LECTURE 20 – FLOW CONTROL VAVLES

SELF EVALUATION QUESTIONS AND ANSWERS

1: A cylinder has to exert a forward thrust of 150 kN and a reverse thrust of 15 kN. The effects of using various methods of regulating the extend speed will be considered. In all the cases, the retract speed should be approximately 5m/min utilizing full pump flow. Assume that the maximum pump pressure is 160 bar and the pressure drops over the following components and their associated pipe work (where they are used):

Filter = 3 bar

Directional control valve (DCV)	**= 2 bar**
Flow control valve (controlled flow)	**= 10 bar**
Flow control valve (check valve)	**= 3 bar**

Determine:

- **(a) The cylinder size (assume 2:1 ratio piston area to rod area)**
- **(b) Pump size, and**
- **(c) Circuit efficiency when using:**

Case 1: No flow controls (calculate extend speed)

Case 2: Meter-in flow control for extend speed 0.5 m/min

Case 3: Meter-out flow control for extend speed 0.05 m/min

2: A flow control valve is used to control the speed of the actuator as shown in the figure 1 and the characteristics of the system are given in the following table. Determine the cylinder force Fp, and the cylinder piston velocity, that is developed in the system when cylinder flow, Q2, is equal to 80% of the pump flow Q.

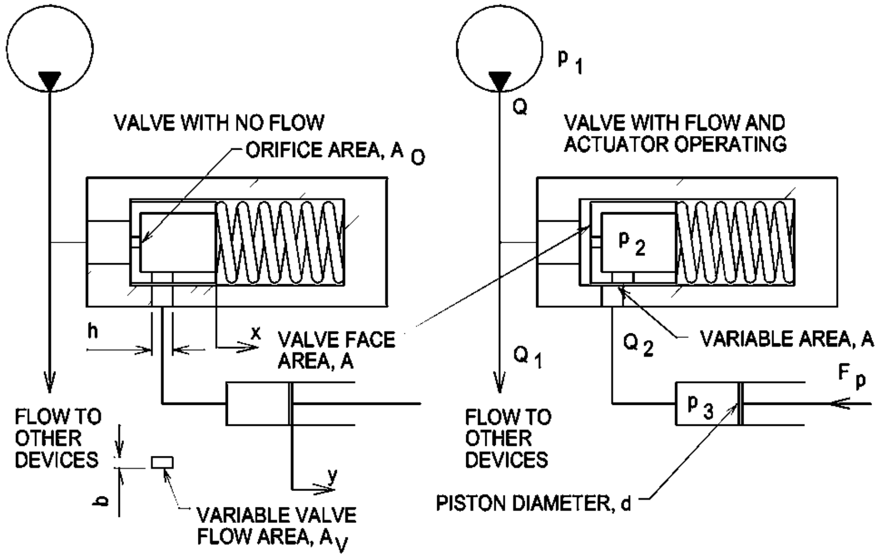

Figure 1

Parameters	Value
Orifice discharge constant (C_d)	0.6
Length of flow area(A_v), h	6 mm
Width of the flow area(A_v), b	1.5 mm
Fixed orifice flow area (A_0)	5 mm^2
Valve face area	130 mm^2
Piston diameter , d	38 mm
Spring preload, F	255 N
Spring constant	65.7 kN/m
Drain side pressure (p_d)	0
System pressure (p_1)	13.97 MPa
Fluid density	$830\frac{kg}{m^3}$
Pump flow	0.32 LPS

3: Determine the flow – rate through a flow control valve that has a capacity coefficient of $\mathbf{3.1}\ \boldsymbol{LPM/\sqrt{kPa}}$ and a pressure drop of 897 kPa. The fluid is hydraulic oil with a specific gravity of 0.95.

