Experimental investigation on system performance using palm oil as hydraulic fluid

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Abstract

Purpose – The aim of this paper is to investigate hydraulic system performance using vegetable-based palm oil as hydraulic fluid.

Design/methodology/approach – The hydraulic system performance test at different operating conditions, such as pressure, speed and oil ageing, was performed using a Yuken vane pump test rig. The endurance system performance test was also conducted for 200 and 400 h. The effect of speed on flow slip coefficient in discrete and continuous tests was studied. In discrete testing, pressure of 35 and 200 bar and speed of 750 and 1,439 rpm were used in determining flow slip coefficient. The instantaneous data were recorded in a computer using an analog-to-digital data acquisition system with respect to time and the parameters stored were reservoir temperature, return line temperature, suction and delivery pressures, instantaneous flow rate, total flow, total running time and torque. The obtained results were interpolated for future prediction of the system performance.

Findings – The experimental and interpolated results showed that slip coefficient decreases with increasing pump speed. The effect of aging condition on volumetric efficiency showed that the efficiency increases with aging period due to increase in oil viscosity.

Practical implications – This vegetable-based palm oil could be a potentially useful substitute for mineral-based energy transport media such as hydraulic fluid.

Originality/value – The investigation of hydraulic system performance using palm oil as hydraulic fluid is scarce in the literature. Therefore, the current study is quite new for the hydraulic system performance and it is hoped that it will provide a high value to researchers for further research before it can be used as hydraulic fluid.

Keywords Oils, Vegetable fats, Hydraulic engineering, Pumps

Paper type Research paper

Nomenclature

C_c	=	Coulomb friction coefficient
C_s	=	Flow slip coefficient
C_v	=	Viscous friction coefficient
D	=	Pump size (m^3/rev)
P	=	System pressure (bar)
Q_a	=	Actual pump flow rate (m^3/s)
Q_c	=	Compressed flow rate (m^3/s)
T_a	=	Actual torque (Nm)
T_v	=	Viscous torque (Nm)
w	=	Pump speed (rpm)
μ	=	Viscosity (Pa.s)
η_v	=	Volumetric efficiency
η_m	=	Mechanical efficiency
Q_A, Q_B, Q_C, Q_D	=	Flow rate for case A, B, C and D, respectively, (m^3/s)

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Introduction

Before nineteenth century, the main lubricating fluids were natural or vegetable esters contained in vegetable oils such as soya and rapeseed or animal fats such as lard and sperm oil (Wilson, 1998). During World War II, other types of based oils were introduced and used. Since, a number of aircraft were used during the World War II, high-temperature resistance fluids were required. The demand for thermally stable oil resulted in a number of synthetic esters produced during this period.

Research in biodegradable and environmental friendly lubricants and hydraulic fluids has recaptured many researchers interest in the 1990s (Cheng *et al.*, 1991; Honary, 1995; Glancey *et al.*, 1996, 1998; Kodali, 2002). During industrial era, there is a lot of development in factory machinery. Heavy-duty machinery such as in steel mill use more and more hydraulic control system rather than mechanical control system. This is due to high power to weight ratio and high accuracy of hydraulic system compared

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to pure mechanical or pure electrical system. Steel mill machinery requires a need for the safest possible media, that is, least flammable hydraulic fluid. Vegetable-based oil is the potential candidate since it has a very high flash and fire point. Furthermore, it possesses high-viscosity indices (VI) compared to mineral oil.

Hydraulic fluid can be regard as blood for a hydraulic system. The characteristics and properties of the fluid can make all the differences to the satisfactory or failure of hydraulic components in specific, to the hydraulic system in general. Function of hydraulic fluid can be summarized as follows (Busch and Backe, 1993; Ahola, 1998):

1. transmitting fluid power;
2. lubricating agent;
3. corrosion protection;
4. transporting contaminants to reservoir; and
5. cooling media.

Out of five main tasks of hydraulic fluid as outlined above, the role in transmitting power is more important than lubricating moving surfaces or other functions. While the oil is transported from reservoir through several hydraulic components by a positive displacement pump and finally returned to the reservoir, the energy is converted into several forms. However, most of the standard methods designed and test benches produced concentrate on testing lubricating performance (ASTM D2882-90, 1991, ASTM D2271-94, 1999). Thus, the role as power transmittance is more poorly understood, except the basic relationship between fluid properties and system performance.

Simple but accepted pump tests are necessary to investigate the power transmission, lubricating and other performance aspects under varying pressure and speed. When the fluid is tested under conditions close to actual use, conclusions on the oil ageing property in practical operation can be drawn more accurately. In the 1990s, several pump testing methods were proposed and used to evaluate the changing properties of rapidly biodegradable hydraulic oils (Perez and Brenner, 1992; Totten and Bishop, 1995). Most of testing methods and equipment were only dedicated for lubricating fluids.

