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ACHIEVING EFFICIENCY GAINS THROUGH HYDRAULIC FLUID SELECTION: LABORATORY PREDICTION AND FIELD EVALUATION

Abstract

Studies completed in medium and high pressure vane and piston pumps showed that their efficiency depends on discharge pressure and on oil viscosity. Recently, a limited field test completed in a medium size excavator confirmed that significant gains in efficiency and productivity even under moderate operating temperatures could be achieved by substituting the OEM-recommended oil with a Maximum Efficiency Hydraulic Fluid (MEHF).

To understand the origin of the efficiency gains observed in the field test, we instrumented an Eaton-Vickers V104C pump to record pressure, temperature, flow rate and power input at the pump shaft. Tests were completed at 69 and 138 bars and temperatures ranging between 30 and 90°C. Candidate hydraulic fluids included the two field test oils and an ISO VG 46 HM oil.

The efficiency gains in the pump stand are consistent with those generated in the field test making the former a valuable tool to assess the potential benefits of MEHF oils in actual service. Thermodynamic models determined earlier can be used to estimate the temperature of the oil leakage stream and the actual oil temperature inside the pump.

1. Introduction

Four key factors are driving the mobile-equipment business today: environment, energy, function, and cost. On the environmental front, tighter exhaust-emission regulations are an ongoing concern for equipment manufacturers and users. Machine noise is also coming under regulatory scrutiny as well. With fuel costs recently at record highs, energy savings is now a key, decisive sales argument for machine manufacturers. There is also increasing demand to automate repetitive functions and improve operating efficiency, enhance productivity, and reduce operator fatigue. Finally, mobile-equipment users are more concerned than ever with total cost of ownership, not just a vehicle's sticker price. This is putting pressure on hydraulic suppliers to drive down component costs, improve reliability, and make them maintenance friendly.

Hydraulic fluids can play a major role in this challenging context. Extensive work conducted on medium and high pressure vane and piston pumps showed that proper selection of a fluid with an optimal viscosity-temperature relationship could help in maintaining volumetric efficiency at a high level and in controlling the rate of temperature increase [1, 2, 3, 4, 5, 6, 7, 8, 9]. An optimum hydraulic fluid should thus bring significant benefits in terms of the environment by decreasing the amount of fuel needed to produce a desired level of work. This oil will also contribute to improve the energy efficiency of the equipment by maintaining a high efficiency under both high and low temperature conditions. It should also help lower the cost of ownership by providing proper pump lubrication under high temperature conditions and limiting the temperature increase in the circuit that promotes oxidation, oil leakage and hose hardening.

The objective of this work is to demonstrate that differences in operating efficiency that were observed in hydraulic pump loops with different fluids could also be observed in actual service. For this purpose, we conducted a limited field test in which we compared the performance of an SAE 10W recommended by the equipment maker with those of a high VI shear stable ISO 46. These two fluids were then evaluated in a pump loop that was fully instrumented to provide accurate measurement of the pump efficiency and of the fluid temperature in various parts of the circuit. A third fluid falling in the ISO VG 46 grade with a VI of 100 was also evaluated. Comparison of the performance of the the two first test oils in the field and in the pump loop will show if this later can be used to estimate the benefit provided by an MEHF oil in actual service.

2. Field Test

2.1 Test equipment

A medium size excavator, Caterpillar 318C L, was used for the evaluation of the effect of lubricants on equipment performance. The excavator has a 1 m^3 bucket corresponding to a capacity of 2 metric tons. It is powered by a Caterpillar 3066T diesel engine producing 93 kW (125 HP) at 2200 rpm and has a typical fuel consumption of 19 to 23 liters per hour (5 to 6 gallons per hour). The equipment has a dual piston pump feeding 3 piston motors that operate the tracks and swivel, plus boom, stick, and bucket cylinders. The two pumps can work at a maximum pressure of 345 bars (5000 psi). Each pump has a nominal flow rate of 95 liters per minute (25.1 gpm) per pump yielding a total flow of 190 liters per minute. The hydraulic circuit contains 255 liters (67.4 gallons) of fluid. The hydraulic fluid reservoir has a capacity of 127 liters (33.6 gallons).Pressure transducers were installed to monitor pump pressure and thermocouples were placed at the stick, boom and hydraulic reservoir location.

