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INFLUENCE OF VISCOSITY ON THE RATE OF TEMPERATURE INCREASE OF HYDRAULIC FLUIDS

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ABSTRACT

A thermodynamic model of a hydraulic circuit has been developed that describes the influence of viscosity and pressure on the oil temperature in different parts of the circuit. We have compared the predictions of this model with actual fluid temperature measurements in two circuits equipped with a medium and a high pressure vane pump respectively. Maximum efficiency is attained when the viscosity of the hydraulic fluid is optimized for the final equilibrium operating temperature. High viscosity index fluids contribute effectively to the control of oil temperature increase.

INTRODUCTION

Modern mobile hydraulic systems operating under high pressure are typically fitted with noise insulation, impact protection, and the smallest possible oil reservoir. This limits the cooling of the fluid by air circulation. These factors contribute to ever-increasing oil temperatures and reduced pump efficiency.

Increasing temperature lowers the hydraulic oil viscosity. In turn, lowering viscosity and increasing pressure result in a reduction of the volumetric efficiency of both pumps and motors. The lower the volumetric efficiency, the higher the energy needed at the pump shaft to produce a given level of work, resulting in a further increase of the oil temperature. To operate most efficiently, modern hydraulic equipment must use a lubricant with a temperature-viscosity profile that can minimize, a) the rate at which temperature increases, and, b) the equilibrium temperature reached under continuous, high pressure operations.

The influence of oil viscosity on the performance of hydraulic equipment has long been recognized. At low temperature, excessive viscosity may result in starting difficulty, high hydro-mechanical losses, sluggish operations, cavitation and pump wear. At high temperature, insufficient viscosity may result in low volumetric efficiency due to high internal leakage, fluid overheating and increased pump wear.

Work conducted in gear and vane pumps [1,2] enabled us to determine the dependence of volumetric efficiency on speed, pressure and fluid viscosity at the inlet of the pump. These models were used to determine the influence of viscosity on the level of energy and time required for a specific pump to deliver a desired amount of work [3,4]. Decreasing volumetric efficiency increases both the power and the time required to deliver a given energy level to a motor. The energy that is lost as a result of low volumetric efficiency contributes to heating the oil, reducing its viscosity and thus further lowering the efficiency of the hydraulic circuit.

In severe operations, the temperature of the hydraulic fluid in mobile equipment rises, decreasing volumetric efficiency. Too low a volumetric efficiency will impair the proper function of the equipment and might even reduce its life. This situation has been made worse over the years by the use of pumps running at higher pressure coupled with a reduction in reservoir size. Mobile equipment hydraulic fluid reservoirs have decreased in size over the last 30 years from as much as 400% of the pump's nominal flow rate (in one minute) to as little as 25% of the pump's flow rate. Auxiliary cooling systems that could alleviate this problem are both costly and, in many instances, impractical on mobile systems because of the limited space available. The objective of this paper is to determine to what extent, the proper selection of a fluid with an optimal viscosity-temperature relationship could help in maintaining volumetric efficiency at a high level and, consequently, in controlling the rate of temperature increase in order to avoid overheating. For this purpose we have first developed a thermodynamic model of a pump circuit and then compared the predictions of this model to actual results generated in a medium and a high pressure circuit equipped with vane pumps.

DEVELOPMENT OF A THERMODYNAMIC MODEL OF THE CIRCUIT

In the work reported in this paper, we have used two hydraulic circuits that are equipped respectively with a medium and a high pressure vane pump. These circuits will be described in more details later in the paper but can be represented by the following diagram under conditions where no cooling is used to control the temperature of the hydraulic fluid.

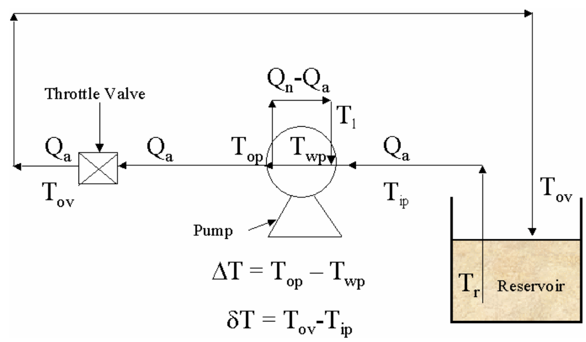

Figure 1: Schematic of the hydraulic circuit.

DEFINITION OF TERMS

Flow Rates

Q_a is the actual flow rate from the pump

Q_n is the nominal flow rate from the pump

V_E is the volumetric efficiency and is equal to Q_a/Q_n

Oil Temperatures

T_{ip} is the temperature at the inlet of the pump

T_l is the temperature of the oil leaking back to the pump inlet

T_{wp} is the oil temperature inside the pump, corresponding to the mixture of the pump inlet flow and the oil leaking back to the pump inlet.