Cheng and Galiano-Roth (1994) reported their findings related to the performance of piston pump running on vegetable-based oil. Except their work, very little information in a published form is available on power transmitting performance, thus, this study focus on investigating hydraulic system performance when vane pump test rig was running on vegetable-based oil.

Experimental details

Materials

The palm oil used in this study was of refined, bleached and deodorized (RBD) palm oil. It was processed for cooking oil and constituents for margarine and shortening. The fatty acid composition for palm oil used is as follows: saturated acids (7 percent), mono-unsaturated acids (63 percent) and poly-unsaturated acids (30 percent).

The density of the RBD palm oil was measured using a 50 ml pycnometer. The viscometric properties of the oil were measured using Brookfield (Stoughton, MA, USA) viscometer model DV -I + . Viscosity was measured in triplicate for each oil sample using spindle SC4-18. The coaxial type cylinder viscometer generated 10 discrete shear rates from 3 to 100 rpm, which is equivalent to 3.9-131.6 s^{-1}. The instrument's accuracy and reproducibility were 1 and 0.2 percent of full scale, respectively. The integrated thermosel provided constant temperatures for experimentation with an accuracy of ± 0.1°C.

Experimental procedure

A Yuken vane pump was chosen to circulate the fluid under investigation. In addition to the 12 vane pieces, the balanced vane pump cartridge contains three other major components, i.e. cam ring, rotor, and side bushing. The rig currently at Kolej Universiti Sains dan Teknologi Malaysia is integrated with LabVIEW Software Version 6.1. The rig can be ON/OFF remotely for safety reason. Using the online software, steady and transient performance of the hydraulic system can also be monitored. Figure 1 shows the circuit diagram of hydraulic system fabricated.

The instantaneous data were recorded in a computer using an analog-to-digital data acquisition system with respect to time and the parameters stored were reservoir temperature, return line temperature, suction and delivery pressures, instantaneous flow rate, total flow, total running time and torque. In addition to that, the condition of the fluid was monitored routinely.

In discrete testing, flow rate from four maximum and minimum pressure and speed combinations were measured. The operated pressure was 35 and 200 bar and operated speed were 750 and 1,439 rpm. This minimum test data was used in determining predicted slip coefficient.

In continuous testing, one operating parameter such as pressure was varied from low to high values while maintaining other operating parameters. In determining the flow slip coefficient, the hydraulic system was operated at different speed conditions and the oil temperature was maintained at 70°C with the help of shell and tube heat exchanger. Pressure was increased from 35 to 200 bar. Based on actual flow rate, volumetric efficiency and flow slip coefficient were calculated.

The test rig was operated continuously up to 400 h to study the effect of oil aging on system performance. The system performance evaluation test was conducted at 200 and 400 h by running the system pressure from low to high speed and pressure.

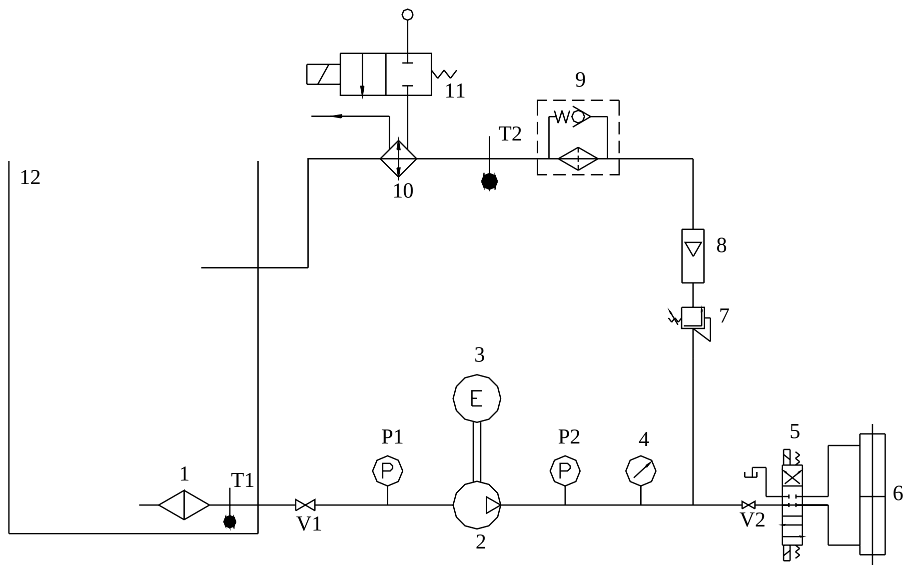
Analysis of system performance

Flow rate and pressure across the vane pump are the main consideration for the flow analysis. Pressure induced (poiseuille) flow and velocity induced (couette) flow govern the flow losses. Owing to these losses, the actual flow rate is less than theoretical flow rate. The actual pump flow coming out from pump outlet, taking into account the compressibility term, can be written as:

$$Q_a = Dw - C_s \frac{\mathrm{DP}}{2\pi\mu} - Q_c. \tag{1}$$

Mechanical losses involved are due to coulomb and viscous friction. Owing to the losses, the actual torque required to run the pump is higher than the theoretical one and is given by:

Figure 1 Schematic drawing of the hydraulic test rig system



Legends: 1 -strainer; 2 -pump; 3 -3 phase electric motor; 4 –pressure gauge; 5 -4/3 way double solenoid spring return directional control valve; 6 –double acting actuator; 7-loading valve; 8 –flowmeter; 9 –return line filter; 10 –heat exchanger; 11 -2/2 way single solenoid spring return valve; 12 –oil reservoir; T1, T2 – thermocouple; P1, P2 – pressure transducer; V1, V2 – shut-off valve

$$T_a = \frac{DP}{2\pi} + C_c \frac{DP}{2\pi} + C_v \mu D w. \qquad (2)$$

Volumetric and mechanical efficiencies can be used to describe the system performance. The leakages as well as the compressibility contribute to the reduction of volumetric efficiency. The volumetric efficiency is defined as:

$$\eta_v = \frac{Q_a}{Dw}. \qquad (3)$$

Using the actual pump flow in equation (1), the volumetric efficiency can be written as:

$$\eta_v = 1 - C_s \frac{P}{2\pi\mu w} - \frac{Q_c}{Dw}. \qquad (4)$$

The mechanical efficiency is defined as the ideal torque divided by the actual torque:

$$\eta_m = \frac{DP}{T_a}. \qquad (5)$$

Using the actual torque expressed in equation (2), taking into account all the torque losses, the mechanical efficiency can be written as:

$$\eta_m = \frac{P}{(P/2\pi) + C_c(P/2\pi) + C_v \mu w}. \qquad (6)$$

The second and third terms in the denominator correspond to losses due to coulomb and viscous frictions, respectively.

Results and discussions

Physical properties of oil

The density of the RBD palm oil with increasing temperature is shown in Figure 2. The density was measured below 60°C, in order not to damage the pycnometer used. Density decreases linearly with temperature. Thus, the density at 70°C was calculated to be 0.881 kg/m^3. Beside viscosity, the density of the oil also influences the suction capability of pumps.

The variation of palm oil dynamic viscosity with temperature is shown in Figure 3. The viscosity decreases exponentially with temperature. The viscosity index of the oil was calculated to be 194. The flow behavior curve of the oil from 30 to100°C is shown in Figure 4. From the flow curves, it can be seen that the fluid behaves as non-Newtonian material at all temperatures.

Flow slip coefficient from discrete performance test

Actual flow rate was measured using an axial turbine type flow meter. Table I shows flow rate measured at four different conditions from discrete test. As expected, the flow rate for 1,439 rpm was higher than of 750 rpm as depicted by equation (1). For the same speed, flow rate for higher pressure was reduced.

Predicted flow slip coefficient was calculated based on constant coefficient linear model and variable coefficient linear model as proposed by McCandlish and Dorey (1984). Based on ratio of flow difference and pressure difference between 200 and 35 bar, flow slip coefficient at high and low speed were calculated as 3.244×10^{-8} and 4.413×10^{-8}, respectively. The average flow slip coefficient for all operating conditions (750 and 1,439 rpm speed; 35 and 200 bar pressure) was calculated as 3.828×10^{-8}.

Based on slip coefficients already calculated using constant coefficient linear model, slip coefficients for other speeds were calculated using variable coefficient linear model. Linear interpolation was performed to determine flow slip coefficients at 1,199 and 899 rpm operation. The coefficient was calculated as a function of speed by:

Figure 2 Density versus temperature for palm oil

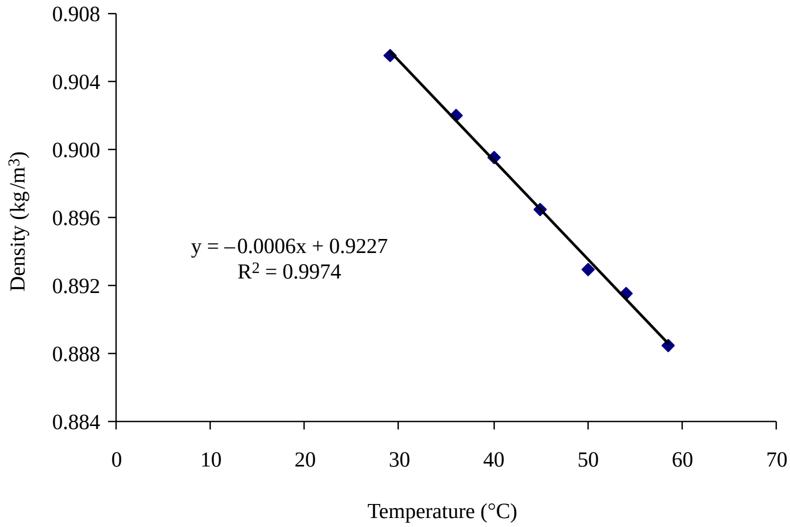
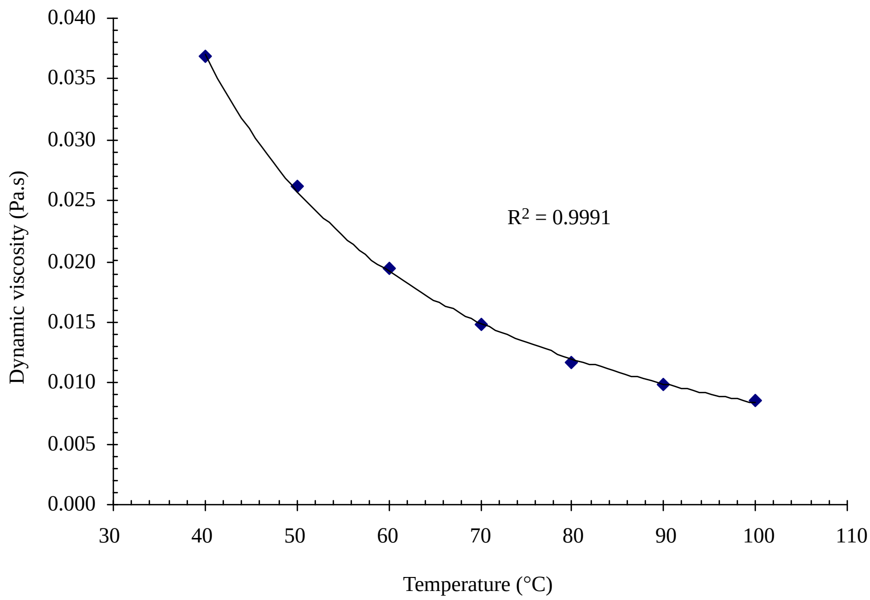


Figure 3 Variation of viscosity with temperature for palm oil



$$C_{s\omega} = C_{sAB} + (C_{sCD} - C_{sAB}) = \frac{(w - w_A)}{(w_C - w_A)} \qquad (7)$$

The interpolation step yields the slip coefficient 3.651×10^{-8} and 4.159×10^{-8} for 1,199 and 899 rpm, respectively. The experimental and interpolated results showed that slip coefficient decreases with increasing pump speed. The high-slip coefficient at low speed was due to the high-leakage flow between pump rotor and cam ring. As the speed increases, pump vane seals better to the inner space of cam ring due to high-centrifugal force, thus results in reduced slip coefficient.

Flow slip coefficient from continuous performance test

Figure 5 shows the variation of flow rate and system loading for pump speed of 600, 899 and 1,199 rpm from continuous testing. System loading greater than 95 and 125 bar cannot be achieved with 600 and 899 rpm, respectively, as system will stall when operated at higher pressure. Based on the highest slope at 600 rpm speed, the calculated values showed that more leakage occurs even with same loading difference. This can be explained by referring to Figure 6. When the rotor rotates, the vane moves inwardly and outwardly in the rotor slot, depending on the balanced pressure between centrifugal and pressure forces. At higher rpm, the pump vane is strongly in contact with pump cam ring due to strong centrifugal force. This action resists the oil from slipping from high-pressure chamber to the low-pressure chamber. However, at lower speed, less centrifugal force exists, thus produce less sealing effect.

In other continuous or steady state testing, pressure was varied from 35 to 200 bar while speed and temperature were