4: A 60 mm diameter sharp edged orifice is placed in a pipe line to measure flow rate. If the measured pressure drop is 500 kPa and the fluid specific gravity is 0.92, find the flow rate in units of $\boldsymbol{m^3/s}$

5: The speed control circuit shown in figure 2 has the following data:

Component A: Capacity 120 ml/rev. Shaft speed 1000 rev/min. Volumetric efficiency = 95 % . Component B: set at 70 bar. No pressure over ride. Component C: Flow equation Q_c = 80 × a × $\sqrt{\Delta P}$, where Q_c = ml/min area(a)=200mm^2.Component D: Capacity 160 ml/rev. Volumetric efficiency = 95 %. Mechanical efficiency =80 %. Load = constant torque (60 Nm).Determine the flow through the valve C under the conditions and maximum speed of motor.

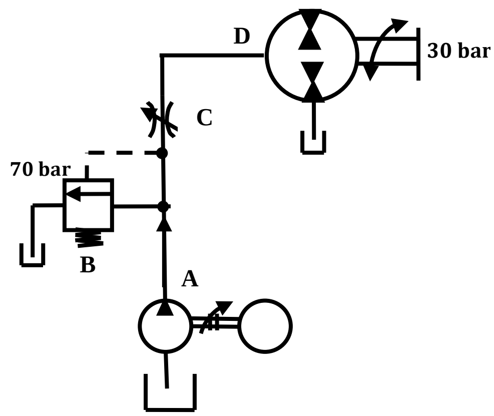

Figure 2 : Speed control of hydraulic motor

6: The system shown below (Figure 3) has a hydraulic cylinder with a suspended load W. The cylinder piston and rod diameter are 60 mm and 30 mm respectively. The pressure relief valve setting is 5000 kPa. Determine the pressure P_2 for a constant cylinder speed,

(a) **W = 10000 N**
(b) **W= 0 (load is removed)**
(c) **Determine the cylinder speeds for parts (a) and (b) if the flow control valve has a capacity coefficient of $0.75\ LPM/\sqrt{kPa}$. The fluid is hydraulic oil with a specific gravity of 0.90**

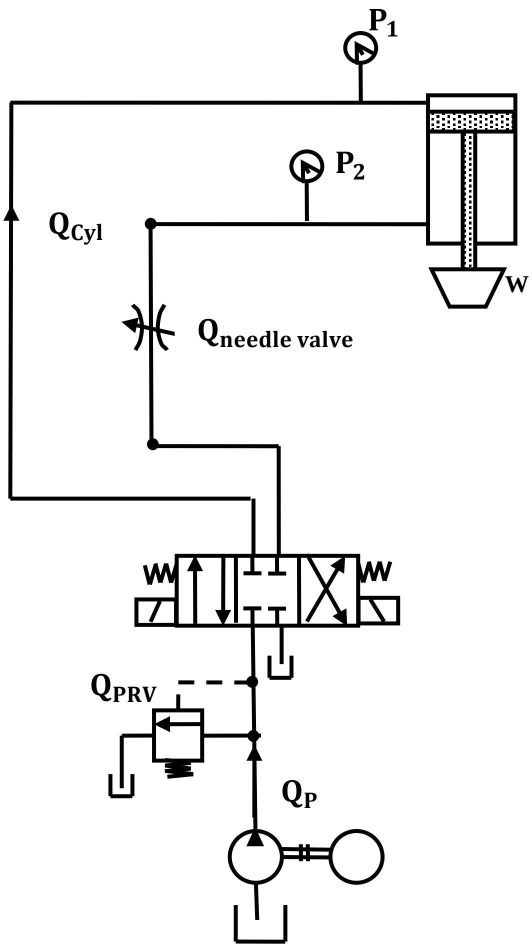

Figure 3 : Hydraulic cylinder with suspended weight

Q1 Solution:

Case 1: No flow controls

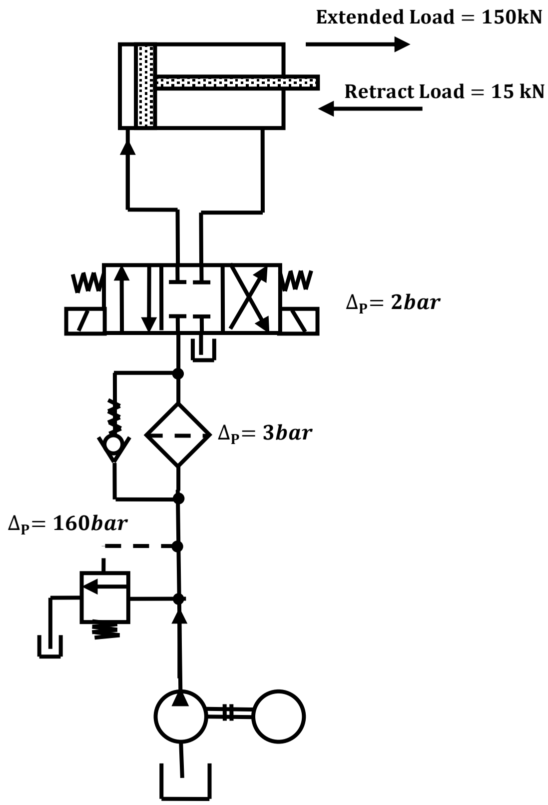

Figure 1 : Hydraulic cylinder with no control

Maximum available pressure at full bore end of cylinder is =160-3-2=155bar

Back pressure at annulus side of cylinder = 2 bar. This is equivalent to 1 bar at the full bore end because of 2:1 area ratio. Therefore, maximum pressure available to overcome load at full bore end is 155-1 = 154 bar

Full bore area = Load/Pressure= $\frac{150 \times 103}{154\ x\ 105}$ =0.00974 m^2

Piston Diameter = $\left(\frac{4 \times 0.00974}{\pi}\right)^{\frac{1}{2}}$

Select a standard cylinder; say with 125 mm bore and 90 mm rod diameter.

Full bore area = 12.27×10^{-3} m^2

Annulus Area $= 6.36 \times 10^{-3}$ m^2

This is approximately a 2:1 ratio

Flow rate for a return speed of 5 m/min =Annular Area x velocity

$= (12.27\text{-}6.36) \times 10^{-3} \times 5$ m^3/min=29.55 LPM

Extend speed $= \frac{29 \times 10^{-3}}{12.27 \times 10^{-3}}$ = 2.41 m/min

Pressure to overcome load on e×tend is$= \frac{150 \times 10^{3}}{12.27 \times 10^{-3}}$= 12.223 MPa = 122.3 bar

Pressure to overcome load on retract is $\frac{15 \times 10^{3}}{6.36 \times 10^{-3}}$ = 2.538 MPa = 25.38 bar

i) Pressure at pump on extend (working back from DCV tank port)

Pressure drop over DCV B to T	2 × ½	1
Load induced pressure		122.23
Pressure drop over DCV P to A		2
Pressure drop over filter		3

Therefore, pressure drop required at pump during e×tend stroke = 128.33 bar

Relief valve setting = 128.23 + 10%= 141.05 bar

ii) Pressure required at the pump on retract (working from DCV port as before)

$= (2 \times 2) + 25.38 + 2 + 3 = 34.38$ bar

Note: The relief valve will not be working other than at the extremities of the cylinder stroke. Also, when movement is not required, pump flow can be discharged to tank at low pressure through the centre condition of the DCV.

iii) System efficiency

$$\frac{\text{Energy required to overcome load on cylinder}}{\text{Total energy into fluid}}$$

$$= \frac{\text{Flow to cylinder} \times \text{Pressure owing to load}}{\text{Flow from pump} \times \text{Pressure at pump}}$$

Efficiency on extend stroke $= \frac{29.551 \times 122.23}{29 \times 141.05} \times 100 \quad = 86.66\ \%$

Efficiency on retract stroke $= \frac{29.551 \times 25.38}{29.55 \times 34.380} \times 100 \quad = 73.822\ \%$

