HYDRAULIC AGGREGATE

1. Hydrostatic drives

1.1. Introduction

The majority of flow machines can be divided into two groups according to the principle of operation:

- turbomachines (e.g., pumps, ventilators, turbocompressors) or
- volume displacement-based volumetric machines (e.g., gear pumps, lamella pumps, axial/radial piston pumps, piston compressors).

While in the case of turbomachines, there is a close relationship between the generated pressure difference and the volume flow rate (through the H(Q) characteristics of the machine), in the case of volumetric machines, the pressure difference and the flow rate are independent with good approximation. This way, for example, a gear pump can deliver approximately the same amount of flow rate in a wide pressure range. Similarly, if we use a gear pump as a motor (a given flow rate passes through the machine, and we use the torque of the shaft), the obtained motor can work in a wide range of lode with the same rotation speed (unlike an internal-combustion engine for example).

Since the volume flow rate of a positive displacement pump is relatively independent of the pressure in the hydraulic system, if the flow demand of the system drops (e.g., a cylinder stops), the pressure rapidly rises, because the pump continues to deliver the same amount of volume flow rate. For this reason, a pressure relief valve has to be implemented directly after volume displacement pumps to prevent overload (increased pressure). These valves open at a preset pressure to release the excess work fluid back to the reservoir. **A volumetric pump and a pressure relief valve together are referred to as a hydraulic aggregate.**

A characteristic of hydrostatic drives is that a hydraulic aggregate drives a cylinder or hydro motor (actuator) through various control units (valves). The use of hydrostatic drives spread because some of their significant benefits:

- the aggregate and the actuator(s) can be installed far from each other (within reasonable limits), and they can be moved easily into a new position using flexible hoses
- high power relative to a unit of machine weight
- easy controlling (e.g., it is quick and easy to reverse the rotation of a hydraulic motor)
- remotely controllable
- simple overload protection
- immediately operational after the termination of overload

- trouble proof operation
- favorable life-span
- free of electric shock protection problems

The most significant disadvantage of hydrostatic drives is that a reservoir is necessary to store the working fluid for operation. Furthermore, the system has to be closed, since the working fluid, which is usually oil (mineral or synthetic) or emulsion, can not be released into the environment. (When environmental considerations have priority or a large amount of work fluid is required, then water is used instead of oil.)

In professional and scientific literature, the shorter term “hydraulic drive” has spread instead of “hydrostatic drives”. Strictly speaking, the word hydraulic is a broader term, since it encompasses hydrostatic and hydrodynamic drives.

Figure 1 shows a schematic diagram of a simple, open-circuit, hydrostatic systems, one with a straight-motion hydro motor (cylinder) and the other one with a rotary-motion hydro motor as the working unit. The used symbols correspond to the MSZ 1981 standard.

