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BUILDING A MATLAB SIMULINK MODEL OF A FIXED DISPLACEMENT PUMP USING TEST PERFORMANCE DATA

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ABSTRACT

An attempt was made in the present paper to build a MATLAB Simulink model of a fixed displacement pump using performance coefficient model. The collected data from the pump performance test was used for graphically determining the performance coefficients. A MATLAB test rig was then build to test the pump model at nominal pressure 100 (bar) and nominal speed 1800 (rpm). The simulation results indicated that no significant difference existed between them and actual pump parameters...

Keywords: ***Pump Performance Test, Fixed Displacement Pump, Lumped Coefficient Model, Performance Coefficient Model, MATLAB test rig.***

1. INTRODUCTION

The purpose of this paper is to build a mathematical model of a fixed displacement pump and testing the behavior of its virtual model created by the use of MATLAB Simulink Software. Regardless of the type of pump and its structure, their mathematical model consists in determining the pump flow and calculation of its energy balance. Mathematical modeling of a fixed displacement pump can be done based on Lumped Coefficient Model or Performance Coefficient Model. The lumped coefficients are derived from a single set of operating conditions while the performance coefficients are derived from a wide range of operation conditions. As a result, the lumped coefficient model may become limited when the operating conditions are varied beyond the preferred range used in deriving the coefficients. Given that the performance coefficients are derived from a wide range of operating conditions, the performance coefficients model will provide a more accurate information for modelling a fixed displacement pump.

The goal is adaptation of this mathematical model of a fixed displacement pump with the mathematical model used in building virtual components of the SimHydraulics library of the MATLAB Simulink Software. Based on this goal, the performance coefficients are calculated for a fixed displacement pump based on the data collected from the performance test. These coefficients are then used to build the virtual model of the pump. Using a test rig, we study its behavior and compare it with the behavior of the real pump.

2. PERFORMANCE COEFFICIENT MODEL:

Output flow rate of a fixed displacement pump is as follows [1], [2]:

$$Q_a = \omega_p D_p - \frac{C_s D_p \Delta P}{\mu} \qquad (1)$$

Where:

ω_p - *input pump speed*
D_p – *pump displacement*
ΔP – *pressure differential across pump*
C_s - *slip coefficient*
μ - *absolute viscosity of fluid*

Applied torque is as follows [1], [2]:

$$T_a = \Delta P D_p + C_v D_p \mu \omega_p + C_f \Delta P D_p + T_c \qquad (2)$$

Where:

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T_a - actual pump input torque
T_c - breakaway torque
C_v – viscous shear coefficient
C_f - mechanical friction coefficient

3. PUMP PERFORMANCE TESTING

The data compiled in Table 1 is the test data acquired from a gear pump performance test with a 2500 RPM and maximum pressure 207 bar. The fluid is MIL-F-87257, which has a kinematic viscosity υ = 7.06 (cSt) at the test temperature (40°C).

Table 1. Pump performance data

Pressure (Pa)	Measured Parameter	Speed (rad·s⁻¹) 78.54	109.95	136.09	167.5	198.92
$34.47 \cdot 10^5$	Flow ($m^3 s^{-1}$) Torque (Nm)	$583.6 \cdot 10^{-6}$ 48.27	$833.9 \cdot 10^{-6}$ 47.19	$1091.5 \cdot 10^{-6}$ 49.80	$1372.2 \cdot 10^{-6}$ 51.11	$1608.8 \cdot 10^{-6}$ 54.91
$68.95 \cdot 10^5$	Flow ($m^3 s^{-1}$) Torque (Nm)	$520.5 \cdot 10^{-6}$ 80.94	$788.6 \cdot 10^{-6}$ 81.90	$1009.4 \cdot 10^{-6}$ 83.01	$1261.8 \cdot 10^{-6}$ 84.60	$1537 \cdot 10^{-6}$ 88.81
$102.4 \cdot 10^5$	Flow ($m^3 s^{-1}$) Torque (Nm)	$473.2 \cdot 10^{-6}$ 113.91	$725.3 \cdot 10^{-6}$ 116.74	$946.5 \cdot 10^{-6}$ 118.83	$1198.7 \cdot 10^{-6}$ 119.33	$1490.2 \cdot 10^{-6}$ 123.18
$137.9 \cdot 10^5$	Flow ($m^3 s^{-1}$) Torque (Nm)	$423.9 \cdot 10^{-6}$ 150.55	$662.4 \cdot 10^{-6}$ 149.14	$883.3 \cdot 10^{-6}$ 150.48	$1133.6 \cdot 10^{-6}$ 151.85	$1390.2 \cdot 10^{-6}$ 154.2
$172.4 \cdot 10^5$	Flow ($m^3 s^{-1}$) Torque (Nm)	$372.2 \cdot 10^{-6}$ 181.68	$624.6 \cdot 10^{-6}$ 183.03	$813.9 \cdot 10^{-6}$ 184.38	$1083.2 \cdot 10^{-6}$ 188.45	$1343.8 \cdot 10^{-6}$ 188.81

