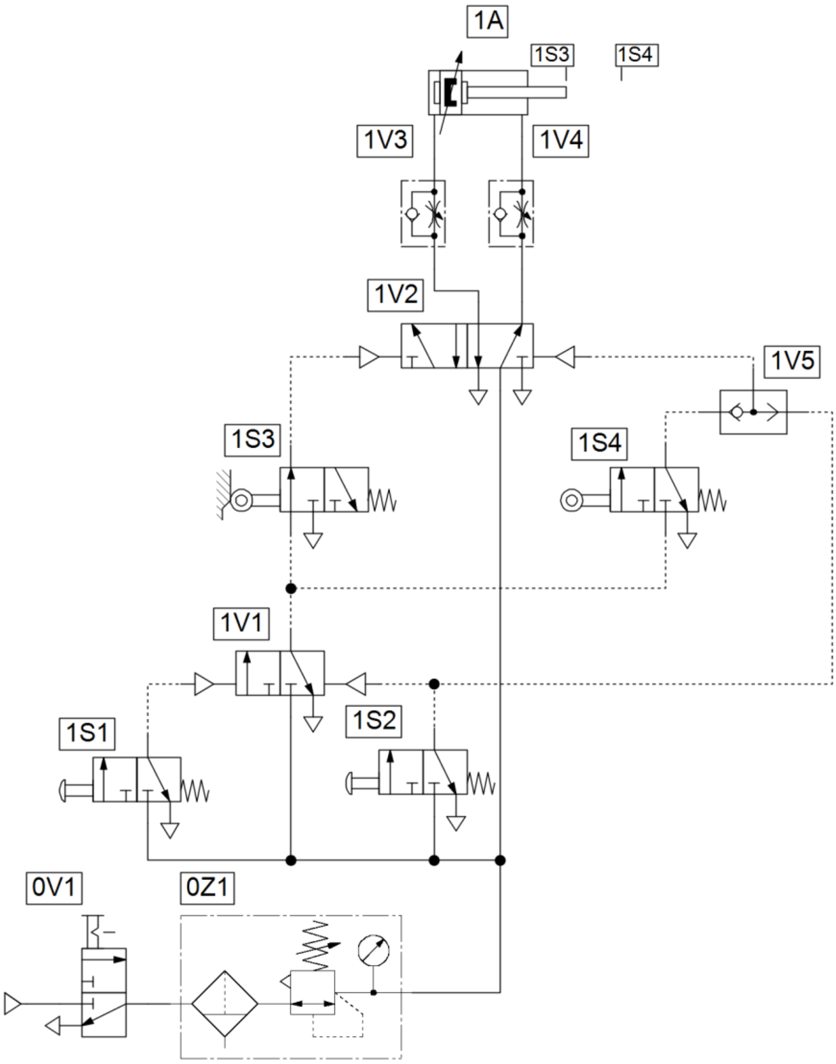



Fig. 11.39 Cleaning the washers - variant 2

Fig. 11.40 shows an example of a pneumatic circuit with two working cylinders performing the following duty cycle: When the start button is pressed, the piston rod of the cylinder is 1A extended, then the piston rod of the cylinder is 2A extended, and subsequently the piston rods are retracted in the same sequence. The individual movements are conditional on the extension or retraction of the piston rod and the actuation of the limit switches, which gives the command for the next movement.

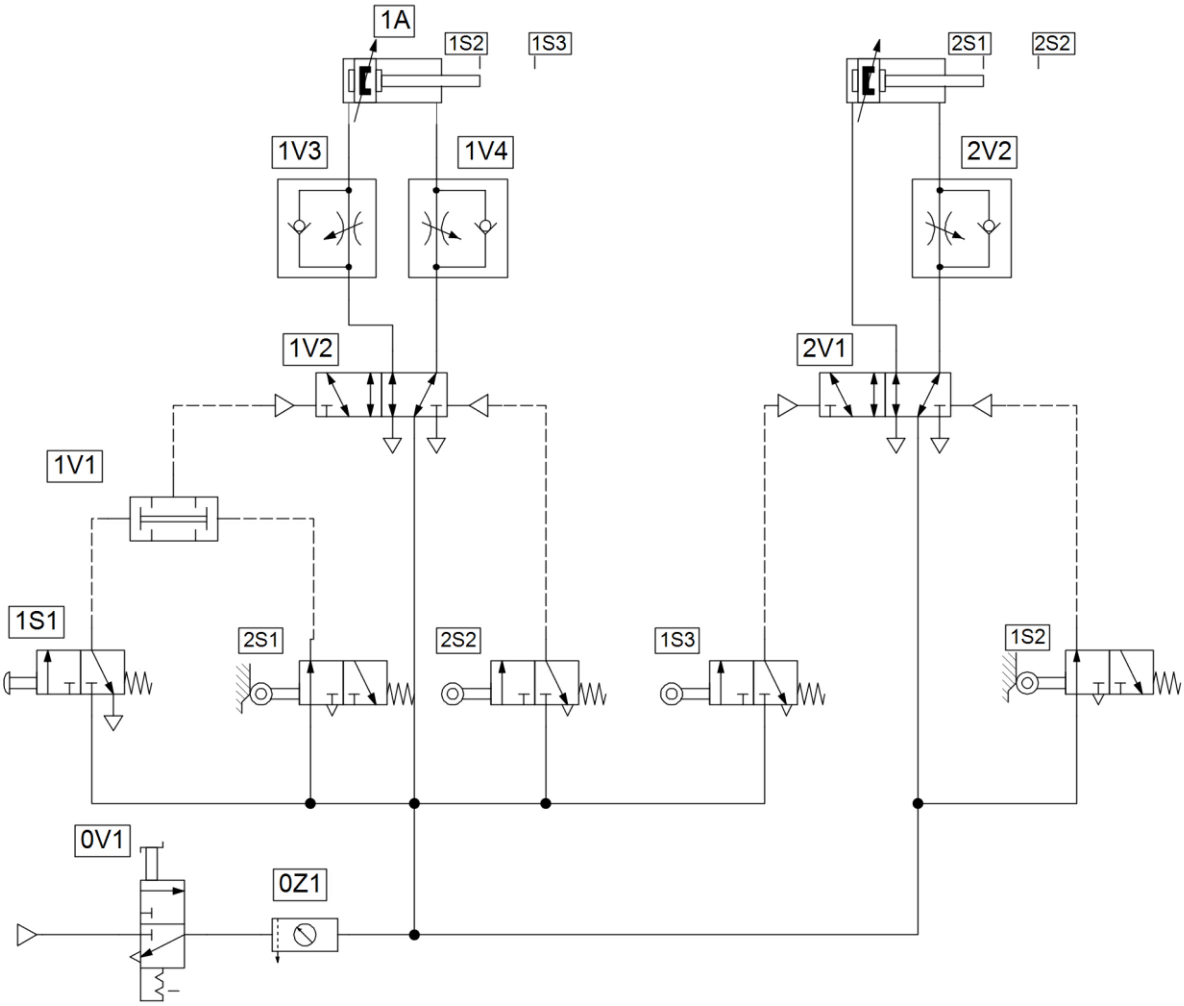



Fig. 11.40 Pneumatic circuit with two cylinders

In the circuit examples, the designation of pneumatic elements according to the VDI 3260 recommendation is used, where the composition of the designation is as follows:

3 – 1 V 2

device number (is not necessary with one device) | the number of whole | unit type designation | unit number

Tab 11.2 Designation of circuit elements

| Designation | Type | Example |
|---|---|---|
| P | compressors | |
| A | actuators | pneumatic cylinder |
| M | motors | electric motor |
| S | signal elements | buttons, limit switches, time relays... |
| V | valves | directional, shut-off, throttle, check valves… |
| Z | other | air treatment elements |

## 11.5 Calculation of basic pneumatic circuit

In addition to the sizing of the working elements mentioned above, the other elements need to be designed correctly. In this case, it is necessary to check that the directional valves, throttle valves, air treatment elements, hoses, etc., will “let through” at least the amount of air required by the actuator to perform the movements at the required velocity. The catalogue values, which express the size of the elements, are used for this check. The most commonly used value is the normal nominal flow rate $Q_N$, or $Q_n$, $Q_{Nn}$, which is determined on the measuring line according to ISO 6358 [50], see Fig. 11.41. The standard prescribes an inlet pressure to the element of 6 bar. The throttle valve is then used to set the resistance so that the pressure behind the element to be measured is 5 bar. The flow measured at these conditions is converted to normal conditions, and this value then defines the element size.

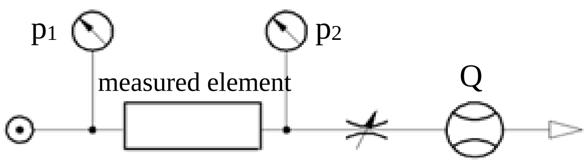



Fig. 11.41 Nominal flow measurement

In addition to the normal nominal flow rate, there are other coefficients that are used not only in Europe, but all over the world.

Their definitions are as follows:

$Q_n$ - Normal nominal flow [$dm^3 \cdot min^{-1}$ (ANR)]. Measured with upstream pressure of 6 bar and the pressure drop of 1 bar through the valve.

*kv* - The kv-value [$dm^3 \cdot h^{-1}$]. Measured with water at a pressure drop of 1 bar through the valve.

*Kv* - The Kv-value [$m^3 \cdot h^{-1}$]. Measured with water at a pressure drop of 1 bar through the valve.

*Cv* - The Cv-value [gal (USwet) $\cdot min^{-1}$]. Measured with water at a pressure drop of 1 psi (0.07 bar) through the valve.

*f* - The f-value [gal (Imp) $\cdot min^{-1}$]. Measured with water at a pressure drop of 1 psi (0.07 bar) through the valve.

*S* - Effective area [$mm^2$]. It corresponds to the cross-section of the orifice at the same flow rate through the element.

Using these coefficients is possible to calculate the so-called flow conductance of a pneumatic system, which will be described later. All of these flow coefficients can be recalculated among themselves. This can be performed, for example, by the diagram in Fig. 11.42, which is provided by SMC in their documentation [51].

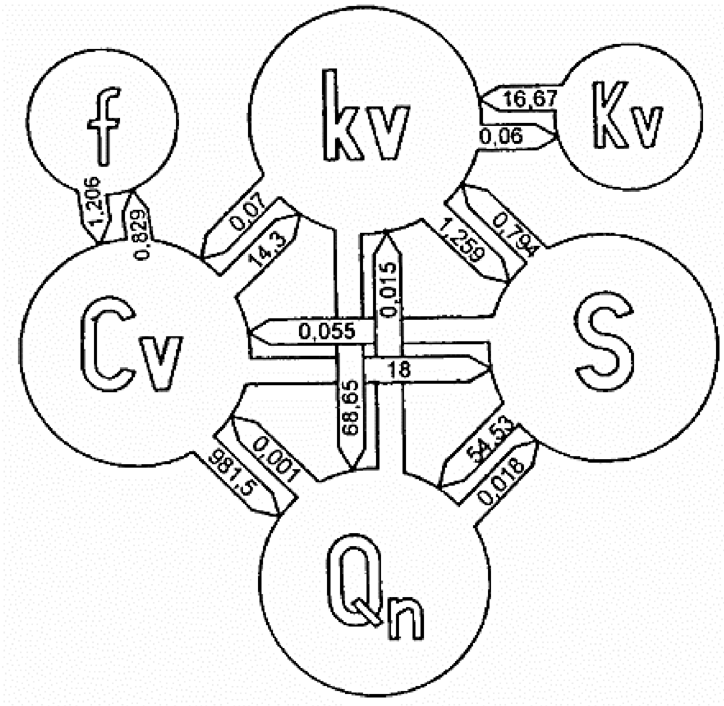



Fig. 11.42 Conversion of coefficients by SMC company [51]

In addition to these coefficients, the catalogue may also contain the so-called Sonic Conductance $C$ [$dm^3 \cdot s^{-1} \cdot bar^{-1}$] and the critical pressure ratio $b$. The measurement of these values is also described according to ISO 6358 and it can be concluded that these values are used for the most accurate calculation of the air flow through the valve and are used, for example, in modelling the dynamics of a pneumatic system.

According to the converter on the Emerson website [52], the conversion to normal nominal flow is:

$$Q_n = 216 \cdot C\,. \tag{11.8}$$

The above coefficients are commonly specified in the datasheets of the individual elements, but it is problematic to determine the flow conductance of hoses. This must be determined from the diagrams provided by manufacturers, e.g., Festo [53], see Fig. 11.43. The Emerson website [52] contains, among other things, a calculation tool for determining the hose flow conductance.

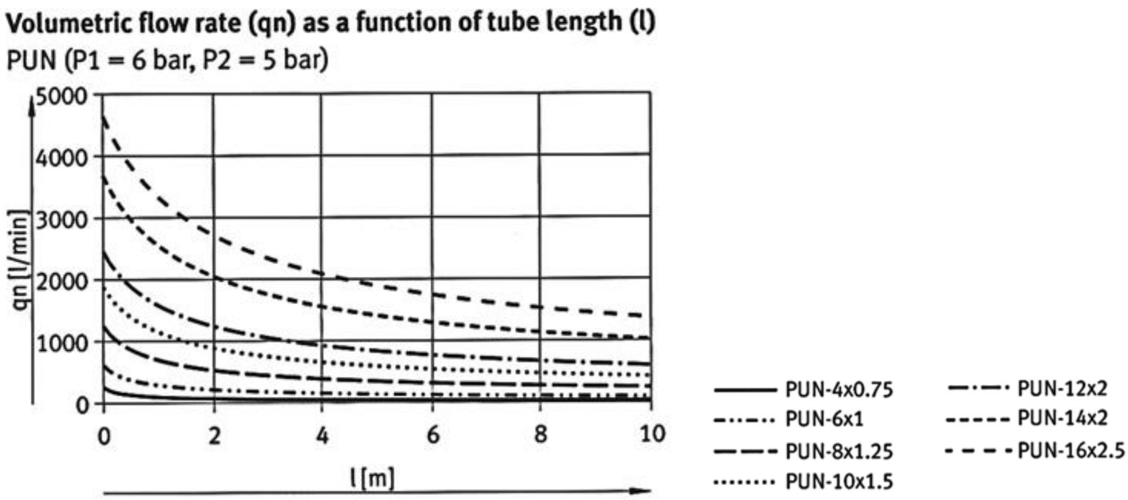



Fig. 11.43 Hose flow conductance [53]

A similar diagram is also provided by SMC company [51], but from this diagram, it is possible to find the effective area, which then needs to be converted to the nominal flow Fig. 11.44.

