CHAPTER 6
Synthesis of Hydraulic Systems

A design process is a search for an engineering solution which will satisfy some needs [18], [19], [14], [13]. These needs are expressed by a set of requirements listed in an engineering specification. The starting point for any design project is to review the requirements; the designer should have a full understanding of the requirements before proceeding with a solution. Having reviewed all of the machine requirements, the designer then develops a concept of the machine, and uses it as a basis for the solution of the design project.

The task of developing a hydraulic system requires knowledge of all the functions which the system must perform and recognition of the various factors (quantitative constraints) and effects (qualitative constraints) which will provide a boundary for the solution space. Clearly established requirements and a specification of the system must precede any attempts to design a system. To successfully design a system the designer must perform the following tasks:

- Identify system requirements and specification.
- Define the required motions, cycle and timing of various functions
- Decide on the concept of a system:
  - type of circuit - open or closed
  - type of control - valve control, pump/motor control
  - method of control - manual, automatic
  - sequence of operation
  - other factors which are dictated by system tasks
- Represent, using graphical symbols, a hydraulic circuit which will perform all the logic functions - i.e. it will direct the hydraulic power to the output devices in the desired sequence
- Define the loads and relate the loads to the machine cycle
- Decide on operating parameters of the system - i.e. pressure and flow requirements
- Select the output devices - motors, actuators
- Select the pumping units
- Select the type of hydraulic fluid and sizes of hydraulic lines
- Select the control elements and the method of control
- Calculate the power losses in the system and calculate the required power of the prime mover
- Check the thermal balance in the system and select the size of the reservoir
- Select filtering system

- Carry out design audit of the proposed system
  - Check that the system meets functional and power requirements
  - Check that system is well protected against excessive loads
  - Determine fault tree of the system and system redundancies
  - Carry out **F**ailure **M**odes and **E**ffects **A**nalysis (FMEA)
- Decide on the method of manufacturing
- Prepare documentation, system description, maintenance and operation manuals
- Determine the cost of the system

The design process is iterative in nature thus the whole design procedure may be repeated a number of times until a satisfactory system is obtained. The task of preparation of a circuit diagram and some other tasks will not be discussed here as they are outside the scope of this book.

6.1 Selection of basic operating parameters

After completion of the circuit diagram of the hydraulic system which shows flow of hydraulic power in the system, and having information on the loads and operating speeds we may select the basic operating parameters of the system. These parameters are operating pressures and flow rates. The flow rates, depend on the capacity of the flow source i.e. hydraulic pump or pumps, determine the operating speeds of output devices (motors, cylinders), whereas pressures in the system are back effect of external loads. At any instance in the machine cycle, the system power at any location in the system is determined by a product of pressure and flow rate at this location.

By *system working pressure* we understand the pump delivery pressure, which is the sum of effective pressure at the output device (motor or actuator) and its hydro-mechanical losses as well as pressure losses in the hydraulic lines and control valves.

The choice of magnitude of working pressure is affected by factors like:

- buckling strength of actuators
- life of pumps and motors
- recovery of energy by using accumulators
- strength of hydraulic lines (specially flexible hoses) and fittings

Other, also important factors, are:

- cost and availability of components
- weight of elements
- degree of overload, equipment safety factors
- qualifications of maintenance and operating staff

As working pressures increase the system requires a better quality of filtration, assembly, and sealing elements. The modern trend of using high pressure systems due to the higher "power density" of such systems (ratio of power/weight) has a tradeoff in the increased costs of manufacturing of these elements. Currently, nominal working pressures are in the range $16 \div 32\,\mathrm{MPa}$.

6.1.1 Selection of motors and actuators

After a nominal working pressure is selected it is necessary to determine the stroke displacement q_m of the motor, stroke displacement is selected using the following relation:

$$q_m = \frac{2\pi T_{load}}{p_n} k_{hm} \tag{6.1}$$

where:

k_{hm}		factor accounting for losses in the motor, $k_{hm} = 1.1 \div 1.3$
T_{load}	-	maximum load torque on a motor
p_n	-	nominal working pressure

We use the value of stroke displacement q_m to select a hydraulic motor from a catalogue, the selected motor should have a stroke displacement slightly higher than calculated above. In a similar way, the initial selection of a cylinder size is based on the relation:

$$A_{1(2)} = \frac{F_{load}}{p_n} k_{hc} \tag{6.2}$$

where:

k_{hm}		factor accounting for losses in the cylinder, $k_{hc} = 1.2 \div 1.3$
F_{load}	-	maximum force on a cylinder
$A_{1(2)}$	-	required piston area

Areas A_1 and A_2 are used to determine diameters of the piston and the piston rod and then, taking into consideration the required working stroke, to select a standard cylinder from a catalogue. Additional check of the cylinder for strength and especially for buckling of the piston rod should be carried out. In the case when a standard cylinder is not available, the cylinder has to be specially designed.

In the above calculations we account for pressure losses in hydraulic lines and control valves by selecting the appropriate values for factor k_{hm} and/or k_{hc}. This is necessary in the preliminary stages of design as the exact calculation of such losses is only possible when other elements of the system are selected.

6.1.2 Selection of nominal flow rate (pump selection)

Flow Q demand of a hydraulic motor is determined from equation:

$$Q = \frac{q_m n_m}{\eta_{vm}} \tag{6.3}$$

where:

q_m	-	stroke displacement of selected hydraulic motor
n_m	-	specified rotation speed of the motor
η_{vm}	-	catalogue value of hydro-mechanical efficiency of a motor at nominal values of n_m and pn

Flow demand of a cylinder is determined from:

$$Q = A_{1(2)}V_{1(2)} \tag{6.4}$$

where:

$v_{1(2)}$	-	specified velocity of the piston
$A_{1(2)}$	-	active area of the piston

Once the flow demands of the output units are determined we select a fluid supply system (a pump or set of pumps), keeping in mind that its flow delivery must satisfy the flow demands of all output units which are intended to operate simultaneously. When selecting a pump from the catalogue we should check that the proposed rotation speed of the pump is within the manufacturer's recommended speed range, this will assure high efficiency of pump operation. The power output of the prime mover selected to drive the pump should be higher than the nominal power of the pump.

6.1.3 Selection of hydraulic fluid

The function of the hydraulic fluid is to enable transmission of hydraulic power in a directionally flexible, but mechanically stiff manner. The supplementary, but very important function of a hydraulic fluid is to carry locally generated heat (friction, energy conversions, pressure expansions) away to a convenient place (reservoir, heat exchanger) for discharge from the system. Inability to dissipate such heat is a severe limitation in most machines (electric motors, turbines, I.C. engines, etc.). The fluid's ability to do this in hydraulic systems is a significant advantage, leading to smaller and lighter components.

Liquids are fluids which are substantially incompressible and can have a free surface. Gases are fluids which are highly compressible and which cannot support a free surface - i.e. they expand to fill their container. The main fluid properties which affect hydraulic system performance are viscosity, density, bulk modulus, volumetric expansion coefficient, specific heat, thermal conductivity, and vapour pressure.

Density is important, as it affects energy transfer throughout the hydraulic system. All motions require accelerations and retardations of the fluid, through which energy transfers take place. Density appears in various fluid state equations, study of which indicates that density should be minimized.

The density of liquids is affected by both temperature and pressure. It reduces with increasing temperature, and increases with increasing pressure. Within the working temperature range $0 \div 65\,^{\circ}\mathrm{C}$ and operating pressure range $0 \div 20$ MPa liquid density can be regarded as constant.

Table 8. Physical properties of fluid affecting performance of hydraulic systems

Fluid property	**Effect on**			
	Steady-state behaviour	Dynamic behaviour	Thermal balance	Wear
Viscosity	▲	▲	▲	▲
Specific gravity	▲	▲		
Vapour pressure	▲	▲		▲
Gas solubility		▲	▲	▲
Bulk modulus		▲		
Neutralisation No.				▲
Pour point	▲		▲	

The selection of hydraulic fluid is very important for the correct operation of the system, table 8. In addition to the usual criteria of proper viscosity, when selecting fluid one should take into consideration the environmental conditions in which the system will operate, safety requirements (e.g. fire hazard), and chemical interaction with the materials of seals and components. The properties of fluid will also have an effect on the dynamic behaviour of a system. Expected effects of physical properties of fluids on a system are shown in fig. 148, [6].