From the tabulated test data, it is possible to plot the graphs in Fig. 1.1, Fig. 2.1, Fig. 3.1, and Fig. 4.1. , and determine the slope of lines, leakage flow and zero-speed torque, as shown in table 2 and table 3.

a) Determination of pump displacement (D_p)

A plot of measured pump flow (Q_a) as a function of input pump speed (ω_p) can be constructed. The slope of any of these parallel lines is pump displacement (D_p) [3] and the leakage flow is the point at which each line crosses the zero-speed axis

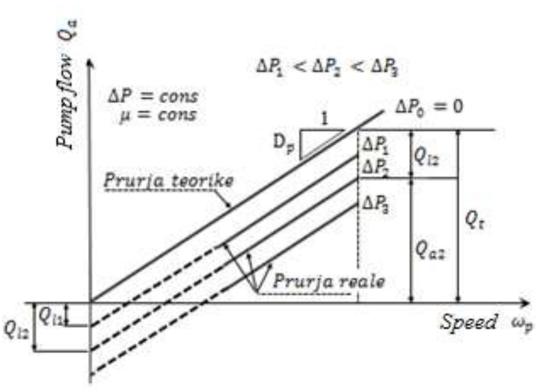

Figure 1. Flow rate vs. input speed

as shown in Fig.. 1.1

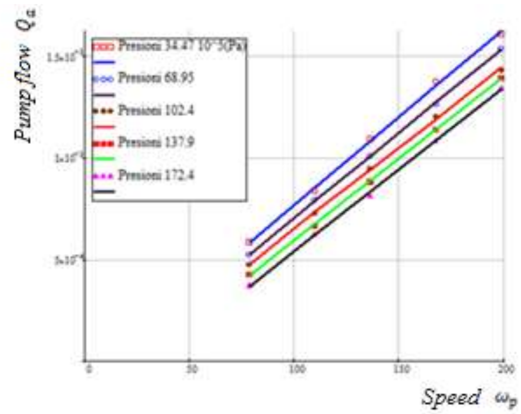

Figure 1.1 Measured flow rate vs. input speed

Table 2. Slope and the intercept of each linear fit at the load pressure

Pressure (Pa)	Slope ($m^3 \cdot rad^{-1}$)	Q_p at 0 speed ($m^3 \cdot s^{-1}$)
$34.47 \cdot 10^5$	$8.658 \cdot 10^{-6}$	$98.44 \cdot 10^{-6}$
$68.95 \cdot 10^5$	$8.500 \cdot 10^{-6}$	$148.50 \cdot 10^{-6}$
$102.4 \cdot 10^5$	$8.352 \cdot 10^{-6}$	$189.20 \cdot 10^{-6}$
$137.9 \cdot 10^5$	$8.203 \cdot 10^{-6}$	$229.80 \cdot 10^{-6}$
$172.4 \cdot 10^5$	$8.060 \cdot 10^{-6}$	$266.1 \cdot 10^{-6}$

The average slope of all lines is numerically equal to the pump displacement.

$$D_p = k_k = 8.35 \cdot 10^{-6}\ (m^3 \cdot rad^{-1}) =$$

$$= 52.4238 \cdot 10^{-6} (m^3/rev) \qquad (3)$$

b) Determination of slip coefficient (C_s)

Once the leakage flow has been determined, a plot

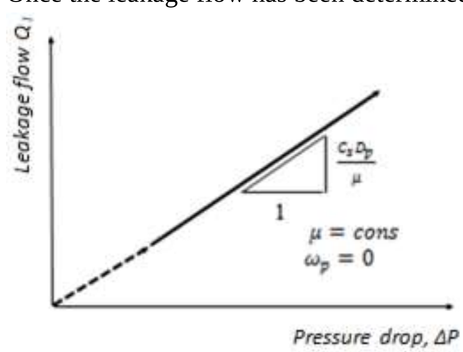

Figure 2. Leakage flow vs. pump load pressure

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of leakage flow versus pump load ΔP can be constructed as shown in Fig. 2.1.

