Determination of the Optimum Steady State Performance of Bent-Axis Piston Motor drive

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Abstract

This article presents the investigation of the steady-state performance of a proportional valve controlled bent-axis axial piston motor drive used in hydrostatic transmission system. In investigating the performance of the motor, non-dimensional approach to characterize the steady-state performance of the system is used presented. The hydrostatic motor has been considered to be driven by proportional valve. Using the well established equation given by Watton **[6]**, the characteristics of the speed, efficiency and power transfer of the motor with respect to the pressure differential across the motor have been investigated. The characteristics are also validated experimentally.

Notations

Q_1, Q_2	Line flow rates
Q_{mean}	Mean flow rate
P_s	Supply pressure
P_1, P_2	Line pressure
P_{load}	Load pressure
T_l	Load torque
T_{losses}	Torque losses
$\omega_m, \omega_m(0)$	Motor speed, no-load motor speed
W_m	Power transfer to motor
i	Valve current
D_m	Motor displacement
k_f	Proportional valve flow constant
$R_{e,}\ R_i, R_m$	Motor external, cross port and total leakage coefficient
α	Motor flow loss coefficient
η	System efficiency

1. Introduction

An electro-hydraulic proportional valve controlled hydraulic motor is a common method of motor speed control in practice where both load pressure and speed change during operation occur. Such a system may be used in industrial equipment, like earth moving equipment for its wide variation of load demand. Determination of the motor loss characteristics play an important role in evaluating the overall performance of the motor. The characteristics of the loss coefficients of the motor are obtained experimentally from the system equations that include flow and torque loss terms [**1-4**].

In the analysis of open loop steady state performance of hydraulic motor drive system, various experiments are conducted extensively to determine its losses and optimum performance [**5**]. Such evaluation of the system's performance may be applicable within the test speed range. However, an analytical approach gives may predict the system's beheviour for its wider range of operation, it may not be always possible to express the motor speed (ω_m), motor efficiency (η), and power transferred to motor (W_m) in explicit forms such as $\eta = f(P_{load})$ or $\omega_m = f(P_{load})$ due to the form of flow equations. It may be possible to determine experimentally the optimum performance, however, an explicit equation in determining the performance would avoid extensive computer simulation and experimental work. Based on the work done by Watton **[6]** on axial piston motor drive, with the help of experiments conducted on test system, is used to obtain explicit characteristics of non-linear steady state performance of Bent-axis piston motor drive system. Various experiments have been conducted on test system to validate the theoretical results.

Figure 1 shows the physical system considered in the present study.

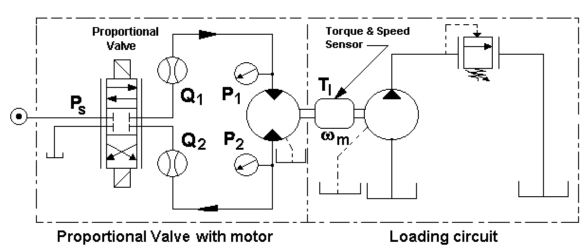


Fig.1 Motor drive system with a loading circuit

A constant pressure is maintained to the proportional directional controlled valve from a stable source of supply that drives a fixed displacement bent axis motor. The motor drives a fixed displacement pump in the loading circuit. The motor load is controlled by adjusting the set pressure of the proportional pressure relief. Both the proportional valves are controlled through micro-computer.

2. Analysis of the system

Considering the steady-state characteristics of the open loop drive

$$Q_1 = k_f\, i\sqrt{P_s - P_1} = D_m\,\omega_m + \frac{P_1 - P_2}{R_i} + \frac{P_1}{R_e} \quad \ldots(1)$$

$$Q_2 = D_m\,\omega_m + \frac{P_1 - P_2}{R_i} - \frac{P_2}{R_e} \quad ..(2)$$

$$D_m\,(P_1 - P_2) = T_l + T_{loss} \quad ..(3)$$

Combining eqns. 1 and 2, the mean flow rate may be expressed as

$$Q_{mean} = \frac{Q_1 + Q_2}{2} = D_m\,\omega_m + \frac{P_1 - P_2}{R_m}$$

where $\frac{1}{R_m} = \frac{1}{R_i} + \frac{1}{2R_e}$

From eqns. 1 and 2, $(Q_1 - Q_2) = \frac{P_1 - P_2}{R_e}$

It is assumed that $P_1 + P_2 = k\,P_s$ and for simplification, it can be taken as $P_1 - P_2 = P_{load}$.
Therefore, in non-dimensionalising the valve flow equations, it can be defined as

$$\overline{Q}_1 = \frac{Q_1}{k_f\, i\sqrt{\frac{P_s}{2}}} = \sqrt{(2-k) - \overline{P}_{load}} \quad ..(4)$$

$$\overline{Q}_2 = \frac{Q_2}{k_f\, i\sqrt{\frac{P_s}{2}}} = \sqrt{(2-k) - \overline{P}_{load}} - \frac{k\,P_s}{R_e\,k_f\, i\sqrt{\frac{P_s}{2}}} \quad ..(5)$$