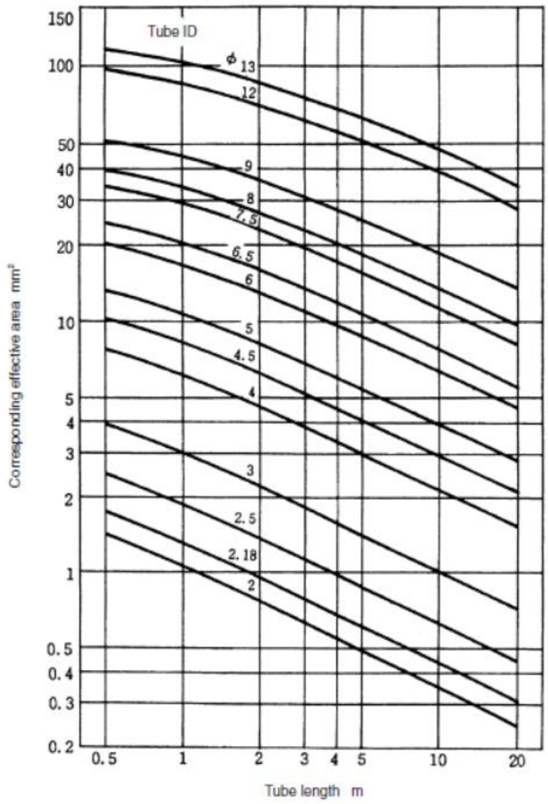



Fig. 11.44 Corresponding effective area of hoses [51]

In addition to the individual system elements and their hose connections, the fittings should not be overlooked when calculating the flow conductance. The flow coefficients of fittings of different diameters can be obtained from Tab. 11.3.

Tab. 11.3 Flow coefficients of fittings

| Nominal clearance of fittings $d$ [mm] | Flow coefficient $K_V$ [$m^3 \cdot h^{-1}$] | Normal nominal flow $Q_N$ [$dm^3 \cdot min^{-1}$(ANR)] |
|---|---|---|
| 2 | 0.21 | 230 |
| 2.5 | 0.33 | 367 |
| 4 | 0.85 | 940 |
| 6 | 1.92 | 2100 |
| 8 | 3.45 | 3700 |
| 10 | 5.33 | 5850 |
| 12 | 7.67 | 8440 |
| 16 | 13.63 | 14960 |
| 20 | 21.32 | 24500 |

**Composite conductance**

After choosing the elements and finding their size (i.e., flow coefficient) from the catalogue, it is possible to check the system flow conductance. In most cases, the elements of the system are arranged in series between the source and the motor Fig. 11.45.

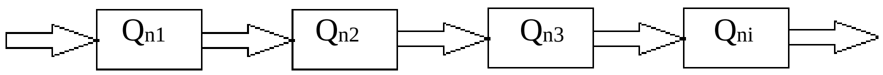



Fig. 11.45 Serial arrangement of elements

For this case, it is possible to use the following formula in order to calculate the so-called equivalent flow rate $Q_{ekv}$ (system flow conductance):

$$Q_{ekv} = \sqrt{\frac{1}{\sum_{i=1}^{n} \frac{1}{Q_{ni}^2}}}, \qquad (11.9)$$

where $Q_{ekv}$ [dm$^3$ · min$^{-1}$(ANR)] is equivalent flow rate of system.

The same can be applied if the size of the elements is defined by another coefficient, e.g. flow coefficient $Kv$:

$$Kv_{ekv} = \sqrt{\frac{1}{\sum_{i=1}^{n} \frac{1}{Kv_{ni}^2}}}, \qquad (11.10)$$

where $Kv_{ekv}$ [m$^3$ · h$^{-1}$] is equivalent flow coefficient.

In case the elements are arranged in parallel (e.g. double air ducts) Fig. 11.46, the equivalent flow rate of this system part is calculated by simply summing the flow rates of its individual parts, e.g., equations (11.11) or (11.12).

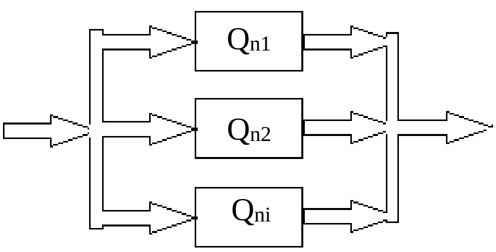



Fig. 11.46 Parallel arrangement of elements

Calculation of equivalent flow rate $Q_{ekv}$ for a parallel arrangement of elements:

$$Q_{ekv} = \sum_{i=1}^{n} Q_{ni} \,. \tag{11.11}$$

Calculation of equivalent flow coefficient $Kv_{ekv}$ for a parallel arrangement of elements:

$$Kv_{ekv} = \sum_{i=1}^{n} Kv_{ni} \,. \tag{11.12}$$

Finally, it is necessary to check if the system flow conductance $Q_{ekv}$ is higher compared to the required flow to the motor $Q_{nM}$.

$$Q_{ekv} \geq Q_{nM} \tag{11.13}$$

### 11.5.1 An example of pneumatic circuit calculation and element selection

Select elements of a basic pneumatic circuit and check their flow capacity. The cylinder is loaded with a force $F$ = 200 N during piston rod extension; the required piston rod extension velocity is $v$ = 0.6 m·s$^{-1}$. The working pressure set on the pressure regulator is $p$ = 5 bar. Consider a pressure drop of $\Delta p$ = 1 bar for air flow through the line and elements. Hose lengths $l_1 = l_2$ = 1 m.

First, the cylinder size is calculated according to the procedure described in chapter 11.2.