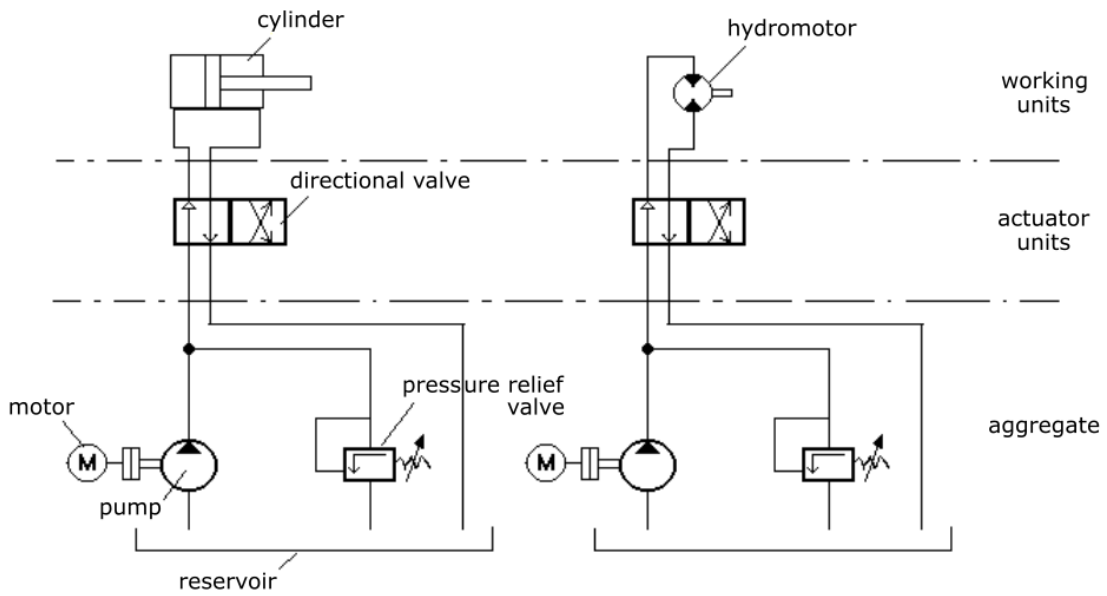


Figure 1. Schematic diagrams of hydrostatic drives

1.2. Characteristics of volumetric pumps and motors

Figure 2 illustrates the operation of a gear pump. The liquid is transported in the volume between the tooth profiles, the bottom land, and the pump casing. Gear motors have similar designs, but the principle of operation is the opposite: fluid that is flowing through the motor turns the gears.

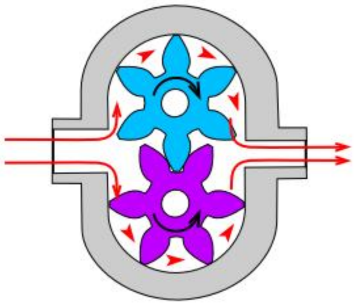


Figure 2. Theoretical operation of a gear pump

Figure 3.a. shows the characteristic curve of a volumetric pump (e.g., gear pump), while in figure 3.b., the characteristic curve of a motor can be seen rotating with different, constant speeds ($n_2 > n_1$). Dashed lines represent the ideal volume flow rate (flow discharge in case of a pump and flow demand in case of a motor). If the counter-pressure increases, then the leakage loss increases as well. This effect is why the deviation between the real and the ideal curves grows at higher pressure. The result of the increasing leakage loss is that, at a given Δp_1 pressure difference, a pump supplies less flow rate, while in the case of a motor, the volume demand increases.

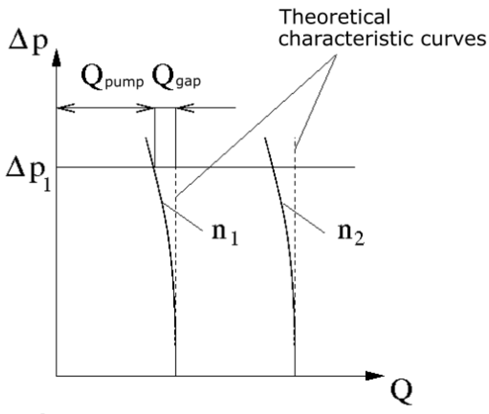


Figure 3.a.

Characteristic curves of a volumetric pump

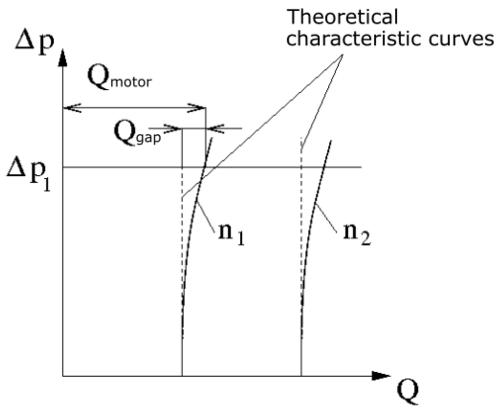


Figure 3.b.

