Efficiency Advantages in Vane, Piston and Gear Pumps – What High VI Hydraulic Fluids Can Do for You!

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ABSTRACT

Increasing demands for reduced hardware costs and improved productivity lead to the design of mobile hydraulic systems which run at higher pressures with smaller pumps, oil reservoirs, and coolers. Such designs tend to operate at higher temperatures and place great stress on the hydraulic fluid. In order to maintain adequate pump efficiency and wear-protection, the proper selection of fluid viscosity grade has become a critical design element.

We have studied the influence of hydraulic fluid viscosity on pump efficiency for vane, gear and piston pumps, and will highlight the impact of viscosity index and shear stability. The analysis of the data demonstrates that a properly selected hydraulic fluid significantly improves overall efficiency: The selection of an appropriate high-VI, highly shear stable oil, can improve pump efficiency by at least five percent compared to a conventional HM (monograde) fluid of the same ISO grade.

Finally, our results allowed us to identify criteria that define a 'Maximum Efficiency Hydraulic Fluid', enabling an equipment operator to easily improve the performance of the system and reduce fuel consumption.

NOMENCLATURE

KV[*XX*]	kinematic viscosity at [*XX*] °C	mm^2/s
DV	dynamic viscosity	mPa·s
Q	flow rate	L/min
D	stroke displacement	cm^3
N	rotational speed	rpm
η	efficiency	%
ΔP	pressure drop across a pump	bar
M	torque	Nm
P	power	kW

1 INTRODUCTION

All industries face a steadily increasing demand for improved profitability and efficiency. In the field of hydraulics, this has led to the design of mobile hydraulic systems which run at higher pressure with smaller pumps and oil reservoirs. Today, standard systems operate at 300 bars, and next generation designs are expected to reach 500 bars. Increasing pressures and reduced fluid volumes result in higher fluid operating temperatures: 80℃ is quite common for mobile equipment, with peak temperatures of 100℃ or higher.

Under these severe conditions, the proper selection of the hydraulic fluid is critical to insure efficient equipment operation and adequate wear-protection. Viscosity is one of the most important selection criteria of a hydraulic fluid since it determines the efficiency of hydraulic pumps and motors. At high temperature, low fluid viscosity may result in excessive overheating and wear that reduces equipment life. Furthermore, insufficient viscosity will increase internal pump leakage, consequently reducing volumetric efficiency and overall productivity.

When considering the environmental aspects of pump efficiency, it has now become even more important to reduce fuel consumption and emissions in order to meet ‘Kyoto Protocol’ targets.

This work will review the influence of hydraulic fluid viscometrics on the pump efficiency of vane, piston and gear pumps. Recommendations on optimum fluid properties that provide improved productivity and fuel efficiency will be offered.

2 ROLE OF VISCOSITY AND OPERATING CONDITIONS ON PUMP EFFICIENCY

In order to estimate the efficiency performance of any hydraulic fluid, it is important to first define the different elements that contribute to the overall efficiency of a hydraulic pump.

2.1 Volumetric efficiency

The nominal flow rate of the pump, Q_n, is equal to the volumetric displacement of the pump, D, multiplied by the rotational speed of the pump, N. The actual flow rate of the pump, Q_a, is never equal to the nominal flow rate since all pumps have some level of internal leakage, Q_l. The actual flow rate, Q_a, of a pump is defined as the nominal flow rate, Q_n, less the leakage flow rate, Q_l. The volumetric efficiency, η_V, of a pump is defined as the actual flow rate divided by the nominal flow rate. The different relations which can be used to express the volumetric efficiency are summarized in Equation 1:

$$\eta_V = \frac{Q_a}{Q_n} = \frac{Q_a}{D \cdot N \cdot 10^{-3}} = \frac{Q_n - Q_l}{Q_n} = \frac{Q_a}{Q_a + Q_l} \qquad (1)$$

Due to the construction of vane (or gear) pumps, internal leakage will take place in the clearance between the vanes (or the gears) and the pump housing from the high pressure to the low pressure chambers. For such pumps, knowing the volumetric displacement and measuring the actual flow rate and rotational speed of the pump allow one to estimate the volumetric efficiency.

In an axial piston pump, the leakage occurs in three different places: in the area between the pump body and the base of the rotating piston/cylinder block, the area between the piston and the cylinder, and the area between the piston shoe and the swivel plate. The resulting leakage flow returning to the fluid reservoir via the case drain line is called external leakage, and can be directly measured. In a case of variable displacement pump like an axial piston pump, as the stroke displacement changes during operation, the volumetric efficiency could be estimated by measuring the actual flow rate and the leakage flow rate.

