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Operational characteristics of a fixed displacement hydraulic pump

With the advancement in technology, a clear shift can be witnessed towards hydraulic power in industrial arena. Hydraulic systems are rapidly replacing mechanical and electrical systems. A hydraulic pump is an indispensable part of any hydraulic system. Fixed displacement pump is the classic variant of hydraulic pumps. This paper provides the numerical description of the operational performance of a fixed displacement pump, taking into account various losses that occur during the operation of the machine. Incorporating the effect of these losses pump efficiency is calculated and analyzed to find out optimum condition.

Keywords: *Fixed displacement pump; flow losses; torque losses; efficiency.*

I. Introduction

Nowadays in industrial scenario fluid power is often preferred over electrical or mechanical power. This development can be owed to the more competitive power-weight ratio of hydraulic systems [1]. In a hydraulic system the starting point is the hydraulic pump, driven by a prime mover i.e. electrical motor or diesel engine. A hydraulic pump may be either fixed displacement type or variable displacement type.

A fixed displacement pump supplies a fixed amount of fluid with each stroke. This property somewhat narrows down its range of applications. Gear pumps are the simplest form of fixed displacement pump. In addition to it vane pumps and axial piston pumps also operate in fixed displacement configuration.

Both fixed (open center system) and variable displacement pumps (closed center systems and closed center load sensing systems) are of the positive displacement type. In fixed displacement pumps the amount of flow which has to be displaced by each pump shaft rotation cannot be altered. Thus, the pump's displacement is varied only by changing the speed of the pump. Since industrial hydraulic systems generally use constant speed electric motors as prime movers, fixed displacement pumps do not find much application.

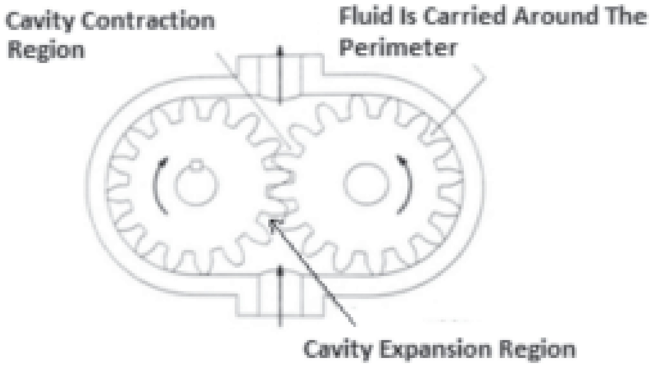

Fig.1 Spur gear pump

Fixed displacement pumps have the advantages of being simple in construction, relatively inexpensive as well as easier to maintain [2]. External gear pumps are typically used as lubrication pumps in machine tools. Screw pumps have large displacement volume and low operating noise. Vane pumps are used in die-casting and injection molding machines.

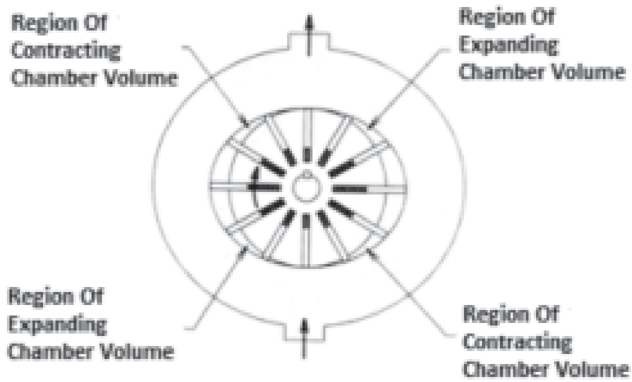

Fig.2 Balanced vane pump

They have low flow pulsation and relatively wide speed range. An axial piston pump, when in fixed displacement configuration, can be used as a pump as well as a motor. They are able to handle large flows at high system pressure, this makes them better suited for use in construction equipment,

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marine auxiliary power and oil-field equipment. Axial piston pumps boast of high volumetric efficiency levels.

Limited number of research has been done on the operational performance of fixed displacement pumps. Out of those available, most of the work is concentrated on the analysis of gear pumps. The dynamic behaviour of a gear pump has been analyzed using simulation by W. Fiebig [3]. E. Mucchi et al provided a detailed vibro-acoustic analysis of gear pumps for automotive applications [4]. Pump flow leakage in axial piston pumps was studied by J M Bergada et al, where it is demonstrated that the main source of leakage is the wear of slipper-swash plate [5]. A detailed case study has been done on vane pumps, where the theoretical and practical aspect of its wear is discussed [6].