Figure 4 Flow behavior curves from 30 to 100°C

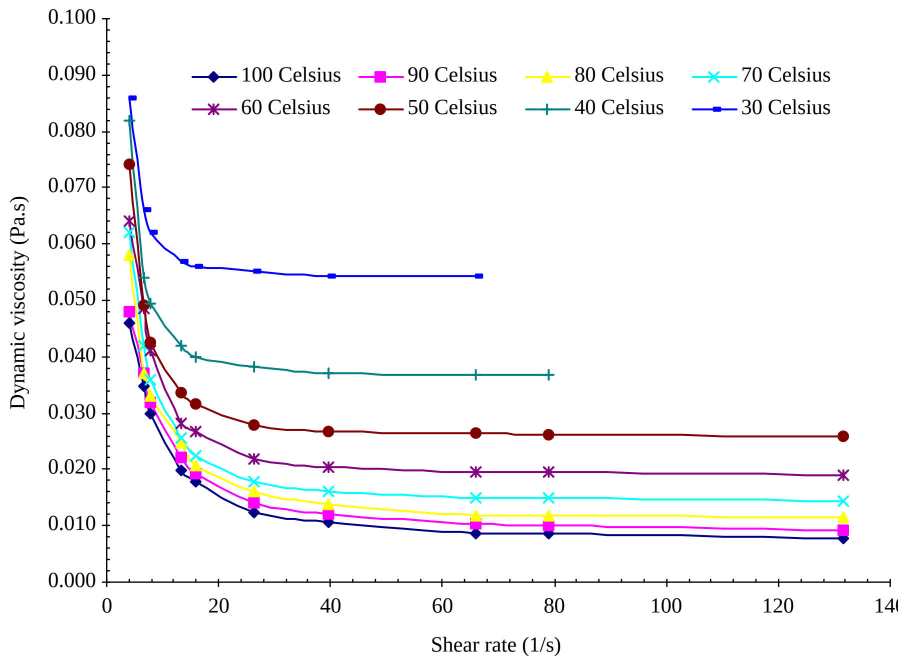


Table I Speed, pressure and flow rate from discrete test

Test	Speed (rpm)	Pressure (bar)	Flow rate (m^3/s)
A	1,439	35	1.83×10^{-4}
B	1,439	200	1.37×10^{-4}
C	750	35	8.22×10^{-5}
D	750	200	2.04×10^{-5}

kept constant throughout the test in order to measure the volumetric efficiency. The test was conducted at a temperature of 70°C. Volumetric efficiency was calculated for a number of test pressures between 35 and 200 bar. Dimensionless parameter values also correspond to the same pressures as was selected for volumetric efficiency.

Plot of volumetric efficiency versus dimensional parameter ($P/\mu w$) is shown in Figure 7, which yields a straight line. The quality of data fitted was shown by correlation coefficient (R^2), where the value of R^2 for 1,439, 1,199, 899 and 600 rpm were 0.9936, 0.9996, 0.9987, and 0.9720, respectively. When the compressibility term was ignored, the slope of the graph was the flow slip factor based on equation (4).

Table II shows the comparison of predicted slip coefficients obtained from both constant coefficient linear model and variable coefficient linear model and actual slip coefficients obtained from continuous test. The values were quite close to each other for 1,439, 1,199 and 899 rpm with 1.23, 1.28 and 0.48 percent error, respectively. This shows that the models fit to the actual experimental data. However, at lower speed (viz. 600 rpm), the error was higher (typically 5.4 percent) when the constant coefficient linear model was performed. Furthermore, the variable coefficient model assumed a linear relationship between the slip coefficient and speed, while in the actual case it was found that the slip coefficient decreases at increasing rate with speed.

Effect of aging time on oil viscosity and system performance

Viscosity of the oil after 200 and 400 h operation at 40, 70 and 100°C is shown in Table III. The table shows that the oil viscosity decreases exponentially with temperature. Oil viscosity increases with aging time as measured at three temperature points. The large increase in viscosity is due to the fact that the oil is plant-based and no viscosity improver was used with the oil. In general, plant or vegetable-based oil degraded faster than mineral oil.

Using the experimental slip coefficient obtained at 1,439 rpm test as presented in the previous section, the volumetric efficiency for ageing operation was predicted. The volumetric efficiency of the system was calculated using equation (4) by neglecting the compressibility effect. The viscosity of 0.024 and 0.030 Pa.s were used to simulate the performance for 200 and 400 h cases, respectively.

The predicted and actual (experimental) volumetric efficiency with pressure is shown in Figure 8.

From the figure, it can be seen that the predicted volumetric efficiency was higher than the actual efficiency for both aging conditions. The figure also shows that the volumetric efficiency of 400 h was higher than that of 200 h. This is because of the increase in oil viscosity. According to equation (4), fluid viscosity can significantly affect the pump volumetric efficiency. Similar observation was reported by Akehurst and Vaughan (2000).

