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Selection of hydraulic components for body hoist control of dumper

Hydraulic system is generally used in off-road vehicles for their efficient operation in wider speed range of operation and easier monitoring. Dumper is one of the important off-road vehicle used in open pit mines for transporting loose material from one place to another. It consists of a body which is mounted on the chassis and raised by means of the hydraulic actuator. The hydraulic system of dumper consists of the hoist, steering, brake and the brake cooling circuits. The selection of the basic hydraulic components of the dumper depends on its maximum dumping capacity. This paper discusses the basic approach of selecting the major hydraulic components for the body hoist control of rear discharge 50 tonne dumper. This paper also highlights the effects of the performance characteristics of the major components like hydraulic pump and the overall efficiency of the proposed hydraulic system.

Keywords: *Hydraulic system; off-road vehicles; dumper; open pit mines; hydraulic actuator; hydraulic pump.*

Notation

F_{max}	:	maximum load for hoisting the cylinder
f_{max}	:	maximum load per cylinder
L	:	stroke length of cylinder
α	:	dump angle
D	:	bore size diameter
d	:	rod size diameter
D_p	:	displacement of the pump
N	:	engine speed
v	:	velocity of the cylinder
t	:	body hoisting time of the cylinder
Q	:	pump flow rate
η_v	:	volumetric efficiency of the pump
η_o	:	overall efficiency of the pump
η_m	:	mechanical efficiency of the pump

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1. Introduction

A dump truck or dumper is a truck used for transporting loose material (such as coal, over-burden, sand, gravel or dirt) for one place to other in mines or construction sites.

A conventional rear dumper is shown in the Fig.1. Its body is mounted on the chassis and raised by means of hydraulic actuator.

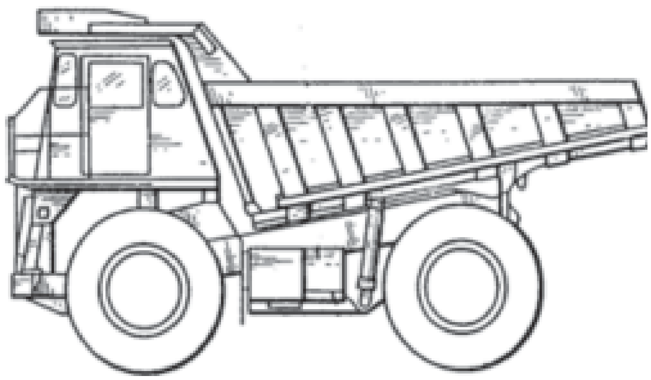

Fig.1 General view of rear discharge dumper

Dumpers up to 85 tonne capacity are powered by diesel engine with power-shift transmission and torque converter. Final gear reduction to the drive wheels is accomplished through a sun geared planetary drive. Fluid torque retarder braking system is incorporated in high capacity dumper system. Suspension systems are leaf spring, hydro-pneumatic or rubber cushioned type.

The diesel power can be transmitted to the drive wheels by either mechanical or electrical means. Usually dumpers in excess of 85 tonne capacity are all powered by diesel electric power trends. Power is supplied to the rear wheels via sun gear planetary driven by independent electric motor, usually mounted within wheel assembly. Electric current is supplied to the motor by means of a diesel driven generator or alternator.

The main sub-systems of the dumper are as follows:

1. Prime mover
2. Power transmission system

3. Braking system
4. Steering system
5. Hoisting system

The prime mover of dumper is a diesel engine and its power varies from 300hp to 1600hp depending upon the capacity. The engines are turbocharged diesel engine.

The steering system is hydraulically operated Ackermann's steering mechanism. Usually hoisting, steering and braking system may be powered from a single hydraulic pump source or for the steering or braking operation there may be a separate hydraulic pump. This depends on the design of the system.

The operation of the hydraulic steering is hydro-static. This means that there is no mechanical connection between the steering columns and steering wheels. Instead there are hydraulic pipe and hoses between the steering unit and steering cylinder. When the steering wheel is turned, the steering unit meters oil volume proportional to the amount of turn. The volume of oil led to the appropriate side of the steering cylinder. The steering unit returns automatically to its neutral position when turning is completed.