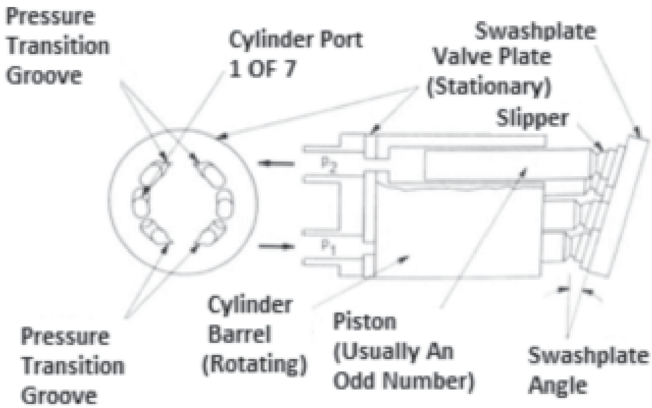

Fig.3 Axial piston pump

II. Operational charactristics

In all the variants of fixed displacement pumps, there are moving elements. The working of these elements decides the performance of the pump. However, in all the pumps an important parameter pump displacement, D defines the operational characteristics of the pump. It is the volume of fluid displaced per radian of revolution and expressed in m^3/rad. Flow rate of a pump is given as

$$Q = D_p.\omega \, m^3/s \qquad \text{... ... (1)}$$

The displacement D_p, for a gear pump is given by Ivantysyn and Ivantysynova [7] as

$$D_p = 2b\pi\left(\frac{2r_p}{n}\right).\omega\left[n+1-\frac{\pi^2\cos^2\psi}{12}\right] \qquad \text{... ... (2)}$$

Whereas for an axial piston pump, the displacement of each piston is given as [8]

$$D_p = \frac{d^2 l}{8} \qquad \text{... ... (3)}$$

Thus it is seen that the theoretical flow rate is a product of pump displacement D and pump speed ω.

But in a practical system, the working of a machine is always accompanied with various kinds of losses. These losses ultimately work towards decreasing the efficiency of the system. Some of these losses occurring in a fixed displacement pumps are listed below.

A. Flow Losses

Flow characteristics of a hydraulic pump suggest that the flow losses increase with increasing pressure [9]. This suggests that effect of individual loss components is over-shadowed by a pressure dependent flow loss. The various flow losses that occur in pump are summarized as follows:

A leakage occurs when the fluid passes through the inner face of the rotating cylinder to the outer face. The flow loss is termed as cross port leakage and is given as

$$Q_{cp} = \frac{P_1 - P_2}{R_{cp}} \qquad \text{... ... (4)}$$

The leakage across the perimeter of the piston and that occurring at the slipper face is combined together to form external loss. It is given as

$$Q_{ext} = \frac{P}{R_{ext}} \qquad \text{... ... (5)}$$

As the fluid in a hydraulic system is pressurized and compressed, a fluid compressibility loss occurs. It is given by

$$Q_c = D_p\omega\left(\frac{P}{\beta}\right) \qquad \text{... ... (6)}$$

To smooth the pressure transition from low inlet pressure P_2 to full pump pressure P_1, a timing groove is required. However it creates a small backflow which results in timing groove loss.

$$Q_{tim} = \frac{P}{R_{tim}} \qquad \text{... ... (7)}$$

The cumulative effect of these losses can be shown in the flow rate equation as given below

$$Q = D_p\omega - P_1\left(\frac{1}{R_{ext}} + \frac{1}{R_{cp}}\right) + \frac{P_2}{R_{cp}} \qquad \text{... ... (8)}$$