Characteristic curves of a volumetric motor

Geometric flow discharge (V_g [cm^3]) is the transported fluid volume over one full rotation if the machine is without load (i.e., the leakage loss is zero). The ideal or geometric volume flow rate, which belongs to the $\Delta p = 0$ pressure difference can be calculated as

$$Q_g = nV_g \,. \tag{1}$$

As per usual, the leakage loss can be taken into account with the use of the volumetric efficiency factor η_v. If the pressure difference is greater than 0 ($\Delta p > 0$), then the real volume flow rate can be calculated as

$$Q_{sziv.} = \eta_v Q_g \tag{2}$$

for a pump with a constant rotation speed. To calculate the flow rate demand of a hydro motor with $\Delta p > 0$ and constant rotation speed, the following formula can be used:

$$Q_{motor} = Q_g / \eta_v \tag{3}$$

The value of the volumetric efficiency depends on the pressure difference, gaps sizes, and the fluid viscosity. Due to the fine machining of the components, the volumetric efficiency at the machine's nominal pressure is around 0.95. Also, this value does not decrease significantly if the machine is overloaded.

1.3. Characteristics of pressure relief valves

Figure 4 shows the characteristic curves of a pressure relief valve at different pressure settings.

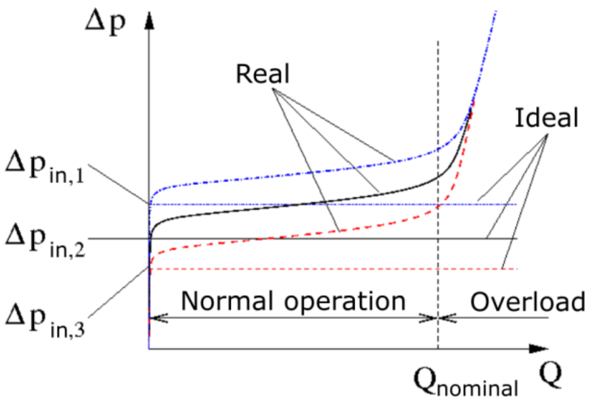


Figure 4. Characteristic curves of a pressure relief valve at different pressure settings

The characteristic curve of an ideal pressure relief valve is horizontal, meaning that it can release any amount of fluid back to the reservoir at the chosen pressure (below this pressure it stays closed). In reality, the curves slowly rise until it reaches the nominal flow rate. At higher flow rates, the valve becomes overloaded, and the pressure rapidly increases with the increasing flow rate (the valve works like a butterfly valve). In this overloaded region, the valve is unable to limit the pressure in the system.

1.4. Characteristic curves of hydraulic aggregates

Plot the resultant characteristics of a system that contains a pump and a pressure relief valve. If we consider the flow rate delivered by the pump as positive and the released flow rate through the parallelly installed pressure relief valve as negative, we gain the characteristic curve of the hydraulic aggregate (figure 5).

If the Δp_1 pressure load is lower than the Δp_{in} set pressure boundary of the relief valve, then the valve remains closed, and the flow rate of the pump supplies the system. If $\Delta p_1 > \Delta p_{in}$, then the relief valve opens, releasing part of the volume flow rate of the pump into the reservoir. This way, the pump supplies less fluid into the system. The characteristic curve of the pressure relief valve determines the quantity of the released volume flow rate.

The pressure relief valve is chosen correctly for a specific pump if the nominal flow rate of the valve is higher than the geometric flow rate of the pump so that the valve can release the total flow rate of the pump during regular operation. In the case of the stalling of the working machine (overload), the relief valve can only release the flow rate of the pump at significantly increased pressure. Thus it can not maintain its intended operation.