In the performance test, volumetric efficiency, η_v, as well as the mechanical efficiency, η_m, as a function of pressure and speed were investigated. For this test, the pump was operated from 600 to 1,440 rpm. Operation at 1,000 rpm could not be made since the hydraulic system vibrated at this speed. This

Figure 5 Flow rate versus system loading

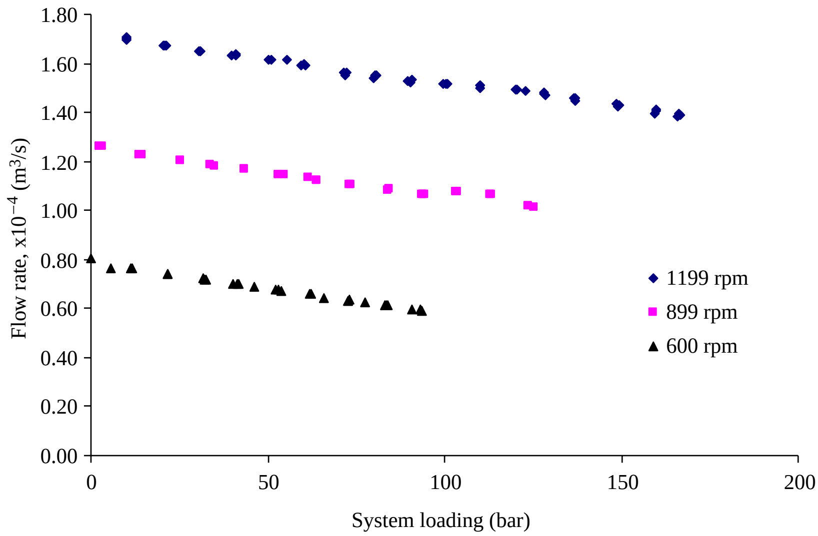


Figure 6 Schematic diagram showing centrifugal and pressure forces acting on vane

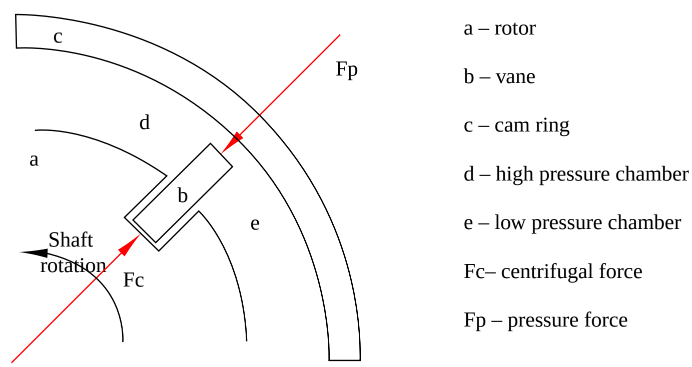


Figure 7 Variation of volumetric efficiency with dimensionless value for speeds of 1,439, 1,199, 899 and 600 rpm

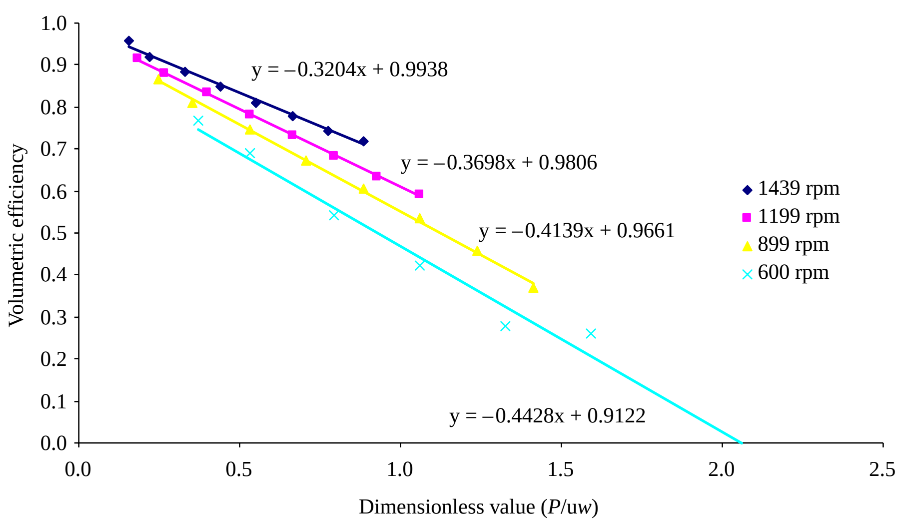


Table II Comparison between predicted and actual flow slip coefficients for four different speeds

Speed (rpm)	Predicted coefficient C_s ($\times 10^{-7}$)	Experimental coefficient C_s ($\times 10^{-7}$)	Coefficient error
1,439	0.3244	0.3204	1.23
1,199	0.3651	0.3698	1.28
899	0.4159	0.4139	0.48
600	0.4667	0.4428	5.40

Table III Viscosity of palm oil at respective running time and temperature

Temperature (°C)	Viscosity (Pa.s) running time 200	At respective (hour) 400
40	0.067	0.094
70	0.024	0.030
100	0.015	0.019

was due to the resonance at system natural frequency. Figure 9 shows that volumetric efficiency increased steadily from around 70 percent to almost 100 percent. On the other hand, the figure shows that the pump speed has no effect on the mechanical efficiency. Mechanical efficiency was not affected since there was no loading change to the system.