Case 2: Meter-in flow control for extend speed of 0.5 m/min

From Case 1,

Cylinder 125 mm bore diameter × 90 mm rod diameter

Full bore area $= 12.27 \times 10^{-3}\ m^2$

Annulus area $= 7.525 \times 10^{-3}\ m^2$

Load induced pressure on extend = 122.23 bar

Load induced pressure on retract = 25.38 bar

Pump flow rate = 29.55 l/min

Flow rate required for extend speed of 0.5 m/min is $= 12.27 \times 10^{-3} \times 0.5$

$= 6.136 \times 10^{-3}\ m^3/min$

= 6.136 l/min

Working back from DCV tank port:

Pressure required at pump on retract is 2 × 2) + (2 × 3) + 25.38 + 2 + 3 = 40.38 bar

Pressure required on pump at extend is (2 × ½) + 122.23 + 10 + 2 + 3 = 138.23 bar

Relief valve setting = 138.23 + 10 % = 152 bar

This is close to the maximum working pressure of the pump (160 bar). In practice, it would be advisable to select either a pump with a higher working pressure (210 bar) or use the next standard size of the cylinder. In the latter case, the working pressure would be lower but a higher flow rate pump would be necessary to meet the speed requirements.

Now that a flow control valve has been introduced when the cylinder is on the extend stroke, the excess fluid will be discharged over the relief valve.

System efficiency on extend is = $\frac{6.136 \times 138.23}{29.551 \times 152.05} \times 100 = 18.877$ %

System efficiency on retract is= $\frac{29.551 \times 25.38}{29.552 \times 40.38} \times 100 = 62.853$ %

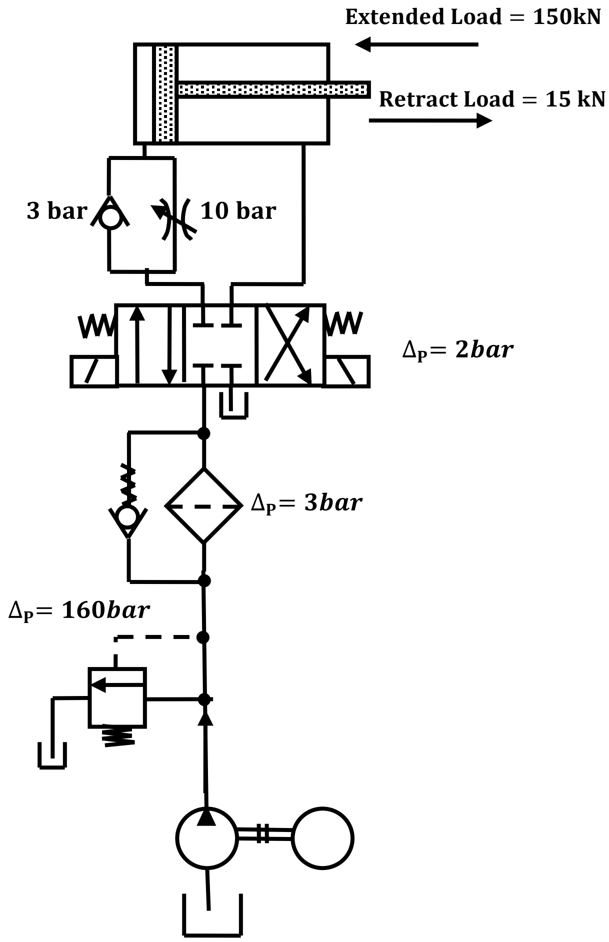

Figure E4 : Hydraulic cylinder with meter in control

Case 3 : Meter-out flow control for extend speed of 0.5 m/min

Cylinder, load, flow rate and pump details are as before. Working back from DCV tank port:

Pressure required at pump on retract is= $(2 \times 2) + 25.38 + 3 + 2 + 3$ = 37.38 bar

Pressure required at pump on extend is $=(2 \times ½) + (10 \times ½) + 122.23 + 2 + 3$ =133.23 bar