2.2 Test fluids

Two hydraulic fluids coded A and B were evaluated in the excavator. Fluid A is an SAE 10W hydraulic fluid recommended by the equipment manufacturer. Fluid B is an ISO VG 46 fluid formulated with a shear stable VI Improver. The NFPA grade [11] and the viscosity of the test fluids are shown in Table 1.

Table 1: Characteristics of the field test fluids

Fluid Code	A	B
SAE or ISO Grade	SAE 10W	ISO VG 46
NFPA Grade	L46-46	L32-100
KV @40 °C, mm^2	38.38	49.94
KV @100 °C, mm^2	6.10	10.14
VI	104	196

2.3 Test protocol

The work day consisted of the following steps:

- 15 minutes for machine warm-up
- refuel, start data logger
- 3 hours of morning work
  - 55 minutes of continuous work, 5 minute break, repeat 3 times
- 1 hour break
  - Refuel, measure fuel weight to 0.1 Kg
- 3 hours of afternoon work
  - 55 minutes of continuous work, 5 minute break, repeat 3 times
- Stop test, refuel, determine total fuel consumption.

Table 2: Field test program

Day	Test fluid	% Throttle
1	A	90
2	A	100
3	fluid and filter change	
4	B	100
5	B	100
6	B	90
7	fluid and filter change	
8	A	100
9	A	90

The work cycle used in this test lasted about one minute. It included four steps.

- Take a full scoop of dirt
- Rotate 180° and travel 30 meters (100 ft)
- Dump the load
- Rotate 180° and return to the starting point

The tests were made under mild climatic conditions. The ambient temperature ranged between 7 and 18 °C, (45 to 65 °F). The equipm ent was operated with the engine running at 90 and 100 % throttle. The complete test program is shown in Table 2 above.

2.4 Test results

The fuel consumption for the two test fluids and the two throttle settings are detailed in Table 3. Fuel consumption increased when going from 90 to 100% throttle setting. Results in Table 3 show that the equipment consumed significantly less fuel per hour with fluid B irrespective of the throttle setting.

Table 3: Fuel consumption per hour

		Fuel Consumption per Hour	
Throttle	Fluid	kg/hour	% Improvement
Full	A	19.50	-
Full	B	16.80	13.8
90%	A	15.20	-
90%	B	13.89	8.6

The number of work cycles per hour was recorded and the average values are shown in Table 4. More cycles per hour were completed with fluid B irrespective of the throttle setting.

Table 4: Work Cycles completed per hour

		Work Cycles per Hour	
Throttle	Fluid	Number of cycles	% Improvement
Full	A	53.5	-
Full	B	56.6	5.8
90%	A	40.0	-
90%	B	49.7	24.3

Combining the fuel consumption per hour and the number of cycles per hour yielded the average fuel consumption per work cycle. The results for the two fluids at the two throttle settings are shown in Table 5. The productivity gains achieved when using fluid B are 26.3% and 18.4% at 90 and 100% throttle respectively.

Table 5: Fuel consumed per work cycle

		Fuel Consumption per Cycle	
Throttle	Fluid	Kg/cycle	% Improvement
Full	A	0.364	-
Full	B	0.297	18.4
90%	A	0.380	-
90%	B	0.280	26.3

3. Pump Tests

3.1 Test stand

The test rig uses an Eaton Vickers V104C vane pump equipped with a Connesstoga cartridge. The pump is driven by a 15 kW electric motor at 1165 rpm. The pump circuit, shown in Figure 1, includes the following elements:

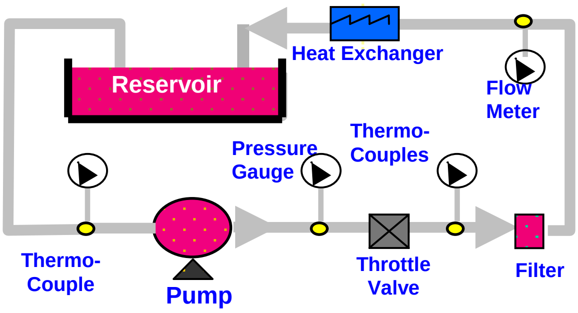
- fluid reservoir
- Eaton Vickers V104C vane pump
- pressure regulator (throttle valve)
- low pressure fluid filter located after the throttle valve
- flow meter
- heat exchanger

Thermocouples are installed at the following locations:

- about 100 mm before the inlet of the pump
- 3 to 5 mm inside the pump suction port
- at the pump outlet
- immediately after the pressure regulator

The power consumed by the electric motor to drive the pump is recorded. It is about equal to the sum of the nominal hydraulic power and of the power needed to overcome the hydromechanical losses taking place in the pump.