T_{op} is the temperature at the pump outlet

T_{ov} is the temperature at the outlet of the throttle valve

T_r is the temperature of the oil in the reservoir

Temperature differentials

ΔT is the temperature increase in the pump and is equal to $T_{op} - T_{wp}$

δT is the temperature increase in the circuit and is equal to $T_{ov} - T_{ip}$

Oil characteristics

$\rho(T)$ = specific gravity of the oil at temperature T

$C_P(T)$ = heat capacity of the oil at temperature T

KV(T) = kinematic viscosity at temperature T

Power

P_{HT} = Theoretical Hydraulic Power

P_{HM} = Hydromechanical power loss in the pump

Circuit

V = volume of oil in the reservoir

CALCULATION OF THE OIL TEMPERATURE IN THE CIRCUIT-

Calculation of the oil temperature in different points of the circuit is based on the oil flow and power dissipated in the pump and in the throttle valve. For this purpose, we assumed that $\rho * C_P$ was constant within the range of temperature encountered by the oil between the pump inlet and the valve outlet.

Table 1: Calculation of the oil temperature in the circuit

Location	Power lost	Flow rate
Pump (1)	P_{HT} *(1-V_E)	Q_n *(1- V_E)
Pump (2)	P_{HM}	Q_n
Valve	P_{HT} * V_E	Q_n * V_E
Circuit	P_{HT} + P_{HM}	Q_n * V_E
Location	**Temperature increase**	
Pump (1)	$T_l - T_{op} = P_{HT} / (\rho * C_P * Q_n)$	
Pump (2)	$\Delta T = T_{op} - T_{wp} = P_{HM} / (\rho * C_P * Q_n)$	
Valve	$T_{ov} - T_{op} = P_{HT} / (\rho * C_P * Q_n)$	
Circuit	$\delta T = T_{ov} - T_{ip}$ $= (P_{HT} + P_{HM}) / (\rho * C_P * Q_n * V_E)$	

Pump (1) corresponds to the high shear zones where internal leakage (Q_l) takes place

Pump (2) corresponds to the volume comprised between the rotor and the pump housing i.e. to the pump displacement per revolution.

Since $T_l - T_{op} = P_{HT} / (\rho * C_P * Q_n) = T_{ov} - T_{op}$ this means that $T_l = T_{ov}$. The temperature of the oil leaking back to the pump inlet is thus equal to the temperature of the oil flowing out of the valve. This is logical since these oil streams carry the same energy per unit of mass.

Conservation of energy in the pump enables us to determine T_{wp}:

$$Q_n * T_{wp} = Q_n * V_E * T_{ip} + Q_n * (1- V_E) * T_l$$

or:

$$T_{wp} = (1- V_E) * \delta T + T_{ip}$$

In this simplified model we assumed that the oil does not exchange any heat with the rest of the circuit. However, in practice, the heat capacity of the hardware must be taken into consideration. Therefore, the further away we are from the pump inlet the less precise are these equations in estimating the actual change of temperature as a function of time. We estimated that for the medium pressure circuit the total heat capacity of the hardware was over 10 times greater than that of the test oil. Therefore, the oil temperature measured in the reservoir is significantly lower than that calculated using the equations. However, this theoretical approach is still useful to estimate the contribution of the oil viscosity and of the operating pressure to the temperature within the pump and at the outlet of the pump that will control the volumetric efficiency of the pump and of the motor respectively. This is due to the fact that the pump body has only a limited heat capacity compared to that of the total circuit. These equations may thus enable us to estimate the relative contribution of the mechanical and viscous elements of the hydromechanical power loss.

APPLICATION OF THE MODEL TO A MEDIUM PRESSURE VANE PUMP CIRCUIT

DESCRIPTION OF THE MEDIUM PRESSURE HYDRAULIC CIRCUIT AND OF THE TEST PROCEDURE

We used an Eaton Vickers V20 vane pump in this circuit. The pump is driven by a 15 kW electric motor at 1200 rpm. The pump circuit, shown in Figure 2, consists of the following elements:

- an oil reservoir
- two filters
- an Eaton Vickers V20 vane pump
- a pressure regulator (throttle valve)
- low and high pressure filters
- a flow meter
- a heat exchanger

Thermocouples were installed about 150 mm before the inlet of the pump and, immediately after the pressure regulator. They provide estimates of T_{ip} and T_{ov} respectively. In this case $T_r = T_{ip}$.