Relief valve setting is 133.33 + 10 % = 146.55 bar

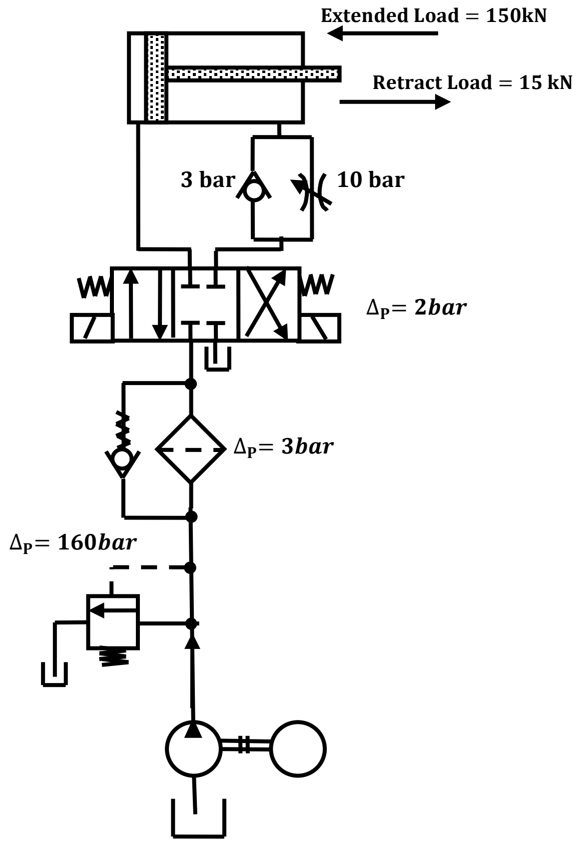

Figure E5 : Hydraulic cylinder with meter out control

System efficiency on extend is= $\frac{6.136 \times 122.23}{29.551 \times 133.23} \times 100 \quad =19.05\ \%$

System efficiency on retract is= $\frac{29.551 \times 25.38}{29.551 \times 37.38} \times 100 \quad = 67.897\ \%$

As can be seen, meter-out is marginally more efficient then meter-in owing to the ratio of piston area to piston rod area. Both systems are equally efficient when used with through rod cylinders or hydraulic motors. It must be remembered that meter-out should prevent any tendency of the load to run away.

Q2 Solution

Flow from pump divides as $Q_1$and Q_2. The pressure drop $P_1 - P_2$occurs acoss office A_o This make valve to move to right against the spring force F. the area of orifice A_v then adjusts to control the flow to the motor.

$$h = \frac{6}{1000}m, b = \frac{1.5}{1000}mm.\ k = 65000N/m\ \ A_o = 5\times 10^{-6}m^2\ A = 130\times 10^{-6}m^2$$

$$D = \frac{38}{1000}, F = 255\ N \quad \rho = 830\frac{kg}{m^3} P_1 = 13970\times 1000\ N/\ m^2$$

$$Q_p = \frac{0.32}{1000} Q_2 = 0.8 \times Q_p = 2.56x\ 10^{-4}m^3/s$$

$$p_3 = \frac{T}{0.97}\times\frac{2\pi}{\frac{40}{100^3}m^3} + p_4 = 1.072 \times 10^7\ Pa$$

$$Q_2 = c_d A_o\sqrt{\frac{2}{830}}\times\sqrt{p_1} - p_2 = 2.56\times\frac{10^{-4}m^3}{s}, solving\ we\ get$$

$$p_2 = 1.095 \times 10^7\ Pa$$

$$p_1 A = p_2 A + K\ X + F\ \ solving\ we\ get\ x = 2.098\times 10^{-3}m = 2.098\ mm$$

Q3 Solution

For a sharp edged orifice we can write,

$$Q = 3.1\sqrt{\frac{\Delta p}{SG}} = 3.1\sqrt{\frac{897}{0.95}} = 95.25\ \ LPM$$

Q4 Solution

For a sharp edged orifice we can write,

$$Q = 0.0851\ A\ C_V\sqrt{\frac{\Delta p}{SG}}$$

Where Q= volume flow rate in LPM

C_V= capacity coefficient = 0.80 for sharp edges orifice, c= 0.6 for square edged orifice.

