PERFORMANCE INVESTIGATION OF A PROPORTIONAL VALVE CONTROLLED HYDRAULIC MOTOR USED IN HYDROSTATIC TRANSMISSION SYSTEM

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Abstract - The hydrostatic transmission systems (HST) are one of the well known and widely acceptable transmission systems. The main purpose of a HST system is to transfer the mechanical input power as a mechanical output power, using a hydraulic system consisting of a hydraulic pump and a motor. This paper concerns an investigation of the performance of a proportional valve controlled hydraulic motor used in hydrostatic transmission system. Various leakage flow and torque losses of the bent-axis hydraulic motor are to be represented as motor parameters like pressure, flow rate. The corresponding losses coefficients are to evaluated and validated experimentally. Various experiments have also been conducted extensively to determine optimum performance of the system.

Index Terms- HST, hydraulic motor, flow & torque losses, leakage characteristics.

I. INTRODUCTION

The hydrostatic transmissions are widely used and well known transmission system [13]. Their main purpose is to transfer the mechanical input power as a mechanical output power, using a hydraulic system consisting of a hydraulic pump and motor. They are often used for their power density and capacity, accuracy, fast response and high operating efficiency. They are mainly an auxiliary system of some other system like mobile machinery and industrial process equipment and their performances are more and more important besides their energy consumption. The total efficiency could be highly improved when the variable displacement pump makes closed loop with the hydraulic motor [1]. The speed of the load is governed by controlling the flow of the pump with good response and accuracy. A hydrostatic transmission involves conversion of power between mechanical energy and hydraulic pressure. The overall system efficiency is the product of the efficiencies of the pump and the motor. Pumps and motors with conversion efficiencies of 90% will lead to a transmission system losing a fifth of its power throughput. For a successful vehicle transmission system it is not tolerable to have an individual machine with losses higher than 5% of throughput. Hydrostatic Transmission System (HST) mainly consists of hydraulic pump and motor/actuator, which are used to generate, control and transmit the power using pressurized fluid with the help of hydraulic control valves with fast response time. In a hydrostatic transmission (HST) system the mechanical energy of the input drive shaft of the hydraulic pump is converted to pressure/hydraulic energy in a nearly incompressible working fluid through either open loop or closed loop system and then reconverted into mechanical energy at the output drive shaft of the hydraulic motor through various control line. The system consists of pipes and valves to regulate flow and its direction. It is used to transmit rotating mechanical power from one source to another without the use of mechanical gears.

In a HST system, motor speed and torque can be controlled by controlling the flow supply through the valve placed in between pump and motor. HST has inherent ability to provide driveline flexibility. This unique characteristic allows the cost of the surrounding machine structure to be changed, both by selecting the size and the type of unit and its placement on the machine. In the performance criteria of positive displacement pumps and fluid motors, Wilson [8] has given the idea for finding out the maximum efficiency of the fluid components based on static analysis. The electro-hydraulic servo valve and proportional valves are widely used in industrial applications, such as machine tools, industrial robots, autonomous manufacturing and various actuators in aircraft. They can be applied in closed loop control system. The dynamic analysis of proportional control valve driven hydraulic motor was performed by Ahmed.et.al. [4]. He has analysed the performance limitations of the system due to various mechanical non-linearities. Manring and Luecke [6] has analysed...
HST system consisting of variable displacement pump and a fixed displacement motor where the system is linearised and the stability range of the system was presented. Watton [9, 10] has analysed hydraulic servo valve controlled motor system based on the liberalized model that is suitable for input of small amplitude.

II. OVERVIEW OF THE PHYSICAL SYSTEM AND EXPERIMENT DESCRIPTION

The experiment was conducted using the test setup shown in Fig. 2.

The computer controlled hydraulic motor setup is a proportional valve controlled system. A brief description of the test setup is given below: A variable displacement pump is driven by electric motor (A.C induction motor). This pump supplies pressurized hydraulic fluid to hydraulic motor through check valve, pressure line filter and directional control valve (either proportional or servo). Check valve doesn’t allow fluid to flow in reverse direction from DC valve to tank. Hence it always maintains some fluid in fluid line. DC valve (proportional) and DC valve (servo) are four ways; open centered and closed centered valves respectively. In proportional or servo controlled valves, spool movement and hence output flow are proportional to input current signal. Input command signal to this valve can be varied by microprocessor

controller. In this system, there are two hydraulic motor setup units which comprises of hydraulic pump and motor. When proportional DC valve is actuated by positive command signal, output flow from port A is supplied to Bent-axis hydro motor (refer Fig. 2). Hence hydraulic motor is driven by this pressurized fluid, which hence drives hydraulic pump. Output flow from hydraulic motor returns to tank through check valve and return line filter. As driven by hydraulic motor, this hydraulic pump sucks oil from tank and delivers pressurized fluid through check valve to direct operated pressure relief valve. Fluid flow through pressure relief valve allows pump to be loaded and hence hydraulic motor will be loaded. But hydraulic power generated by pump is mostly wasted as heat in pressure relief valve. Output flow from pressure relief valve passes to tank. Manually load on pump can be varied by changing pressure setting of direct operated pressure relief valve. Hence load on the Bent-axis hydro motor can be varied. A torque sensor and stroboscope are used for measuring the torque and speed of motor respectively. In this way this hydraulic system is operated to determine performance characteristics of DC valves & Hydraulic motor at different load. The hydraulic system is protected from overload by unloading pressure relief valve. It serves purpose of pressure relief valve when DC solenoid valve is actuated. In normal position of DC valve, it serves the purpose of unloading valves. The function of unloading valve is to allow pump to be unloaded at lower pressure (approx. tank pressure) when required. Hydraulic pumps and motors have external leakage line connected to tank. Pressure relief valve and Gauge isolator has external leakage line connected to tank.