In literature, a model is often given to describe the leakage in a vane or a gear pump. This model indicates that the leakage is proportional to the pressure rise and inversely proportional to the viscosity in the pump. By consequence, the volumetric efficiency of a pump decreases with increasing operating pressure, and increases with increasing viscosity.

In previous papers /Her02/ and /Her02A/, we have verified the validity of this model in two vane pumps and one gear pump using monograde mineral oil fluids. The model accurately predicts pump volumetric efficiency by considering the kinematic viscosity of the fluid at the pump inlet and pressure measured at the pump outlet. Very good correlation has been obtained; the results are summarized in the ***Table 1***.

Type / Pump	**Volumetric Efficiency in %**	**R^2**
Vane Pump / Eaton Vickers V-104	$\eta_v = 100 \cdot (1 - 0{,}0173 \cdot \Delta P/KV)$	0,96
Vane Pump / Eaton Vickers V-20	$\eta_v = 100 \cdot (1 - 0{,}0138 \cdot \Delta P/KV)$	0,96
Gear Pump / Bosch	$\eta_v = 100 \cdot (1 - 0{,}027 \cdot \Delta P/DV)$	0,99

Table 1: Verifications from previous studies of the model for the volumetric efficiency

Similar models can be used to describe the leakage of a piston pump, where volumetric efficiency is proportional to the ratio of pressure and operating viscosity.

2.2 Hydro-mechanical efficiency

A hydraulic pump always requires more input torque than expected in theory to generate rotational speed. The hydro-mechanical efficiency, η_{HM}, is an expression which considers the extra torque needed by the pump to overcome friction and viscous drag. This efficiency is the ratio of the theoretical input torque to the pump, M_t, by the actual input torque, M_a. The hydro-mechanical efficiency also represents the excess of actual mechanical power supplied to the pump compared to theory. It is equal to the ratio of the theoretical mechanical power, P_{tM}, and the actual mechanical power provided to the pump, P_{aM}. (Equation 2).

$$\eta_{HM} = \frac{M_t}{M_a} = \frac{P_{tM}}{P_{aM}} \tag{2}$$

The theoretical mechanical power is equal to the product of the nominal flow rate, Q_n, and the pressure drop across the pump, ΔP (Equation 3).

$$P_{tM} = \frac{\Delta P \cdot Q_n}{600} = \frac{\Delta P \cdot D \cdot N \cdot 10^{-3}}{600} \tag{3}$$

The actual mechanical power delivered to the pump is equal to sum of the theoretical mechanical power and of hydro-mechanical power provided to overcome the friction in the pump, P_{HM}. (Equation 4).

$$P_{aM} = P_{tM} + P_{HM} \tag{4}$$

The frictional resistance occurring in a pump is the sum of four terms: mechanical friction occurring in the pump (proportional to the pressure across the pump), viscous friction of the laminar flow of the fluid layers within the pump (proportional to the viscosity and the speed of the moving parts), viscous friction of the turbulent flow of the fluid in the pump (proportional to the square of the flow), and the static friction inside the pump (constant).

Based on these terms, it follows that the hydro-mechanical efficiency of a pump increases with the operating pressure at constant rotational speed and viscosity, decreases with the rotational speed at constant operating pressure and viscosity, and decreases with the viscosity at constant operating pressure and rotational speed.

2.3 Overall efficiency

When comparing the volumetric and the frictional losses occurring in a pump, or the influence of two different hydraulics fluids working in the pump, the overall efficiency , $\eta_{Overall}$, is the best parameter to consider. Overall efficiency is equal to the product of the volumetric efficiency and the hydro-mechanical efficiency. The overall efficiency is also equal to the ratio of the hydraulic power produced by the pump, P_{aH}, by the actual mechanical power provided to the pump, P_{aM} (Equation 5). The hydraulic power produced by the pump, P_{aH}, is equal to the product of the actual flow rate of the pump and the pressure drop across the pump (Equation 6).

$$P_{aH} = \frac{\Delta P \cdot Q_a}{600} \tag{5}$$

$$\eta_{Overall} = \eta_V \cdot \eta_{HM} = \frac{Q_a}{Q_n} \cdot \frac{P_{aH}}{P_{aM}} = \frac{\Delta P \cdot Q_a}{600} \cdot \frac{600}{\Delta P \cdot Q_n} \cdot \frac{P_{aH}}{P_{aM}} = \frac{P_{aH}}{P_{tM}} \cdot \frac{P_{tM}}{P_{aM}} = \frac{P_{aH}}{P_{aM}} \tag{6}$$

In simple terms, the overall efficiency of a pump is a measure of its ability to transform mechanical energy into hydraulic energy. The critical factors which influence overall efficiency are pressure drop across the pump and the viscosity of the fluid in service. In ***Figure 1*** /Tes00/, the influence of fluid viscosity on the overall efficiency at constant pressure has been described.