Transmission in dumpers fall into two categories: mechanical and electric drives. The brake-off point between the two systems is 85 tonne capacity. Mechanical drive dumper predominates below this level and diesel electric dumpers above it.

Fully automatic power-shift transmission generally consist of a torque converter, retarder and hydraulic clutch operated planetary gear box. The planetary gear sets are shifted under full power to the wheel. Final drive is affected by the mechanical linkage through the drive shaft to differential and final planetary gear box.

Power-shift transmissions normally have 3 to 6 gear ratio, including lock-up clutch which locks the torque converter into direct drive. The engagement or disengagement of this clutch is an automatic operation and depends on converter, output shaft and oil pressure for activation.

II. The physical system

Fig.2 shows the hydraulic system for body hoist control of dumper, considered for the analysis.

This system consists of a pressure compensated variable displacement pump which is driven by diesel engine at the speed of 2100 RPM. The following information discusses basic operation and oil flow through the entire system when the engine is running and all controls are in the neutral position. The hydraulic tank (1) provides the oil supply for the entire system. The pressure compensated pump (3) is supplied oil through the strainer (2) to the identical two stage double acting telescopic cylinders (8). The flow is equally divided between the two identical telescopic cylinders. A pressure reliefvalve (4) is set at the pressure 20% higher than the setting pressure of pressure compensator. A four port, four position solenoid controlled directional control valve (5) is used in this system. A counterbalance valve (7) is used to provide a cushion effect at extreme end position and protect the cylinders from damaging. [1]

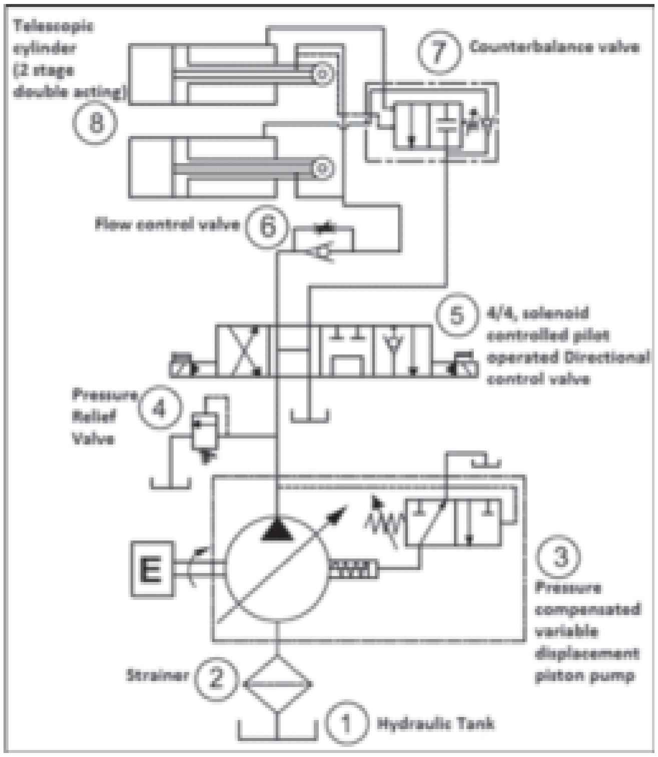

Fig.2 Hydraulic system for body hoist control of dumper

III. Design of hydraulic system

The methodology used for designing the hydraulic system for body hoist control of dumper is given below:

Centre of Gravity of Dump Body

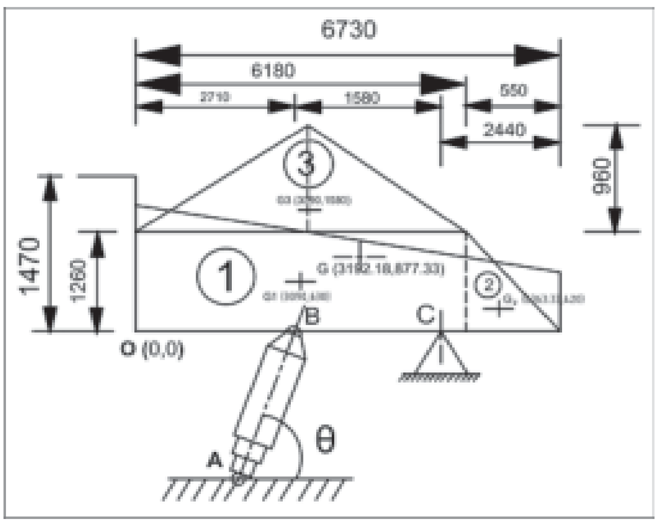

Fig.3 Calculation of centre of gravity of dump body

The dump body is in loading condition. The centre of gravity of the dump body is in loading condition is given below:

$$X = \frac{(A_1 \times Y_1) + (A_2 \times Y_2) + (A_3 \times Y_3)}{(A_1 + A_2 + A_3)}$$

$$X = \frac{(1260\times 6180)\times 3090 + (\frac{1}{2} 550\times 1260)\times (6180 + \frac{550}{3}) + (\frac{1}{2}\times 6180\times 960)\times 3090}{(1260\times 6180) + (\frac{1}{2}\times 550\times 1260) + (\frac{1}{2}\times 6180\times 960)}$$

X = 3192.18 mm

$$Y = \frac{(A_1 \times Y_1) + (A_2 \times Y_2) + (A_3 \times Y_3)}{(A_1 + A_2 + A_3)}$$

$$X = \frac{(1260\times 6180)\times 630 + (\frac{1}{2} 550\times 1260)\times (\frac{1260}{2}) + (\frac{1}{2}\times 6180\times 960)\times 1580}{(1260\times 6180) + (\frac{1}{2}\times 550\times 1260) + (\frac{1}{2}\times 6180\times 960)}$$

Y = 877.33 mm

The center of gravity acts at point G (3192.18, 877.33).

Force analysis of dump body

The free body diagram of dump body at loading condition is given below. For calculating the maximum force some assumptions are considered:

- Cylinder is fixed at an angle of 50 degree with horizontal.
- Floor of dump body is friction less.
- The dump angle (á) is the minimum angle where no sliding of material takes place.

For calculating the maximum force for hosting the cylinder, take moment about point C.

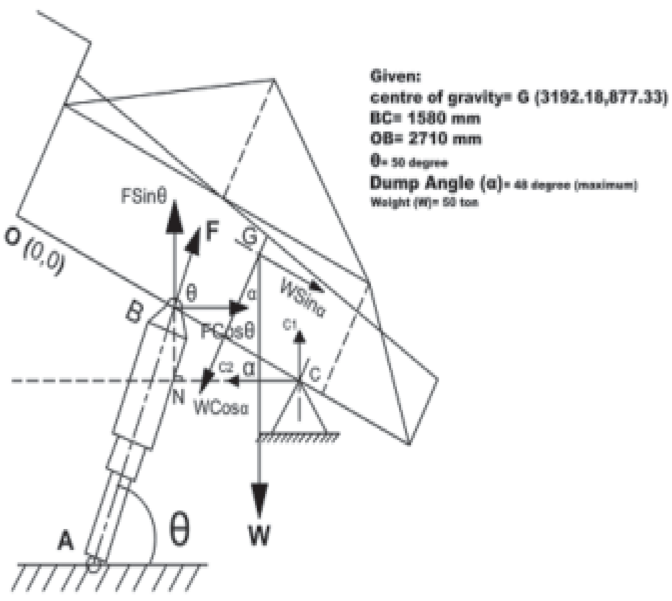

Fig.4 Free body diagram of dump Body

$$[F \sin\theta \times BC \cos\alpha + F \cos\theta \times BC \sin\alpha - W \cos\alpha \times (OB + BC - OG') + W \sin\alpha \times GG'] = 0$$

$$F = \frac{W[(OB + BC - OG') \times \cos\alpha - GG' \sin\alpha]}{BC \times [\sin(\theta + \alpha)]} \qquad \text{... ... (1)}$$

Putting the numerical values in equation (1),

$$F = \frac{50000 \times 9.81[1097.82 \cos\alpha - 877.33 \sin\alpha]}{1580 \times [\sin(50 + \alpha)]} [N]$$

$$F = \frac{310.443[1097.82 \cos\alpha - 877.33 \sin\alpha]}{1000 \times [\sin(50 + \alpha)]} [kN] \qquad \text{... ... (2)}$$

For different values of dump angle (á), the following values of forces are obtained which is shown in Fig.5.