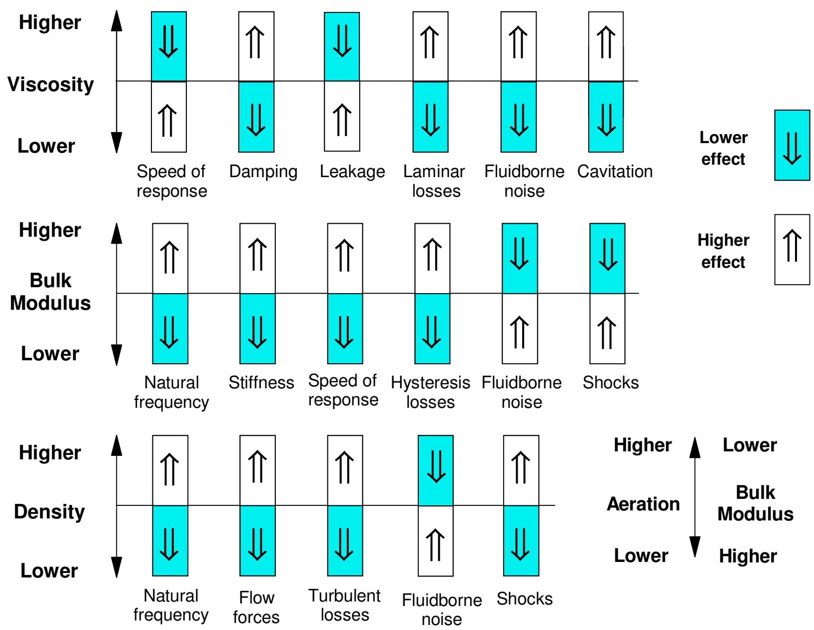

Fig. 148. Expected effects of physical properties of fluid on system dynamics

6.1.3.1 Desirable characteristics of hydraulic fluids

Some desirable characteristics of fluids which are used in hydraulic systems are briefly discussed below:

- *Good lubricity:* lubrication of the working parts of system components is essential. e.g. pistons in pumps and motors; bearing pads of swash-plate pumps; spool type valves; side plates on gear pumps. These are close fit, heavily-loaded, high speed sliding components.
- *Stable chemically and physically::* fluid chemical and physical characteristics should remain unchanged during an extended useful life, and during storage. The fluid in a working hydraulic system is subjected to violent usage - large pressure fluctuations, shock, turbulence, aeration, cavitation, water and particulate contamination, high shear rates, and large temperature variations.
- *Compatibility with system materials*: hydraulic fluid should be compatible with metals used in construction of components, seal materials, filter media etc.. Fluid should not cause rusting, corrosion, or material degeneration.
- *Good thermal conductivity:* an important requirement of the fluid is the ability to carry heat away from working parts. Pressure drops, mechanical friction, fluid friction and leakages all contribute to generation of heat in hydraulic systems. The fluid must carry the generated heat away and be able to readily transfer it to the atmosphere or a heat exchanger. The fluid should have high conductivity, and high specific heat.
- *High bulk modulus:* high bulk modulus is an essential requirement for fast and stiff response. A typical values of bulk modulus for a hydraulic, mineral oil based fluid are $B = 1400 \div 1700$ MPa. The value of bulk modulus is strongly affected by the presence of free air in the fluid (under atmospheric pressure air dissolved in a mineral oil will be approx. 8% of the volume of fluid). It must be remembered that air dissolved in oil will be released when local pressure is lower than atmospheric, this can happen, for example, during flows through throttle valves. Usually, fluid in a hydraulic system may contain between $0.5 \div 5\%$ of free air which lowers effective bulk modulus of the fluid, and increases its viscosity [1] The value of bulk modulus of mineral oils in usual operating conditions (pressure below 40 MPa and temperature below $80°C$) increases slightly with the increase of pressure and decreases with the increase of fluid temperature (thus temperature via bulk modulus can have some affect on the system's dynamic behaviour).
- *Good viscosity characteristics:* Viscosity low enough that power is not lost nor excessive heat generated by shearing, but high enough so that leakage at seals is not excessive. The viscosity of liquids decreases with increasing temperature. Hydraulic fluids need a reasonably high Viscosity Index. Typically, the viscosity of an industrial hydraulic oil will fall to about 25% of its ambient temperature value when the system temperature is within the normal operating rage $50 \div 60\,°$C. The viscosity of liquid increases with pressure, although this effect is not as dramatic as the temperature effect; a typical mineral hydraulic oil will more than double its viscosity when raised in pressure from ambient to 40 MPa, table 9.

Table 9. Viscosity increase with pressure - mineral oils

Pressure in MPa	7.0	15.0	20.0	40.0	60.0
% increase in viscosity	$20 \div 35$	$35 \div 40$	$50 \div 60$	$120 \div 160$	$250 \div 350$

- *Cavitation Resistance:* the Cavitation Number describes the tendency of liquids to cavitate.

$$N_c = \frac{p - p_v}{Y \rho v^2} \tag{6.5}$$

where:

p	-	static pressure
p_v	-	vapour pressure
ρ	-	fluid density
v	-	local fluid velocity
Y	-	constant for oil

- Low N_c indicates a high tendency of liquid to cavitate. Hence we need low p_v, high v, low p. Vapour pressure p_v increases with temperature, hence N_c is temperature dependent. Cavitation causes damage by:
  - tearing metal from surfaces, when cavitation intensity is high.
  - causing fatigue from lower intensity cavitation
  - destroying protective films on surfaces, allowing then corrosion and wear.
- *Low Inertia:* the fluid must be accelerated and retarded at very high rates, thus low fluid density is advantageous.
- *Fire Resistance:* there is always a danger of fire when flammable fluid is used in a hydraulic system. Fluid leaks, hydraulic component or line failures and system overheating, e.g. due to incorrect viscosity of fluid, can potentially cause a fire. The high fluid flash point is one criteria of fluid suitability for a high fire risk application. The common hydraulic liquids are petroleum derivatives, and consequently they burn vigorously once past the flash point. For critical applications fire resistant hydraulic fluids (FRF), which have very high fire resistance, can be used. These are:
  - water-in-oil emulsions
  - water glycol solutions
  - synthetic fluids (chlorinated hydrocarbons, phosphate esters)
  - blends of synthetic fluids and petroleum oils.

When a fire resistant fluid is proposed for use in an existing system which uses a mineral-base fluid, the first step in determining the fluid/component compatibility, is to compare the properties of the new fluid to those of the mineral-base fluid. This comparison will provide an intuitive feeling as well as factual data concerning the effect of the fluid on the system and on individual components. If the fluid has obvious advantages in fire resistance and other properties, then the second step is to implement the component alterations necessary to eliminate any serious incompatibility with fluid. Often these alternatives cannot be determined by analysis and then component testing is necessary. While this will eventually yield a satisfactory design, it will usually increase the cost associated with the testing due to lost time and damaged equipment.

Once component compatibility has been established, the next step is an overall evaluation of system performance, culminating with the successful field conversion of an existing system. If the fluid is proven to be reliable in an existing system, new systems can then be designed specifically to employ fire-resistant fluids. The effects of properties of water based fluids on hydraulic system components are summarized in tables 10 and 11, and the effects of synthetic fluids on system components are presented in tables 12 and 13, [6].

The effect of the component's working environment may have a considerable effect on fluid selection. The problem is compounded since temperature has a similar effect on the specific gravity and the vapour pressure of the fluid - both of which also exhibit extreme effects on pump and plumbing design. Environmental conditions such as, for example, dusty or dirty conditions, extremely humid climate must be considered in the selection of a fluid, These conditions must be considered in the final evaluation of the fluid/component compatibility even though some limitations may result from the lack of familiarity with the physical properties of fire-resistant fluids and the scarcity of a working knowledge of their use.

Most of the synthetic or water based fluids have serious drawbacks when compared to mineral oil, including some combination of:

- high cost
- poor lubricity
- low N_c due to high p_v
- high density and hence inertia
- low Viscosity Index (VI)
- incompatibility with common hydraulic system materials etc.

Table 10. Effects of water-based fluids on component design

Effect on	**Fluid property**		
	Corrosivity	Lubricity	Thermal Conductivity
High temperature applications	Low	Low	High
Low temperature applications	Low	Medium	Medium
Pump/motor design	Medium	High	Medium
Fittings	Medium	Low	Low
Valves/actuators	Medium	High	Low
Filter design	High	Low	Low
Reservoir design	Medium	Low	Medium
Seals and hoses	High	Medium	Low

Table 11. Effects of water-based fluids on component design

Effect on	**Fluid property**		
	Specific gravity	Viscosity	Vapour pressure
High temperature applications	Low	Medium	High
Low temperature applications	Low	High	Low
Pump/motor design	High	High	High
Fittings	High	High	High
Valves/actuators	Medium	High	Medium
Filter design	High	High	Medium
Reservoir design	Medium	Low	High
Seals and hoses	Low	Medium	Low

Table 12. Effects of synthetic fluids on components design

Effect on	**Fluid property**		
	Corrosivity	Lubricity	Thermal Conductivity
High temperature applications	Low	Low	Medium
Low temperature applications	Low	Low	Low
Pump/motor design	Low	Medium	Low
Fittings	Medium	Medium	Low
Valves/actuators	Medium	Medium	Low
Filter design	High	High	Low
Reservoir design	Medium	Low	Low
Seals and hoses	High	Medium	Low

Table 13. Effects of synthetic fluids on components design

Effect on	**Fluid property**		
	Specific gravity	Viscosity	Vapour pressure
High temperature applications	Low	Low	Low
Low temperature applications	Low	Low	Low
Pump/motor design	High	Medium	Low
Fittings	High	Low	Low
Valves/actuators	Medium	Medium	Low
Filter design	High	Medium	Low
Reservoir design	Medium	Low	Low
Seals and hoses	Low	Low	Low

The effect of the component's working environment may have a considerable effect on fluid selection. The problem is compounded since temperature has a similar effect on the specific gravity and the vapour pressure of the fluid - both of which also exhibit extreme effects on pump and plumbing design. Other environmental conditions must be considered in the selection of a fluid, such as dusty or dirty conditions, and extremely humid climate. These must be considered in the final evaluation of the fluid/component compatibility even though some limitations may result from the lack of familiarity with the physical properties of fire-resistant fluids and the scarcity of a working knowledge of their use.