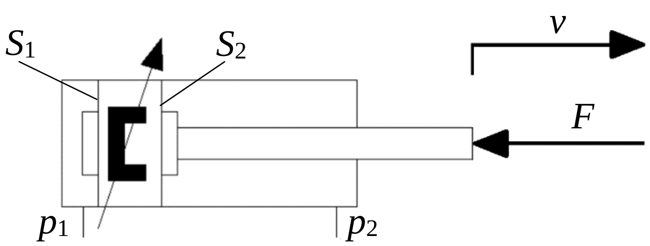



Fig. 11.47 Figure for cylinder calculation

The calculation is based on the equation of force balance on the piston:

$$p_1 \cdot A_1 = p_2 \cdot A_2 + F + F_{fr} \,. \tag{11.14}$$

In the equation, $p_1$ is the pressure on the piston area $A_1$ during movement, i.e. the working pressure set on the pressure regulator reduced by the supposed pressure drop $p_1 = p - \Delta p$. The pressure drop of 1 bar is supposed. We will suppose that the pressure on the other piston side (in the piston rod chamber) $p_2$ = 0. $F_{fr}$ is the frictional force generated by the friction of the piston seal and piston rod seal. It can be replaced by multiplying the loading force by coefficient a = (0.1 ÷ 0.3), i.e., $F_{fr} = F \cdot a$. Then the force balance equation is in the form:

$$(p - \Delta p) \cdot A_1 = F \cdot (1 + a) \,. \tag{11.15}$$

From the equation (11.16), it is possible to calculate the required piston area $A_1$:

$$A_1 = \frac{F \cdot (1 + a)}{p - \Delta p} = \frac{200 \cdot (1 + 0.2)}{(5 - 1) \cdot 10^5} = 0.0006\, m^2 \,. \tag{11.16}$$

Subsequently, the required piston diameter $D_r$ is:

$$D_r = \sqrt{4 \cdot \frac{A_1}{\pi}} = \sqrt{4 \cdot \frac{0.0006}{\pi}} = 0.028\, m \,. \tag{11.17}$$

It is now possible to select the cylinder of the nearest greater diameter. From the catalogue of standardized cylinders, e.g., [54] a cylinder with a diameter $D$ = 32 mm is selected.

Furthermore, as far as sizing and checking the cylinder is concerned, depending on other conditions such as stroke, there is a check for buckling strength, damping or the amount of lateral force, but this is not part of this example.

Besides the cylinder selection, the aim of this example is to design other elements and to check the selection, i.e., to calculate the flow conductance. First, it is necessary to calculate the required air flow $Q_{nM}$ into the cylinder converted to normal conditions. The absolute working pressure must be included in the conversion; the normal atmospheric pressure is 1 bar. The operating temperature can be considered 20 °C, which is also the normal temperature and therefore, the temperature will not affect the calculation:

$$Q_{nM} = v \cdot \frac{\pi \cdot D^2}{4} \cdot \frac{p_{abs}}{p_n} = 0.6 \cdot \frac{\pi \cdot 0.032^2}{4} \cdot \frac{6}{1} = 2.9 \cdot 10^{-3}\, m^3 \cdot s^{-1} (ANR) \,, \tag{11.18}$$

$$Q_{nM} = 174\, dm^3 \cdot min^{-1} (ANR) \,.$$

To achieve the required velocity, the elements between the compressed air source (compressor with air receiver or central distribution) and the cylinder must therefore let through a minimum of 174 normal litres per minute. But the problem is that the nominal flow of all elements, which is stated in the catalogue, is determined at a pressure of 6 bar at the inlet and 5 bar at the outlet. Because of the given pressure of 5 bar and the assumed pressure drop of 1 bar, the required flow rate has to be multiplied by the correction coefficient $c_f$ given by SMC [51] in its documents, see Fig. 11.48.

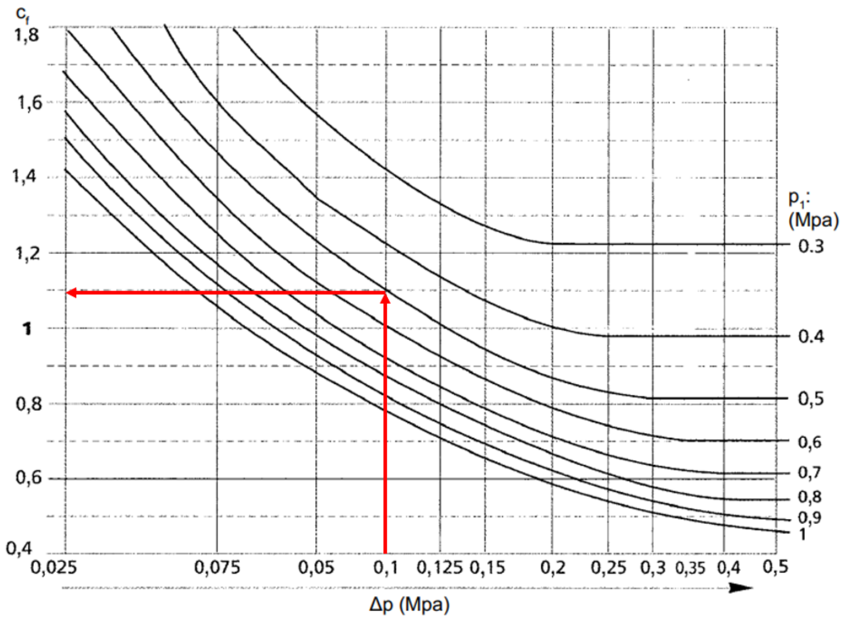



Fig. 11.48 Corrective flow coefficient $c_f$ [51]

The correction coefficient $c_f$ value is 1.1, and therefore, the system flow conductance must be higher compared to the corrected flow rate $Q_{Mc}$:

$$Q_{Mc} = Q_{nM} \cdot c_f = 174 \cdot 1.1 = 191.4 \ dm^3 \cdot min^{-1}(ANR)\,. \tag{11.19}$$

From the point of view of controlling the system flow conductance, the elements that are lined up between the compressed air source and the cylinder working chamber should be included in the calculation, as indicated in red in the diagram in Fig. 11.49.

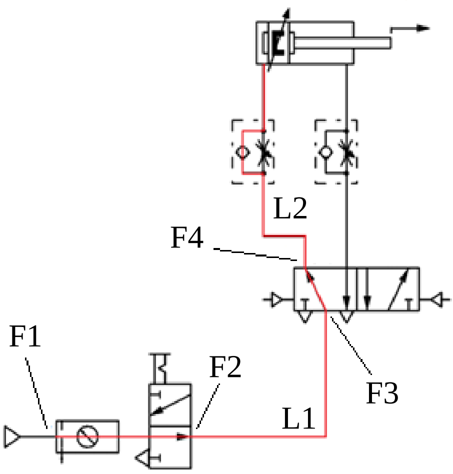
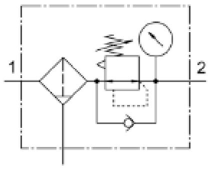



Fig. 11.49 Basic pneumatic circuit diagram

Since the elements are arranged in series, the size of the elements must be chosen several times larger compared to the required flow rate. The first elements are the filter regulator (Fig. 11.50) and the 3/2 mechanically operated switching valve (Fig. 11.51), which can be mounted directly on the filter by means of a connecting element.