B. Torque Losses

A hydraulic pump basically converts mechanical energy into hydraulic energy. It draws input torque from prime mover and converts it into output hydraulic torque. During the process it has to overcome torque losses. The component of a pump rotating around a viscous fluid, experiences resistance from the fluid, this is due to viscous friction. The resulting torque loss is proportional to the pump speed. To overcome the initial friction, a torque is necessary to bring the pump shaft into motion. As the shaft starts rotating, it now has to overcome coulomb friction. Therefore, for a pump operating at a given speed and pressure, torque losses is given as

$$T_{loss} = B_v \omega + T_{sc} \quad \text{... ... (9)}$$

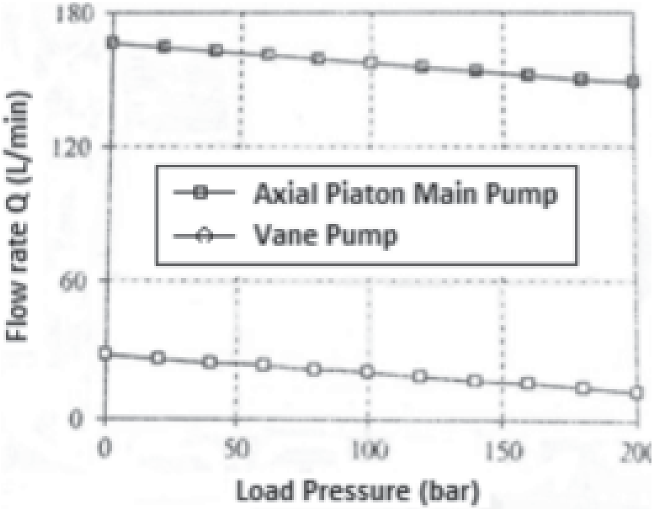

Fig.4 Flow characteristics of axial piston pump and vane pump

C. EFFICIENCY

Efficiency is a factor of prime importance in any system. It shows the overall view of the effectiveness of that system. The overall efficiency of a pump is the product of volumetric and mechanical efficiencies.

Volumetric efficiency is expressed as the ratio of output hydraulic flow rate to mechanically generated input flow rate, mathematically

$$\eta_v = \frac{D_p \omega_p - \frac{P_1}{R_p} + \frac{P_2}{R_i}}{D_p \omega_p}$$

$$= 1 - \frac{P_1}{D_p \omega_p R_p} + \frac{P_2}{D_p \omega_p R_i} \quad \text{... ... (10)}$$

Neglecting the inlet pressure effect and introducing a leakage pressure loss term as

$$P_q = D_p \omega_p R_p \quad \text{... ... (11)}$$

Eqn. (10) becomes

$$\eta_v = 1 - \frac{P_1}{P_q} \quad \text{... ... (12)}$$

Mechanical efficiency is given as the ratio of output hydraulic torque to input mechanical torque, mathematically

$$\eta_m = \frac{D_p (P_1 - P_2)}{(1+\alpha) D_p (P_1 - P_2) + B_v \omega_p + T_c}$$

$$= \frac{1}{(1+\alpha) + \frac{(B_v \omega_p + T_c)}{D_p (P_1 - P_2)}} \quad \text{... ... (13)}$$

Neglecting inlet pressure effect and defining a friction pressure loss term

$$P_f = \frac{(B_v \omega_p + T_c)}{D_p} \quad \text{... ... (14)}$$

Eqn. (13) can be written as

... ... (15)

The overall pump efficiency is

$$\eta_{pump} = \eta_v \eta_m = \frac{1 - \frac{P_1}{P_q}}{(1+\alpha) + \frac{P_f}{P_1}}$$

$$(1+\alpha)\eta_{pump} = \frac{\overline{P_1} - \overline{P_1}^2}{\overline{P_1} + \frac{1}{k_p}} \quad \text{... ... (16)}$$

Where, $\overline{P_1} = \frac{P_1}{P_q}$, $k_p = \frac{(1+\alpha) P_q}{P_f} = \frac{(1+\alpha) D_p^2 \omega_p R_p}{(B_v \omega_p + T_c)}$

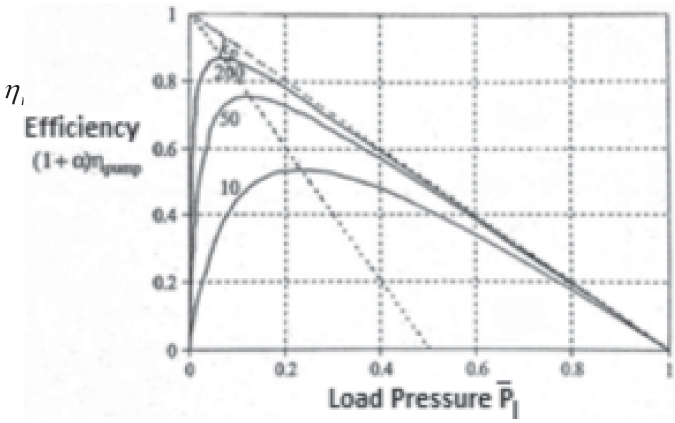

Fig.5 Pump efficiency variation with load pressure

Differentiating Eqn. (16) with respect to load pressure $\overline{P_1}$, we obtain the maximum value of pump efficiency, that is