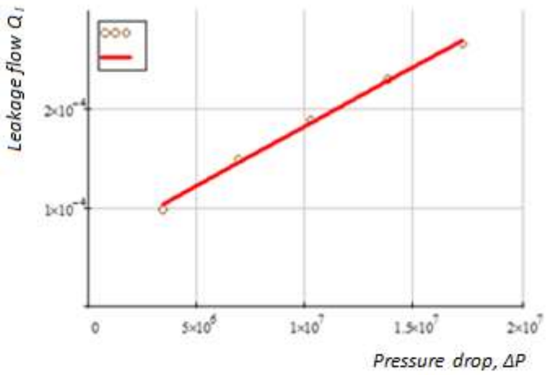

Figure 2.1 Graphically determined leakage flow vs. pump load pressure

The value of slip coefficient is determined by calculating the slope of the line [3] and solving Eq. (4) for C_s [3].

$$k'_k = \frac{C_s D_p}{\mu} = 1.208 * 10^{-11} \left(\frac{m^3}{Pa\ s}\right) \quad (4)$$

$$C_s = 8.5653 * 10^{-9} \quad (5)$$

c) Determination of viscous shear coefficient (C_v).

A plot of the measured actual torque (T_a) as a function of input pump speed (ω_p) can be constructed as shown in Fig. 3.1.

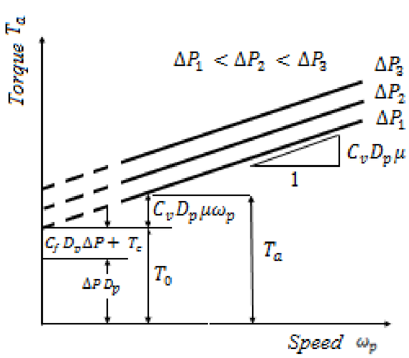

Figure 3. Applied torque vs. input speed

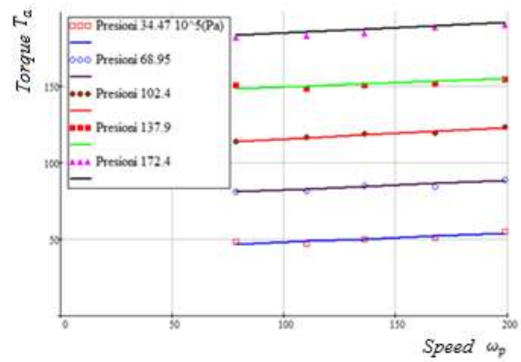

Figure 3.1 Measured applied torque vs. input speed

Table 3. Slope and the intercept of each linear fit at the load pressure

Pressure (Pa)	Slope (N·m·s·rad⁻¹)	Torque at 0 speed (N·m)
$34.47 \cdot 10^5$	0.0577	42.30
$68.95 \cdot 10^5$	0.0616	76.22
$102.4 \cdot 10^5$	0.0723	108.38
$137.9 \cdot 10^5$	0.0549	144.25
$172.4 \cdot 10^5$	0.0728	175.41

The value of viscous shear coefficient is determined by calculating the average slope of the lines [3] and solving Eq. (6) for Cv.

$$k''_k = C_v D_p \mu = 0.0638 \ (\text{N·m·s}\cdot rad^{-1}) \quad (6)$$

$$C_v = 1.2916 * 10^6 \quad (7)$$

d) Determination of mechanical friction coefficient (C_f) and the breakaway torque (T_c).

Using the plot of Fig. 3.1, a second plot of graphically determined zero-speed torque (T_o) as a function of pump load pressure ΔP can be constructed as shown in Fig. 4.1. Breakaway torque (T_c) can also be determined as the point at which line crosses the zero-pressure drop axis.

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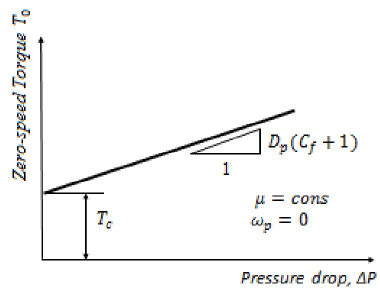

Figure 4. Zero-speed torque vs. pressure drop

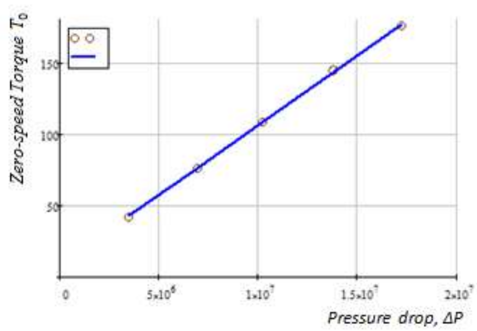

Figure 4.1 Graphically determined zero-speed torque vs. pressure drop

The value of viscous shear coefficient is determined by calculating the slope of the line [3] and solving Eq. (8) for C_f.