**filter regulator**
**MS4-LFR-1/8-D6-CRM-AS**
Part number: 529160

FESTO



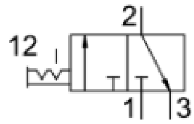

## Data sheet

| Feature | Value |
|---|---|
| Size | 4 |
| Series | MS |
| Actuator lock | Rotary knob with detent<br>can be closed with accessories |
| Mounting position | Vertical +/-5° |
| Grade of filtration | 5 µm |
| Condensate drain | Manually rotating |
| Design | Filter regulator with pressure gauge |
| Max. condensate volume | 19 ml |
| Controller function | Output pressure constant<br>With secondary venting<br>With return flow function |
| Bowl guard | Plastic bowl guard |
| Symbol | 00991589 |
| Pressure gauge (ANALOG) or Pressure display (DIGITAL) | With pressure gauge |
| Operating pressure | 0.8 bar ... 14 bar |
| Pressure regulation range | 0.3 bar ... 7 bar |
| Max. pressure hysteresis | 0.25 bar |
| Standard nominal flow rate | 900 l/min |
| Operating medium | Compressed air to ISO 8573-1:2010 [-:4:-]<br>Inert gases |

Fig. 11.50 Technical data of the filter-regulator [55]

**On/off valve**
**MS4-EM1-1/8-S**
Part number: 541263

FESTO




## Data sheet

| Feature | Value |
|---|---|
| Design | Rotary gate valve |
| Type of actuation | Manual |
| Exhaust-air function | Without flow control option |
| Type of piloting | Direct |
| Symbol | 00991670 |
| Valve function | 3/2 double solenoid |
| Operating pressure | 0 bar ... 14 bar |
| C value | 4.6 l/sbar |
| b value | 0.51 |
| Standard nominal flow rate | 1200 l/min |
| Operating medium | Compressed air to ISO 8573-1:2010 [7:4:4]<br>Inert gases |

Fig. 11.51 Technical data of 3/2 switching valve [56]

Since no specifying or limiting conditions (e.g., temperature) have been defined, a universal directional valve (B52 - bistable in version 5/2) can be selected Fig. 11.52.

## Data sheet

Function
2x3/2C, 2x3/2U, 2x3/2H
5/2-way, monostable
5/2-way, bistable
5/3C, 5/3U, 5/3E

Width 14 mm

Flow rate
500 ... 780 l/min

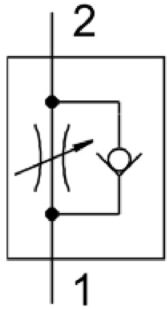
| General technical data | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Valve function | | | T32-A | | | T32-M | | | M52-A | B52 | M52-M | P53 | | |
| Normal position | | | C[1] | U[2] | H[4] | C[1] | U[2] | H[4] | – | – | – | C[1] | U[2] | E[3] |
| Pneumatic spring return | | | Yes | | | No | | | Yes | – | No | No | | |
| Mechanical spring return | | | No | | | Yes | | | No | – | Yes | Yes | | |
| Vacuum operation at port 1 | | | No | | | Yes | | | No | Yes | | | | |
| Vacuum operation at port 3/5 | | | Yes | | | | | | | | | | | |
| Design | | | Piston spool | | | | | | | | | | | |
| Lap | | | Overlap | | | | | | | | | | | |
| Sealing principle | | | Soft | | | | | | | | | | | |
| Actuation type | | | Pneumatic | | | | | | | | | | | |
| Type of control | | | Direct | | | | | | | | | | | |
| Flow direction | | | Reversible with restrictions | | | Reversible | | | Reversible with restrictions | Reversible | Reversible | Reversible | | |
| Exhaust air function | | | Can be throttled | | | | | | | | | | | |
| Type of mounting | | | Optionally via through-holes[6] or on manifold rail | | | | | | | | | | | |
| Mounting position | | | Any | | | | | | | | | | | |
| Standard nominal flow rate | | [l/min] | 650 | 600 | 650 | 550 | 500 | | 780 | | | 650 | 600 | |
| Switching time on/off | | [ms] | 6/19 | | | 9/13 | | | 12/22 | – | 12/32 | 8/30 | | |
| Changeover time | | [ms] | – | | | | | | | 6 | – | 16 | | |
| Width | | [mm] | 14 | | | | | | | | | | | |
| Connection | 1, 2, 3, 4, 5 | | G1/8 | | | | | | | | | | | |
| | 12, 14 | | M5 | | | | | | | | | | | |
| Product weight | | [g] | 81 | | | 77 | | | 75 | 81 | 67 | 81 | | |
| Corrosion resistance class CRC[5] | | | 2 | | | | | | | | | | | |

Fig. 11.52 Technical data of the 5/2 directional valve [57]

The velocity control will be realized by throttle valves Fig. 11.53 for mounting in the cylinder thread. These designs of the combinate check and throttle valves are usually smaller in size (the nominal flow) than in-line valves. If, after calculation, the system flow conductance is less than the required, it may be sufficient in some cases to replace the throttle valve with the in-line version. For this case, we choose a valve with 1/8" thread (thread in the cylinder) and 8 mm hose connection.

**One-way flow control valve**
**GRLA-1/8-QS-8-MF-D**
Part number: 537076

FESTO




## Data sheet

| Feature | Value |
|---|---|
| Valve function | Flow control non-return function |
| Pneumatic connection 1 | QS-8 |
| Pneumatic connection 2 | G1/8 |
| Type of mounting | Screw-in<br>With external thread |
| Standard nominal flow rate in flow control direction | 475 l/min |
| Standard nominal flow rate in non-return direction | 325 l/min ... 500 l/min |
| Ambient temperature | -10 °C ... 60 °C |
| Maritime classification | See certificate |
| Mounting position | Any |
| Symbol | 00991452 |
| Operating pressure for entire temperature range | 0.2 bar ... 10 bar |
| Standard flow rate in flow control direction 6 -> 0 bar | 720 l/min |
| Standard flow rate in non-return direction at 6 -> 0 bar | 610 l/min ... 760 l/min |
| Operating medium | Compressed air as per ISO 8573-1:2010 [7:4:4] |
| Information on operating and pilot media | Operation with oil lubrication possible (required for further use) |
| LABS (PWIS) conformity | VDMA24364-B1/B2-L |
| Storage temperature | -10 °C ... 40 °C |
| Temperature of medium | -10 °C ... 60 °C |
| Nominal tightening torque | 3 Nm |
| Tolerance for nominal tightening torque | ± 10% |
| Product weight | 32 g |
| Note on materials | RoHS-compliant |
| Seals material | NBR |
| Hollow bolt material | Wrought aluminum alloy<br>Anodized |
| Releasing ring material | POM |
| Material of adjusting screw | Brass |
| Swivel joint material | Die-cast zinc |

Fig. 11.53 Technical data of combined check and throttle valve [58]

As already indicated, the hose will have an outer diameter of 8 mm and an inner diameter of 5 mm. To find the flow conductivity of the hose (the length of 1 m and the inner diameter of 5 mm) we can use, for example, the dependencies in Fig. 11.54.