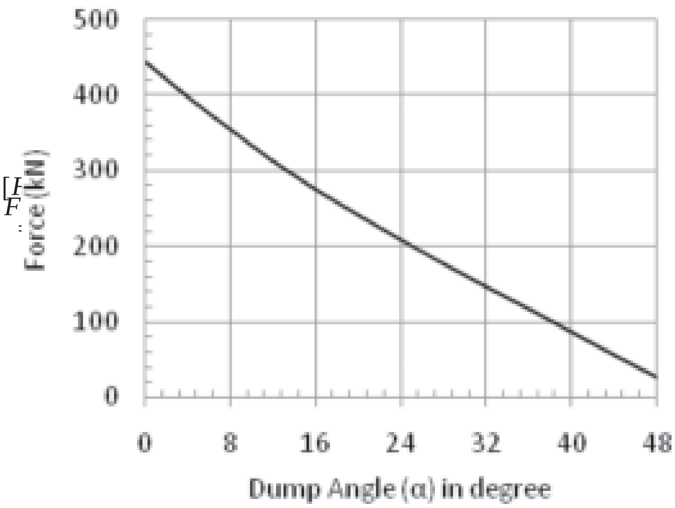

Fig.5 Graph between force versus dump angle

From Fig.5 it is clear that maximum force occurs at when dump angle is 0^o and the value of maximum force is:

F_{max} = 444.896 kN

Assume 30% factor of safety.

Then, maximum load

F_{max} = 444.896 × 130 kN

F_{max} = 578.36 kN

F_{max} = 580 kN

This system has used two telescopic cylinders. Then maximum load is distributed on both the cylinders equally.

Maximum load per cylinder; $f_{max} = \frac{580}{2} = 290kN$

STROKE LENGTH OF CYLINDER

In triangle BB'C apply sine rule,

$$\frac{BC}{\sin\theta} = \frac{L}{\sin\alpha} = \frac{B'C}{\sin[180^\circ - (\theta+\alpha)]}$$

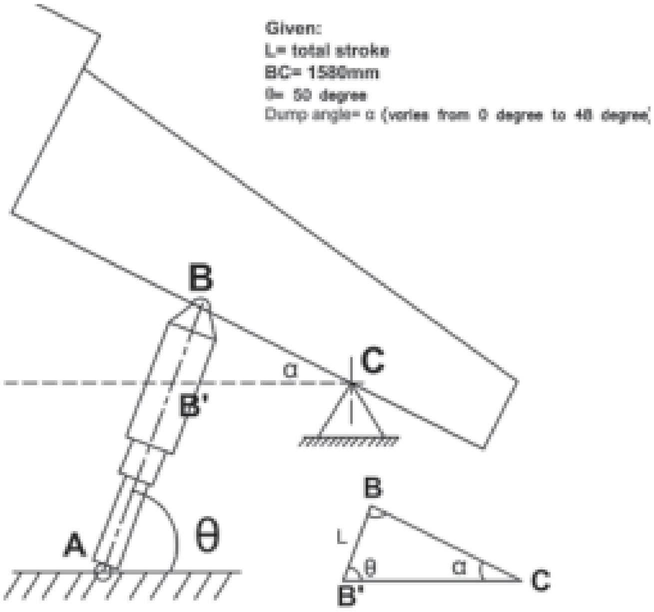

Fig.6 Calculation of total stroke length of cylinder

Taking only;

$$\frac{BC}{\sin\theta} = \frac{L}{\sin\alpha}$$

$$L = \frac{BC \times \sin\alpha}{\sin\theta}$$

$$L = \frac{1580 \times \sin\alpha}{\sin 50}$$

$$L = 2062.54 \times \sin\alpha \ \ [mm] \qquad \text{... ... (3)}$$

For different values of dump angle (á), the following values of stroke length are obtained which is shown in Fig.7.