Figure 1: Schematic of the Vane Pump Circuit



3.2 Test procedure

After thoroughly flushing the circuit and installing a new filter, the reservoir is filled with 18.9 liters (5 US gallons) of test fluid. All tests are started with the fluid and circuit at room temperature. The discharge pressure is increased to 138 bars (2000 psi) until the desired fluid temperature is attained. The pressure is then either maintained at 138 bars or reduced to 69 bars (1000 psi). The temperature is controlled with the heat exchanger. Data are collected for a period of 30 to 60 minutes. During that period, pump flow rate, power consumed by the electric drive motor and temperatures are collected at one second intervals and averaged once a minute. Each candidate fluid is tested with the pump inlet temperature ranging between 30 to 90 ℃ at 69 bars and 40 to 80 ℃ at 1 38 bars. The complete evaluation of a test fluid lasted over 60 hours.

3.3 Test fluids

We evaluated in the pump rig the two fluids that were tested in the field and one additional ISO VG 46 hydraulic fluid with a VI of 101. The characteristics of the fresh test fluids are detailed in Table 6. The viscosity of fluid B was monitored during the test. We observed a maximum loss of viscosity of only 3% after 60 hours in the pump circuit.

Table 6: Characteristics of the fresh test fluids

Fluid Code	A	B	C
SAE or ISO Grade	SAE 10W	ISO VG 46	ISO 46
NFPA Grade	L46-46	L32-100	L46-46
KV @40 ℃, mm^2	38.38	49.94	45.1
KV @100 ℃, mm^2	6.10	10.14	6.71
VI	104	196	101

The viscosity of the test fluids at the inlet and inside the pump were calculated using the Walther MacCoul equation [10].

3.4 Test results

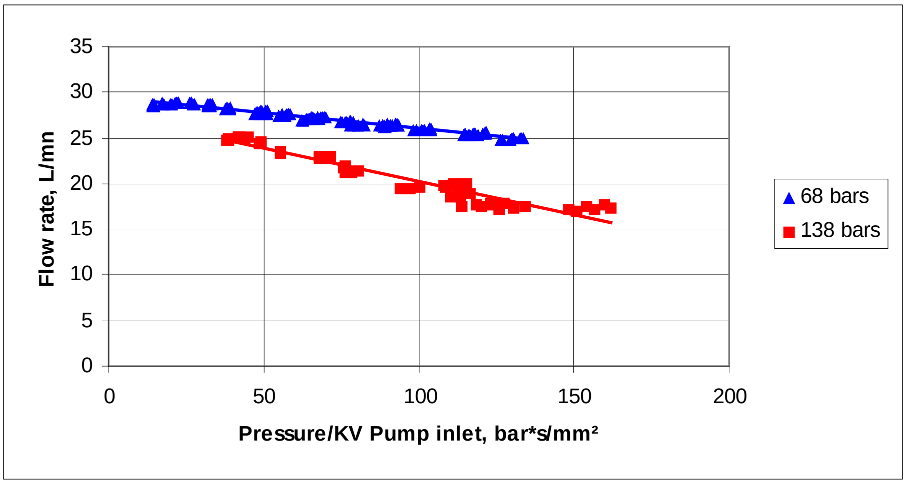
3.4.1 Dependence of flow rate on pressure and viscosity

Previous work [1,3,4,5] showed that for Newtonian fluids, the leakage (Q_l) in the V104C pump equipped with a standard cartridge was proportional to pressure and inversely proportional to viscosity at the pump inlet temperature. Since the actual flow rate is equal to the nominal flow rate (Qa) less the fluid leakage (Q_l) we have:

$Q_a = Q_n - c*P/KV$ at pump inlet [1]

Plotting the flow rate for the three test fluids as a function of the ratio pressure over viscosity at the pump inlet temperature showed that this model does not apply to the V104C pump equipped with the new cartridge. Figure 2 shows a distinct line for each test pressure.