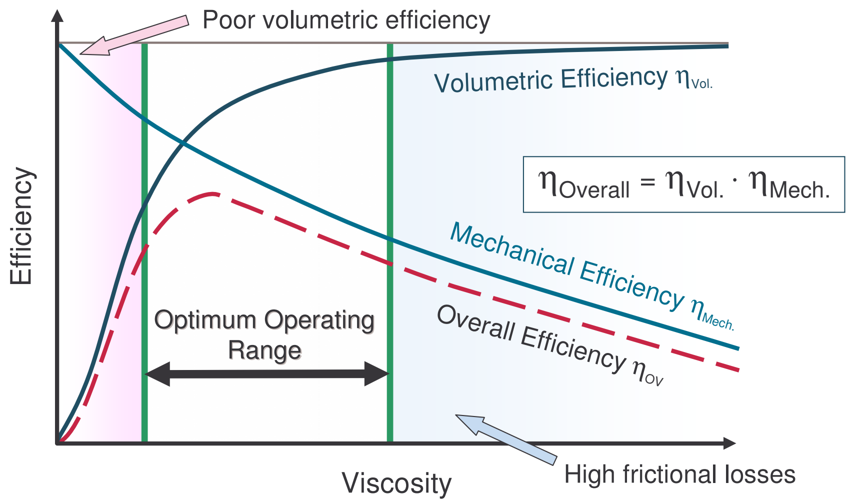

Figure 1: Effect of viscosity on overall efficiency

In the low viscosity regime, the volumetric efficiency term dominates overall efficiency, whereas in the area of the high viscosity, the mechanical efficiency term is most critical /Pla03/. Overall efficiency is maximized when both terms are balanced.

2.4 Kinematic viscosity and "in-service" viscosity

The volumetric efficiency of a monograde HM fluid can be estimated based on its kinematic viscosity at operating temperature, which is the same as its "in-service" viscosity. Previous

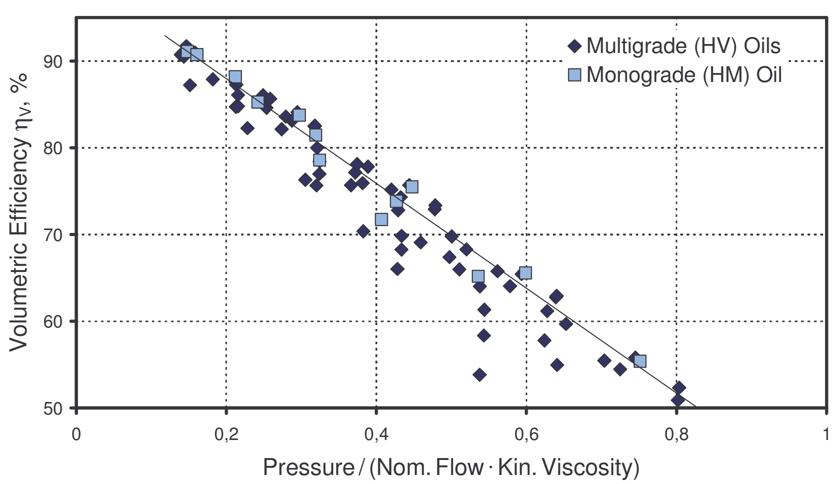

Figure 2: Volumetric efficiency η_V in a vane pump as a function of the pressure drop across the pump, the nominal flowrate and kinematic viscosity

work /Gör06/ has demonstrated that pump efficiency models for high VI fluids formulated with viscosity index improver additives require the use of high temperature high shear viscosity, or after shear kinematic viscosity measurements. In this study, the volumetric efficiency of a vane pump using monograde HM mineral oil fluids are compared to multigrade fluids containing Viscosity Index Improver as a function of pressure and kinematic viscosity at the pump inlet temperature (***Figure 2***).

This figure clearly shows that the kinematic viscosity of monograde HM fluid is sufficient to estimate the volumetric efficiency of the pump. The "in-service" viscosity of Newtonian monograde HM fluids does not vary with the shearing conditions in the pump. However, the kinematic viscosity of fluids containing viscosity index improvers must be measured after shear in order to get a good estimate of "in-service" viscosity" of the fluid in the pump. The same document /Gör06/ also shows that volumetric efficiency can be accurately predicted for high VI fluids based on viscosity measured under high shear conditions, or kinematic viscosity following a standard laboratory shear stability test. ***Figure 3*** shows the excellent correlation:

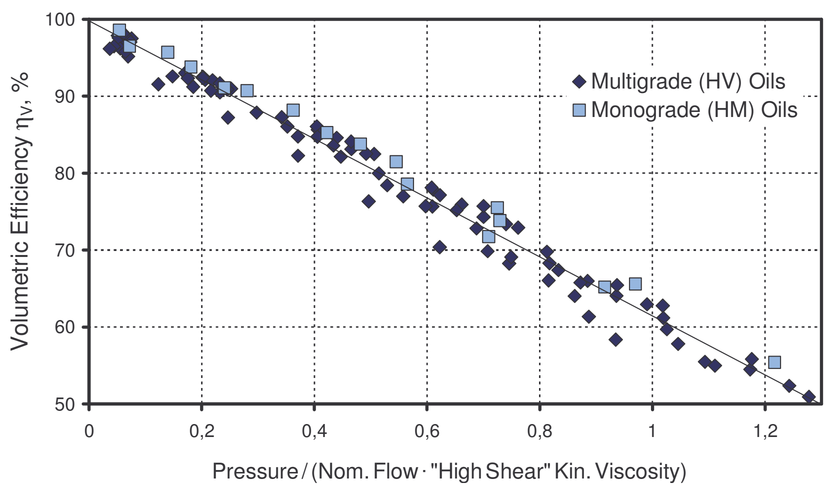

Figure 3: Volumetric efficiency η_V in a vane pump as a function of the pressure drop across the pump, the nominal flowrate and "high shear" kinematic viscosity

3 PUMPING STUDIES CONDUCTED IN PUMPS WITH BOTH CONVENTIONAL AND HIGH VI FLUIDS

In order to compare the performance of high viscosity index fluids (HV) and monograde (HM) fluids we have conducted a large number of tests in vane, gear, and piston pumps. For this purpose, hydraulic fluids with different ISO viscosity grades, VI and shear stability have been tested in hydraulic circuits.

3.1 Experiments run in a gear pump

3.1.1 Gear pump model and observations

The earliest and most fundamental work /Her02A/ was conducted in a pump loop with a Bosch gear pump at 1500 rpm and fluids at 100°C. Pump flowrate as a function of pressure and viscosity can be modeled to predict volumetric efficiency at any desired operating condition. ***Figure 4*** shows the good correlation:

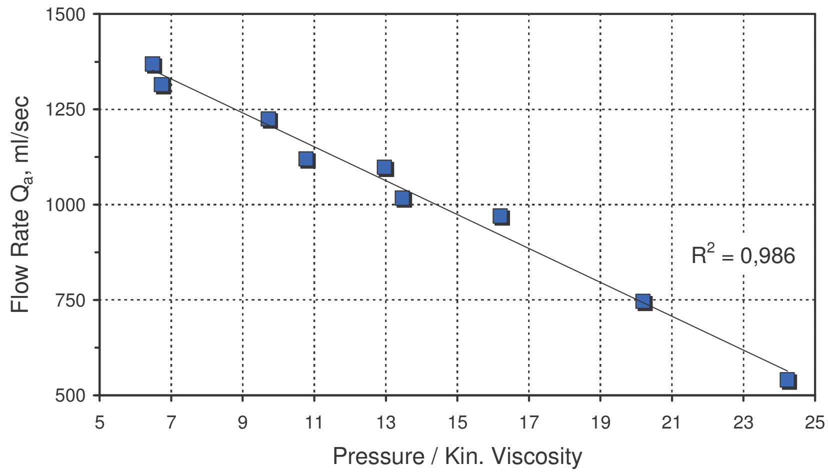

Figure 4: Pump flow rate Q_a as a function of pressure and "in-service" viscosity (mPa·s)

This simple example demonstrates that a shear stable ISO 46 fluid with VI = 150 would have a 7.6% volumetric efficiency advantage over a monograde HM ISO 46 oil with VI = 100 (comparison made @150 bar, 80°C). The volumetric efficiency equation from ***Table 1*** can be applied to the data in ***Figure 4***.

3.2 Experiments run in a high pressure vane pump

3.2.1 Experimental conditions and fluids studied

In this hydraulic circuit, we used a Parker-Denison T6CM vane pump designed for mobile equipment. This pump can operate at 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, the vane pump had a nominal flow rate of 31.9 liters per minute according to the pump manufacturer's data sheet. The other elements of the circuit were a reservoir, two filters (high and low pressure), a flow meter, a pressure valve, a heat exchanger and different sensors to measure the operating conditions (temperatures, pressures, torque, rotational speed).

The tests were conducted according to the following procedure: the vane pump was run until a pump inlet temperature of 80°C, then the heat exchanger was activated in order to maintain this temperature and the performance of the pump was measured at six different steps of pressure during five minutes (15, 50,100, 150, 200 and 250 bars).