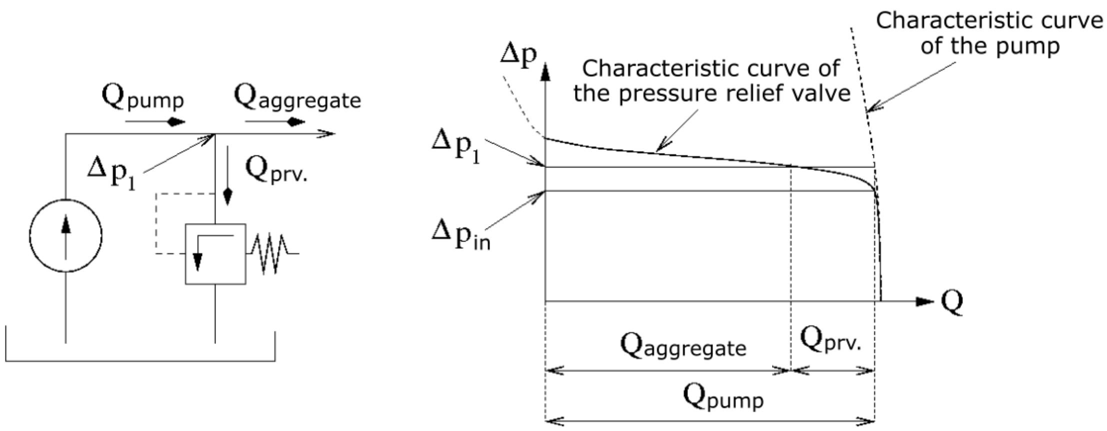


Figure 5. Hydraulic aggregate and its characteristic curve

2. Measurements of hydraulic aggregates

2.1. Objective of measurements

The measured hydraulic aggregate consists of a gear pump and a pressure relief valve. During the measurements, the following characteristic curves have to be determined:

- pressure difference as a function of volume flow rate $\Delta p(Q)$
- input power as a function of volume flow rate $P_{in}(Q)$
- volumetric efficiency as a function of pressure difference $\eta_v(\Delta p)$
- total efficiency of the machine group as a function of volume flow rate $\eta_t(Q)$