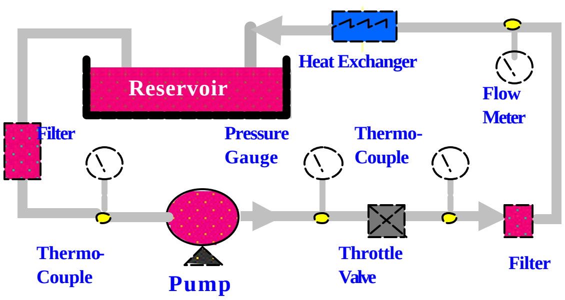

Figure 2: Schematic of the Medium Pressure Pump Circuit

The test procedure used to study the change of temperature in the circuit is as follows:
After thoroughly flushing the circuit and installing a new filter, the reservoir was filled with 18.9 liters (5 US gallons) of test oil. All tests were started with the oil and circuit at room temperature (approximately 20 °C) The discharge pressure was progressively increased to the desired level and maintained until the inlet oil temperature reached about 100 °C. This maximum temperature was selected to avoid excessive wearing of the pump and degradation of the seals that might have modified the nominal flow rate. Flow rate and temperature measurements were made every minute. Each candidate oil was tested at three different pressures 69, 103 and 138 bars (1000, 1500, and 2000 psi).

TEST OILS

Four Newtonian fluids falling in the ISO 22, 32, 46 and 68 grades were evaluated. They all contain 0.8% of a standard hydraulic fluid additive package. Their viscometric characteristics are detailed in Table 2.

Table 2: Viscometric characteristics of the oils tested in the medium pressure circuit.

	ISO Grade			
	22	32	46	68
Kinematic Viscosity @				
40°C, mm²/s	22.3	32.2	46.28	68.35
100°C, mm²/s	4.33	5.32	6.75	8.71
VI	101	95	99	99

DATA ANALYSIS

Analysis of δT - For each of the four test oils we have computed the difference between the temperature at the inlet of the pump and at the outlet of the throttle valve. This provides an estimate of δT. In this circuit we have T_{ip} equal to T_r.
The values of δT obtained for the ISO 22 oils as a function of time for each of the three pressures have been plotted in Figure 3.

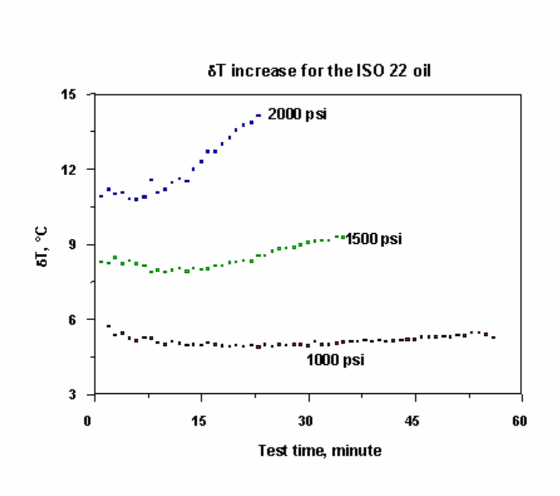

Figure 3: Plot of dT for the ISO 22 oil in the medium pressure circuit as a function of time and pressure

It can be seen that δT first decreases and then increases with time. The rate at which δT rises in this second stage increases with pressure. In order to explain this behavior we have used the equation for δT obtained from Table 1.

$$\delta T = T_{ov} - T_{ip} = (P_{HT} + P_{HM}) / (\rho * C_P * Q_n * V_E) \quad (1)$$

The theoretical hydraulic power is defined by:

$$P_{HT} = 100 * P * Q_n$$
(2)

Where P is the pressure in bars in the circuit before the throttle valve, Q_n is the flow rate in liters per second, and P_{HT} is the theoretical hydraulic power in Watts.

The hydromechanical power, P_{HM}, is the sum of four terms [8]. The first one corresponds to the mechanical friction occurring in the pump and is proportional to pressure. The second term represents the viscous friction of the laminar flow of the fluid layers within the pump. It is proportional to the rotational speed, n, and to the fluid viscosity, η. The third and fourth terms, static friction and viscous friction of the turbulent flow, are usually small compared to the two first components of P_{HM} and have not been considered in our model. Therefore, the equation used for P_{HM} is

$$P_{HM} = \alpha * P + \beta * n * \eta$$
(3)

Where α and β are pump constants. For ease of calculation, we considered the kinematic viscosity in place of the dynamic viscosity.

$$\delta T = (100 * P * Q_n + \alpha * P + \beta * n * \eta) / (\rho * C_P * Q_n * V_E)$$

(4)

According to this equation, δT is composed of three elements. The first one corresponds to the dissipation of the hydraulic power, P_{HT}. The second one comes from the mechanical part of the hydromechanical power losses, P_{HM}. The last one is associated with the viscous component of P_{HM}.

When the pump starts running, the oil temperature increases resulting in a decrease of the contribution of the viscous component of P_{HM}. However, decreasing viscosity also results in a decrease of the volumetric efficiency and since δT is proportional to the reciprocal of V_E, the contribution of P_{HT} and of the mechanical portion of P_{HM} increases with time.
When the oil temperature is high enough, the contribution of the viscous component of the hydromechanical power loss to δT becomes small compared to the two other sources. In this case, according to equation [4], δT should be directly proportional to the ratio $P/(\rho * C_P * V_E)$.

$$\delta T = P/(\rho * C_P * V_E) * (100 + \alpha / Q_n)$$
(5)

To verify this point, we have plotted δT as a function of $P/(\rho * C_P * V_E)$ in Figure 4 and Figure 5, first for all the data points and, second, after eliminating the results collected during the first ten minutes of the test respectively.