Most of the synthetic or water based fluids have serious drawbacks when compared to the mineral oil, including some combination of:

- high cost
- poor lubricity
- low N_c due to high p_v
- high density and hence inertia
- low Viscosity Index (VI)
- incompatibility with common hydraulic system materials etc.

Several unrelated in-field and laboratory tests have established a variety of component modifications that have been found to be effective when using fire-resistant fluids. A summary of these modifications is given in table 14.

6.1.4 Selection of hydraulic lines

Once the proposed system's flow and operating pressure are determined we proceed to select the nominal diameters of hydraulic lines. The usual practice is to maintain a uniform nominal diameter of passages for all components (valves, fittings) in a particular line.

Selection of the size of a hydraulic line is to some degree a compromise between a need to minimize pressure losses in fluid lines and a desire to keep the size and

Table 14. Components modifications upon changeover to fire resistant fluids

<table>
<tr><th>Fluid</th><th>Component</th><th>Modification</th></tr>
<tr><td rowspan="9">Water-based</td><td rowspan="3">Pump/Motor</td><td>Increase size of suction lines</td></tr>
<tr><td>Reduce displacement, speed or operating pressure</td></tr>
<tr><td>Isolate rolling element bearings</td></tr>
<tr><td rowspan="3">Valve</td><td>Decrease clearances</td></tr>
<tr><td>Use stainless steel</td></tr>
<tr><td>Use elastomer seals</td></tr>
<tr><td rowspan="2">Cylinder</td><td>Line with corrosion resistant material</td></tr>
<tr><td>Use stainless steel</td></tr>
<tr><td>Reservoir</td><td>Minimize capacity</td></tr>
<tr><td rowspan="2">Synthetic-based</td><td>Pumps/Motor</td><td>Increase size of suction lines</td></tr>
<tr><td>Reservoir</td><td>Minimize capacity</td></tr>
</table>

the weight of a system as small as possible. The preliminary selection of the line diameter is based on permissible flow velocities, which are:

- suction lines - $v = 0.5 \div 2.0\,\mathrm{ms}^{-1}$
- return lines - $v = 1.0 \div 3.0\,\mathrm{ms}^{-1}$
- delivery lines - $v = 3.0 \div 6.0\,\mathrm{ms}^{-1}$

We may also use the data in table 15 which relates flow rate, at recommended flow velocities, to nominal diameters of suction and delivery lines.

6.2 Pressure losses

To calculate system power requirements we must know pressure losses in all elements of the system, i.e. in pumps, motors, actuators, control valves, filters and hydraulic lines. These losses are calculated using Bernoullie's equation [4]:

$$\rho\frac{v_1^2}{2} + p_1 + \gamma h_1 = \rho\frac{v_2^2}{2} + p_2 + \gamma h_2 + \Sigma p_{loss} \tag{6.6}$$

where:

p_1, p_2	-	static pressures in line section **1** and **2**
v_1, v_2	-	flow velocity in line section **1** and **2**
h_1, h_2	-	elevation of sections **1** and **2** above reference plane
$\sum p_{loss}$	-	sum of energetic losses (pressure lossses) between sections **1** and **2**
ρ, γ	-	density and specific weight of the fluid

Pressure loss is caused by friction (fluid viscosity) and changes of fluid momentum. The magnitude of the losses depends, to a large degree, on the character of the flow which can be either laminar or turbulent, although in many cases a transitional flow regime may occur.

The Reynold's Number Re which characterizes the flow regime is defined by equation:

$$\mathrm{Re} = \frac{v d_h}{\nu} \tag{6.7}$$

Table 15. Recommended nominal diameters of hydraulic lines vs flow

Flow Q			**Nominal diameter D_n [mm]**		
$[\mathrm{Lmin}^{-1}]$	$[\mathrm{m}^3\mathrm{s}^{-1}]$	$[\mathrm{Ls}^{-1}]$	**suction line**	**return line**	**delivery line**
10	0.00017	0.17	16	10	8
20	0.00033	0.33	20	16	10
40	0.00066	0.66	32	20	16
80	0.00133	1.33	40	32	20
160	0.00267	2.67	63	40	32
320	0.00533	5.33	80	63	40
640	0.01066	10.66	100	80	63

where:

d_h - hydraulic diameter (characteristic length)
ν - kinematic viscosity of the fluid
v - average flow velocity

For circular flow sections characteristic length d_h is simply the inside diameter of the flow section. For non-circular hydraulic diameter d_h is calculated using the following equation:

$$d_h = \frac{A_h}{S_h} \tag{6.8}$$

where:

A_h - flow area
S_h - length of flow perimeter

For example, hydraulic diameter d_h of a square cross section is equal to:

$$d_h = \frac{a^2}{4a} = 0.25a \tag{6.9}$$

where a is the length of the side of the square.

In cylindrical lines critical Reynold's Number Re, determined by experiments, is usually accepted to be in the range:

$$\mathrm{Re}_{critical} = 2300 \div 2500 \tag{6.10}$$

and we may expect turbulent flow to be for $\mathrm{Re} \geq \mathrm{Re}_{critical}$ and laminar flow to be for $\mathrm{Re} < \mathrm{Re}_{critical}$. There is no strict criteria which would determine what kind of flow regime exists for flows through control valves, fittings etc. However, on the basis of the results of many experiments it is generally accepted that during flow through various flow restrictions and clearances laminar flow condition will occur when:

$$\mathrm{Re} < 150 \div 400 \qquad \text{usually} \qquad \mathrm{Re} < 200 \tag{6.11}$$

Pressure losses are classified as:

- *Local pressure losses* occurring in localities of flow disturbance, e.g. in valves, tee-junctions, sudden changes of flow area, flow restrictors.
- *Line pressure losses* occurring in fluid lines in which the ratio of length to diameter is large.

6.2.1 Pressure losses - turbulent flow

Local pressure losses, which are mainly due to the loss of kinetic energy of the fluid, are defined by equation:

$$\Delta p = p_{loss} = \xi \rho \frac{v^2}{2} = \xi \frac{\rho}{2} \frac{Q^2}{A^2} \tag{6.12}$$

where:

ξ - coefficient of local losses
Q - flow rate
A - flow area
v - average flow velocity

When assessing local pressure losses we must determine values of loss coefficients ξ. Some values of ξ for various hydraulic elements quoted in literature are shown in table 16.

Typical values of coefficient ξ for 90° pipe bends in terms of ratio of bend radius to inside diameter of the pipe are shown in fig. 149, fig. 150 shows loss coefficients for subplate connections.

Table 16. Average values of loss coefficient for various hydraulic elements.

No.	**Element**	**Loss coeff. ξ**	**Velocity v**
1	Flapper-nozzle	$1.6 \div 1.9$	$\frac{Q}{A_{min}}$
2	Directional control valve	$2 \div 1.6$	$\frac{Q}{A_{min}}$
3	Directional control valve- tapered spool	$1.0 \div 1.6$	$\frac{Q}{A_{min}}$
4	Increased flow area	$\left(\frac{d_2^2}{d_1^2} - 1\right)^2$	$\frac{Q}{A_{min}}$
5	Reduced flow area - sharp entry	$\frac{z-1}{1.7z+1}$ where: $z = \frac{d_1^2}{d_2^2}$ 0.5 for $z > 10$	$\frac{Q}{A_{min}}$
6	Sharp entry $z > 10$	$1.0 \div 1.3$	$\frac{Q}{A_{min}}$
7	Chamfered entry $z > 10$	$0.06 \div 0.2$	$\frac{Q}{A_{min}}$
8	Rounded entry $z > 10$	0.012	$\frac{Q}{A_{min}}$
9	Right angle bend	1.0	
10	Straight fitting	0.5	
11	Angle fitting	$2.5 \div 3.0$	

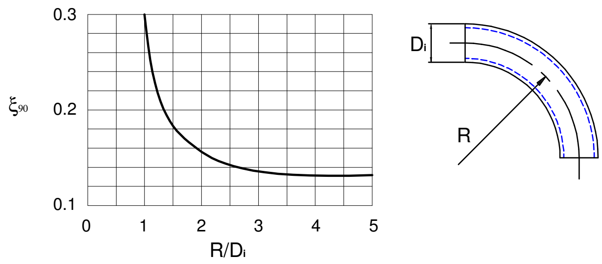

Fig. 149. Values of coefficient ξ for 90° bends

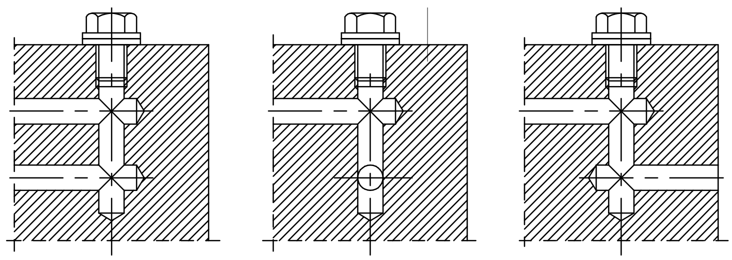

Fig. 150. Values of coefficient ξ for subplate connections

Hydraulic line pressure losses are proportional to line length, for line pressure loss is:

$$\Delta p = p_{loss} = \lambda \frac{l}{d} \rho \frac{v^2}{2} = \frac{\lambda l}{d} \frac{\rho}{2} \frac{Q^2}{A^2} \tag{6.13}$$

where:

λ	-	coefficient of line loss
l, d	-	length and diameter of line

Values of coefficient λ for hydraulic lines varies with k/d of the line walls (where k is the average roughness height) and Re, however as hydraulic lines are, in general, very smooth thus for flows in the turbulent range we can accept value $\lambda = 0.025$.