Seven hydraulic fluids falling in the ISO 46 viscosity grade have been tested. One of them was monograde HM oil, the other test fluids were formulated using three different PAMA VI Improvers, having either a VI of 150 or 200 with different levels of shear stability. The shear test used to estimate the shear stability performance of the fluids was the 40 minute sonic test (ASTM D 5621). The viscometric properties of the fluids are summarized in ***Table 2***.

			Fresh Oil			After 40min sonic test (ASTM D 5621)	
		Polymer	**KV40**	**KV100**	**VI**	**KV100**	**%KV100 losses**
ISO 46	**VI 100**	-	42,65	6,57	105	-	-
ISO 46	**VI 150**	A	45,92	8,05	153	7,52	**7,8**
ISO 46	**VI 150**	B	45,49	8,14	153	7,24	**11,0**
ISO 46	**VI 150**	C	44,07	7,94	153	6,79	**14,6**
ISO 46	**VI 200**	A	46,18	9,57	198	8,10	**15,4**
ISO 46	**VI 200**	B	45,36	9,63	205	7,29	**22,1**
ISO 46	**VI 200**	C	45,29	9,86	212	6,76	**31,4**

Table 2: Hydraulic fluids tested in the Parker Denison T6CM vane pump

3.2.2 Results obtained for the different fluids

The overall efficiency obtained for the different tested fluids at a pump inlet temperature of 80℃ and at pressures of 150, 200 and 250 bars has been plotted in the ***Figure 5***.

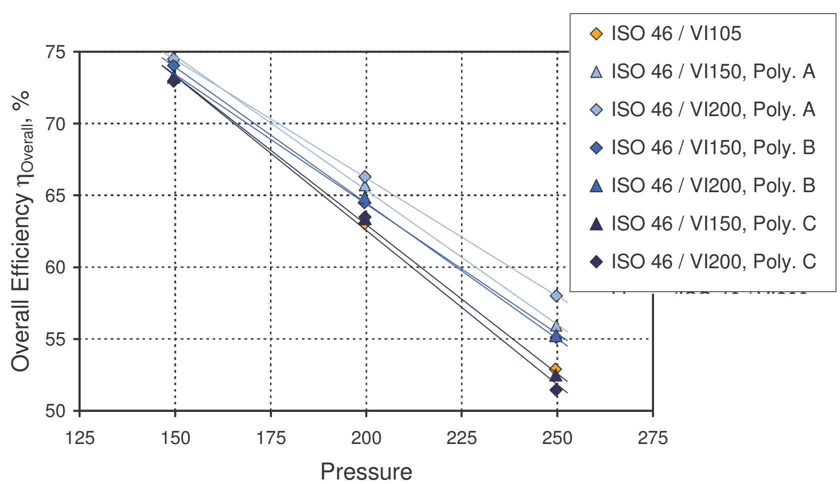

Figure 5: Overall efficiency measured in the Parker Denison T6CM vane pump as a function of pressure

From data in ***Table 2***, it can be seen that although all the high VI fluids containing polymers have approximately the same VI and similar kinematic viscosity at 80℃, they do not provide the same overall efficiency. Furthermore, the two oils containing the shear-unstable polymer

C blended to a VI of 150 and 200 performs more or less like the monograde HM fluid having a VI of 105 .

This difference in performance can be explained by the shear stability of each fluid (illustrated in our studies by the percentage of kinematic viscosity losses at 100℃ given in the ***Table 2***) as discussed in section 2.4. In order to illustrate the advantage of shear stable high VI fluids more clearly, the percentage of overall efficiency improvement for each high VI fluid has been plotted in the ***Figure 6***.

It can be observed that when the shear stability decreases, the percentage of improvement also decreases and that for a certain level of shear stability no gain could be achieved (fluids containing the polymer C). It is also important to note that for oils containing the polymer A and B, the higher the operating pressure, the higher the gain in overall efficiency.