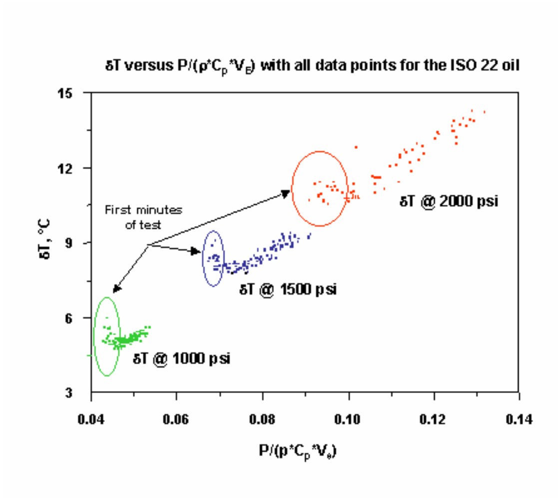

Figure 4: Dependence of d**T on P/(**r *** CP*VE) for the ISO 22 at all times**

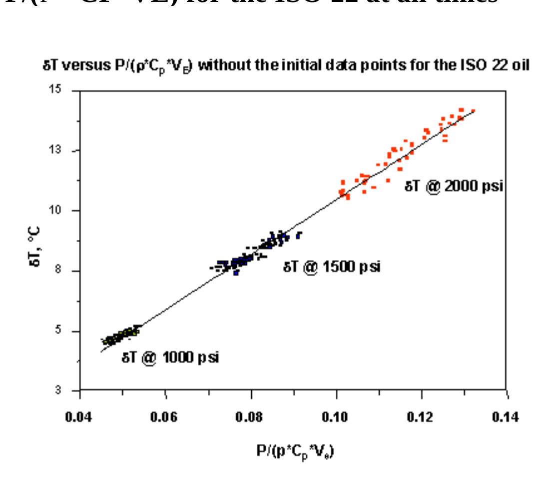

Figure 5: Dependence of d**T on P/(**r *** CP*VE) for the ISO 22 after 10 minutes.**

Regression analysis yielded the following least square equations.

For all the data points collected from the start of the test
$\delta T = -0.1 + 109.28 * P/(\rho * C_P * V_E)$ $R^2 = 0.977$

For the data points collected after 10 minutes of testing
$\delta T = -0.3 + 110.03 * P/(\rho * C_P * V_E)$ $R^2 = 0.995$

The second least square equation, with a higher coefficient of determination, better fits the theoretical model. For the ISO 22, the value of the slope, 110.03, corresponds to the sum of the contribution of the hydraulic power, 100, and of the hydromechanical losses, 10.03.

In order to estimate the coefficients α and β, Equation [4] can be rewritten as:

$$\delta T = (100 + \alpha/Q_n) * X + \beta * Y \qquad (6)$$

with $X = P/(\rho * C_P * V_E)$ and $Y = n * \eta/(\rho * C_P * Q_n * V_E)$

By computing X and Y for each data point, we can obtain by linear regression analysis a value of $100 + \alpha/Q_n$ and of β that are the two coefficients of equation [6] required to calculate the hydromechanical losses in the pump. To obtain X and Y we calculated the value of ρ and C_P at the temperature T_f using equations detailed in Appendix 1.

The kinematic viscosity of the oil at the temperature at the inlet of the pump was calculated using the MacCoull, Walter, Wright (MWW) relationship [5]:

$$LogLog(m) = -m \times Log(T) + b \qquad (7)$$

To compute X and Y, we also needed an estimation of the nominal flow rate of the pump at 1200 rpm. For this purpose, we used flow rate measurements made with the throttle valve fully open that were obtained on three mineral oils at four different temperatures, 20, 50, 65 and 80 °C. This experiment was reported in an earlier paper [6]. We have plotted in Figure 6 the flow rate as a function of the reciprocal of the kinematic viscosity, KV at the pump inlet.

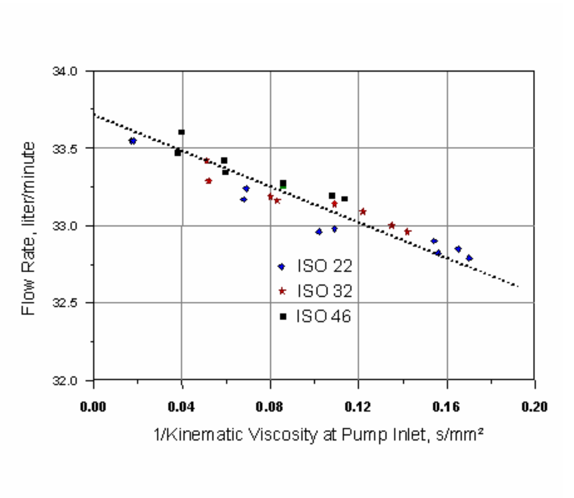

Figure 6: Flow rate of mineral oils with valve fully open

By linear regression analysis, we obtained the following least square equation with a coefficient of determination, R^2, of 0.89.
Flow rate = 33.64 –4.94/KV
The intercept, 33.64 liter per minute, was used as the value of Q_n when computing the values of X and Y.