2.2. The measurement rig

Figure 6 shows the circuit diagram of the measured equipment

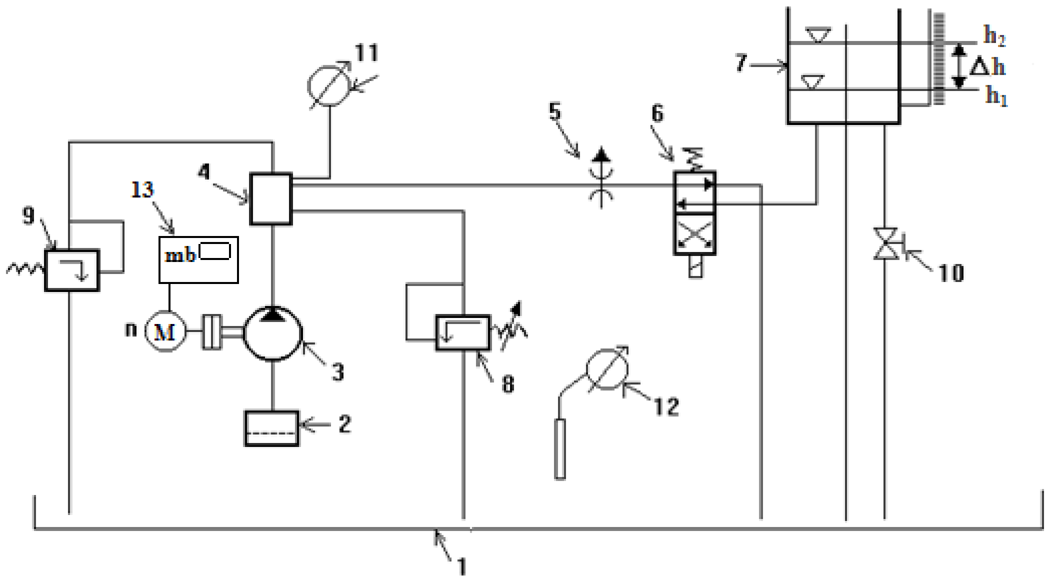
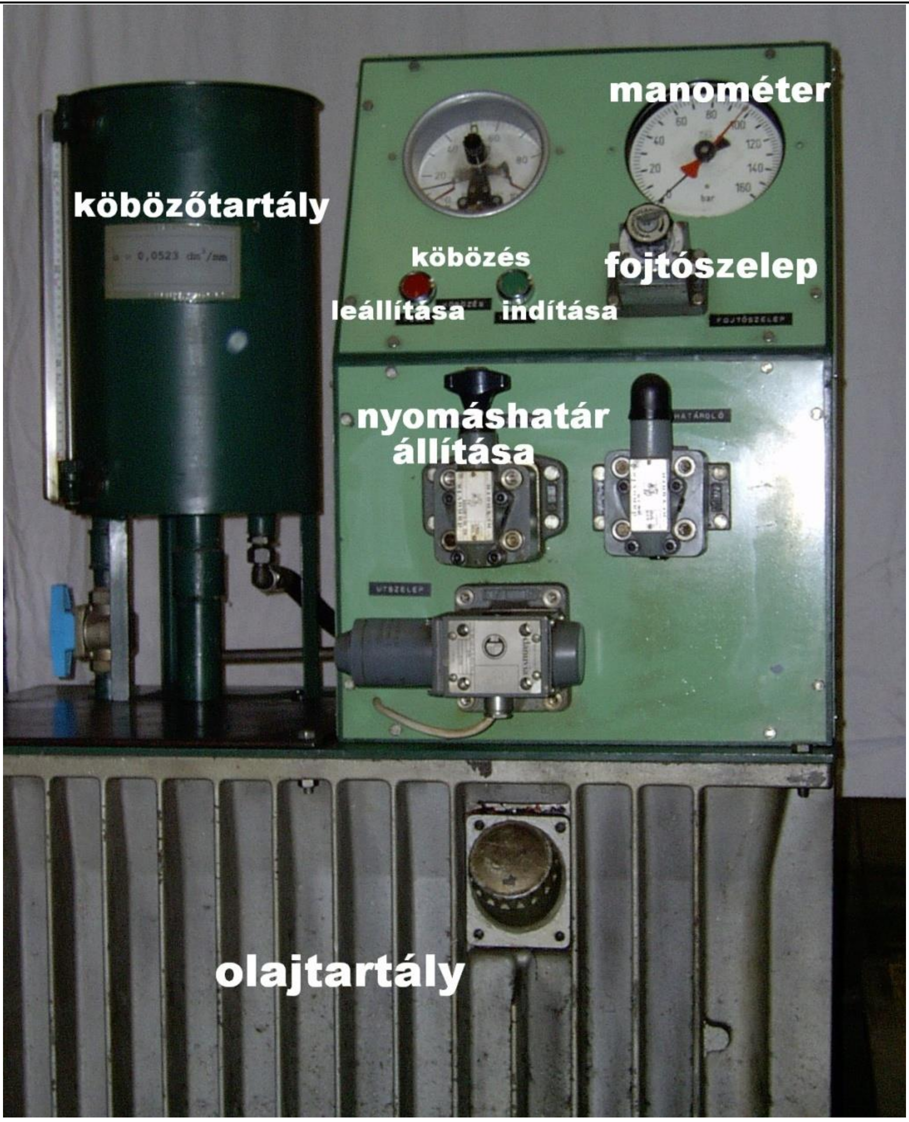


Figure 6. Diagram of the measured equipment

The pump (3) draws oil through the filter (2). The conveyed oil goes through the distributor block (4), the variable throttle (5), and the electromagnetic actuator (6) and returns into the reservoir if the valve is in its base position. In this case, the pipe leading to the displacement meter is closed. If the electromagnet of the valve (6) is activated, then the oil flows into the metering tank (7). After the volumetric flow measurement, the oil from the metering tank can be released back to the reservoir (1) by opening the tap (10). The variable throttle (5) adjusts the discharge pressure of the pump. The Bourdon type manometer measures the set pressure (picture 1).