Figure 2: Dependence of flow rate on pressure and viscosity at pump inlet for the three test fluids



By linear regression analysis we obtained the following least square equation:

All pressures: Flow Rate = 30.126 - 0.0618 P/KV Inlet $R^2 = 0.5817$

The poor coefficient of determination obtained when the results for all fluids and pressures are considered shows the need for identifying a new, more appropriate empirical equation describing the dependence of flow rate on pressure and viscosity. The search for a new model was first conducted on the Newtonian fluid C for which the largest number of data points (143) had been collected. Equations of the form Flow Rate = $Q_n + c*P^a/(KVinlet)^b$ have been tested and the results are shown in T. 7.

Table 7: Equations describing the dependence of flow rate on pressure and viscosity for fluid C

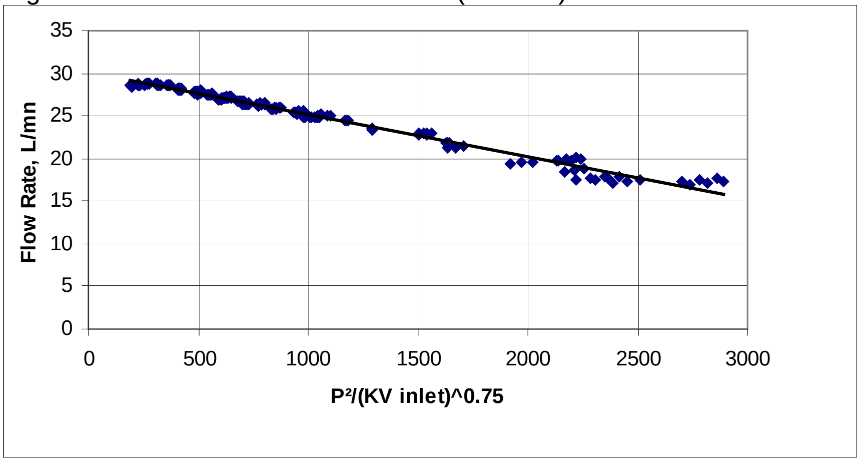
	Qa = Qn - c* P / KV Inlet^0.5			Qa = Qn - c*P² /(1000 * KV Inlet)		
Pressure	Qn	c	R^2	Qn	c	R^2
69 bars	31.2	0.248	0.96	29.5	0.248	0.98
138 bars	37.8	0.628	0.99	29.7	0.628	0.99
All	34.1	0.464	0.90	30.3	0.464	0.97

Using the ratio of pressure by the square root of viscosity or the ratio of the square of pressure by viscosity gave significant increases of the coefficients of determination for all pressures. Non-linear regression analysis was used to determine the value of the parameters "a" and "b" in the equation $P^a/(KVinlet)^b$.Values for "a" and "b", very close to 2 and 0.75 respectively, were obtained leading to the following empirical model:

Flow Rate = $30.06 - 0.0049*P^2/(KV\ Inlet)^{0.75}$ $R^2 = 0.9851$

Plotted in Figure 3 is the flow rate as a function of $P^2/(KV\ Inlet)^{0.75}$.

Figure 3: Flow rate as a function of $P^2/(KV\ Inlet)^{0.75}$



3.4.2 Effect of pressure and viscosity on volumetric efficiency

Using the flow rates determined on the test fluids (Q_a) and a value of the nominal flow rate (Q_n) of 30 L/mn, the volumetric efficiency can be determined as a function of the pump inlet viscosity at each of the two test pressures. The value of Q_n determined by the empirical model is close to the flow rate of fluid C in the pump rig with no applied back pressure.