Multiple linear regression analysis was conducted for the four mineral oils at the three pressures considered. We obtained the following equation in which Q_n is expressed in liter per second:

$\delta T = (100 + 3.554/Q_n)*X + 0.0175*Y \quad R^2 = 0.978$

It should be noted that the values of α and β obtained for the Eaton Vickers V20 vane pump are higher than those of the Denison T6CM vane pump that we used in the high pressure circuit. The coefficients for the T6CM pump that are shown in Table 3 were calculated using information published by the pump manufacturer.

Table 3: Value of a and b for Equation (6)

α, Watt/Bar		β, Watt/(rpm*mm²/s)	
T6CM	V20	T6CM	V20
5.4	3.6	0.0246	0.0175

In order to assess the adequacy of our model, we have plotted in Figure 7 the actual versus the predicted increase in temperature for the ISO 32 oil for the three test pressures. As it can be seen, the calculated values are slightly above the actual values. This is likely due to the fact that some of the heat is dissipated in the pump and in the throttle valve.

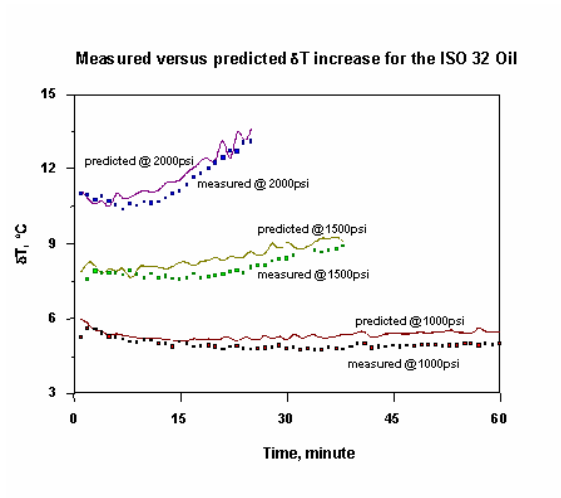

Figure 7: Adequacy of the model for dT

INCREASE OF THE OIL TEMPERATURE AT THE OIL RESERVOIR WITH TIME

The energy dissipated in the circuit results in an increase of temperature in the reservoir, T_r. The major contributor to the increase of T_r is the hydraulic power, P_{HT} that is independent of the oil viscosity. The second and much smaller contributor to the rise of T_r is the hydromechanical power loss, P_{HM}. If all the energy were to remain in the oil, the oil temperature should rise by at least 14 °C per minute at 2000 psi. However, in our experiments, the rate at which the oil temperature rises at the inlet of the pump is much lower. This is due to the fact that a) a significant part of the energy is used to heat the hardware and b) the hardware losses energy to the surroundings.

We have plotted he temperature of the oil at the inlet of the pump as a function of test time at two of the three pressures and for the three oils considered in our experiment in Figure 8.

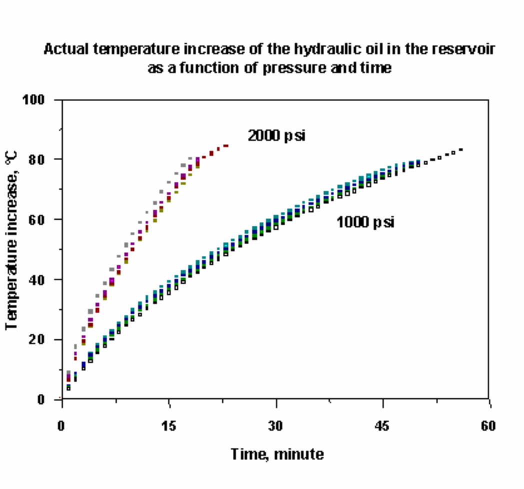

Figure 8: Increase of oil temperature in the reservoir as a function of time and pressure.

The rate at which temperature rises increases with pressure. Furthermore, the rate at which temperature changes decreases with time. This possibly results from the fact that the rate of heat transfer to the surroundings increases when the temperature of the circuit increases. To represent this phenomenon, we used a second order equation of the form:

$$T_r = P*(2.3*10^{-3}+2.29*10^{-3}*t - 1.703*10^{-5}*t^2) \qquad R^2 = 0.9830$$

With t in minute and P in psi.

Since the hydromechanical losses represent only about 10% of the energy transferred to the oil by the electric motor and that only a fraction of these losses are due to the viscous drag that takes place in the pump, estimating the effect of the oil viscosity on T_r is impossible with the data in hand.