At no-load, the flow output from the proportional valve can be approximately expressed as

$$Q_1 = k_f\, i\sqrt{\frac{P_s}{2}} \cong D_m\,\omega_m(0) \quad ..(6)$$

Using eqn. 6,

$$\overline{Q}_2 = \sqrt{(2-k) - \overline{P}_{load}} - \frac{k\,P_s}{R_e\,D_m\,\omega_m(0)} \quad ..(7)$$

Considering eqns. 4 and 7, for ideal case, i.e. k = 1, non-dimensional mean flow rate is defined as

$$\frac{\overline{Q}_1 + \overline{Q}_2}{2} = \sqrt{1 - \overline{P}_{load}} - \frac{L}{2} \quad ..(8)$$

where, $L = \frac{P_s}{R_e\,D_m\,\omega_m(0)} = Cons\tan t$

In most of the practical purposes, the assumption k =1, i.e, $P_1 + P_2 \cong P_s$ may be justified. Using eqns. 6, 7 and 8, the mean flow rate can be expressed as:

$$Q_{mean} = \frac{Q_1 + Q_2}{2} \cong D_m\,\omega_m(0)\left(\sqrt{1 - \overline{P}_{load}} - \frac{L}{2}\right) \quad ..(9)$$

From the above flow relations, the following steady-state system equations are derived by Watton [6] in explicit forms [6], that are used in the present analysis for investigating the motor performance:

Motor speed

$$\omega_m = \omega_m(0)\left[\sqrt{1-\overline{P}_{load}} - \alpha\left(\frac{R_m}{2R_e} + \overline{P}_{load}\right)\right] = f(\overline{P}_{load}) \quad .(10)$$

Power transfer to motor

$$W_m = (\overline{P}_{load})\left[\sqrt{1-\overline{P}_{load}} - \frac{L}{2} + \alpha\left(\frac{R_m}{2R_e} + \overline{P}_{load}\right)\right] P_s\,D_m\,\omega_m(0) = \phi(\overline{P}_{load}) \quad ..(11)$$

Efficiency of motor drive system

$$\eta = (\overline{P}_{load} - \overline{T}_{loss})\left\{1 - \frac{\alpha\,\overline{P}_{load}}{(\sqrt{1-\overline{P}_{load}} - \frac{L}{2})}\right\} = \varphi(\overline{P}_{load}) \quad ..(12)$$

where,

$$\overline{T}_{loss} = \frac{T_{loss}}{P_s\,D_m}, \quad \overline{P}_{load} = \frac{P_{load}}{P_s}, \quad \alpha = \frac{P_s}{R_m\,D_m\,\omega_m(0)},$$

while solving the systems equations, the experimental data of the motor resistance coefficients, losses and the displacement of the motor are used.

3. Experimental Investigation

The values of the leakage coefficients and the constant torque loss (T_{loss}) of the motor at its three different speed of operation were obtained experimentally from its steady state performance.
During experiment the temperature of the fluid was nearly kept constant at 40 ± 2^0C. The commercially available bent axis motor and the power pack were used in the test rig. The major parameters of the components are given below:
$D_m = 0.777 \times 10^{-6}$ m^3/rad, P_s = 86 bar and
T_{loss} = 1.13 Nm at 1983 rpm,
T_{loss} = 1.2 Nm at 2618 rpm
T_{loss} = 1.28 Nm at 3296 rpm.
$R_e = 1.57 \times 10^{12}$ Nm^{-5}/s, $R_i = 4.56 \times 10^{12}$ Nm^{-5}/s, $R_m = 1.86 \times 10^{12}$ Nm^{-5}/s.
The above parametric values are determined experimentally.
In the test set-up, supply from a constant pressure source is given to the inlet of a proportional direction control. The outlet port of the electro-hydraulic proportional valve is connected to Bent-axis piston motor that is loaded by a loading circuit. The loading circuit mainly consists of a fixed displacement pump and a proportional pressure relief valve. By controlling the proportional pressure relief valve, the load torque and speed of the hydro-motor are controlled. The load torque (T_l) and motor speed (ω_m) are measured by torque-speed sensor.
The photographic view of the test rig is shown in Fig. 2a and 2b.