Figure 4: Volumetric efficiency at 69 bars

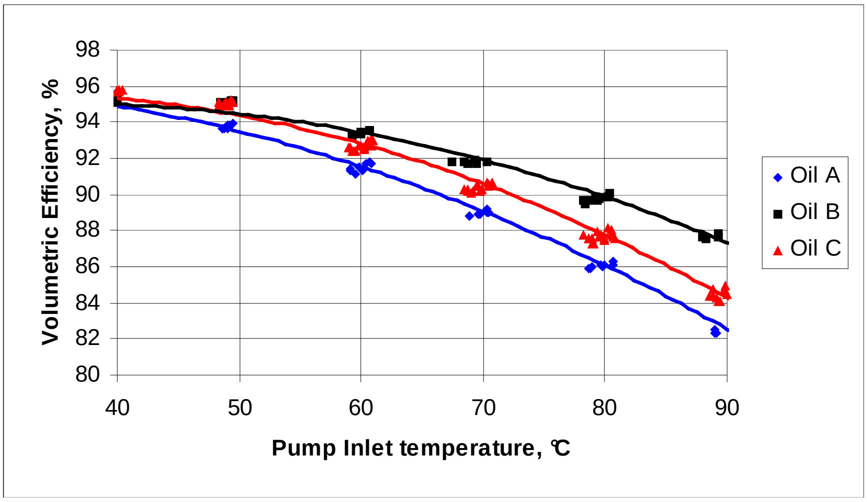
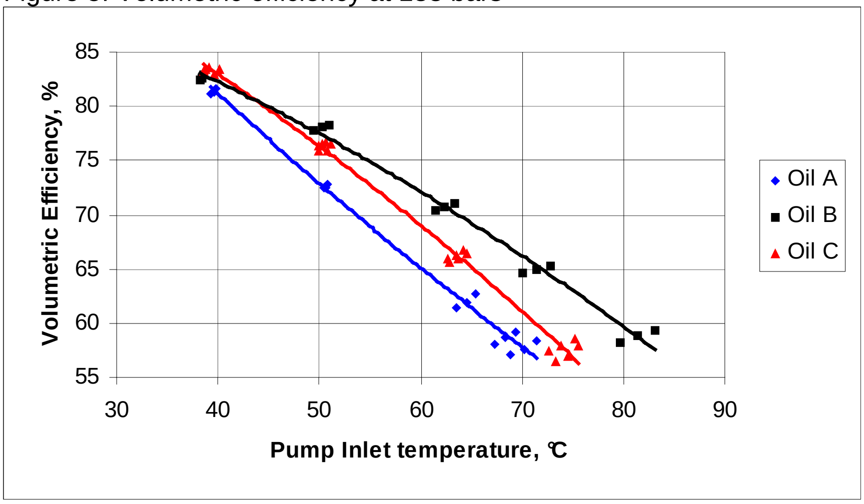


Figure 5: Volumetric efficiency at 138 bars



Volumetric efficiency = Actual flow rate/Nominal flow rate or

$\eta_{Volumetric} = Q_a/Q_n$ [2]

Plotted in Figures 4 and 5 is volumetric efficiency for the three test fluids as a function of pump inlet temperature at 69 and 138 bars.

3.4.3 Effect of viscosity and pressure on global efficiency

Global efficiency is the ratio of the actual hydraulic power (P_{Ha}) coming out of the pump by the power delivered to the pump shaft. The hydraulic power delivered by the pump can be calculated as follows:

Actual hydraulic power = Actual flow rate* Pressure/0.6 or

$P_{Ha} = Q_a*P/0.6$ [3]

P_{Ha}, actual hydraulic power in watts

Q_a, flow rate in liters per minute

P, pressure in bars

The power delivered by the electric motor is about equal to the power needed to deliver the nominal flow (Q_n) at the pump discharge pressure plus that needed to overcome the hydromechanical losses (P_{HM}).

Total Power = $P*Q_n/0.6 + P_{HM}$ [4]

Therefore, mechanical and global efficiency can be obtained by the following equations:

$\eta_{Mechanical} = (P*Q_n/0.6)/ (P*Q_n/0.6 + P_{HM})$ [5]

$\eta_{Global} = \eta_{Volumetric} * \eta_{Mechanical}$ [6]

$\eta_{Global} = [P*Q_a/P*Q_n]* [PQ_n/ (P*Q_n/0.6 + P_{HM})]$ [7]

The average of total power consumption for each fluid and test pressure is shown in Table 8.