However, according to equation (6), speed term is available in the denominator in calculating mechanical efficiency. Insensitivity of the mechanical efficiency to the speed change might due to large value of viscous coefficient, C_v. Thus, the viscous friction coefficient was investigated. If viscous torque term in equation (2) is divided by theoretical torque, the ratio can be written as:

$$T_v/\mathrm{DP} = C_v(\mu w/P). \quad (8)$$

If dimensionless parameter T_v/DP is plotted against dimensionless parameter $\mu w/P$, viscous friction coefficient can be determined. In order to determine the viscous friction coefficient, a test was conducted at 70°C. Speed of the pump was maintained at 1,199 rpm. Pressure was varied from 35 to 210 bar. Actual torque was recorded. Theoretical, coulomb and viscous torques as well as dimensionless parameter $\mu w/P$ were calculated. Using these values, dimensionless parameter T_V/DP was plotted against dimensionless parameter $\mu w/P$ and is shown in Figure 10.

From this graph, the viscous friction coefficient was determined as 3.61×10^5. This result compares well with other researchers results. The C_v as conducted on gear pump was 2.05×10^5 (McCandlish and Dorey, 1981). The large value of C_v gives rise to insensitivity of mechanical efficiency even the speed was varied from 600 to 1,440 rpm as shown in Figure 9.

Conclusion

This study showed that with limited operating data, flow slip coefficient for other operating conditions can be predicted with small error using constant and variable coefficient linear models. The experimental and interpolated results showed that slip coefficient decreases with increasing pump speed. The results indicate that viscosity of energy transport media has significant effect on volumetric efficiency. Vegetable oil which is environmental friendly degrades faster than mineral oil, and thus has significant increase in viscosity with aging condition. This property influences the overall system performance and should be considered when considering vegetable-based oil as energy transport media in hydraulic system.

Figure 8 Variation of predicted and experimental volumetric efficiency with pressure