Fig .2a

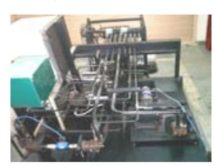
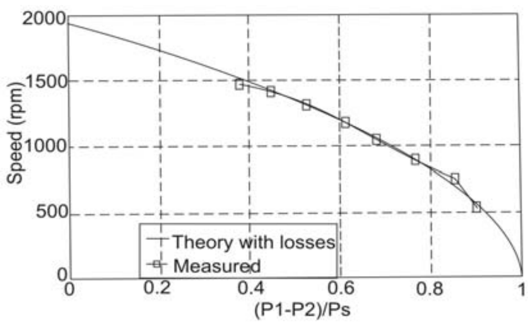


Fig .2b

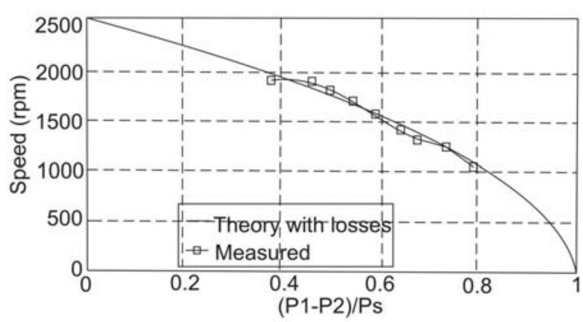
As Bent-axis piston motor is of low-torque and high speed drive unit, its performance at low speed is not investigated. In particular, its operation at low speed (below 600 rpm) is unstable. The measurement of its performance at low speeds becomes difficult. The performance of the motor drive system was investigated at higher speed range.

4. Results & highlights of impointant points

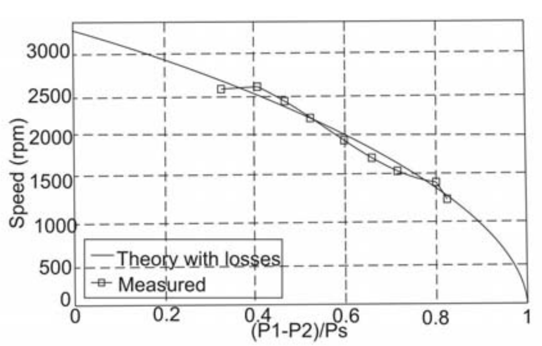
Figures 3 show the results from the practical tests at no-load motor speeds $\omega_m(0)$ at 1983 rpm, 2618 rpm and 3296 rpm which are compared with the theoretical results by solving eqns. 1 through 12. Both the motor speed and efficiency fall to zero at a load pressure less than supply pressure as dictated by the flow loss coefficient α. The power transfer (W_m) to the motor is not zero at this load pressure since the power has to be supplied to overcome the external flow loss.