From Fig.7 it is cleared that maximum stroke length occurs when dump angle is 48^o and minimum stroke length occurs when dump angle is 0^o. The value of maximum stroke length is:

$$L = 2062.54 \times \sin\alpha \ \ [mm]$$

$$L = 2062.54 \times \sin 48^\circ \ \ [mm]$$

L = 1532.77mm = 153.277cm = 154cm

3.4 SELECTION OF HYDRAULIC COMPONENTS

3.4.1 Selection of cylinder

For designing of the cylinder following assumptions are considered:

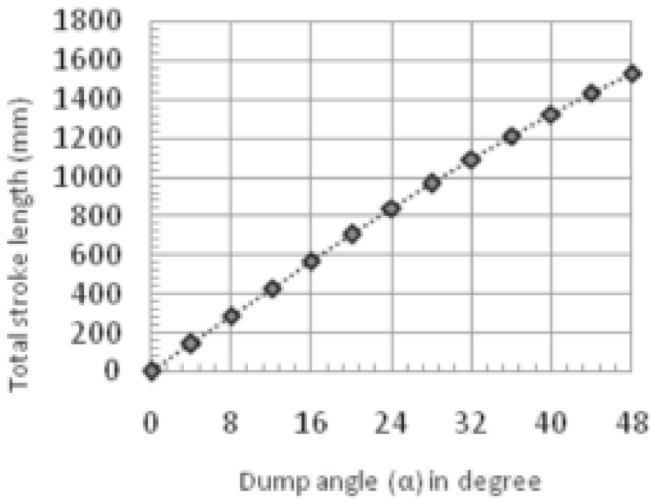

Fig.7 Graph between total stroke length versus dump angle

- The system has two-stage double acting telescopic cylinder.
- Maximum required pressure at full bore end of cylinder is = 2750 psi = 190 bar
- Maximum load per cylinder; f_{max} = 290kN

From using formula, $P = \frac{f_{\max}}{A}$ and putting the numerical values in equation:

$$190 \times 50^5 = \frac{290 \times 10^3 \times 4}{\pi \times D^2}$$

$$D^2 = \frac{290 \times 10^3 \times 4}{190 \times 10^5 \times \pi}$$

D = 0.139405m

D = 13.94cm

D = 5.49 inch

This system has used standard size, Prince (Model: PMC/SAE-53), 2-STAGE, double acting telescopic cylinder of bore sizes of 5.5×4.5 inches and rod sizes of 5×4 inches. [3]

3.4.2 Selection of pump

Total stroke length is, L = 1532.77mm = 153.277cm = 154cm

Hoisting cylinder is two-stage double acting telescopic cylinder and stroke of each stage is supposed to equal.

Then, first stage stoke length = second stage stroke length = $\frac{154}{2} = 77cm = 0.77m$

Assume total body raise time is 14 seconds and for raising first and second stages it takes 8 seconds and 6 seconds respectively.

Then, first-stage extended speed $= v_1 = \frac{L_1}{t_1} = \frac{0.77}{8}$

= 0.09625 m/s.

Second-stage extended speed $= v_2 = \frac{L_2}{t_2} = \frac{0.77}{6}$

$= 0.1283$ m/s.

Now, required flow rate for first stage extend stroke = first-stage extend speed × first-stage extend area

$Q_1 = v_1 \times A_1$

$$Q_1 = 0.09625 \times \frac{\pi \times D^2}{4}$$

$$Q_1 = 0.09625 \times \frac{\pi \times (0.1397)^2}{4}$$

$$Q_1 = 1.4753 \times 10^{-3} \frac{m^3}{s}$$

$Q_1 = 88.52\ LPM$

Required flow rate for second-stage extend stroke = second-stage extend speed × second-stage extend area

$$Q_2 = v_2 \times A_2$$

$$Q_2 = 0.1283 \times \frac{\pi \times D^2}{4}$$

$$Q_2 = 0.1283 \times \frac{\pi \times (0.1143)^2}{4}$$

$$Q_2 = 1.31646 \times 10^{-3} \frac{m^3}{s}$$

$Q_2 = 78.99\ LPM$

For designing the pump, maximum required flow rate is considered.