Picture 1. Measurement equipment

The purpose of the pressure relief valve (8) is to release some of the oil back into the reservoir by opening when the pressure on the discharge side exceeds the pressure set by preloading the valve spring. This prevents the overloading of the working unit (cylinder/hydraulic motor) supplied by the aggregate. The pump is protected by a pressure relief valve (9), which opens at a pressure higher than that set by the other pressure relief valve (8). The mercury-filled tubular spring thermometer (12) monitors the temperature of the oil. The power meter built into the measurement suitcase displays the input electrical power (13).

2.3. Equipment specifications

Type of the gear pump:	A 18X
The serial number of the gear pump:	11/1984
Geometric flow discharge of the gear pump:	V_g = 8,25 cm^3
Nominal (maximum) pressure of the gear pump:	160 bar
Type of the motor:	VZP 112 M4
The serial number of the motor:	3745/9
Nominal power and rotation speed of the motor:	4 kW, 1440/min
Type of the pressure relief valve:	Danuvia DM10-2.10/315
Type of the throttle:	Danuvia F10 G3 21/40L
Constant of the used metering tank:	$\alpha = 5{,}227 \cdot 10^{-2}\,\frac{\mathrm{dm}^3}{\mathrm{mm}}$

2.4. Calculation of the quantities on the characteristic curves

The geometric volume flow rate of the pump can be calculated as $Q_g = nV_g$, where n is the rotation speed of the pump. If the pressure relief valve is closed, then the effective volume flow rate of the pump can be determined from the change of the fluid level in the metering tank ($\Delta h = h_2 - h_1$) and the measured time (Δt):

$$Q_{sziv.} = \alpha \frac{\Delta h}{\Delta t}\,. \tag{4}$$

Thus, the volumetric efficiency can be calculated as

$$\eta_v = \frac{Q_{sziv.}}{Q_g}\,. \tag{5}$$

Change in Bernoulli enthalpy:

$$\Delta i_B = \left[\frac{c^2}{2} + \frac{p}{\rho} + gh\right]_1^2, \qquad \text{ha } c_2 = c_1 \text{ és } g(h_2 - h_1) \ll (p_2 - p_1)/\rho \quad \text{,akkor}$$

$$\Delta i_B = (p_2 - p_1)/\rho = \Delta p/\rho \;,$$

The effective power of the pump:

$$P_h = \dot{m}\Delta i_B = Q\rho\Delta i_B = Q\Delta p. \tag{6}$$

Taking into account the effective pressure of the machine, we can neglect the suction pipe resistance, and the height difference of the pressure gauge taps in the calculation. With this in mind, the Δp overpressure read from the manometer is also the pressure between the pump's suction and discharge sides, since the pressure in the suction pipe is approximately atmospheric.

The input power of the drive motor is measured using a power meter built into a measurement suitcase. The input electrical power can be read directly (P_{in} $[kW]$). Finally, the overall efficiency of the motor-pump machine group:

$$\eta_{ö} = \frac{P_h}{P_{be}}. \tag{7}$$

2.5. Measurement points

The power supply is put into operation by the instructor, so the measuring group starts work on the operation equipment. The various operating conditions required for recording the measurement points are set with the throttle valve (5).

The starting fluid level in the metering tank can be adjusted by opening the tap (10). The oil in the level indicator class tube should still be visible after the draining.

The recording of the measurement points has to happen in the following order:

1. Set the operation point (see below for details).
2. Record the starting fluid level (h_1) in the cistern.
3. Switch the transfer to the metering tank by actuating the magnet of the directional valve and start the stopwatch at the same time. After a measurement time of approximately 20-30 seconds, stop the stopwatch and read the oil level at the same time. Switch the transfer back to the reservoir.
4. Read the manometer (11), the valve position (5) and the electrical power (13). Measure the rotation speed of the motor.
5. Plot the measurement point on the control chart.
6. Set the next operation point.