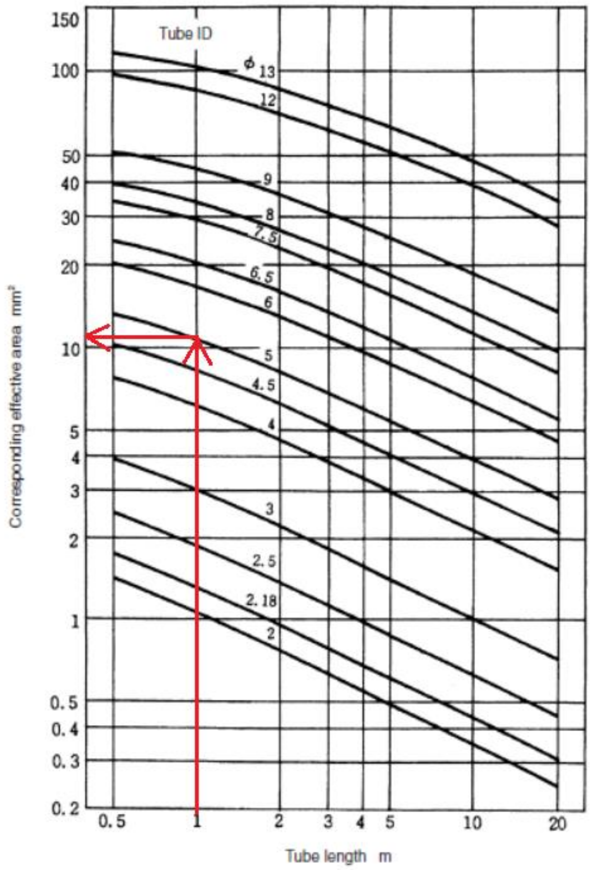



Fig. 11.54 Effective area of hose *S*

The effective area of the hose is $S$ = 11 mm$^2$. To convert to the nominal flow, it is necessary to multiply it by the constant of 54.53 (see Fig. 11.42.). The flow coefficient of the hose $Q_H$ can be calculated as follows:

$$Q_H = S \cdot 54.53 = 11 \cdot 54.53 = 600\ dm^3 \cdot min^{-1} (ANR)\ . \tag{11.20}$$

In addition to the above-mentioned elements and the hose, there are also connection fittings in the air flow way to the cylinder. One at the inlet to the filter regulator unit, the second at the outlet from the 3/2 switching valve (the connection of these two elements does not need to be considered - the connection plate does not affect the throughput), the third fitting at the inlet and the fourth at the outlet from the 5/2 directional valve. The fitting on the throttle valve does not need to be considered. The fitting is a fixed part of the valve, and the catalogue already specifies the flow coefficients of the valve, including the fitting. All elements have a 1/8" thread and a fitting inner diameter of approximately 4 mm. The flow coefficients of fittings are available in Tab. 11.3 above.

Tab 11.4 gives an overview of all the selected elements and their size determined from the catalogue. The flow coefficient of the hoses is determined from the graph.

Tab 11.4 The sizes of selected elements

| ***Element*** | ***Flow coefficient*** **[dm³ · min⁻¹(ANR)]** |
|---|---|
| Air treatment MS4-LFR-1/8-D6-CRM-AS | $Q_{AT} = 900$ |
| Switching valve 3/2 MS4-EM1-1/8-S | $Q_{SV} = 1200$ |
| Directional valve  5/2 VUWG-L14-B52-Q6-QN | $Q_{DV} = 780$ |
| Combined check and throttle valve GRLA-1/8-QS-8-MF-D | $Q_{CV} = 325$ |
| Hose TU Ø8 / Ø5, length 1 m, 2 pcs | $Q_H = 600$ |
| Fitting  Ø4-1/8“, 4 pcs | $Q_F = 940$ |

Since all the elements through which the air flows from the source to the cylinder are arranged in series, the equivalent flow, i.e. (*10.9*). The system flow conductance $Q_{ekv}$ is calculated using the following equation:

$$Q_{ekv} = \sqrt{\frac{1}{\frac{1}{{Q_{AT}}^2} + \frac{1}{{Q_{SV}}^2} + \frac{1}{{Q_{DV}}^2} + \frac{1}{{Q_{CV}}^2} + \frac{2}{{Q_H}^2} + \frac{4}{{Q_F}^2}}},$$

$$Q_{ekv} = \sqrt{\frac{1}{\frac{1}{900^2} + \frac{1}{1200^2} + \frac{1}{780^2} + \frac{1}{325^2} + \frac{2}{600^2} + \frac{4}{940^2}}}, \tag{11.21}$$

$$Q_{ekv} = 209\ dm^3 \cdot min^{-1}(ANR)\,.$$

Now it is necessary to compare the flow conductance $Q_{ekv}$ with the required flow rate $Q_{Mc}$ to motor:

$$Q_{ekv} \geq Q_{Mc}\,, \tag{11.22}$$

$$209 > 191.4\,.$$

Since the system flow conductance is higher compared to the required flow into the cylinder, or its value corrected for other pressure conditions, it can be concluded that the elements are dimensioned sufficiently, and the velocity of piston rod $v = 0.6\ \mathrm{m \cdot s^{-1}}$ will be achieved.

Should this not be the case, other, larger elements should be selected.

# References

[1] ČSN 01 3624. *Značky pro kreslení hydraulických a pneumatických schémat*. Praha: Československá státní norma, 1985.

[2] ČSN ISO 1219-1. *Hydraulika a pneumatika - Grafické značky a obvodová schémata - Část 1: Grafické značky*. Praha: Česká technická norma, 1999.

[3] ISO 1219-1:2012. *Fluid power systems and components — Graphical symbols and circuit diagrams — Part 1: Graphical symbols for conventional use and data-processing applications.* Geneve: International Organization for Standardization, 2012.

[4] KOPÁČEK, Jaroslav and Bohuslav PAVLOK. *Tekutinové mechanismy.* Ostrava: VSB – Technical University of Ostrava, 2016. ISBN 978-80-248-3930-1.

[5] GROLLIUS, Horst-Walter. *Grundlagen der Hydraulik. 4., aktualisierte Aufl.* München: Fachbuchverlag Leipzig im Carl Hanser Verlag, 2008. ISBN 978-3-446-41596-6.

[6] MATTHIES, Hans Jürgen and Karl Theodor RENIUS. *Einführung in die Ölhydraulik. 6., aktualisierte und erw Aufl.* Wiesbaden: Vieweg + Teubner, 2008. Studium (Vieweg + Teubner). ISBN 978-3-8351-0238-5.

[7] PAVLOK, Bohuslav. *Hydraulické prvky a systémy. Díl 2. Řídicí prvky hydrostatických systémů příslušenství hydraulických obvodů.* Ostrava: VSB – Technical University of Ostrava, 2008. ISBN 978-80-248-1827-6.

[8] WILL, Dieter a Norbert GEBHARDT, ed. *Hydraulik: Grundlagen, Komponenten, Schaltungen. 4.* neu bearb. Aufl. Berlin: Springer, 2008. ISBN 978-3-540-79534-6.

[9] PIVOŇKA, Josef et al. Tekutinové mechanismy. Praha: SNTL – Nakladatelství technické literatury, 1987.

[10] KOPÁČEK, Jaroslav et al. *Projektování hydraulických systémů*. Ostrava: Dům techniky ČSVTS Ostrava, 1980.