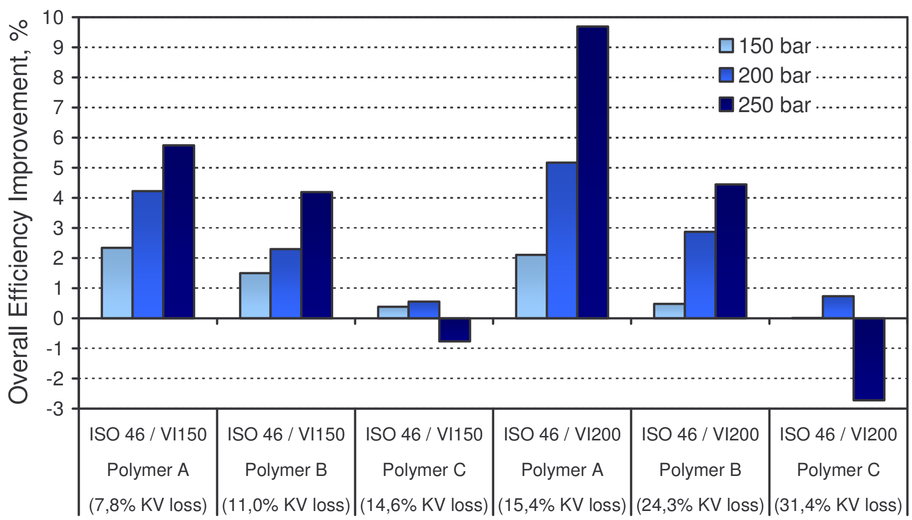

Figure 6: Percentage of overall efficiency improvement compared to the monograde fluid at a pump inlet temperature of 80℃

This figure also shows that the level of efficiency gain obtained with a given polymer does not automatically increase with the VI. For example, the polymer B offers more or less the same gain (about 4%) at a pressure of 250 bars for the fluids having a VI of 150 and 200, whereas the gain increases with the VI for the polymer A (from 6% to 10%).

3.2.3 Comparison of the efficiency gain with after sonic shear kinematic viscosity

When modeling and estimating the possible improvement brought by using a high VI fluid, it is best to consider the kinematic viscosity of the fluid after a shear test, as shown in ***Figure 7***. In this figure, the percentage of overall efficiency improvement measured at 250 bars and 80℃ has been plotted as a function of the kinematic viscosity at 100℃ after sonic shear for all the fluids containing a viscosity index improver.

It can be observed in this graph that the percentage of improvement in overall efficiency increases in a linear manner with the after shear kinematic viscosity at 100°C. Furthermore, it should be noted that using an oil having an after shear viscosity higher than 7.5 mm²/s (such as the ISO 46 fluid containing the polymer A and having a VI of 150) will provide an improvement of overall efficiency (compared to an ISO 46 monograde fluid) higher than 5 percent. This 5 percent improvement in pump efficiency would be realized as a 5 percent improvement in energy consumption compared to an ISO 46 monograde fluid for an equal amount of hydraulic work at 80°C and 250 bars.

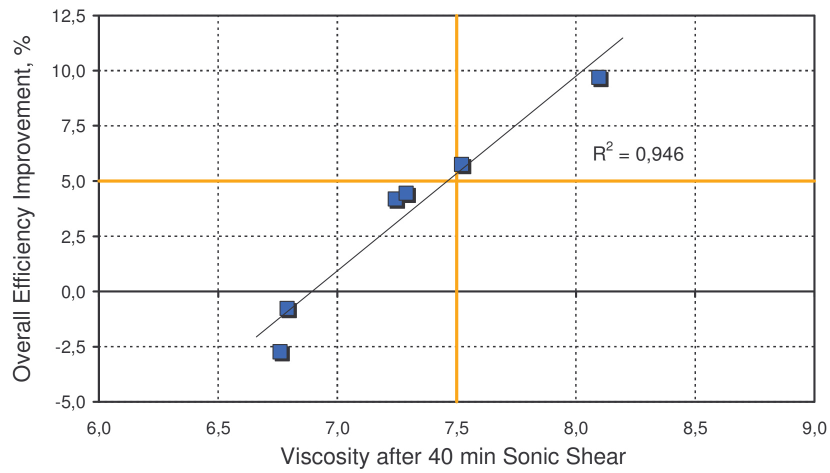

Figure 7: Percentage of overall efficiency improvement at 250 bars and 80°C vs. viscosity at 100°C after 40 minutes of sonic shearing for all high VI fluids

3.3 Experiments run in a piston pump

3.3.1 Experimental conditions and fluids studied

The piston pump selected for this study was a Komatsu axial variable dual piston pump HPV35+35. Only one of the two pumps was used for our experiment, the other one was just run idle at no pressure (i.e. minimum pressure). This pump can operate up to 350 bars on a continuous basis when using a proper lubricant. It is driven by a 22 kW electric motor at 1700 rpm. The other elements of the circuit were a reservoir, two filters (high and low pressure), two flow meters (to measure the actual flow and the leakage), a pressure valve, a heat exchanger and different sensors to measure the operating conditions (temperatures, pressures, torque, rotational speed).

The tests were conducted according to the following procedure: the piston pump was run until a pump inlet temperature of 100°C was reached, then the heat exchanger was activated in order to maintain this temperature and the performance of the pump was measured at six different steps of pressure (30, 70,140, 210, 280 and 350 bars).

Seven hydraulic fluids with ISO 46 viscosity grade viscometrics have been tested (***Table 3***). One of them was a monograde HM fluid, the other test fluids were formulated using six different PAMA VI Improvers. Four of these fluids (polymers A, B, D and E) were formulated in order to have a VI of 120, 140, 160 and 200, respectively, and a percentage of after sonic shear kinematic viscosity losses at 100℃ lower than 10%. The other fluids were formulated to have a VI of 160 and different levels of shear stability.