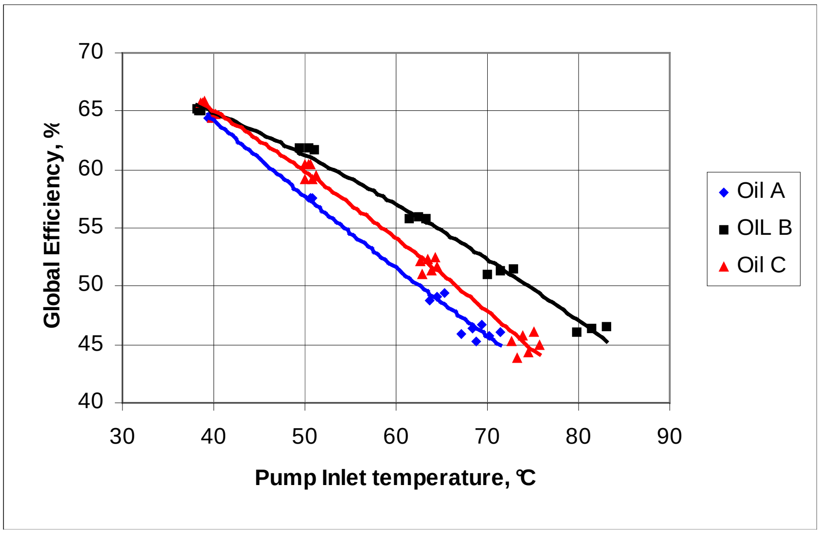
Table 8: Average power consumption in watts

	A	B	C	Average
69 bars	4681	4728	4702	4704
138 bars	8734	8755	8821	8770

Little difference in total power consumed can be observed between the test fluids. At a given pressure, the largest deviation from the average is less than 1%. In order to explain this finding, we need to consider that the total power consumed is the sum of two terms according to Equation [4]. The first one, $P*Q_n/0.6$, is independent from the fluid and temperature. The second one (P_{HM}) has been shown in earlier work [6] to be the sum of two terms. The first one is proportional to pressure and the second is proportional to viscosity. Data from Table 8 indicate that the viscous contribution to the total power consumed is small and, consequently, that mechanical efficiency is not significantly dependent on fluid viscosity under the operating conditions we selected.

Since mechanical efficiency is independent from the fluid viscosity, the influence of viscosity on global efficiency comes only from the volumetric efficiency. Therefore, the relative differences in global efficiency between the test fluids are essentially the same as those observed for the volumetric efficiency. The global efficiency as a function of the pump inlet temperature is shown in Figure 6.

Figure 6: Global efficiency at 138 bars as a function of pump inlet temperature



Using the data from Figure 6, it is possible to estimate the gain in global efficiency relative to fluid A at 138 bars as a function of the pump inlet temperature. The results are detailed in Table 9.

Table 9: Percent gain in global efficiency relative to oil A

Temperature, ℃	35	40	45	50	55	60	65	70	75
	% Gain in Global Efficiency								
Oil B	-1.5	1.3	3.9	6.5	8.9	11.2	13.3	15.1	16.7
Oil C	-0.1	1.4	2.8	3.9	4.8	5.3	5.5	5.2	4.3

It is also possible to estimate at which temperature a fluid reaches a given level of global efficiency at 138 bars. The results are detailed in Table 10.

Table 10: Global efficiency as a function of pump inlet temperature

	Oil A	Oil B	Oil C
Global Efficiency, %	Temperature, ℃		
65	38.8	40.0	40.2
60	46.3	53.4	49.8
55	54.2	64.9	58.6
50	62.4	75.0	66.9
45	71.1	84.2	74.6

3.4.4 Dependence of the hydromechanical losses on pressure and viscosity

Using Equation [4], it is possible to calculate the hydromechanical losses.

P_{HM} = Total power- $P*Q_n/0.6$ [8]

Total power = watts consumed by the electric motor
Q_n = nominal flow rate calculated earlier in liters per minute
P = pressure in bar

Figure 7 shows the hydromechanical losses determined for fluid B. No effect of viscosity on mechanical efficiency can be noted.

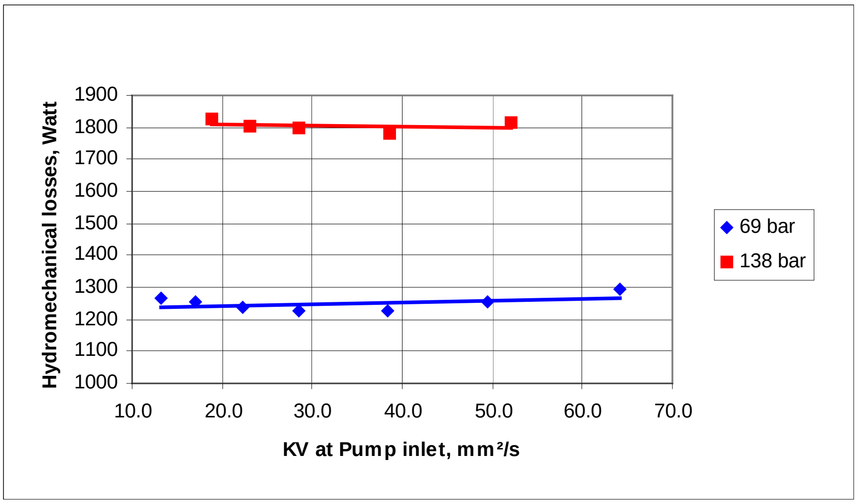
By multiple linear regression analysis on the three test fluids, the following equation was obtained.