6.2.2 Pressure losses - laminar flow

Local pressure losses are proportional to flow velocity v. Pressure loss due to flow Q, e.g. through a laminar restrictor which has a circular cross section, is defined by the following equation:

$$\Delta p = p_{loss} = \frac{128 \mu l Q}{\pi d^4} = \frac{32 \mu l v}{d^2} \tag{6.14}$$

where:

μ	-	dynamic viscosity of the fluid
v	-	average flow velocity
d	-	inner diameter of a passage
l	-	length of passage

In practice, although flow through the system may be laminar, there are disturbances during flow through local restrictions, thus very often local losses are calculated using equation identical to one for turbulent flow:

$$\Delta p = p_{loss} = \xi \frac{\rho}{2} v^2 \tag{6.15}$$

Line pressure losses are proportional to flow velocity, but in practice we calculate pressure losses using the same equation as for turbulent flow:

$$\Delta p = p_{loss} = \lambda \frac{l}{d} \rho \frac{v^2}{2} \tag{6.16}$$

where coefficient of pressure losses λ is dependent only on the Reynold's Number Re. For long hydraulic lines, under assumption that there is no exchange of heat between the fluid and the environment, λ is equal to:

$$\lambda = \frac{64}{\mathrm{Re}} \tag{6.17}$$

however, as in practice there is heat transfer between the fluid and the environment, the value λ is defined by:

$$\lambda = \frac{75}{\mathrm{Re}} \tag{6.18}$$

Coefficient λ in transition and turbulent zone ($\mathrm{Re} \geq \mathrm{Re}_{critical}$) is obtained from (which is valid for Re in range $2 \times 10^3 \div 8 \times 10^4$):

$$\lambda = 0.3164\,\mathrm{Re}^{-0.25} \tag{6.19}$$

thus, as in hydraulic systems using mineral oils as working medium, Re is ordinarily less than 2×10^4 then Blasius formulae is generally applicable.

6.2.3 Calculation of pressure losses

To calculate pressure losses in hydraulic lines we must perform the following tasks:

- Determine, at operating temperature of the system, density ρ and dynamic viscosity μ of the fluid keeping in mind that system temperature may substantially vary during systems's operation due to environmental factors.
- Calculate Re for all flow paths in the system.

- Assume or determine relative smoothness k/d of hydraulic lines
- Calculate, or determine from a chart, values of λ and ξ
- Calculate pressure losses using appropriate equations.

A total pressure loss in a hydraulic flow path is a sum of the line and local losses

$$\Sigma p_{loss} = \frac{\rho}{2} v^2 \left(\Sigma \xi_i + \lambda \frac{\Sigma l_j}{d} \right) + \Sigma \Delta p_k \tag{6.20}$$

where:

ξi	-	local pressure loss in element i
l_j	-	length of line j
λ_j	-	coefficient of pressure loss in line j
Δp_k	-	pressure loss in element k

We may also use an equivalent length of line for values of coefficients of local losses λ. An equivalent length of line l_{eq} corresponds to the length of hydraulic line whose pressure loss is equal to the pressure loss in a local restriction, thus:

$$\lambda \frac{l_{eq}}{d} \frac{\rho}{2} v^2 = \xi \frac{\rho}{2} v^2 \tag{6.21}$$

and

$$l_{eq} = \frac{\xi d}{\lambda} \tag{6.22}$$

If we use the equivalent lengths of line for the local restrictions then a total pressure loss, for the lines and line elements connected in series, can be calculated from the formula:

$$\Sigma p_{loss} = \lambda \frac{\Sigma l_i + \Sigma l_{eqj}}{d} \frac{\rho}{2} v^2 + \Sigma \Delta p_k \tag{6.23}$$

where:

l_i	-	length of straight section i of the hydraulic line
l_{eqj}	-	equivalent line length for a local restriction j in series with the line

Calculations of pressure losses should be carried out for all fluid flow paths in the system, for example in an open type circuits we may identify:

- $\sum p_{1fr}$ - sum of pressure losses in a suction line (reservoir - pump)
- $\sum p_{2fr}$ - sum of pressure losses in a delivery line (pump - output unit)
- $\sum p_{3fr}$ - sum of pressure losses in a return line (output unit - reservoir)

6.2.3.1 Suction line

We are concerned with pressure p_1, at the pump suction port, which is a deciding factor affecting pump suction condition. This pressure cannot be lower than a permissible suction pressure p_{1cat} for the selected pump. The condition for correctly selected suction line is:

$$p_1 = p_r + \gamma h_s - \Sigma p_{1fr} > \Sigma p_{1cat} \tag{6.24}$$

where:

γ - specific weight of the fluid
h_s - pump suction head
p_r - reservoir pressure less atmospheric pressure

In open-to-air reservoirs pump suction head is the difference in elevation of the fluid level in the reservoir and the suction port of the pump (when the suction port is below the level of fluid in the reservoir $h_s < 0$). Depending on the type of pump, permissible (catalogue) vacuum pressure p_{1cat} ($-ve$ sign) at the pump suction port is in a range -40 to $-10\,\text{kPa}$.

6.2.3.2 Delivery line

Total pressure loss in a delivery line affects the actual pressure p_3 at the output unit (motor or actuator). As, in addition to line losses, there are also local pressure losses due to various control valves which are included in the delivery line, the total pressure loss in the delivery line may reach a large value and have a marked effect on the efficiency of the system. Thus, to calculate pressure losses in the delivery lines we need to know the pressure - flow characteristics of the control elements included in the delivery line.

6.2.3.3 Return lines

When calculating pressure losses in a return line we should remember that the return flow may be much higher than the delivery flow (e.g. in a system with double acting, single rod cylinders during return strokes of pistons).

For motors which rotate in either direction, the calculation should be done for both directions of rotation. For hydraulic systems employing several pumps and/or motors calculations should be carried out for all possible variants of the system operation.

6.2.4 Power losses

Knowledge of pressure losses in the system is needed in order to determine, for each phase of the operation cycle, a balance of power in the system and its overall efficiency η. The overall efficiency of the system is the ratio of the output power P_2 to input power P_1:

$$\eta = \frac{P_2}{P_1} \tag{6.25}$$

Input power P_1 is the power delivered by a prime mover to the pumping unit (which may consist of one or more pumps). In the case of a single pump system, operating in an open circuit, this power is determined from equation:

$$P_1 = \frac{Q_p \Delta p_{12}}{\eta_p} \tag{6.26}$$

where:

Q_p - actual flow delivery of the pump
η_p - overall efficiency of the pump
Δp_{12} - pressure differential across pump ports

Overall efficiency of the pump, during operation with delivery flow Q_p and with pressure differential Δp_{12}, can be determined from its pressure-flow characteristics. As, in practice, suction pressure p_1 compared to delivery pressure p_2 is very small, pressure difference Δp_{12} is determined from equation:

$$\Delta p_{12} = p_2 = \Delta p_{34} + \Sigma p_{2loss} + \Sigma p_{3loss} \tag{6.27}$$

where:

Δp_{34} - pressure difference across the output unit (motor or actuator)
$\sum p_{2fr}$, $\sum p_{3fr}$ - total losses in the delivery and return lines

6.3 Thermal calculations

The properties of hydraulic fluids, especially viscosity, and the materials of seals limit the range of the operating temperature of hydraulic systems. The temperature of hydraulic fluid is usually maintained between $60 \div 80\,^{\circ}\mathrm{C}$. Thus, after calculation of the system efficiency and the power loss in the circuit, we must calculate the expected operating temperature of the fluid and if the fluid temperature is too high, provide a heat exchanger (a cooler).

Thermal calculation of hydraulic systems is carried out under the following assumptions:

1. the locally generated heat is quickly transmitted to other parts of the system, thus thermal balance is determined for a whole system and not for its individual elements.
2. fluid temperature has a constant value throughout the system.
3. the system operates in a certain average operation conditions.
4. only the reservoir and the heat exchanger (if it is installed) transfer heat to the environment (heat transferred to the environment by other system elements will compensate for the occasional thermal overload).