APPLICATION OF THE MODEL TO A HIGH PRESSURE VANE PUMP LOOP

DESCRIPTION OF THE HIGH PRESSURE HYDRAULIC CIRCUIT AND OF THE TEST PROCEDURE

In this circuit we used a Denison T6CM vane pump designed for mobile equipment. This pump can operate up to 250 bars on a continuous basis when using a proper lubricant. It is driven by a 15 kW electric motor at 1500 rpm. With the cartridge selected this corresponds to a nominal flow rate of 31.9 liters per minute according to the pump manufacturer data sheet.

The temperature of the test oils at the inlet of the pump was adjusted to about 80 °C using a heat exchanger located between the reservoir and the pump inlet. Another heat exchanger located after the throttle valve was used to cool the oil to about 85 °C. Temperature of the oil was measured at the pump inlet (T_{ip}), at the pump outlet (T_{op}) and after the throttle valve (T_{ov}).

Experiments were conducted at the beginning of the test, just after stabilizing the inlet temperature to 80 °C, and after 3, 6 and 24 hours of operation under the cyclic conditions specified in the test method A-TP 30283 used by the OEM to test the anti-wear performance of lubricants.

TEST OILS

Three oils containing 0.6% of a standard hydraulic fluid additive package were evaluated in this circuit. One falls in the ISO 46 grade and the two others in the ISO 32 grade. Their characteristics are detailed in Table 4. The three test oils have almost the same kinematic viscosity at 80 °C that is the temperature prevailing at the inlet of the pump.

Table 4: Characteristics of the oils tested in the high pressure circuit

Fresh oil

Oil Code	VII	KV40°C mm^2/s	KV80°C mm^2/s	KV100°C mm^2/s	VI
1	None	43.95	10.87	6.68	104
2	9.2% of P1	35.24	11.07	7.20	174
3	5.1% of P2	34.55	10.92	7.18	178
	After KRL 20 hour test				
Oil Code	KV100°C mm^2/s			VI	
1	6.68			104	
2	6.00			150	
3	5.27			132	

Polymer P1 is more shear stable than polymer P2 and shows a lower loss of viscosity after the KRL 20 hour test (CEC L-45-A-99).

TEST PROGRAM

We conducted a limited test program using this high pressure circuit. This initial testing was aimed at optimizing the hardware and the software used to acquire and treat the critical test parameters. Temperature measurements were made at six different back pressures, 0, 50, 100, 150, 210 and 250 bars and four different times, 0, 3, 6 and 24 hours. The ISO 46 oil (Oil 1) was tested in duplicate.

DATA ANALYSIS

In all the tests we conducted, the highest flow rate we obtained was 33.8 l/min. We used this value as an estimate of Q_n. In the following we used ΔP that is equal to the difference between the outlet and the inlet pressure of the pump.

<u>Analysis of T_{ov}- T_{ip}</u>

According to equation [4] discussed earlier, we have:

$$\delta T = (100*\Delta P*Q_n + \alpha*\Delta P + \beta*n*\eta)/(\rho* C_P*Q_n*V_E) \quad (4)$$

This equation can be written as:

$$\delta T = A*\Delta P/V_E + C/V_E \quad (8)$$

with $A = 100/(\rho * C_P) + \alpha/(\rho * C_P *Q_n)$ and $C = \beta*n*\eta/(\rho* C_P *Q_n)$

We conducted a non-linear regression analysis to force the intercept to zero and obtained the results summarized in Table 5.

Table 5: Dependence of T_{ov}- T_{ip} on DP/V_E and $1/V_E$

Oil Code	A	C	R^2
Oil 1	0.0398	1.01	0.9805
Oil 2	0.0440	0.40	0.9931
Oil 3	0.0339	2.11	0.9906
All	0.0395	1.16	0.9879

The high values of the coefficient of determination (R^2) indicate that equation [4] can be used for all the oils to estimate the increase of temperature taking place in the circuit. The low values of coefficient C, that represents the effect of viscous component of P_{HM}, confirm that under the conditions selected, the oil viscosity has only a modest effect on the increase of temperature in the circuit.
We have plotted in Figure 9, the increase of temperature for the three test oils as a function of $\Delta P/V_E$. It can be seen that the rate at which temperature increases with $\Delta P/V_E$ is essentially the same for all the oils.

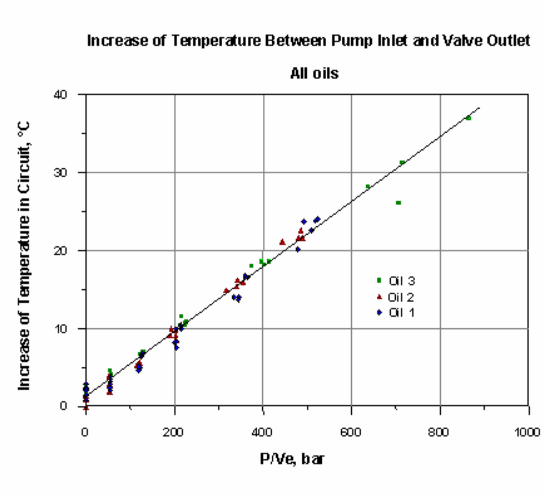

Figure 9: Dependence of the increase of temperature in the circuit on P/VE.