A = area of orifice opening in mm^2

Δp = pressure drop across the orifice (kPa.

SG= specific gravity of flowing fluid = 0.92

$$A_{Orifice=}\frac{\pi}{4}\left(D_{orifice}{}^{2}\right)=\frac{\pi}{4}(60^2)=2827.4mm^2$$

$$Q(LPM)=0.0851x2827.4\,x0.80\sqrt{\frac{500}{0.92}}=4487.35\text{ LPM}$$

Q5 Solution

HP = 2;

$mM_t = 9 \times p \times m$

$M_t = \frac{9 \times p}{0.8}$

Torque exerted by the motor $= \frac{60}{0.8} = 75$ Nm

$M_t = \frac{Q \times p}{2} = 160 \times p/2$

$p = 75 \times 2 \times \pi \times 10/5160$

$= 30$ bar

$\Delta P = 40$ bar

$Q_c = 80 \times 200 \times 40^{1/2} = 102000$ ml/min = 102 l/min

Actual flow to the motor = 96.9 l/min

Motor speed = 96.9 × 10 / 160 = 605 rpm

Q6 Solution

For constant cylinder speed, the summation of the forces on the hydraulic cylinder must be equal to zero. Thus we have.

$$-W - P_1 A_P + P_2(A_P - A_R) = 0$$

$$where\ P_1 = pressure\ relief\ valve\ setting = 5000\ kPa$$

$$A_{P=} \frac{\pi}{4}\left(D_P{}^2\right) = \frac{\pi}{4}(0.06^2) = 0.0028273\ m^2$$

$$A_{R=} \frac{\pi}{4}\left(D_R{}^2\right) = \frac{\pi}{4}(0.03^2) = 0.000707\ m^2$$

$$A_P - A_R = 0.00212\ \ m^2$$

$$\boldsymbol{Case\ 1: if\ \ W = 10000\ N}$$

$$-W - P_1 A_P + P_2(A_P - A_R) = 0$$

$$-10000 - 5000x10^3 \frac{N}{m^2} x2.8273x10^{-3} m^2 + P_2(0.00212\ m^2) = 0$$

$$P_2 = 11{,}382\ kPa$$

$$\boldsymbol{Case\ 2: if\ \ W = 0}$$

$$0 - 5000x10^3 \frac{N}{m^2} x2.8273x10^{-3} m^2 + P_2(0.00212\ m^2) = 0$$

$$P_2 = 6668\ kPa$$

$$\boldsymbol{Case\ 3: cylinder\ speed\ for\ case\ 1}$$

For a sharp edged orifice we can write,

$$Q = C_v \sqrt{\frac{\Delta p}{SG}} = 0.75 \sqrt{\frac{11382{,}700}{0.9}} = 84.3\ LPM$$

Where $\Delta p = P_2$ since the flow control valve discharges directly to the oil tank. This is the flow rate through the flow control valve and thus the flow rate of the fluid leaving the hydraulic cylinder. Thus we have

$$V_p(A_P - A_R) = Q$$

$$V_p({}^{m}/_{s})(0.00212)m^2 = 84.3\ \frac{L}{min} x \frac{1m^3}{10^3 L} x \frac{1\ min}{60\ s}$$

$$V_p = 0.66\ \ m/s$$

$\boldsymbol{Case\ 4: cylinder\ speed\ for\ case\ 2}$

$$Q = C_v \sqrt{\frac{\Delta p}{SG}} = 0.75 \sqrt{\frac{6668}{0.9}} = 64.5\ LPM$$

$$V_p({}^{m}/_{s})(0.00212)m^2 = 64.5\ \frac{L}{min} x \frac{1m^3}{10^3 L} x \frac{1\ min}{60\ s}$$

$$V_p = 0.5\ \ m/s$$