Heat balance of a system at its thermal equilibrium can be written as:

$$\Phi_s = \Phi_e \tag{6.28}$$

where:

Φ_s - heat flux generated by the system
Φ_e - heat flux to the environment

According to assumption 4, heat generated in the system is equal to the average power loss, thus:

$$\Phi_s = P_{fra} = \frac{\sum_{i=1}^{n} P_{fri} x_i}{\sum_{i=1}^{n} x_i} \tag{6.29}$$

where:

P_{fri} - power loss during time interval x_i for $i = 1, 2, ...n$,

$\sum_{i=1}^{n} x_i$ - total duration of operation cycle

P_{fra} - average power loss in the system

In general, heat Φ_e transmitted to the environment is equal to:

$$\Phi_e = \Phi_{res} + \Phi_{he} \tag{6.30}$$

where:

Φ_{res} - heat radiated by the reservoir

Φ_{he} - heat transmitted by the heat exchanger

Usually we will initially try to find the system's equilibrium temperature assuming that the system operates without a heat exchanger and that heat will be transferred to the environment through the reservoir wall. Thus:

$$\Phi_e = \Phi_{res} = (T_f - T_e) k_c A_{res} \tag{6.31}$$

where:

T_f - fluid temperature, K

T_e - environment temperature, K

k_c - heat transmission coefficient from the reservoir to the environment

A_{res} - surface area of the reservoir

Average values of coefficient k_c are, [8]:

$k_c = 10 \div 14 \text{ Wm}^{-2}\text{K}$ – for steel reservoirs

$k_c = 6 \div 9 \text{ Wm}^{-2}\text{K}$ – for cast iron reservoirs

Thus, using the above equation we may either calculate equilibrium temperature of the fluid:

$$T_f = \frac{\Phi_s}{k_c A_{res}} + T_e \tag{6.32}$$

or the required surface area of reservoir:

$$A_{res} = \frac{\Phi_s}{k_c(T_f - T_e)} \tag{6.33}$$

If the calculated temperature of the fluid is too high or if maintaining of the acceptable fluid temperature would require a reservoir with an excessive surface area then we should include a cooler in the system. Heat removal capacity of the cooler can be calculated from equation:

$$\Phi_{he} = \Phi_s - \Phi_{res} = \Phi_s - (T_f - T_e)A_{res}k_c.$$

The calculation of heat exchangers and their design is widely discussed in literature.

Examples of Calculations - Synthesis of Hydraulic systems

Problem 6.1 Hydraulic drive for air-compressor

A diagram of a hydraulic system driving an air-compressor on a cement mixer is shown in fig. 151. Select the pump and calculate overall efficiency and power loss in the proposed system. Rotation speed of the compressor is $n_m = 1250$ rpmand compressor torque is $T_m = 260$ Nm.

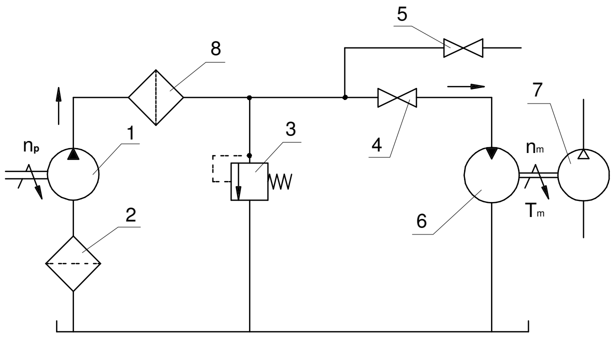

Fig. 151. Hydraulic drive for air-compressor

Answer: We follow the procedure outlined in the previous chapter. Thus, we start with selection of main components and hydraulic lines and follow with calculation of pressure losses in the circuit and eventually we calculate power requirements and power losses in the circuit.

- *Selection of motor size*

We choose nominal operating pressure $p_n = 12$ MPa. Thus, the required stroke displacement of the motor driving the air-compressor, with 20% error margin to accommodate volumetric losses in the system, is:

$$q_m = 1.2\frac{2\pi T_m}{p_n} = 1.2\frac{2\pi \times 260}{12 \times 10^6} = 163.4 \times 10^{-6}\ \mathrm{m^3rev^{-1}}$$

The motor selected from a catalogue has stroke displacement $q_m = 175.3\ \mathrm{cm^3rev^{-1}}$. According to manufacturer's data, for output torque $T_m = 260$ Nm and rotation speed $n_m = 1250$ rpm, pressure difference across motor ports should be $\Delta p_{34} =$ 10.3 MPa and actual motor flow demand is $Q_m = 3.8\ \mathrm{Ls^{-1}}$.

- *Selection of pump size*

When the motor is driving the compressor, shut-off valve **5** is closed and thus full flow from pump **1** is directed to hydraulic motor **6**. Thus, the required minimum flow rate capacity of the pump is $Q_p = Q_m = 3.8$ $\mathrm{Ls^{-1}}$. The chosen pump is of the same size as the motor, i.e. $q_p = q_m = 175.3$ $\mathrm{cm^3rev^{-1}}$. For this pump to deliver flow $Q_p = 3.8$ $\mathrm{Ls^{-1}}$ at pressure $p_{12} = 12\,\mathrm{MPa}$ the required rotation speed is $n_p = 1350\ rpm$.

- *Selection of hydraulic fluid*

We select a hydraulic fluid which has kinematic viscosity $\nu = 30$ $\mathrm{mm^2s^{-1}}$(30 cSt) at temperature t= 40°C and its density is $\rho = 900$ $\mathrm{kgm^{-3}}$.

- *Selection of nominal diameters of hydraulic lines*

Using table 15 we select the following nominal internal diameters of hydraulic lines: $D_s = 63\,\mathrm{mm}$ for the suction line, actual flow velocity:

$$v_s = \frac{4 \times Q}{\pi \times D_s^2} = \frac{4 \times 3.8 \times 10^{-3}}{\pi \times 0.063^2} = 1.22\ \mathrm{ms^{-1}}$$

$D_d = 32\,\mathrm{mm}$ for the delivery and $D_r = 40\,\mathrm{mm}$ return lines, actual flow velocities:

$$\begin{aligned} v_d &= \frac{4 \times Q}{\pi \times D_d^2} = \frac{4 \times 3.8 \times 10^{-3}}{\pi \times 0.032^2} = 4.72\ \mathrm{ms^{-1}} \quad \text{for delivery line} \\ v_r &= \frac{4 \times Q}{\pi \times D_r^2} = \frac{4 \times 3.8 \times 10^{-3}}{\pi \times 0.040^2} = 3.02\ \mathrm{ms^{-1}} \quad \text{for return line} \end{aligned}$$

The velocity of fluid in the return line is on the high side but still within accepted limits.

- *Selection of suction filter*

  Suction filter data (100 μm filter) catalogue data:

nominal flow through the filter	$Q_n = 4.0\ \mathrm{Ls^{-1}}$
nominal diameter of filter ports	$D_n = 63\,\mathrm{mm}$

Pressure losses for flow $Q_p = 3.8$ $\mathrm{Ls^{-1}}$ through the suction filter at various viscosities of fluid are:

$$\begin{aligned} \Delta p_1' &= 0.0035\ \mathrm{MPa} \quad \text{for viscosity} \quad \nu = 45\ \mathrm{mm^2s^{-1}}(45\ \mathrm{cSt}) \\ \Delta p_1'' &= 0.0040\ \mathrm{MPa} \quad \text{for viscosity} \quad \nu = 110\ \mathrm{mm^2s^{-1}}(110\ \mathrm{cSt}) \end{aligned}$$

- *Selection of high pressure filter*

High pressure filter data (10 μm filter) catalogue data:

nominal flow through filter	$Q_n = 5.25\ \mathrm{Ls^{-1}}$
nominal diameter of filter port	$D_{ft} = 32\,\mathrm{mm}$

Pressure loss for flow $Q_p = 3.8$ $\mathrm{Ls^{-1}}$ through the filter is $\Delta p_2 = 0.08$ MPa

- *Restrictor valve*

Nominal internal diameter $D_{rv} = 32\,\mathrm{mm}$. Loss coefficient $\xi = 0.8$ (assumed)

- *Relief valve*

Maximum relief pressure 20.00 MPa, adjustment range 2 − 16 MPa in steps of 0.4 MPa

- *Calculation of pressure losses*

a. *Suction line*

Local losses in the suction line are due to:

reduction of flow area	$\xi_1 = 0.5$
90-degree bend	$\xi_2 = 1.0$
2 straight fittings	$\xi_3 = 0.5$
suction filter	$\Delta p_1 = 5.0$ kPa

Pressure drop on the suction filter corresponds to "loaded" filter or cold start-up. Total length of straight sections of the suction line is $l = 1600\,\text{mm}$. We assume that internal diameter of the suction pipe is equal to the previously selected nominal diameter of line $D_s = 63\,\text{mm}$. Thus average flow velocity in suction line at pump flow $Q_p = 3.8\ \text{Ls}^{-1}$ is equal to:

$$v = \frac{4Q}{\pi D_s^2} = \frac{4 \times 3.8 \times 10^{-3}}{\pi \times 0.063^2} = 1.22\ \text{ms}^{-1}$$

We assume the average temperature of fluid $t = 40^0 C$ at which viscosity of the selected fluid is $\nu = 30 \times 10^{-6}\ \text{m}^2\text{s}^{-1}$(30 cSt). Thus Reynold's Number for suction line is:

$$\text{Re} = \frac{vD_s}{\nu} = \frac{1.22 \times 0.063}{30 \times 10^{-6}} = 2562 > Re_{critical} = 2300 - 2500$$

As the flow in the suction line is turbulent, the pressure loss coefficient is equal to:

$$\xi_4 = \frac{0.3164}{\sqrt[4]{\text{Re}}} \frac{l}{D_s} = \frac{0.3164}{\sqrt[4]{2562}} \frac{1600}{63} = 1.13$$