Then, $Q = 88.52\ LPM$

This required flow rate is only for single cylinder. Then the total flow rate required for both the cylinder is = 88.52 × 2 = 177.04 = 178 LPM

Assume total flow loss in the circuit due to leakage is 10%. [2]

Then, maximum required flow rate for the system is

$Q_{max} = 178 \times 1.10 = 195.8\ LPM = 196\ LPM$

And pump is driven by diesel engine at 2100 RPM.

Therefore, displacement of the pump:

$$D_p = \frac{Q}{N}$$

$$D_p = \frac{196 \times 10^{-3} \times 10^6 cm^3}{2100} \frac{cm^3}{rev} = 93.33 \frac{cm^3}{rev} = 94 cm^3/rev$$

Before selection of standard size pump following points are taken into consideration:

- Assume total pressure drop in the system due to flow control valve, DCV, counter-balance valve and pipe work is 15 bar.
- Maximum required pressure for the system is = 190+15= 205 bar
- Maximum required flow rate for the system is = 196 LPM
- Maximum required displacement of the pump is = 94 cm^3/rev
- Maximum required speed = 2100 RPM

This system has used standard size, Parker: PVAC 100 series, pressure compensated variable displacement piston pump of displacement 100 cm^3/rev and speed rating varies between 600 RPM to 2600 RPM. [4]

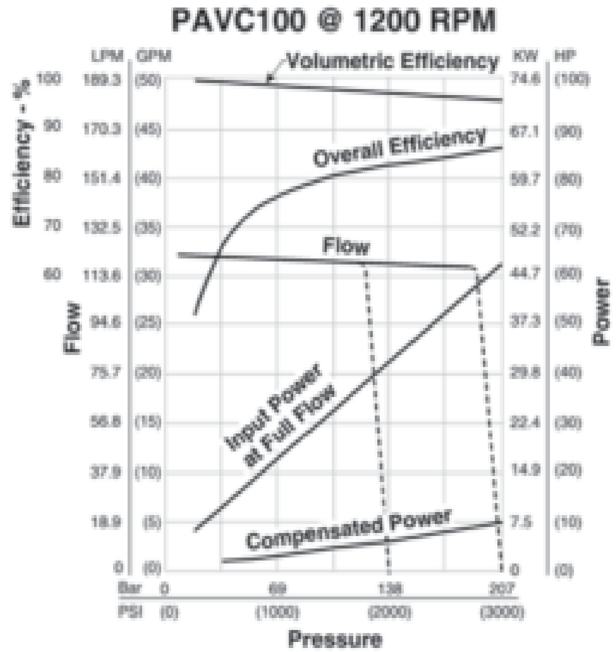

Fig.8 Characteristics curve of pump at 1200 RPM

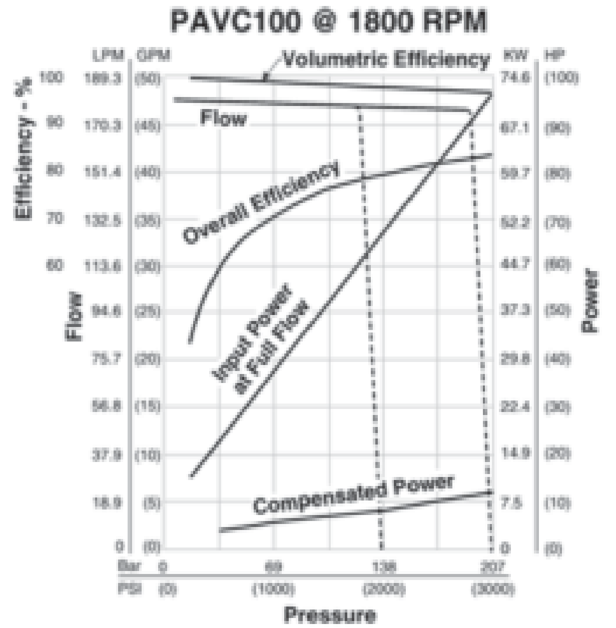

Fig.9 Characteristics curve of pump at 1800 RPM

Efficiency of pump

Volumetric efficiency (η_v)