[11] KUČÍK, Pavol, Igor STRÁŽOVEC a Peter KRIŠŠÁK. *Hydraulický prenos energie: mobilné pracovné stroje.* Žilinská univerzita, 2000. ISBN 80-7100-725-0.

[12] PAVLOK, Bohuslav. *Hydraulické prvky a systémy. Díl 1. Kapaliny v hydraulických mechanismech. Hydrostatické převodníky.* Ostrava: VSB – Technical University of Ostrava, 2005. ISBN 80-248-0857-9.

[13] PAVLOK, Bohuslav and Lumír HRUŽÍK. *Základy hydrauliky* [online]. Ostrava: VSB – Technical University of Ostrava, 2009. www.vsb.cz [cit. 2022-12-1]. Available from: https://cdrive.vsb.cz/index.php/s/gKZjIdp8uiti7uh

[14] BAROŠKA, Ján. *Hydrostatické mechanizmy.* Žilina: Hydropneutech, 2012. ISBN 978-80-970897-2-6.

[15] PARKER. *2 Way Flow Control Valve* [online]. www.ph.parker.com [cit. 2022-12-1]. Available from: https://www.parker.com/content/dam/Parker-com/Literature/Industrial-Systems-Division-Europe/Catalogues/Industrial-Valves-UK/05/GFG-UK.pdf Parker Hannifin Corporation, 2022.

[16] HYDAC Electronic. *Portable data recorder HMG 3000 – User manual* [online]. www.hydac.com [cit. 2022-12-1]. Available from: https://www.hydac.com/uploads/tx_userhysoftware/User_manual_HMG_3000_V03_E_2012-09-21.pdf Saarbrücken: HYDAC Electronic GmbH, 2012.

[17] HYDAC International. *Pressure Transmitter HDA 4400* [online]. www.hydac.com [cit. 2022-12-1]. Available from: https://www.hydac.com/shop/media/catalog/crossbase/PRD_DOC_PRO/PRD_DOC_PRO_18305-00001__SEN__AIN__V6.pdf Saarbrücken: HYDAC Electronic GmbH, 2002.

[18] HYDAC International. *Electronic Flow Rate Transmitter with HSI-Sensor Recognition* [online]. www.hydac.com [cit. 2022-12-1]. Available from: https://www.hydac.com.au/pub/media/productattach/e/_/e.18.334-2-11-13_evs3100.pdf Saarbrücken: HYDAC Electronic GmbH, 2011.

[19] BOSCH REXROTH. *Fluidní technika: průmyslová hydraulika – základy. 3. doplněné vydání.* Brno: Bosch Rexroth, spol. s.r.o. (školení a poradenství), 2012.

[20] HYDAC. *Hydraulics – basics and components.* Sulzbach/Saar: HYDAC Service GmbH, training Center 2012.

[21] DREXLER, Peter et al. Hydraulik Trainer – Band 3 - *Projektierung und Konstruktion von Hydroanlagen.* Wertheim/Wartberg: Mannesmann Rexroth GmbH, 1988.

[22] EXNER, Herbert et al. *Hydraulics: basics principles and components. 3rd edition/revision and editing H Kempf.* Würzburg: Bosch Rexroth AG, Drive & Control Academy, 2011. The hydraulic trainer, volume 1. ISBN 978-3-9814879-3-0.

[23] OLAER. *Hydropneumatický vakový akumulátor OSP 035 – Všeobecné informace - použití* [online]. www.cdn.kovaz.cz [cit. 2022-12-1]. Available from: https://cdn.kovaz.cz/data/user-content/katalogy/Akumul%C3%A1tory%20a%20chladi%C4%8De/Souvisej%C3%ADc%C3%AD%20podklady%20k%20akumul%C3%A1tor%C5%AFm/Pou%C5%BEit%C3%AD%20vakov%C3%BDch%20akumul%C3%A1tor%C5%AF.pdf

[24] PARKER. *Accumulators, Accessories and Spares* [online]. www.seall.cz [cit. 2022-12-1]. Available from: http://www.seall.cz/uploads/files/european-accumulator-catalogue/HY10-4004-GB_European_Accumulator_Catalogue_240114_v3_.pdf Parker Hannifin Corporation, 2014.

[25] MURRENHOFF, Hubertus. Fundamentals of fluid power: lecture notes: translation of the 7. revised German edition 2012. Part 1, Hydraulics. Aachen: Shaker Verlag, 2017. Reihe Fluidtechnik, 007. ISBN 978-3-8440-2826-3.

[26] HYDAC International. *Bladder Accumulators – Standard design* [online]. www.hydac.com [cit. 2022-12-1]. Available from: https://www.hydac.com/shop/media/catalog/crossbase/PRD_DOC_PRO/PRD_DOC_PRO_3201-00001__SEN__AIN__V31.pdf Sulzbach/Saar: HYDAC Technology GmbH, 2011.

[27] KOPÁČEK, Jaroslav. *Pohony a převody*. Ostrava: VSB – Technical University of Ostrava, 2012. ISBN 978-80-248-2780-3.

[28] PAVLOK, Bohuslav, Lumír HRUŽÍK and Miroslav BOVA. *Hydraulická zařízení strojů – učební text* [online]. Ostrava: VSB – Technical University of Ostrava, 2007. www.vsb.cz [cit. 2022-12-1]. Available from: https://cdrive.vsb.cz/index.php/s/K0ABHFEN8xPYcIX

[29] HRUŽÍK, Lumír, Kamil Fojtášek and Adam Bureček. *Pohony a převody – Řešené příklady do cvičení.* Ostrava: VSB – Technical University of Ostrava, 2019. www.vsb.cz [cit. 2022-12-1]. Available from: https://cdrive.vsb.cz/index.php/s/TCBqW70dEBHLRMm

[30] PARKER. *PWO Water/oil cooler* [online]. www.parker.com [cit. 2022-12-1]. Available from: https://www.parker.com/Literature/Accumulator%20&%20Cooler%20Division%20-%20Europe/Cooler%20Literature/PWO_Water%20Oil%20Coolers,%20EMDC.%20HY10-6010.UK.pdf Parker Hannifin Corporation, 2017.

[31] PARKER. *CADE – Cylinder and Accumulator Division Europe Distribution Catalogue 2022* [online]. www.parker.com [cit. 2022-12-1]. Available from: https://www.parker.com/parkerimages/Parker.com/Divisions-2011/Accumulator%20and%20Cooler%20Division%20-%20Europe/Support/distributioncatalogue22.pdf Parker Hannifin Corporation, 2021.

[32] PARKER. *SWO Water/oil cooler* [online]. www.parker.com [cit. 2022-12-1]. Available from: https://www.parker.com/Literature/Accumulator%20&%20Cooler%20Division%20-%20Europe/Cooler%20Literature/SWO_Water%20Oil%20Coolers,%20EMDC.%20HY10-6011.UK.pdf Parker Hannifin Corporation, 2013.