Considering that the contribution of the viscous component of P_{HM} was small, we conducted a linear regression analysis on the data collected for the three test oils yielded the following least square equation.

$\delta T = 1.1 + 0.0421*\Delta P/ V_E$ $\qquad R^2 = 0.9879$

The intercept of 1.1 °C corresponds to the average effect of the viscous component of P_{HM} in our tests.

<u>Analysis of T_{ov}- T_{op}</u>

According to the energy balance described earlier, T_{ov}- T_{op} should be directly proportional to pressure.

$T_{ov}- T_{op} = 100*\Delta P/(\rho *C_P) = B*\Delta P$ (9)

with $B= 100/(\rho *C_P)$

By linear regression analysis we obtained the results shown in Table 6.

Table 6: Adequacy of the model T_{ov}- T_{op} = a + B*DP

Oil Code	a	B	R^2
Oil 1	1.03	0.0282	0.9713
Oil 2	0.98	0.0273	0.9804
Oil 3	1.27	0.0272	0.9703
All	1.15	0.0273	0.9648

Here again, the high values of the coefficient of determination (R^2) indicate that equation (9) can be used for the three test oils to estimate the increase of temperature taking place between the outlet of the pump and the outlet of the valve. Furthermore, this analysis shows that the rate at which temperature increases with pressure between the outlet of the pump and the outlet of the valve is essentially the same for all the oils.

We have plotted in Figure 10 the increase of temperature between the outlet of the pump and the outlet of the valve as a function of ΔP. It can be seen that this graph confirms that this increase of temperature is proportional to ΔP.

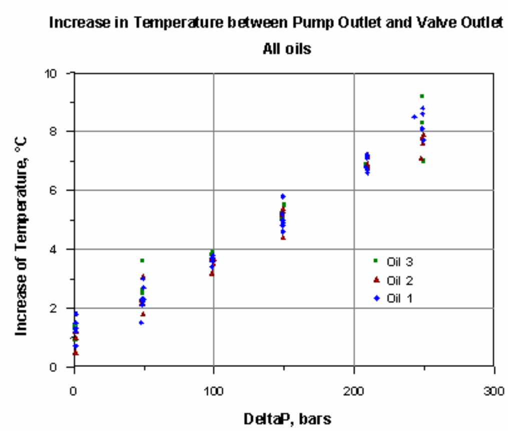

Figure 10: Increase of temperature between the outlet of the pump and the outlet of the valve as a function of D P

<u>Analysis of T_{op}- T_{ip}</u>

Considering that the energy lost in the pump is equal to $P_{HT}*(1-V_E)+P_{HM}$ with a flow rate of Q_n*V_E, we can determine that:

$T_{op}- T_{ip} = (P_{HT}*(1-V_E)+P_{HM})/(\rho * C_P* Q_n*V_E)$

(10)

Replacing P_{HT} and P_{HM} as per equations [2] and [3] yields:

$T_{op}- T_{ip} = A*\Delta P/V_E - B*\Delta P + C/V_E$ (11)

With A = 100/(ρ*C_P) + α/(ρ* C_P *Q_n),
B = 100/(ρ* C_P) and C = β*n*η/(ρ* C_P * Q_n)

The value of A, B and C can be obtained either by combining the models obtained for T_{ov} - T_{op} and T_{ov} - T_{ip} or by non-linear regression analysis of the results. This latter yielded the results summarized in Table 7.

Table 7: Adequacy of the model
T_{op} - T_{ip}= A*DP/V_E - B*DP + C/V_E

Oil Code	A	B	C	R^2
Oil 1	0.0511	0.0483	0.38	0.9815
Oil 2	0.0487	0.0339	0.38	0.9941
Oil 3	0.0380	0.0236	0.53	0.9957
All	0.0424	0.0289	0.08	0.9648

Use of the models

The coefficients A, B and C obtained by regression analysis of the increase of temperature between a) the pump outlet and the pump inlet, b) the valve outlet and the pump outlet and, c) in the circuit cannot be used to calculate the value of α and β. This results from the fact that the energy dissipated in the circuit is used to heat both the oil and the hardware. Furthermore, because of the high temperature at which we run our tests, the viscous part of P_{HM} is small and high heat losses to the ambient are taking place. Therefore, we would need to use different values of the ρ*Cp characterizing the heat capacity of the oil and of the hardware in each of the three models considered.

The interest of theses models comes from the fact that we need only a limited number of measurements to determine the values of the coefficients of the three equations and, therefore, to characterize the dependence of the temperature in this circuit on pressure and volumetric efficiency.