Total pressure loss in the suction line is:

$$\begin{aligned} \Sigma p_{1fr} &= \frac{\rho v^2}{2}(\xi_1 + \xi_2 + 2\xi_3 + \xi_4) + \Delta p_1 \\ \Sigma p_{1fr} &= \frac{900 \times (1.2)^2}{2} \times (0.5 + 1.0 + 2 \times 0.5 + 1.13) + 5 \times 10^3 = 7.3\ \text{kPa} \end{aligned}$$

Pump suction head is equal to:

$$p_h = \rho g h$$

where:

h	-	vertical distance between fluid level in the reservoir and pump inlet port
g	-	acceleration of gravity
ρ	-	fluid density

for our system we assume $h = 1.5\,\mathrm{m}$, thus suction head is:

$$p_h = 900 \times 9.81 \times 1.5 = 13.2 \text{ kPa}$$

The value of pressure losses in suction line, in comparison to pump suction head, is relatively low - thus we will reduce the nominal diameter of suction line to $D_s = 50\,\mathrm{mm}$ (we use however the same filter). The average flow velocity is now:

$$v = \frac{4 \times 3.8 \times 10^{-6}}{\pi \times 0.050^2} = 1.94 \text{ ms}^{-1}$$

and the Reynold's Number:

$$\mathrm{Re} = \frac{vD_s}{\nu} = \frac{1.94 \times 0.05}{30 \times 10^{-6}} = 3233 > \mathrm{Re}_{critical}$$

and a new loss coefficient is:

$$\xi_4 = \frac{0.3164}{\sqrt[4]{\mathrm{Re}}}\frac{l}{D_s} = \frac{0.3164}{\sqrt[4]{3233}} \times \frac{1600}{50} = 1.34$$

The total pressure loss in the suction line is equal to:

$$\begin{aligned} \Sigma p_{1fr} &= \frac{\rho}{2}v^2(\xi_1 + \xi_2 + 2\xi_3 + \xi_4) + \Delta p_1 \\ \Sigma p_{1fr} &= \frac{900 \times 1.94^2}{2} \times (0.5 + 1.0 + 2 \times 0.5 + 1.34) + 5 \times 10^3 = 11.50 \text{ kPa} \end{aligned}$$

The pressure at the pump inlet port is equal to:

$$p_1 = p_h - \Sigma p_{1fr} = \rho g h - \Sigma p_{1fr}$$

Thus:

$$p_1 = 900 \times 9.81 \times 1.5 - 11.50 \times 10^3 = 1.7 \text{ kPa}$$

and as the pump inlet pressure is above atmospheric pressure the suction line is satisfactory.

b. *Delivery line*

Local losses in the delivery line are due to:

4 straight fittings	$\xi_1 = 0.5$
3 90^0 bends	$\xi_2 = 1.0$
tee fitting	$\xi_3 = 0.6$
restrictor valve **4**	$\xi_4 = 0.8$
high pressure filter	$\Delta p_2 = 100$ kPa ("loaded" filter or cold start-up)

Total length of straight sections of the delivery line is $l = 2300$ mm and the internal diameter of the delivery line is again assumed to be equal to the, previously selected, nominal diameter $D_d = 32$ mm. Then average flow velocity in delivery line is equal to:

$$v = \frac{4 \times 3.8 \times 10^{-1}}{\pi \times 0.032^2} = 4.7 \text{ ms}^{-1}$$

and as Reynold's Number is:

$$\text{Re} = \frac{4.7 \times 0.032}{30 \times 10^{-6}} = 5013 > \text{Re}_{critical}$$

thus loss coefficient for straight lines is equal to:

$$\xi_5 = \frac{0.3164}{\sqrt[4]{5013}} \times \frac{2300}{32} = 2.7$$

and the total pressure loss in the delivery line is equal to:

$$\begin{aligned}
\Sigma p_{2fr} &= \frac{\rho v^2}{2}(4\xi_1 + 3\xi_2 + \xi_3 + \xi_4 + \xi_5) + \Delta p_2 \\
\Sigma p_{2fr} &= \frac{900 \times 4.7^2}{2}(4 \times 0.5 + 3 \times 1.0 + 0.6 + 0.8 + 2.7) + 100 \times 10^3 = 190 \text{ kPa}
\end{aligned}$$

c. *Return line*

Local losses in the return line are due to:

2 straight fittings	$\xi_1 = 0.5$
2 90^0 bends	$\xi_2 = 1.0$

Total length of straight sections of return line $l = 1200$ mm, and line nominal diameter $D_r = 40$ mm. An additional pressure loss occurs in the line due to an enlarged flow area at the entry to the reservoir - we assume loss coefficient $\xi_3 = 1.0$. The average flow velocity in the return line is 3.02 ms^{-1} and the Reynold's Number is:

$$\text{Re} = \frac{4.7 \times 0.040}{30 \times 10^{-6}} = 6266 > \text{Re}_{critical}$$

thus line pressure loss coefficient is equal to:

$$\xi_4 = \frac{0.3164}{\sqrt[4]{6266}} \times \frac{1200}{40} = 1.07$$

thus total losses in the return line are:

$$\Sigma p_{3fr} = \frac{\rho v^2}{2}(2\xi_1 + \xi_2 + \xi_3 + \xi_4)$$
$$\Sigma p_{3fr} = \frac{900 \times 4.7^2}{2} \times (2 \times 0.5 + 2 \times 1.0 + 1.0 + 1.07) = 50.4\,\mathrm{kPa}$$

- *Power loss and efficiency of the system*

Pressure difference across the pump is defined by equation:

$$\Delta p_{12} = \Delta p_{34} + \Sigma p_{2fr} + \Sigma p_{3fr}$$
$$\Delta p_{12} = 10.3 \times 10^6 + 190 \times 10^3 + 50.4 \times 10^3 = 10.54\ \mathrm{MPa}$$

From the pump catalogue (not shown) we determined that to deliver flow rate $Q_p = 3.8\ \mathrm{Ls}^{-1}$ at pressure differential $\Delta p_{12} = 10.4$ MPa we need input power $P_p = 45\,\mathrm{kW}$. Thus system efficiency is:

$$\eta = \frac{P_m}{P_p} = \frac{T_m \pi n_m}{30 P_p} = \frac{260 \times \pi \times 1250 \times 10^{-3}}{30 \times 45} = 0.76 \qquad \text{answer!}$$

and the power loss is equal to:

$$P_{loss} = P_p(1 - \eta) = 45 \times (1 - 0.76) = 10.8\,\mathrm{kW} \qquad \text{answer!}$$

Problem 6.2 Thermal balance of hydraulic system

Evaluate thermal balance of a hydraulic system driving a cement mixer shown in Problem 6.1. It takes approximately one hour of system operation to empty the cement container. The volume of fluid in the system is $V_{fl} = 400$ L, the reservoir dimensions are 1200 mm ×1200 mm ×300 mm and the permissible fluid temperature $T_{fmax} = 80\,°\mathrm{C}$.

Answer: Temperature of the fluid at the end of a working cycle is calculated from equation:

$$\Phi_s = \Phi_{res} + \Phi_f$$

where:

Φ_s	-	heat flux generated in the system
Φ_{res}	-	heat flux radiated by the reservoir to the environment
Φ_f	-	heat flux absorbed by the fluid

Power loss was calculated to be $P_{loss} = 10.8\,\mathrm{kW}$ thus heat flux generated in the system during its continuing operation is equal to:

$$\Phi_s = P_{loss} = 10.8\,\mathrm{kW}$$

whereas heat flux radiated by the reservoir Φ_{res} and heat flow rate Φ_c absorbed by the fluid are:

$$\begin{aligned}\Phi_{res} &= k_c A_{res}(T_f - T_e)\\ \Phi_c &= m_f c\frac{dT_f}{dt}\end{aligned}$$

where:

m_{fl}	-	fluid mass
c	-	specific heat of the fluid
$\frac{dT_f}{dt}$	-	temperature gradient
k_c	-	heat transfer coefficient
A_{res}	-	surface area of the reservoir
T_f, T_e	-	fluid and environment temperatures

Thermal balance during transient is defined by equation:

$$\Phi_s = k_c A_{res}(T_f - T_e) + m_{fl} c\frac{dT_f}{dt}$$

and after rearranging:

$$\frac{dt}{d\tau} = \frac{m_{fl} c}{\Phi_s - k_c A_{res}(t - t_e)}$$

and integrating this equation we obtain:

$$T_f(t) = T_e + \frac{\Phi_s}{k_c A_{res}}\left(1 - e^{\dfrac{-k_c A_{res} t}{m_{fl} c}}\right)$$

For our calculations we assume $k_c = 12.0\ \mathrm{Wm^{-2}K}$ and $c = 2.0\ \mathrm{kJkg^{-1}K^{-1}}$ and as the area of the reservoir is equal to:

$$A_{res} = 2 \times 1.2^2 + 4 \times 1.2 \times 0.3 = 4.32\ \mathrm{m}^2$$

and mass of the fluid ($\rho = 900\,\mathrm{kg\,m^{-3}}$) in the reservoir is:

$$m_{fl} = V_{fl}\rho = 0.4 \times 900 = 360\,\mathrm{kg}$$

then after one hour continuous operation ($t = 3600\,\mathrm{s}$), and at assumed ambient temperature $T_e = 30\,^\circ\mathrm{C}$, the system temperature will reach:

$$T_f = 30 + \frac{10800}{12 \times 4.32} \times \left\{ 1 - e^{\left(-\frac{12 \times 4.32 \times 3600}{360 \times 2 \times 10^3} \right)} \right\} {}^\circ\mathrm{C}$$

$$T_f = 77.6\,^\circ\mathrm{C} \qquad \text{answer!}$$

The temperature of the fluid does not exceed the permissible temperature, thus the system can operate as it is without an additional heat exchanger.