III. EXPERIMENTAL ANALYSIS AND TEST RESULTS

The experiment was conducted using the test setup shown in Fig. 2. The computer controlled hydraulic motor setup is a proportional valve controlled system. Various test analysis and experimental calculations are done to investigate the performance of the proportional valve controlled hydraulic motor used in hydrostatic transmission system.

A. Determination of Motor Resistance Coefficient

The experimental data were obtained at different constant speed of motor and from the help of those experimental data, various performance curves are drawn as shown below in Figures 3, 4, 5, 6 & 7:
Performance Investigation Of A Proportional Valve Controlled Hydraulic Motor Used In Hydrostatic Transmission System

(i) Calculation of external leakage coefficient:

It can be seen from above flow characteristics that the difference between the input flow rate and output flow rate is sufficiently constant for a range of pressure difference and speeds, since we have the equation:

$$R_e = \frac{Q_1 + Q_2}{P_1 - P_2}$$  \hspace{1cm} (1)

Where, $R_e$ = External leakage coefficient.

The flow rate difference is remarkably constant $\approx 0.43$ lpm for a range of pressure differentials and speed. The sum of line pressure is $\approx 86$ bar with sufficient accuracy, and therefore:

$$R_e = \frac{86 \times 10^5}{43 \times 10^{-3}} = 1.20 \times 10^{12} \text{Nm}^{-2}/\text{m}^3\text{s}^{-1}$$

(ii) Calculation of internal leakage coefficient $R_i$:

Since, we have:

$$\frac{Q_1 + Q_2}{2} = D_m \omega + (P_1 - P_2) \left( \frac{1}{R_i} + \frac{1}{2R_e} \right)$$  \hspace{1cm} (2)

Referring Fig. 3, the inverse of slope of the (mean flow rate)/(pressure differential) graph gives:

$$\left( \frac{1}{R_i} + \frac{1}{2R_e} \right) \approx 0.61 \times 10^{-12}$$

or,

$$\left( \frac{1}{R_i} + \frac{1}{2 \times 1.2 \times 10^{12}} \right) \approx 0.61 \times 10^{-12}$$

or, $R_i \approx 5.26 \times 10^{12} \text{Nm}^{-2}/\text{m}^3\text{s}^{-1}$

(iii) Calculation of mean leakage coefficient $R_{m}$:

Since, we have the equation for torque loss:

$$\text{Net Torque} (T) = D_m (P_1 - P_2) - B_\omega \omega - T_{sc} \hspace{1cm} (4)$$

With the help of Fig. 5 (torque)/(pressure differential) graph, it is seen that the variation is linear at different speed. It means that the variation of torque with respect to pressure difference is following linear characteristic and also depends on speed having nearly same slope $= 0.6$ as shown in graph.

Considering the variation of constant term $(k)$ of this linear equation with respect to speed, it can be represented as:

<table>
<thead>
<tr>
<th>Rpm ($\omega$)</th>
<th>K</th>
</tr>
</thead>
<tbody>
<tr>
<td>2093</td>
<td>0.0566</td>
</tr>
<tr>
<td>2213.14</td>
<td>0.1468</td>
</tr>
<tr>
<td>2303.16</td>
<td>0.3172</td>
</tr>
</tbody>
</table>

Table I: Table showing linear constant of Fig. 8 vs speed.
The flow and torque across the motor may be represented in a generalized linear form of speed and pressure (inlet and outlet). For flow, external and internal leakage constant for motor is same for all range of bent-axis motor operating condition. For torque, viscous torque loss coefficient gives linearly dependency with speed of motor and stiction-coulomb torque loss is constant for all range of bent-axis hydro motor operating conditions.

CONCLUSION

The deduction that the ideal flow equation is sufficiently accurate for evaluating motor speed, drive efficiency and motor power transfer in practice is particularly significant and useful, since it lead to ability to drive a directly set of explicit design equation. It is validated that at higher load pressure (P1 – P2) across bent-axis hydro motor, the supply pressure (P1 + P2) comes constant (Fig. 6). The torque loss is linear in nature having direct dependency on the speed and pressure differential across the motor which has been validated experimentally (Fig. 7). The flow across bent axis hydro motor has linear dependency at different speed with pressure difference across the motor (Fig. 3). The flow difference across the motor (leakage flow) is constant for all operating conditions (Fig. 4) keeping the line pressure constant. The derived equations are applicable in determining the motor performance for its wide range of operation. The flow across bent axis hydro motor has linear dependency at different speed with pressure difference across the motor. The system design equations allow direct determination of the conditions for maximum efficiency. These derived design equations become more accurate in determining the power transfer particularly for higher no-load speeds. The derived equations are applicable in determining the motor performance for its wide range of operation. Dynamic performance of bent axis hydro motor can be studied as a future work of this research.

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