P_{HM} = 579 + 0.594*P + 0.971*KV inlet $R^2 = 0.9777$

According to the model, 579 watts are needed to move the fluid in the circuit with no applied back pressure. This compares to 652 watts measured with fluid C with the throttle poppet removed.

Considering that the viscosity of the test fluids at the pump inlet ranged between 9 and 70 mm²/s, the viscous contribution of the fluid to P_{HM} is, according to the least square equation above, ranging between 10 and 70 watts. This compares to 4700 and 8400 watts consumed by the system at 69 and 138 bars respectively.

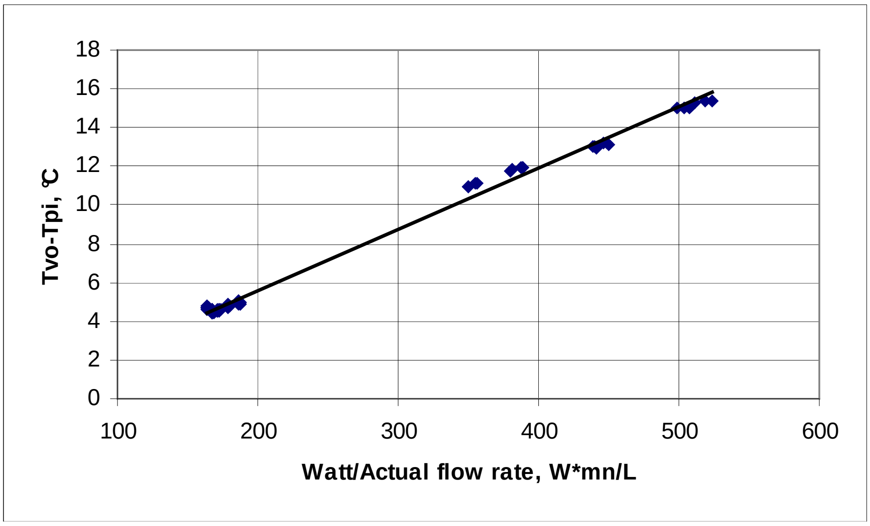
Figure 7: Hydromechanical losses for fluid B



3.4.5 Analysis of the fluid temperature in the circuit

Earlier work by the authors [6] showed that, in a first approximation, the increase of the fluid temperature between the pump inlet (T_{pi}) and the valve outlet (T_{vo}) is proportional to the total energy input to the system and to the reciprocal of the flow rate. The higher the volumetric efficiency, the lower the increase in fluid temperature. Figure 8 shows the difference between T_{vo} and T_{pi} as a function of the ratio of the total energy consumed by the motor by the actual flow rate.

Figure 8: T_{vo}-T_{pi} as a function of the ratio of the energy consumed by actual flowrate



By linear regression analysis of all the data, the following equation was obtained.

T_{vo}-T_{pi} = 0.0317*Energy/Qa – 0.7 R^2 = 0.9917

The negative constant term may result from the loss of energy from the hydraulic circuit to the surrounding environment.

In the study mentioned earlier [6], the difference of temperature between the pump outlet (T_{po}) and the throttle valve outlet (T_{vo}) was shown, in a first approximation, to be proportional to the fluid pressure.

By linear regression analysis of all the data, the following least square equation was obtained.

T_{vo}-T_{po} = 0.042*P + 0.3 R^2 = 0.9552

Finally, the temperature within the pump can be determined by combining the thermal energy of the leakage stream and that of the fluid coming from the reservoir using the following equation.