Problem 6.3 Hydraulic drive for a crane

The circuit diagram shown in fig. 152 represents a hydraulic system operating a crane. Select a pump and cylinder for the system and determine overall efficiency of the system when piston is extending. Assume that the maximum effective force of the cylinder is $F_c = 600$ kN, and that extension speed of the piston is $v = 0.05\ \mathrm{ms}^{-1}$. The return stroke of the cylinder is due to gravity force, and is controlled by counterbalance valve **4**. Power to the hydraulic system is supplied by a diesel engine.

Answer: We will again follow the procedure outlined in the previous chapter. Thus, we start with the selection of main components and hydraulic lines and follow with calculation of pressure losses in the circuit and eventually we calculate the overall efficiency of the circuit.

- *Selection of cylinder*

Arbitrarily we choose operating pressure $p_n = 16$ MPa, then the area of the piston **2** (with 20% allowance for losses) is:

$$A_p = \frac{1.2F_c}{p_n} = \frac{1.2 \times 600 \times 10^3}{16 \times 10^6} = 0.045\ \mathrm{m}^2$$

thus required diameter of the piston is:

$$D = \sqrt{\frac{4A_p}{\pi}} = \sqrt{\frac{4 \times 0.045}{\pi}} = 239\,\mathrm{mm}$$

we will chose cylinder with dimensions 250/150 × 1000, i.e. diameter of the piston $D = 250\,\mathrm{mm}$, rod diameter $d = 160\,\mathrm{mm}$ and stroke $S = 1000\,\mathrm{mm}$. Then the working areas of the cylinder are:

$$A_1 = \frac{\pi D^2}{4} = \frac{\pi \times 0.250^2}{4} = 4.91 \times 10^{-2}\ \mathrm{m}^2 \text{ - piston area}$$

and

$$A_2 = \frac{\pi \left(D^2 - d^2\right)}{4} = \frac{\pi \times \left(0.250^2 - 0.160^2\right)}{4} = 2.9 \times 10^{-2}\ \mathrm{m}^2 \text{ - annulus area}$$

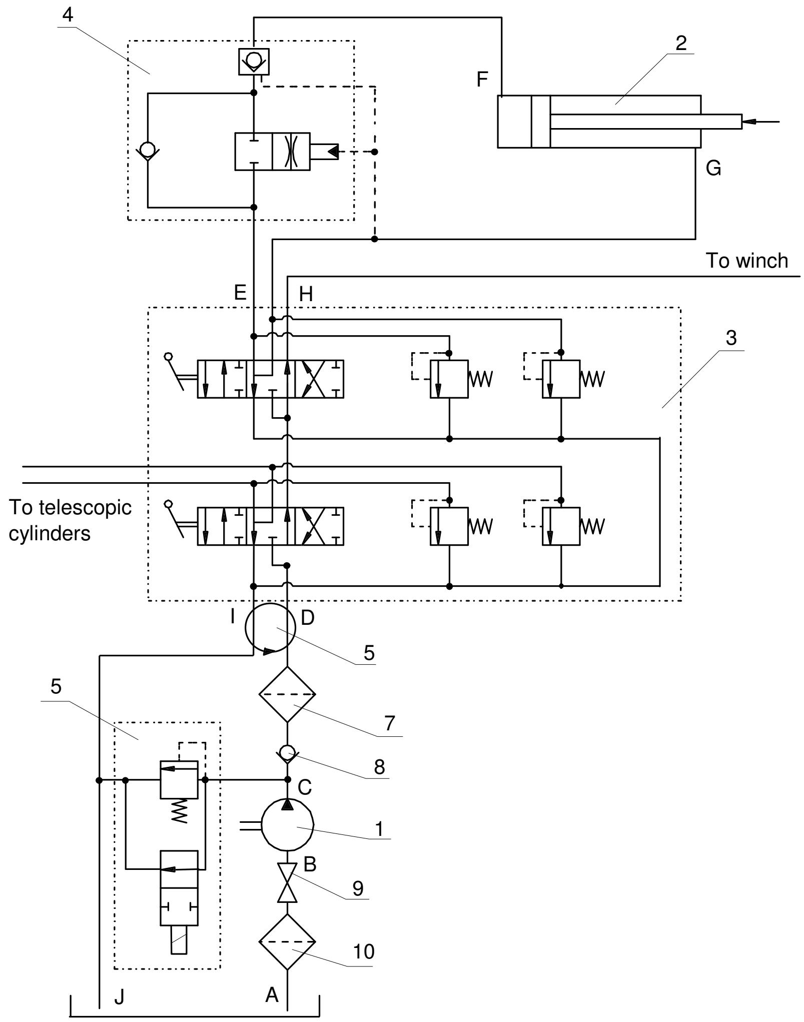

Fig. 152. Crane hydraulic system

Piston area of the selected cylinder is larger than the required area, thus the system will operate at lower pressure. Flow rate Q_c required to move piston with velocity $v = 0.05\ \mathrm{ms}^{-1}$ is calculated using equation:

$$Q_c = \frac{vA_1}{\eta_{vc}}$$

we assume that the volumetric efficiency of the cylinder is $\eta_{vc} = 0.99$, then required flow is equal to:

$$Q_c = \frac{0.05 \times 4.91 \times 10^{-2}}{0.99} = 2.48 \times 10^{-3}\ \mathrm{m^3s^{-1}} = 2.48\ \mathrm{Ls^{-1}}$$

- *Selection of pump*

According to the circuit diagram cylinder **2** is supplied by pump **1**, which also supplies other systems on the crane - the winch and the telescopic cylinder (not shown on the diagram). When the upper valve in the valve block **3** is shifted from its neutral position then full flow of the pump is directed to cylinder **2.** Required stroke displacement of the pump can be determined from equation:

$$q_p = \frac{Q_c}{n_p \eta_{vp}}$$

and if we assume volumetric efficiency to be $\eta_{vp} = 0.96$ and choose maximum rotational speed of the prime mover as pump input speed i.e. $n_p = 2200$ rpm (36.7 revs^{-1}) then the required stroke displacement of the pump is:

$$q_p = \frac{2.48 \times 10^{-3}}{36.7 \times 0.96} = 7.04 \times 10^{-5}\ \mathrm{m^3rev^{-1}}$$

The pump selected from the catalogue is a gear pump which has a stroke displacement of $q_p = 70.5\ \mathrm{cm^3rev^{-1}}$ and using catalogue data we determined that at nominal pressure $p_n = 16.0$ MPa and rotation speed $n_p = 2200$ rpm the actual pump flow is $Q_p = 2.41\ \mathrm{Ls^{-1}} \approx Q_c$.

- *Selection of hydraulic fluid*

Hydraulic fluid selected for this application has kinematic viscosity $\nu = 50\ \mathrm{mm^2s^{-1}}$ (50 cSt) at temperature t= 50 °C and fluid density is $\rho = 900\ \mathrm{kgm^{-3}}$

- *Calculation of pressure losses*

a. *Suction line (line **A**-**B**)*

Local losses in the suction line are due to the following components::

hydraulic tube $50 \times 3 \times 1700$ mm	ξ_1
hydraulic tube $50 \times 3 \times 1200$ mm	ξ_2
filter - port diameter $D_f = 50$ mm	$\Delta p_f = 5$ kPa
shut-off valve - port diameter $D_{sv} = 50$ mm	$\xi_3 = 0.7$
one straight fitting	$\xi_4 = 0.5$
3 90° bends	$\xi_5 = 1.0$

In addition to pressure losses on these components an additional pressure loss will occur due to decreased flow area when fluid enters suction line, $\xi_6 = 0.5$. Average velocity in the suction pipe is equal to:

$$v_s = \frac{4Q_p}{\pi D_s^2} \qquad D_s = D_{so} - 2s$$

where:

D_{so} - outside diameter of the tube

s - wall thickness

thus, $D_s = 50 - 2 \times 3 = 44\,\text{mm}$ and velocity of fluid in the tube:

$$v_s = \frac{4 \times 2.41 \times 10^{-3}}{\pi 0.044^2} = 1.59\ \text{ms}^{-1}$$

Reynold's Number for flow in the suction line is:

$$\text{Re} = \frac{v_s D_s}{\nu} = \frac{1.59 \times 0.044}{50 \times 10^{-6}} = 1400 < \text{Re}_{crit} = 2300 - 2500$$

as Re is in the laminar range thus flow in the suction line is laminar and loss coefficients ξ_1 and ξ_2 are calculated using:

$$\xi_1 = \lambda \frac{l}{D_i} = \frac{75}{\text{Re}} \frac{l}{D_s} = \frac{75 \times 1.7}{1400 \times 0.044} = 2.07$$

and

$$\xi_2 = \frac{75 \times 1.2}{1400 \times 0.044} = 1.46$$

The average flow velocity through other line elements is calculated using selected nominal diameter $D_n = 50\,\text{mm}$, thus:

$$v_n = \frac{4Q_p}{\pi D_n^2} = \frac{4 \times 2.41 \times 10^{-3}}{\pi \times 0.05^2} = 1.23\ \text{ms}^{-1}$$