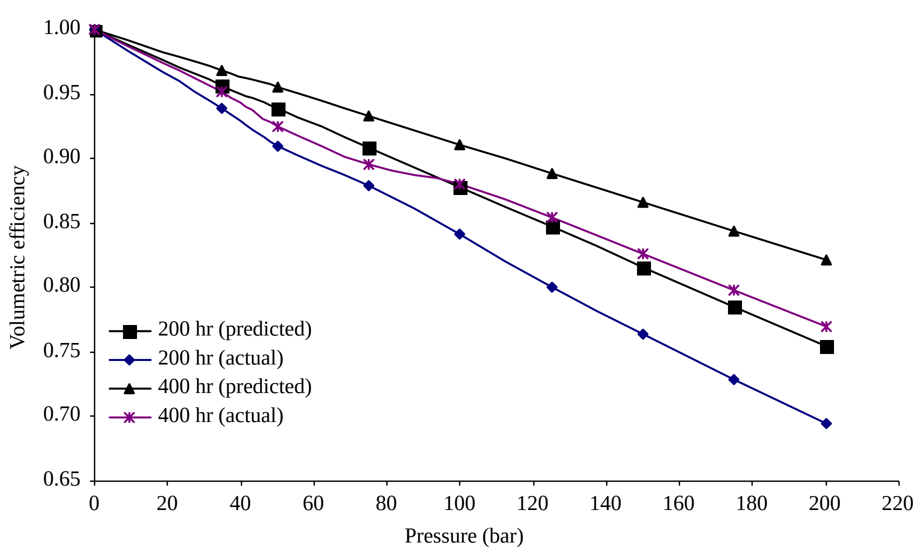


Figure 9 Volumetric and mechanical efficiencies versus pump speed

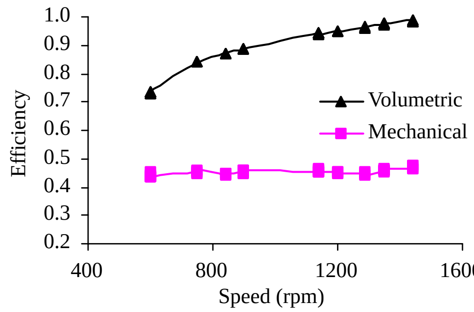
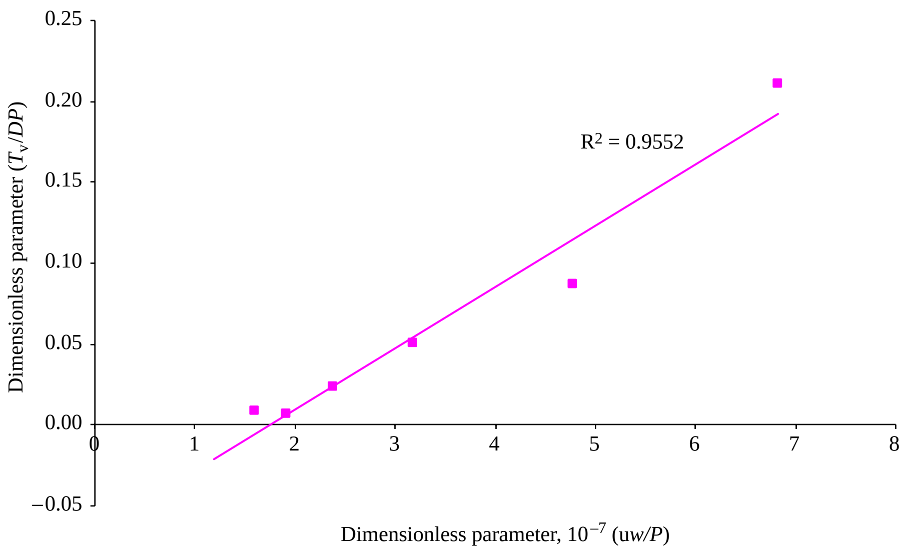


Figure 10 Determination of viscous friction coefficient



References

Ahola, P.M. (1998), "Biodegradable hydraulic fluids in the forest", SAE Technical Paper 981517, SAE, Warrendale, PA.

Akehurst, S. and Vaughan, N.D. (2000), "The effect of lubricant temperature on the loss mechanisms associated with an automotive metal V-belt CVT", SAE Technical Paper 2000-01-1872, SAE, Warrendale, PA.

ASTM D2882-90 (1991), "Standard test method for indicating wear characteristics of petroleum and non-petroleum hydraulic fluids in a constant volume vane pump", *Annual Book of ASTM Standards, Section 5, 05.02, Petroleum Products and Lubricants*, American Society of Testing and Materials, Philadelphia, PA.

ASTM D2271-94 (1999), "Standard test method for preliminary examination of hydraulic fluid (wear test)", *Annual Book of ASTM Standards, Section 5, 05.02, Petroleum Products and Lubricants*, American Society of Testing and Materials, Philadelphia, PA.

Busch, C. and Backe, W. (1993), "Development and investigation in biodegradable hydraulic fluids", SAE Technical Paper 932450, SAE, Warrendale, PA.

Cheng, V.M. and Galiano-Roth, A.S. (1994), "Vegetable-based oil performance in piston pumps", SAE Technical Paper 941079, SAE, Warrendale, PA.

Cheng, V.M., Wessol, A.A., Baudouin, P., BenKinny, M.T. and Novic, N.J. (1991), "Biodegradable and nontoxic hydraulic fluids", SAE Technical Paper 910964, SAE, Warrendale, PA.

Glancey, J.L., Benson, E.R. and Knowlton, S. (1996), "A low volume fluid power system for the evaluation of genetically modified vegetables as industrial fluids", SAE Technical Paper 961725, SAE, Warrendale, PA.

Glancey, J.L., Knowlton, S. and Benson, E.R. (1998), "Development of a high oleic soybean oil-based hydraulic fluid", SAE Technical Paper 981999, SAE, Warrendale, PA.

Honary, L. (1995), "Performance of selected vegetable oils in ASTM hydraulic tests", SAE Technical Paper 952075, SAE, Warrendale, PA.

Kodali, D.R. (2002), "High performance ester lubricants from natural oils", *Industrial Lubrication and Tribology*, Vol. 54 No. 4, pp. 165-70.

McCandlish, D. and Dorey, R. (1981), "Steady state losses in hydrostatic pumps and motors", paper presented at 6th International Fluid Power Symposium, Cambridge, UK, April 1981, Paper C3, pp. 131-44.

McCandlish, D. and Dorey, R.E. (1984), "The mathematical modelling of hydrostatic pumps and motors", *Proceedings of Instn Mech Engrs*, Vol. 198B No. 10, pp. 165-74.

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Perez, R.J. and Brenner, M.S. (1992), "Development of a new constant volume vane pump test for measuring wear characteristics of fluids", *Lubrication Engineering*, Vol. 48, pp. 354-9.

Totten, G.E. and Bishop, R.J. Jr (1995), "Hydraulic pump testing procedures to evaluate lubrication performance of hydraulic fluids", SAE Technical Paper 952092, SAE, Warrendale, PA.

Wilson, B. (1998), "Lubricants and functional fluids from renewable sources", *Industrial Lubrication and Tribology*, Vol. 50 No. 1, pp. 6-15.

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