For low pressures (at the vertical part of the characteristic curve), set the operating point based on the pressure indicated by the manometer (11). In the operating range of the pressure relief valve (the horizontal part of the characteristic curve), since the pressure changes only slightly, adjust the operating point evenly according to the scale on the throttle valve (5). Make sure to set measurement points more frequently around the breakpoint of the curve. During the measurement, draw the control diagram continuously and, if necessary (the measurement points are too far apart), set further operating conditions accordingly.

There is a flow through the valve, even in its fully closed condition. Because of this leakage, we can not measure curve points below a minimum flow rate (approx. 2 lit/min).

When evaluating the volumetric efficiency, make sure to evaluate only the points measured with the closed pressure relief valve. With the pressure relief valve open, the metering tank measures the Q_{pump}-Q_{prv} volume flow rate instead of the flow rate of the pump.

2.6. Preparing for the measurement

a) Prepare a table for recording the measurement results (15 measurement points)

b) Prepare the control chart on a millimeter paper for the $\Delta p = \Delta p(Q)$ characteristic curve. Maximum values: $\Delta p_{max} = 100\,bar \quad Q_{max} = 15 lit/min$.

3. Control questions

1. What is a hydraulic aggregate? What is the function of a pressure relief valve?

2. What are the advantages and disadvantages of hydrostatic drives?

3. Draw the schematic diagram of a simple, open-loop hydrostatic drive with a straight-acting hydro motor (cylinder) as the working unit! Mark each element in the circuit diagram!

4. Draw the ideal and real characteristic curves of a volumetric pump at two different, constant rotation speeds!

5. Draw the ideal and real characteristic curves of a volumetric motor at two different, constant rotation speeds!

6. Draw the ideal and real characteristic curves of a pressure relief valve with two different opening pressure! In the figure, indicate the operating ranges and the nominal flow rate!

7. Describe the purpose of the measurement (characteristic curves)!

8. Sketch the measuring equipment and name its parts!

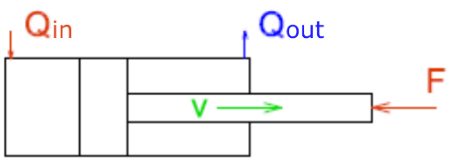
9. Group the flow machines according to the principle of operation! What is the relationship between the pressure difference and the flow rate created by the machine for each type? Give 2-2 examples!

10. Draw a hydraulic aggregate and name its parts! Draw the characteristic curve of the aggregate (Q-Δp) and describe each section!

11. Draw approximately correct characteristic curves of a pressure relief valve for two different opening pressure! Describe each section of the curve! What is the ideal shape of the characteristic curve?

4. Measurement tasks

1. The measured aggregate operates a cylinder. Determine the v_m speed of the cylinder piston based on the nominal load of the cylinder! (F – load force, D – cylinder diameter, d – piston rod diameter, F_f – frictional force in the cylinder.) The cylinder moves in the direction shown in the picture.



	F [N]	D [mm]	d [mm]	F_s [N]
a.)	3000	40	28	270
b.)	5500	50	36	350

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2. The measured aggregate operates a hydro motor. Determine the n_m rotation speed of the hydro motor shaft based on the nominal load of the hydro motor! (M – load torque, η_{vm} – volumetric efficiency of the hydro motor, V_{gm} – geometric flow discharge of the hydro motor, $\eta_{öm}$ – total efficiency of the hydro motor.)

	M [Nm]	V_{gm} [cm^3]	$\eta_{öm}$	η_{vm}
a.)	5,0	10	0,75	0,92
b.)	8,0	15	0,80	0,94

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3. Draw the characteristic curve of the hydraulic aggregate, if the rotation speed of the motor is reduced by

   a.) 20 %

   b.) 40 %.