			Fresh Oil			After 40min sonic test (ASTM D 5621)	
		Polymer	KV40	KV100	VI	KV100	%KV100 losses
ISO 46	**VI 100**	-	44,98	6,81	105	-	-
ISO 46	**VI 120**	D	44,69	7,20	122	6,87	**4,7**
ISO 46	**VI 140**	B	46,08	7,89	141	7,18	**8,9**
ISO 46	**VI 160**	A	47,75	8,57	158	7,79	**9,0**
ISO 46	**VI 200**	E	46,01	9,66	202	9,01	**6,7**
ISO 46	**VI 160**	F	45,99	8,40	160	7,85	**6,5**
ISO 46	**VI 160**	G	45,16	8,33	162	7,13	**14,5**

Table 3: Hydraulic fluid tested in the Komatsu HPV35+35 dual piston pump

3.3.2 Dependence of efficiency on the viscosity index

Although only one pump was delivering work, the torque measured corresponds to the total torque consumed by both the operating and the idle pump (the un-pressurized pump was operating with a constant mechanical power). Only the flow of the tested pump was measured, therefore, the “overall efficiency” measured here is the ratio of the hydraulic power provided by the tested pump divided by the mechanical work delivered by the electric engine to both pumps.

In ***Figure 8*** the overall efficiency at an operating pressure of 350 bars and a pump inlet

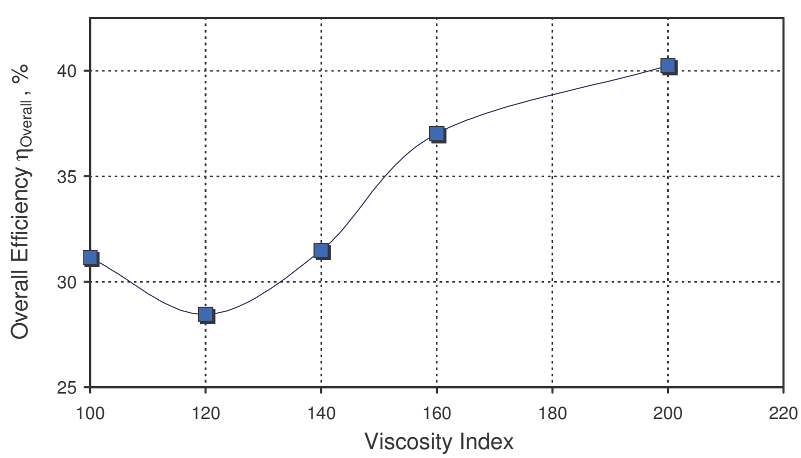

Figure 8: Overall efficiency of the tested ISO 46 viscosity grade fluids vs. VI

temperature of 100℃ has been plotted as a function of the VI for the monograde fluid and the four fluids having a VI of 120, 140, 160 and 200, which contain the polymers D, B, A and E, respectively. As these last four fluids have very good shear stability (their percentage of kinematic viscosity losses at 100℃ after sonic shear were lower than 10%), the influence of the VI on the overall efficiency can be related in this graph.

It can be observed that the use of a fluid having a VI of 120 results in a lower efficiency than the monograde fluid and a fluid with a VI of 140 offered equivalent performance. This behavior can be explained, and again is due to "in-service" viscosity as discussed in section 2.4. The fluid with VI = 120 was formulated with a moderately shear stable polymer, and will suffer in service from viscosity shear losses. The "in-service" viscosity is lower than its fresh kinematic viscosity, and it therefore does not perform as well as the monograde HM fluid. In the same way, the relatively shear stable fluid of VI 140 has an "in-service" viscosity at 100℃ which is approximately equal to the kinematic viscosity of the monograde fluid. Finally, it can be concluded that to a have a significant improvement in overall efficiency in severe conditions (350 bars and 100℃), a minimum VI of 150 should be used.

3.3.3 Results obtained for fluids having a constant viscosity index of 160 and different shear stability levels

The results obtained during the test of the three ISO 46 fluids having a VI of 160, are presented by plotting the percentage of overall efficiency improvement compared to the monograde fluid. The results obtained at 210, 280 and 350 bar at a pump inlet temperature of 100℃ are presented in ***Figure 9***.