From using formula,

$$\eta_v = \frac{Q_{actual}}{Q_{theoretical}}$$

$$Q_{theoretical} = D_p \times N_p$$

$$Q_{theoretical} = 100 \times 10^{-6} \times 10^{-3} \times 2100\, LPM$$

$Q_{theoretical} = 210\ LPM$

From Fig.8 and Fig.9;

PAVC @ 1200 RPM and 205 bar pressure, Q_{actual} = 114 LPM

PAVC @ 1800 RPM and 205 bar pressure, Q_{actual} = 172 LPM

PAVC @ 2100 RPM and 205 bar pressure,

$$Q_{actual} = \left(\frac{172-114}{1800-1200}\right) \times 2100 = 203\ LPM$$

Therefore, Q_{actual} = 203 LPM

$$\eta_v = \frac{Q_{actual}}{Q_{theoretical}} = \frac{203}{210} = 0.96666 = 96.67\%$$

Overall efficiency (η_0)

From using formula,

$$\eta_0 = \frac{output\ \ power}{input\ \ power}$$

Output power = P×Q

$$Output\ \ power = \frac{205 \times 10^5 \times 203 \times 10^{-3}}{1000 \times 60} kW$$

Output power = 69.36 *kW*

From Figure 8 and 9;

PAVC @ 1200 RPM and 205 bar pressure, *Input power* = 47 *kW*

PAVC @ 1800 RPM and 205 bar pressure, *Input power* = 71.2 *kW*

PAVC @ 2100 RPM and 205 bar pressure, *Input power* =

$$\left(\frac{71.2-47}{1800-1200}\right) \times 2100$$

=84.7 *kW*

Therefore, *Input power* = 84.7 *kW*

$$\eta_0 = \frac{output\ \ power}{input\ \ power} = \frac{69.36}{84.70} = 0.8189 = 81.89\%$$

Mechanical efficiency (η_m)

From using formula,

$$\eta_0 = \eta_v \times \eta_m$$

$$\eta_m = \frac{\eta_0}{\eta_v}$$

$$\eta_m = \frac{81.89}{96.67} = 0.8471 = 84.71\%$$

Selection of directional control valve (DCV)

For required flow rate 203 LPM,this system has used standard size, Yuken (model no: DSHG-04-3C60 and spool type "60"), solenoid controlled pilot operated directional control valve which allows maximum flow rate of 300 LPM. For required flow rate 203 LPM, pressure drop in DCV is (calculated from Figure 11) P→A = 3 bar, B→T = 4 bar, P→B = 3 bar, A→T = 3 bar and P→T = 5 bar. [5]

Spool Type	Pressure Drop Curve Numbers					Spool Type	Pressure Drop Curve Numbers				
	P→A	B→T	P→B	A→T	P→T		P→A	B→T	P→B	A→T	P→T
2	5	4	5	6	—	60	7	5	7	7	2
3	5	5	5	5	7	7	5	4	5	6	—
4	5	5	5	5	—	9	5	4	5	6	—
40	5	4	5	6	—	10	5	2	5	6	—
5	7	4	5	5	5	11	6	4	5	6	—
6	5	5	5	6	1	12	5	4	5	5	—

Fig.10 Pressure drop curve numbers versus spool type

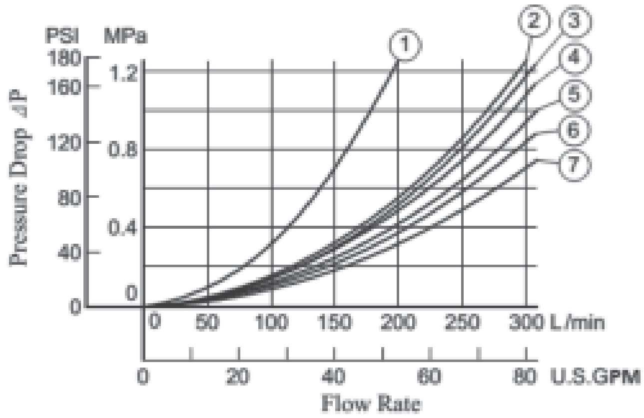

Fig.11 Characteristics curve for directional control valve

Selection of flow control valve

For required flow rate 203 LPM, this system has used standard size, Parker series F (model no: F 2020), flow control valve which allows maximum flow rate of 265 LPM and operated at maximum operating pressure of 207 bar (3000 psi). For required flow rate 203 LPM, pressure drop in flow control valve is (calculated from Fig.12) 2.3 bar. [6]