$$k'''_k = D_p(C_f+1) = 9.69628 * 10^{-6}\left(\frac{N\cdot m}{Pa} = m^3\right) \quad (8)$$

$$C_f = \frac{k'''_k}{D_p} - 1 = 0.16059 \quad (9)$$

From Fig. 4.1 the breakaway torque is:

$$T_c = 9.04\ (N\cdot m) \quad (10)$$

e) Calculation of volumetric, mechanical and overall efficiency of the pump at 100 (bar) and 1800 (rpm)

Volumetric efficiency is:

$$\eta_{vp} = 1 - C_s\left[\frac{\Delta P}{\mu\omega_p}\right] = 0.923 \quad (11)$$

Mechanical efficiency is:

$$\eta_{mp} = \frac{(P_1-P_2)D_p}{(P_1-P_2)D_p + C_v D_p \mu\omega_p + C_f(P_1-P_2)D_p + T_c} = 0.743 \quad (12)$$

Overall efficiency is:

$$\eta_{op} = \eta_{vp}\cdot\eta_{mp} = 0.686 \quad (13)$$

f) Output flow rate of pump

Theoritical flow rate is:

$$Q_t = \omega_p \cdot D_p = 94.36\ \text{(l/min)}$$

Actual flow rate is:

$$Q_a = Q_t \cdot \eta_{vp} = 87.12\ \text{(l/min)}$$

g) Input torque:

$$T_a = \Delta P D_p + C_v D_p \mu\omega_p + C_f \Delta P D_p + T_c = 117.94\ \text{(N·m)}$$

4. MATLAB SIMULINK TEST RIG OF THE PUMP MODEL

Based on the determined performance coefficients, it is possible to parameterize the fixed-displacement pump block of the SimHydraulics library.

A test rig, as shown in Fig.5 is then modeled to test the accuracy of the performance coefficients model.

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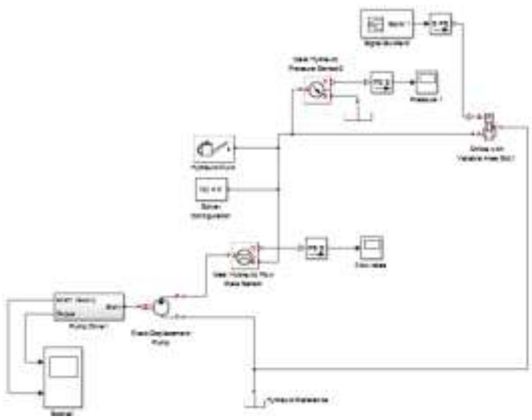

Figure 5. Test rig of the pump model

Simulation results:

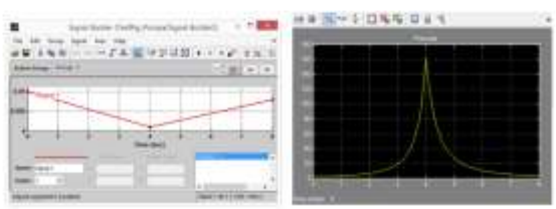

Figure 6. Pressure drop vs. opening of orifice with variable area slot

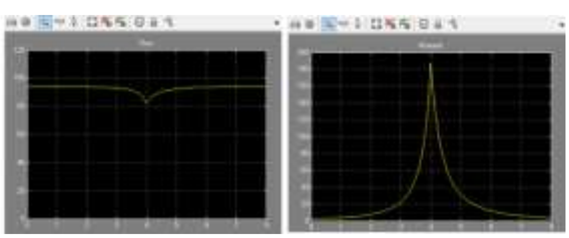

Figure 7. Output flow rate and input torque vs. opening of orifice with variable area slot

5. CONCLUSIONS

The graphs Fig. 6 and Fig. 7 show:

- For the pump pressure drop of 100 (bar), the output flow rate of pump is 86.91 (l/min) very close to the actual flow rate value of 87.12 (l/min) performed by the pump, resulting in an inaccuracy of only 0.24%.
- For the pump pressure drop of 100 (bar) and output flow rate of 86.91 (l/min), the input torque value is 116.3 (Nm), very close to its actual value of 117.94 (Nm) resulting in an inaccuracy of 1.39%.
- The foregoing simulation results prove the accuracy of the model introduced.

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