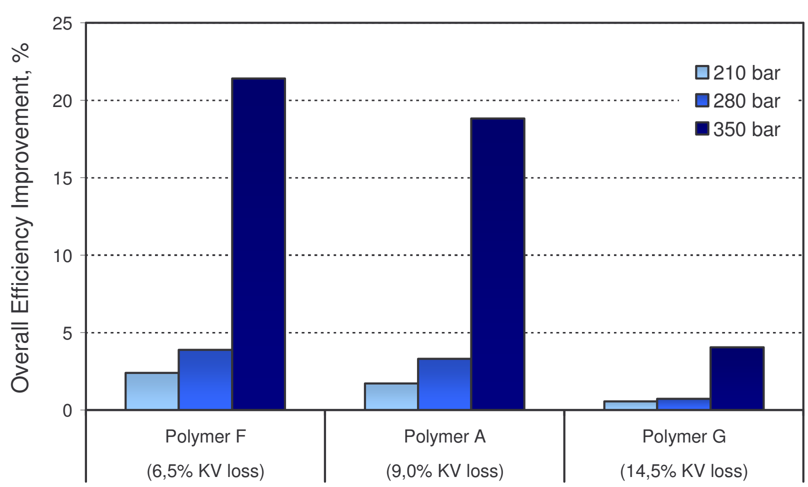

Figure 9: Overall efficiency improvement (in %) compared to the monograde fluid for the ISO viscosity grade 46 fluids having a VI of 160 at pump inlet temperature of 100℃

It should be noted that the level of improvement increases with the operating pressure in this piston pump, similar to the vane pump. The same conclusion regarding the influence of the shear stability can be drawn: When the shear stability increases the percentage of improvement also increases. As a consequence, a high VI fluid formulated with shear unstable polymers (such as polymer G) will not provide pump efficiency benefits.

3.3.4 Comparison of the efficiency gain with after sonic shear kinematic viscosity

The after shear kinematic viscosity of a high VI fluid gives the best indication of piston pump performance, similar to what has been observed for gear and vane pumps. The percentage of improvement in overall efficiency increases with higher levels of kinematic viscosity at 100℃ after sonic shear. In the piston pump, using an ISO 46 viscosity grade fluid having an after sonic shear viscosity higher than 7.5 mm^2/s and a VI of at least 150 delivers larger improvements of overall efficiency (compared to an ISO 46 monograde HM fluid). Significant improvements of more than 17.5 percent at 100℃ and 350 bars have been measured. This boost in pump efficiency would be realized as greater than 15 percent of energy savings or fuel consumption reduction, compared to an ISO 46 monograde HM fluid delivering equal amounts of hydraulic work.

4 SUMMARY OF PERFORMANCE ADVANTAGES AND DEFINITION OF MEHF

For a hydraulic system at constant pressure, “in-service” fluid viscosity determines the overall system efficiency. Actual flow rate data measured in gear, vane, and piston pumps demonstrates that this relationship is valid in all types of pumps at all conditions. It is desirable to have the highest possible viscosity at high temperature operating conditions, and the lowest possible viscosity at cold start-up conditions (***Figure 1***).

Overall efficiency at high temperature can be improved by substituting a higher ISO viscosity grade fluid, but caution must be taken to insure that the fluid has sufficient low temperature fluidity. This can be achieved through the use of fluids with high viscosity index and good shear stability. The use of a higher ISO grade monograde HM fluid will negatively impact low temperature viscosity and air release properties.

Mineral oil fluids formulated with viscosity index improvers can provide higher “in-service” viscosity at high operating temperature, and lower viscosity at low start-up temperature, compared to a monograde HM fluids from the same ISO viscosity grade. Selecting a fluid with high viscosity index will help to improve pump efficiency at both low and high temperature. However, based on the results obtained for all pump types, it is important to select only shear stable high VI fluids if improved overall efficiency and reduced fuel consumption are desired. More precisely, we have observed that a minimum VI and a minimum viscosity after shear are necessary to improve pump performance.

For example, the use of shear stable high VI ISO 46 fluid was shown to improve the overall efficiency of a vane pump by more than 5 percent, under typical operating temperatures and pressures, compared to a monograde HM ISO 46 fluid. To insure a consistent level of high performance, such fluids should meet the following criteria as derived from the results of our studies:

- a viscosity index higher than 150,
- a minimum after shear kinematic viscosity at 100℃ (ASTM D 5621) of 7.5 mm²/s.

The overall pump efficiency advantage of high VI shear stable fluids were also found to be significant in a gear pump and a piston pump at typical operating conditions, compared to monograde HM fluids.

Based on these observations, a new performance level definition for hydraulic fluids called 'Maximum Efficiency Hydraulic Fluid' (MEHF) has been defined /Ham05/ which takes into consideration all the critical technical issues that affect pump efficiency. Such levels of performance have also been defined for other ISO viscosity grades.

To achieve the highest levels of pump efficiency, equipment designers and operators should select fluids that meet the MEHF performance level definition.

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