$$(1+\alpha)\eta_{pump} = 1 - 2\overline{P_1} \quad \text{... ... (17)}$$

The load pressure at the condition of maximum efficiency is given as

$$\overline{P_1} = \frac{\sqrt{1+k_p} - 1}{k_p} \quad \text{... ... (18)}$$

III. Conclusions

This paper has conceptually described the operating characteristics of a fixed displacement pump, outlining the various losses occurring in the form of flow loss and torque loss during the operation of the pump. Numerical expressions in the form of mathematical equations have been provided to analyze the characteristics of the pump. Expressions for volumetric as well as mechanical efficiencies have been generated to calculate the overall efficiency, which has been presented in a graphical form. Ultimately the optimum condition pertaining to maximum efficiency has been laid out.

Nomenclature

b	=	gear width
B_v	=	viscous friction coefficient
d	=	piston diameter
D_p	=	pump displacement
l	=	stroke length
P_1	=	outlet pressure
P_2	=	inlet pressure
$\overline{P_l}$	=	load pressure
P_f	=	friction pressure loss term
P_q	=	leakage pressure loss term
Q	=	pump flow rate
Q_c	=	fluid compressibility flow loss
Q_{cp}	=	cross-port leakage loss
Q_{ext}	=	external flow loss
r_p	=	pitch circle radius
R_{cp}	=	cross-port resistance
R_{ext}	=	external resistance
R_{tim}	=	timing groove resistance
T_c	=	coulomb friction running torque
T_{loss}	=	torque loss
T_{sc}	=	striction-coulomb friction torque loss function
α	=	torque loss constant
β	=	fluid effective bulk modulus
η_m	=	mechanical efficiency
η_v	=	volumetric efficiency
ψ	=	pressure angle
ω	=	pump speed

References

[1] Banks DS and Banks DD (1988): "Industrial Hydraulic Systems", Prentice-Hall.

[2] Watton, J. (2009): Fundamentals of fluid power control (Vol. 10). Cambridge University Press.

[3] Fiebig, W., and Korzyb, M. (2015): Vibration and dynamic loads in external gear pumps. Archives of Civil and Mechanical Engineering, 15(3), 680-688.

[4] Mucchi, E., Rivola, A., and Dalpiaz, G. (2014): Modelling dynamic behaviour and noise generation in gear pumps: Procedure and validation. Applied Acoustics, 77, 99-111.

[5] Bergada, J. M., Kumar, S., Davies, D. L., and Watton, J. (2012): A complete analysis of axial piston pump leakage and output flow ripples. Applied Mathematical Modelling, 36(4), 1731-1751.

[6] Kunz, A., Gellrich, R., Beckmann, G., and Broszeit, E. (1995): Theoretical and practical aspects of the wear of vane pumps Part B. Analysis of wear behaviour in the vickers vane pump test. *Wear*, 181, 868-875.

[7] Ivantysyn, J., and Ivantysynova, M. (2001): Hydrostatic pumps and motors. New Dehli: Academic Books International.

[8] Nesbitt, B. (1999): System effects in positive displacement pumping. *World pumps*, (397), 40-43.

[9] Totten, G. E. (2011): Handbook of hydraulic fluid technology. CRC Press.

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[5] K. Dasgupta et al, 2014 Study of different types of accumulators and their characteristics and applications, International journal of research in Aeronautical and Mechanical Engineering, Vol. 2, Issue 2, pp 56-63.

[6] Mohsen Saadat, Farzad A. Shirazi, Perry Y. Li 2014 Modelling and control of an open accumulator Compressed Air Energy Storage (CAES) system for wind turbines, Elsevier ltd.

[7] David Kuhm, Dominique Knittel, 2011, New mathematical modelling and simulation of an industrial accumulator for elastic webs, Elsevier ltd.

[8] PierpaoloPuddu, Maurizio Paderi, 2013 Hydro-pneumatic accumulators for vehicles kinetic energy storage: Influence of gas compressibility and thermal losses on storage capability, Elsevier ltd.