[33] ČSN 11 9101. *Hydrostatické mechanismy. Montážní otvory pro ventily pro zašroubování do otvorů s upevňovacím závitem pro jmenovitý tlak do 32 MPa. Připojovací rozměry*. Praha: Československá státní norma, 1985.

[34] BOSCH REXROTH. *Hydraulic cylinder Tie rod design – Series CDT3* [online]. www.boschrexroth.com [cit. 2022-12-1]. Available from: https://www.boschrexroth.com/cs/cz/media-details/bfba4841-51ea-4275-8ba9-39f162032c8b Lohr am Main: Bosch Rexroth AG, Industrial Hydraulics, 2022.

[35] VYBO Electrics. *Elektromotor 2LC160L-4* [online]. www.vyboelectric.cz [cit. 2022-12-1]. Available from: https://vyboelectric.cz/obchod/elektromotor-2lc160l-4-15kw-ie2-1470-ot-min/

[36] BOSCH REXROTH. *Axial piston fixed pump AA2FO* [online]. www.boschrexroth.com [cit. 2022-12-1]. Available from: https://www.boschrexroth.com/cs/cz/media-details/0d03f624-3f73-46d8-83c1-bc164b3f6e0f Southchase Court: Bosch Rexroth Corporation, Mobile Applications, 2014.

[37] ARGO HYTOS. *4/2 and 4/3 Directional Control Valve, Solenoid Operated RPE3-06* [online]. www.argo-hytos.com [cit. 2022-12-1]. Available from: https://www.argo-hytos.com/products/valves/directional-control-valves/rpe3-06.html , 2016.

[38] ARGO HYTOS. *Check Valve, Poppet-type, Modular MVJ3-06* [online]. www.argo-hytos.com [cit. 2022-12-1]. Available from: https://www.argo-hytos.com/fileadmin/user_upload/Datasheet_MVJ3-06_ha5018_EN.pdf , 2016.

[39] ARGO HYTOS. *Pressure Relief Valve, Poppet Type, Direct Acting VPP1-06(10)* [online]. www.argo-hytos.com [cit. 2022-12-1]. Available from: https://www.argo-hytos.com/fileadmin/user_upload/Datasheet_VPP1-06_10_ha5061_EN.pdf , 2019.

[40] BOSCH REXROTH. *Pressure relief valve, pilot-operated – Type ZDB and Z2DB* [online]. www.boschrexroth.com [cit. 2022-12-1]. Available from: https://store.boschrexroth.com/Hydraulics/Valves/Pressure-valves/Pressure-relief-mechanical/PRESSURE-RELIEF-VALVE--Z_R900423274?cclcl=en_US Lohr am Main: Bosch Rexroth AG, Industrial Hydraulics, 2022.

[41] ARGO HYTOS. *Return Filters E 043 – E 072* [online]. www.argo-hytos.com [cit. 2022-12-1]. Available from: https://www.argo-hytos.com/fileadmin/user_upload/Datasheet_E043_E072_20.20_EN_US.pdf , 2018.

[42] KOPÁČEK, Jaroslav. *Pneumatické mechanismy. Díl 1., Pneumatické prvky a systémy*. 2. vydání. Ostrava: VSB – Technical University of Ostrava, 2005. ISBN 80-248-0879-X.

[43] DVOŘÁK, Lukáš. *Vlastnosti tekutin – Sylabus předmětu.* Ostrava: VSB – Technical University of Ostrava, 2010. www.vsb.cz [cit. 2022-12-1]. Available from: https://cdrive.vsb.cz/index.php/s/N6SRCEIWBkEhP2O

[44] ISO 5598:2020. *Fluid power systems and components — Vocabulary*. Geneve: International Organization for Standardization, 2020.

[45] JIS B 0142. *Fluid power systems and components — Vocabulary*. Tokyo: Japanese Standards Association (JSA), 2015.

[46] MURRENHOFF, Hubertus and Oliver REINERTZ. *Fundamentals of fluid power: lecture notes: 2nd revised German edition of 2014*. Part 2, Pneumatics. Aachen: Shaker Verlag, 2014. Reihe Fluidtechnik, 008. ISBN 978-3-8440-3213-0.

[47] ISO 8573-1:2010. *Compressed air — Part 1: Contaminants and purity classes*. Geneve: International Organization for Standardization, 2010.

[48] CAMOZZI COMPETENCE CENTRE. *Pneumatic Automation From Basic Principles to Practical Techniques.* Brescia: Camozzi Automation S.p.A., 2019. www.automation.camozzi.com/ [cit. 2022-12-1]. Available from: https://en.automation.camozzi.com/literature/didactics/didactics.kl

[49] PNEUMAX. *Didactic Handbook – Modern Pneumatics, Components for Pneumatic Automation.* Lurano: PNEUMAX S.p.A., 2015. www.pneumaxspa.com/ [cit. 2022-12-1]. Available from: https://pneumaxspa.com/pneumax-cz/wp-content/uploads/Pneumatics-handbook.pdf

[50] ISO 6358-2:2019. *Pneumatic fluid power — Determination of flow-rate characteristics of components using compressible fluids — Part 2: Alternative test methods.* Geneve: International Organization for Standardization, 2019.

[51] SMC. *SMC Training - Stlačený vzduch a jeho využití*. Brno: SMC Industrial Automation CZ s.r.o.

[52] EMERSON. *Calculation tools* [online]. www.emerson.com [cit. 2022-12-1]. Available from: https://www.emerson.com/catalog/ConfiguratorToolsView?key=CONF-START&catalogId=20051&langId=-1&storeId=20151&additionalParams=%26area%3Dcomputation

[53] FESTO. *Festo Pneumatic Components Catalogue. 2014/06*. Praha: Festo

[54] FESTO. *Válce dle norem DSBC, ISO 15552* [online]. www.festo.com [cit. 2022-12-1]. Available from: https://www.festo.com/cat/cs_cz/data/doc_cs/PDF/CZ/DSBC_CZ.PDF , 2020.

[55] FESTO. *Filter regulator MS4-LFR-1/8-D6-CRM-AS* [online]. www.festo.com [cit. 2022-12-1]. Available from: https://www.festo.com/cz/cs/a/download-document/datasheet/529160 , Festo SE & o KG, 2022.

[56] FESTO. *On / off valve MS4-EM1-1/8-S* [online]. www.festo.com [cit. 2022-12-1]. Available from: https://www.festo.com/gb/en/a/download-document/datasheet/541263 , Festo SE & o KG, 2022.

[57] FESTO. *Pneumatic valves VUWG* [online]. www.festo.com [cit. 2022-12-1]. Available from: https://www.festo.com/media/pim/393/D15000100123393.PDF , Festo SE & o KG, 2017.

[58] FESTO. *One-way flow control valve GRLA-1/8-QS-8MF-D* [online]. www.festo.com [cit. 2022-12-1]. Available from: https://www.festo.com/us/en/a/download-document/datasheet/537076 , Festo SE & o KG, 2022.

**Manufacturer's product catalogues and internet resources:**

Argo-Hytos, Bosch Rexroth, Camozzi, Emerson, Festo, Hydac, Olaer, Parker Hannifin, Pneumax, SMC.