Selection of counterbalance valve

For required flow rate 203 LPM, this system has used standard size, Parker series E (model no: E2A300 and E2C300), counterbalance valve which allows maximum flow rate of 350 LPM and operated at pressure range of 50–350 bar. For required flow rate 203 LPM, pressure drop in counterbalance

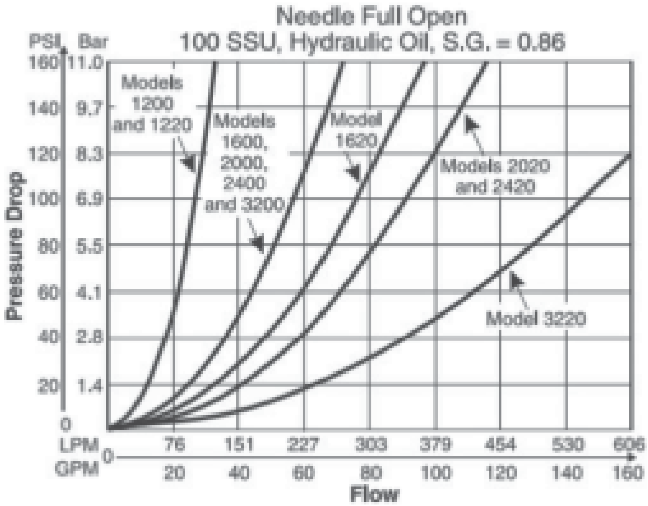

Fig.12 Characteristics curve of flow control valve

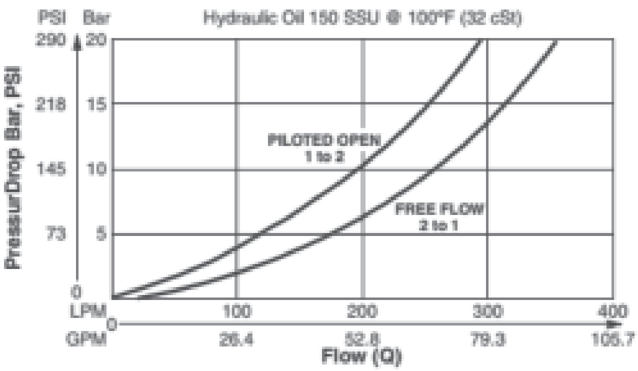

Fig.13 Characteristics curve of counterbalance valve

valve is (calculated from Fig.13): 6.5 bar for free flow and 10bar for piloted open condition. [7]

Efficiency of proposed hydraulic system

- Efficiency of telescopic cylinder:

  Efficiency on extend stroke

  $$= \frac{\text{energy on overcome load on cylinder}}{\text{total energy into liquid}}$$

  Efficiency on extend stroke

  $$= \frac{\text{flow to cylinder} \times \text{pressure owing to load}}{\text{flow from pump} \times \text{pressure at pump}}$$

Efficiency on extend stroke = $\frac{203 \times 190}{203 \times 205} \times 100 = 92.68\%$

- Overall efficiency of pump = 81.89%
- Assume the efficiency of other components is 100%

  Then, Efficiency of proposed system = 92.68% × 81.89% × 100% = 75.89%.

It is obtained that the efficiency of proposed hydraulic system is 76%.

IV. Conclusion

In this paper, a simple analytical method has been shown to estimate the required size of the pump for hoisting the 50 tonne capacity dumper. Also, the selection of other hydraulic components has been made for the optimum efficiency of the hydraulic system. In this work, the calculations made are very straight forward, i.e., taking moments about any fixed points. For better result and more efficient operation the system has been selected pressure compensated variable displacement piston pump, a “meter out” flow control valve and a two stage double acting telescopic cylinder.The proposed hydraulic system also uses a counterbalance valve which provides a “cushion” of hydraulic oil in the hoist cylinder annular area to prevent damage to the cylinders. The approach shown here for the selection of the hydraulic component may be useful for the initial design and development of the experimental set up of body hoist control of dumper.

References

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