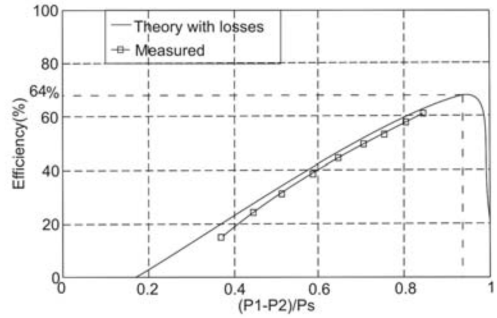
(a) At no load speed of 1983



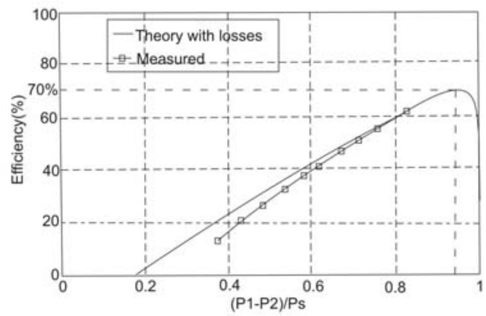
(b) At no load speed of 2618



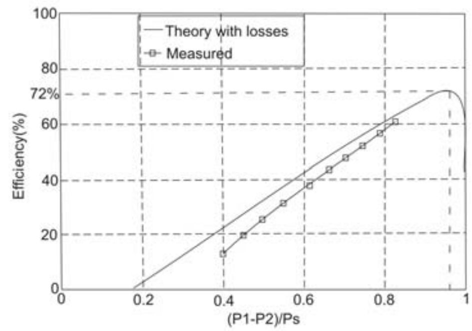
(c) At no load speed of 3296



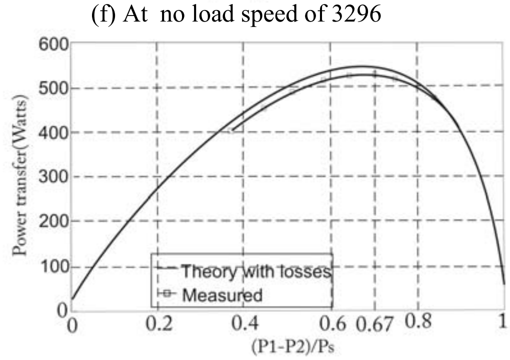
(d) At no load speed of 1983



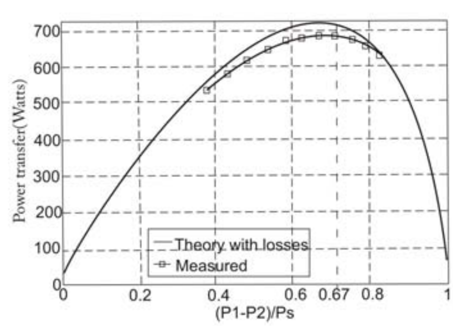
(e) At no load speed of 2618



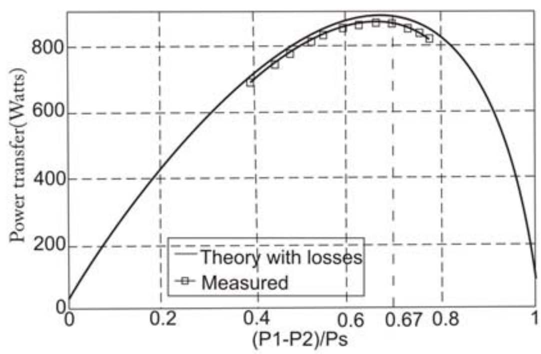
(f) At no load speed of 3296



(g) At no load speed of 1983



(h) At no load speed of 2618



(i) At no load speed of 3296

Fig.3 Measured and theoretical speeds, power transfers and efficiencies for open loop drive

The results for power transfer also confirms extremely well that for an open loop drive the maximum power is transferred when the load pressure is nearly two-third of the supply pressure. Three initial motor speeds only, over the speed range are considered sufficient to validate the theoretical results. The graphs illustrate very good comparisons with the postulated theory over a wide speed range. The maximum efficiency depends on the motor speed; the higher the motor speed, the higher the maximum efficiency attainable mainly due to the lower flow losses. The maximum efficiency also occurs at increasingly higher load pressure as the motor speed increases. However, at higher speeds the load pressure required for maximum efficiency becomes a problem due to the large drop in motor speed as given by eqn. 10. As the speed decreases, it will be seen that the maximum load pressure that can be achieved becomes severely limited as the supplied flow increasingly has to compensate the leakage losses of the motor.

5. Conclusion

In the present article the steady-sate performance of an axial piston bent axis hydrostatic motor is investigated through non-dimensional approach [6]. The following observations are made from the system analysis.

- 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.
- For open loop operation, the maximum power transferred to the load occurs at a load pressure $(P_1 - P_2) \cong \frac{2}{3} P_s$ (As shown by dotted line in Figs. 3 (g), (h)) and (i)) and it is independent of the motor leakage losses.
- The derived equations are applicable in determining the motor performance for its wide range of operation.

The analysis carried out in this article can be extended for closed-loop system with speed feedback that may improve the performance of the system indicated above. Due to the limitations of the test set-up, it was difficult to get the data for its low speed operation. However, in line with the approach presented, it may be possible to investigate the low speed performance of the Low speed high torque hydrostatic motor. The investigation carried out in this article may be useful for the control theoretical aspects of the plant where such motor is an integrated component.

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7. References

[1] Wilson , W., *"Performance criteria for positive displacement pumps and fluid motors", Trans. ASME,1949.*

[2] Schlossor, W. M. J., *"A mathematical model for displacement type pumps and motors", Oelhydraulik und Pneumatik, 1961*.

[3] McCandlish, D. and Dorey, R., *"Steady-state losses in hydrostatic pumps and motors", In Proceedings of the Sixth BHRA Fluid Power Symposium, Cambridge, UK, 1981, pp. 133-144*.

[4] McCandlish, D. and Dorey, R. E., *"The mathematical modeling of hydrostatic pumps and motors", Proc. Instn Mech. Engrs, 1984, 198B(10), 165-174.*

[5] Mandal. S. K., `*Hydrostatic transmission system with low speed high torque hydraulic motor – a quest for energy efficient performance and design'*, an unpublished Ph.D thesis. 2009.

[6] Watton, J., *"An explicit design approach to determine the optimum steady state performance of axial piston motor drives", J. Systems and Control Engineering: Proc. IMechE Vol. 220 Part- I. 2008.*