We can also use the thermodynamic model of the circuit to estimate the temperature of the oil within the pump.

$$T_{wp} = (1-V_E)* \delta T + T_{ip} \qquad (12)$$

Replacing δT by its value as per equation (4):

$$T_{wp} = (1-V_E)*(100*\Delta P*Q_n + \alpha*\Delta P + \beta*n*\eta) / (\rho* C_P*Q_n*V_E)+ T_{ip}$$

(13)

This may be simplified using the parameters A and C that were introduced previously:

$$\delta T = A*\Delta P/V_E + C/V_E \qquad (8)$$

We obtain:

$$T_{wp} = (1-V_E)*(A*\Delta P/V_E + C/V_E) + T_{ip} \qquad (14)$$

Using the values A and C obtained for the 3 test oils, we have computed T_{wp}- T_{ip} as a function of pressure and volumetric efficiency. The results of this exercise are plotted in Figure 11.

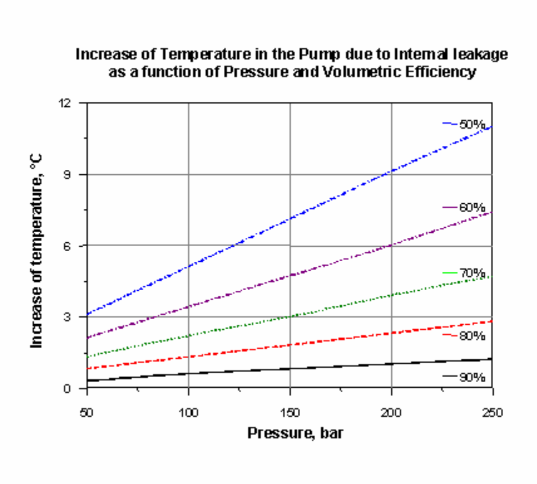

Figure 11: Increase of temperature in the pump as a function of pressure and volumetric efficiency

CONCLUSIONS

Flow rate and temperature measurements were completed in a medium and in a high pressure hydraulic circuit equipped with vane pumps from two different manufacturers. These data were compared to predictions derived from a simplified thermodynamic model of the hydraulic circuit in which we considered that all the heat corresponding to the dissipation of the hydraulic power, P_{HT}, and hydromechanical power loss, P_{HM}, remained in the oil.

MEDIUM PRESSURE CIRCUIT

Using the model for the increase of temperature taking place in the pump and in the throttle valve, we have been able to estimate the contribution of the mechanical and of the viscous part of P_{HM} and to calculate their relative effect on the increase of the temperature of the oil.

The value of α and β of the equation representing P_{HM} determined by analysis of the experimental data in the medium pressure circuit are close to, but still higher than, those determined for the high pressure vane pump using data published by the pump manufacturer.

Using the value of α and β computed in our work and the oil temperature in the reservoir, we can predict the actual temperature increase between the pump inlet and the throttle valve outlet for four mineral oils with an ISO viscosity grade ranging from 22 to 68.

Our results confirmed that after a few minutes of operation, the increase of temperature between the pump inlet and the throttle valve outlet is proportional to pressure and inversely proportional to the volumetric efficiency.
The actual increase of temperature of the oil in the reservoir is much smaller than that calculated with our model because the oil exchanges heat with the hardware. The heat capacity of the hardware is several times higher than that of the oil. The rate of temperature increase in the reservoir is proportional to pressure and decreases with time.

HIGH PRESSURE CIRCUIT

We could not estimate precisely the parameters α and β of P_{HM} because under the conditions selected a) the contribution of the viscous component of P_{HM} is small, about 1 °C, and b) the oil and hardware lose heat to the surroundings.

We confirmed that $T_{ov} - T_{op}$ is proportional to ΔP and that, if we neglect the viscous component of P_{HM}, $T_{ov} - T_{ip}$ is proportional to $\Delta P/V_E$..

The oil temperature within the pump is proportional to the reciprocal of V_E and can be over 10 °C higher than the temperature at the pump inlet under high pressure (250 bars) and low volumetric efficiency (50%).

ALL CIRCUITS

Results obtained in the medium and high pressure circuits confirmed that the temperature of the oil depends on the operating pressure and on the volumetric efficiency. The higher the volumetric efficiency, the less heat is lost in the oil and the lower the increase of oil temperature. One option for limiting the rate at which the oil temperature increases is to select a lubricant that provides the highest volumetric efficiency over the range of pressure and temperature encountered by the circuit. Hydraulic fluids with high viscosity index are well suited for this purpose.

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APPENDIX 1

Determination of $\rho(T)$ in Kg per liter – as per reference [7]:

$\rho(T) = \rho(15)-6.61*10^{-4}*(T-15)$

with $\rho(15)$ =KV(100°C)/(1.1032*KV(100°C + 0.2338)

Determination of $C_p(T)$ in Joule per gram per °C – as per reference [7]

$C_p(T) = 4.1868*(0.415/\rho(15)0.5+9*10^{-4}*(T-15))$