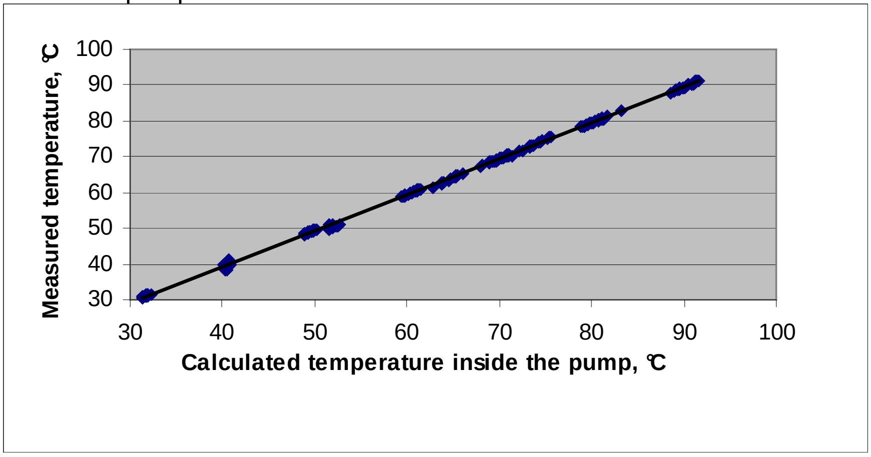
$Q_n*T_{inside}=Q_l*T_{leakage} + Q_a*T_{ip}$ [9]

According to the work discussed earlier [6], the temperature of the fluid leakage should be equal to that of the fluid after the throttle valve since these two streams carry the same energy. By replacing $T_{leakage}$ with T_{vo}, the temperature inside the pump can be calculated. Plotted in Figure 9 is T_{inside} calculated according to Equation [9] and the temperature measured 3 mm inside the pump.

T_{inside} calculated = 1.004* T_{inside} measured - 0.9 R^2 = 0.9997

The temperature measured 3 mm inside the pump shows a strong 1 to 1 correlation to the calculated fluid temperature inside the pump. It is, however, about 0.9 °C lower possibly because the temperature probe is not located exactly at the point where the fluid flowing from the reservoir and the fluid leaking back from the high pressure area are mixing.

Figure 9: Calculated temperature inside the pump versus that measured 3 mm inside the pump



4. Comparison of the efficiency gains in the field and pump test

We compiled in Table 11, the gain achieved with oil B over oil A in the field test in terms of fuel consumption and number of work cycles per hour.

Table 11: Percent gain achieved with Oil B over oil A.

	% Gain achieved with Oil B over Oil A		
Throttle	Fuel Consumption per Hour	Work Cycles per Hour	Fuel Consumption per Cycle
Full	13.8	5.80	18.4
90%	8.6	24.30	26.3

It is interesting to note that the gain in fuel consumption per cycle under full throttle is close to that achieved in the pump loop with the oil inlet temperature set at 75 °C. Using data from Table 9, we can estimate that the gain in global efficiency in the pump loop with the oil pump inlet set at 80 °C will be essentially the same as the gain in fuel consumption per cycle under full throttle measured in the field test with an average oil tank temperature of about 80 °C.

5. Conclusions

Results obtained in a medium size excavator operating under mild climatic conditions indicate that significant differences in terms of fuel consumption per hour and per test cycle could be obtained with two hydraulic fluids having widely different NFPA grading (L46-46 and L32-100).

These two fluids showed also significant differences in terms of volumetric efficiency, global efficiency and temperature increase in a hydraulic circuit equipped with a V104C vane pump.

The gain in global efficiency in the pump loop with the oil pump inlet set at 80 °C is essentially the same as the gain in fuel consumption per cycle under full throttle measured in the field test with an average oil tank temperature of about 80 °C. This suggests that the pump loop can be used to estimate the improvement in efficiency in actual service of a candidate hydraulic oil when ccompared to a SAE 10W.

The temperature measurements made in the pump loop are in line with the thermodynamic models discussed in an earlier paper. They confirm that increasing volumetric efficiency results in a lower fluid temperature. If the hydraulic energy is delivered to the system, it does not increase the fluid temperature

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.004.13	gledište učinkovitosti	efficiency viewpoint
.004.15	stvarni učinak	actual efficiency
532.55	gubici energije tlaka i strujanja	pressure and flow energy loss
.001.41	ispitivanje u laboratoriju	testing, laboratory
.001.55	ispitivanje u pogonskim uvjetima	testing in exploatation

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