The total pressure loss in the suction line is now equal:

$$\begin{aligned}
\Delta p_{1fr} &= \frac{\rho}{2}\left[v_s^2(\xi_1 + \xi_2) + v_n^2\,(\xi_3 + \xi_4 + 3\xi_5 + \xi_6)\right] + \Delta p_f \\
\Delta p_{1fr} &= \frac{900}{2}\left[1.59^2 \times (2.1 + 1.4) + 1.23^2 \times (0.7 + 0.5 + 3 \times 1.0 + 0.5)\right] + 5 \times 10^3 = \\
&= 12\ \text{kPa}
\end{aligned}$$

b. *Delivery line (line **C** - **F**)*

Local losses in the delivery line are due to:

check valve **8**	$\Delta p_{ch} = 200$ kPa
high pressure filter **7**	$\Delta p_f = 60$ kPa
rotary joint **5**	$\Delta p_r = 120$ kPa
DCV control valve **3**	$\Delta p_{dcv} = 700$ kPa
Counterbalance valve **4**	$\Delta p_{cb} = 200$ kPa
12 straight fittings	$\xi_1 = 0.5$
17 90° bends	$\xi_2 = 1.0$
tee fitting	$\xi_3 = 0.3$
hydraulic tube $28 \times 2.5 \times 6500$ mm	ξ_4
flexible hose 25×800 mm	ξ_4

Total length of straight sections of $\phi 28 \times 2.5$ mm steel tubing is $l = 6500$ mm, the delivery line also includes a $\phi 25$ mm flexible hose which is 800 mm long. The average flow velocity in the steel tube (internal diameter $D_d = 0.028 - 2 \times 0.0025 = 23$ mm) is:

$$v_d = \frac{4Q_p}{\pi D_d} = \frac{4 \times 2.41 \times 10^{-3}}{\pi \times 0.023^2} = 5.80 \text{ ms}^{-1}$$

and Reynold's Number is equal to:

$$\text{Re} = \frac{v_d D_d}{\nu} = \frac{5.82 \times 0.023}{50 \times 10^{-6}} = 2677$$

Thus as $\text{Re} > \text{Re}_{crit} = 2500$ the flow is turbulent and the pressure loss coefficient is obtained from equation:

$$\xi_4 = \frac{0.3164}{\sqrt[4]{\text{Re}}} \frac{1}{D_d} = \frac{0.3164}{\sqrt[4]{2677}} \times \frac{6500}{23} = 12.4$$

The average flow velocity in the flexible hose which has nominal internal diameter $D_n = 25$ mm is equal to:

$$v_h = \frac{4Q_p}{\pi D_n^2} = \frac{4 \times 2.41 \times 10^{-3}}{\pi \times 0.025^2} = 4.91 \text{ ms}^{-1}$$

we assume that average velocity v_n through other components in the delivery line is the same as through the flexible hose, thus $v_n = v_h = 4.91 \text{ ms}^{-1}$.

Reynold's Number for flow in the flexible hose is:

$$\text{Re} = \frac{4.91 \times 0.025}{50 \times 10^{-6}} = 2455 \approx \text{Re}_{crit}$$

we again consider flow in the flexible hose to be in the turbulent regime and the coefficient of pressure losses is equal to:

$$\xi_5 = \frac{0.3164}{\sqrt[4]{2455}} \frac{800}{25} = 1.44$$

The total pressure loss in the delivery line is obtained from equation:

$$\Delta p_{2fr} = \frac{\rho}{2}\left[v_d^2\xi_4 + v_n^2(12\xi_1 + 17\xi_2 + \xi_3 + \xi_5)\right] + \Delta p_{ch} + \Delta p_f + \Delta p_r + \Delta p_{dcv} + \Delta p_{cb}$$

and finally:

$$\begin{aligned}
\Delta p_{2fr} &= \frac{900}{2}\left[5.80^2 \times 12.4 + 4.91^2 \times (12 \times 0.5 + 17 \times 1.0 + 0.3 + 1.44)\right] + \\
&\quad + (200 + 60 + 120 + 700 + 200) \times 10^3 \\
\Delta p_{2fr} &= 456 \times 10^3 + 1280 \times 10^3 = 1736 \text{ kPa}
\end{aligned}$$

c. *Return line (line* ***G*** *- J)*
The flow rate in the return line is calculated from the following equation:

$$\begin{aligned}
Q_r &= Q_p\frac{A_2}{A_1} = \frac{Q_p(D^2 - d^2)}{D^2} \\
Q_r &= \frac{2.41 \times 10^{-3} \times (0.25^2 - 0.16^2)}{0.25^2} = 1.42 \times 10^{-3} \text{ m}^3\text{s}^{-1} = 86 \text{ Lmin}^{-1}
\end{aligned}$$

Local losses in the return line are due to:

DCV control valve **3**	$\Delta p_{dcv} = 300$ kPa
rotary joint **5**	$\Delta p_r = 80$ kPa
7 straight fittings	$\xi_1 = 0.5$
13 90-degree bends	$\xi_2 = 1.0$
3 tee fittings	$\xi_3 = 0.3$
hydraulic tubing 28×2.5 × 7660 mm	ξ_4
flexible tubing 25×1400 mm	ξ_5

Total length of straight sections of $\phi 28 \times 2.5$ mm steel tubing is $l = 7660$ mm, and the $\phi 25$ mm flexible hose is 1400 mm long. The average flow velocity in the steel tubing (internal diameter $D_r = 23$ mm) is:

$$v_r = \frac{4 \times 1.42 \times 10^{-3}}{\pi \times (0.023)^2} = 3.42 \text{ ms}^{-1}$$

and the Reynold's Number is:

$$\text{Re} = \frac{3.42 \times 0.023}{50 \times 10^{-6}} = 1573 < \text{Re}_{crit} = 2300 - 2500$$

Flow in the return line is in the laminar range thus the coefficient of pressure loss in the steel tube is equal to:

$$\xi_4 = \frac{75}{1573}\frac{7660}{23} = 15.88$$

The average flow through the elements with nominal diameter $D_n = 25\,\text{mm}$ is equal to:

$$v_n = \frac{4 \times 1.42 \times 10^{-3}}{\pi \times 0.025^2} = 2.89 \text{ ms}^{-1}$$

Because calculation of pressure loss in the flexible hose is based also on the nominal diameter, Reynold's Number for the flexible hose is:

$$\text{Re} = \frac{2.89 \times 0.025}{50 \times 10^{-6}} = 1445 < \text{Re}_{crit} = 2300 - 2500$$

and the pressure loss coefficient is then:

$$\xi_5 = \frac{75}{1445}\frac{1400}{25} = 2.91$$

Finally, total loss in the return line is equal to:

$$\begin{aligned}
\Delta p_{3fr} &= \frac{\rho}{2}\left[v_r^2\xi_4 + v_n^2(7\xi_1 + 13\xi_2 + 3\xi_3 + \xi_5)\right] + \Delta p_{dcv} + \Delta p_r \\
\Delta p_{3fr} &= \frac{900}{2}\left[3.42^2 \times 15.88 + 2.89^2 \times (7 \times 0.5 + 13 \times 1.0 + 3 \times 0.3 + 2.91)\right] + \\
&\quad + (300 + 80) \times 10^3 \\
\Delta p_{3fr} &= 159 \times 10^3 + 380 \times 10^3 = 539 \text{ kPa}
\end{aligned}$$

- *System operating conditions*

Pressure differential across the cylinder is calculated using equation:

$$\Delta p_{34} = \frac{F_c}{A_1}\eta_{hmc}$$

If we assume that the hydro-mechanical efficiency of the cylinder is $\eta_{hmc} = 0.95$, then Δp_{34} is equal to:

$$\Delta p_{34} = \frac{600 \times 10^3}{0.049 \times 0.95} = 12.9 \text{ MPa}$$

thus, the output pressure of the pump is:

$$\begin{aligned}
p_2 &= \Delta p_{34} + \Delta p_{2fr} + \Delta p_{3fr} \\
p_2 &= 12.9 \times 10^6 + 1.737 \times 10^6 + 0.539 \times 10^6 = 15.17 \text{ MPa}
\end{aligned}$$

The pump is working at slightly lower pressure than initially assumed, however the effect on pump delivery Q_p will be minimal and is ignored. Thus, the actual speed of the piston is equal to:

$$v_c = \frac{Q_p}{A_1}\eta_{vc} = \frac{2.41 \times 10^{-3}}{0.049} \times 0.99 = 0.0487 \text{ ms}^{-1}$$

Again using pump characteristics; we determined that pump input power, at delivery pressure $p_2 = 15.17$ MPa and rotation speed $n_p = 37$ revs^{-1}, is equal to $P_p = 40\,\text{kW}$. The overall efficiency of the system is then:

$$\eta = \frac{F_c v_c}{P_p} = \frac{600 \times 10^3 \times 0.0487}{40 \times 10^3} = 0.73 \qquad \text{answer!}$$

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