# The Importance of Viscosity for Hydraulic Fluid Efficiency – What Can We Learn from a Decade of Fluid Development?

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# Abstract

Nowadays, fuel economy and improving efficiency of equipment are key parameters and drivers in the development and design of modern equipment. This trend now also becomes important for hydraulic equipment: more efficient pumps and motors, hydraulic hybrids, and high performance hydraulic fluids all contribute to improved efficiency.

In this paper, we will focus on the hydraulic fluid which is often overlooked. However in the last years, the industry has started to recognize the importance of a careful selection of hydraulic fluid. Indeed in the past decade, extensive work based on field tests, pump tests and laboratory studies has been conducted to improve equipment efficiency by using multigrade shear stable oils (HV) in comparison to conventional, monograde (HM) oils.

The purpose of this paper is to provide a review of the results and conclusions published during the last ten years. It will show the enormous influence of hydraulic fluid properties on the efficiency of hydraulic systems.

KEYWORDS: hydraulic fluids, efficiency

# 1. Motivation

Energy efficiency and fuel economy are the key development parameters for all types of modern equipment. Efficiency of hydraulic systems has been receiving more and more attention over the last few years. Most of the efforts to increase efficiency happen through improvements on design of the machine parts of the system.

As mobile and industrial equipment becomes lighter and smaller, pump reservoirs also are getting smaller leading to sub-optimal cooling from short residence times and low fluid volumes. Pumps are operating at ever-increasing pressures (up to 6000 psi), and fluid temperatures of 80°C and peak temperatures of 100°C are common, especially in mobile equipment such as excavators and skid steer loaders. These changes require and extended performance of the hydraulic fluid which on the other hand strongly influences the performance of the hydraulic system. Therefore the lubricant must be seen as an integral part of the entire hydraulic system that inherently can contribute to increased efficiency.

Modern hydraulic lubricants are formulated to a high viscosity index to minimize the loss of viscosity at increased temperatures. While the viscometric properties of the fresh fluid are relatively easy to adjust, more and increased interest is now drawn on the performance of the lubricant during operation.

In the following chapters an overview of the work of the past ten years will be given. It comprises fundamental studies, multiple bench tests, rig tests and field trials which all conclude that a high performance hydraulic fluid can provide immediate efficiency gains independent on the type of hydraulic pumps used.

# 2. Influence of oil viscosity on hydraulic pump efficiency

Fundamental studies were conducted to evaluate the influence of oil viscosity on both volumetric and hydromechanical efficiency of hydraulic pumps, independently on each other. Prior to examination of these results, the definition of the key technical terms will be given.

# 2.1. Hydraulic pump efficiencies: definitions

The overall efficiency of hydraulic pumps is affected by two major contributors.

 Volumetric efficiency (1) depends on the losses that result from internal leakage. It is the ratio of the actual flow delivered by a pump and its nominal flow rate. Indeed, if the pump operating pressure is high enough, internal leakage takes place between the vanes and the pump housing. The term leakage describes a recirculation of the fluid in the pump in a direction opposite to the rotation of the pump (**Figure 1**). Leakage corresponds to volumetric power losses for the pump. The lower the volumetric efficiency the higher the volumetric power losses.



Figure 1: Schematic of internal leakage in a hydraulic pump

2. Mechanical efficiency depends on the losses that result from friction, either within the pump or caused by viscosity of the hydraulic fluid. It can be described as the ratio of the theoretical and the actual input torque needed to drive the pump (2). The difference of both corresponds to the extra input torque required to overcome the frictional losses within the pump. This extra input torque is associated to the hydromechanical power losses of the pump.

$$\eta_{\rm M} = M_{\rm theoretical} / M_{\rm actual} = M_{\rm theoretical} / (M_{\rm theoretical} + M_{\rm friction})$$
(2)

The overall efficiency of the pump is equal to the product of volumetric and mechanical efficiency.

# 2.2. Pump leakage and volumetric efficiency: influence of oil viscosity

The internal leakage in a pump is often described in the literature through the following model (3). It is the product of a geometrical factor associated to the pump with the pressure across the pump divided by the dynamic viscosity within the pump.

$$Q_{\text{leakage}} = \alpha \cdot \Delta P / \eta \tag{3}$$

By combining (1) and (3), the volumetric efficiency can be described as (4):

$$\eta_{V} = Q_{actual} / Q_{nominal} = (Q_{nominal} - Q_{leakage}) / Q_{nominal} = 1 - \alpha \cdot \Delta P / (\eta \cdot Q_{nominal})$$
(4)

It is difficult to measure the dynamic viscosity of the fluid in the pump at operating temperature directly. Several studies /Her02/ and /Nev02/ in vane and gear pumps were carried out to find models based on the fresh kinematic viscosity of the fluid at the

pump inlet temperature (using monograde mineral fluids (Newtonian fluids)). These studies concluded that the kinematic viscosity of monograde mineral fluids is a suitable parameter to estimate the volumetric efficiency of hydraulic pumps as described by the high correlation coefficient in **Table 1**.

Pump	Туре	Volumetric Efficiency	R <sup>2</sup>
Vane Pump	Eaton Vickers V-104	$\eta_V = 1 - 0.0173^* \Delta P/KV$	0.96
Vane Pump	Eaton Vickers V-20	$\eta_V = 1 - 0.0138^* \Delta P/KV$	0.96
Gear Pump	Bosch	η <sub>V</sub> = 1 - 0.027*Δ <i>P/KV</i>	0.99

Table 1: Models for the volumetric efficiency of three hydraulic pumps

However, the behavior of non-Newtonian fluids like high viscosity index mutigrade hydraulic oils follows more complex models /Gör06/. Indeed since these fluids are sensitive to shear, several studies based on different shear tests were evaluated to simulate the shear taking place in the pump loop.

For example an Eaton Vickers V-20 vane pump was used at an operating temperature of 65.6°C and operating pressures of 103.4 bar and 137.9 bar. Three HV hydraulic fluids, two engine oils, and one automatic transmission fluid were evaluated. All of these fluids contained PAMA viscosity index improvers. The model described in the equation (3) was applied to the actual flow rate of the pump. Fresh oil kinematic viscosity, kinematic viscosity of the fluid after 100 hours of use, and kinematic viscosity of the fluid after 40 minutes sonic shear according to the ASTM D 5621 were examined towards their correlation to the actual flow rate.

It was found in this study that the fresh oil viscosity cannot be used to predict the pump leakage. Kinematic viscosity of the used oil is suited to adequately predict the pump flow rate as a function of pressure.

Furthermore, the study also demonstrates that kinematic viscosity after 40 minute sonic shear provides the best correlation and can be used to obtain a precise estimate of the pump leakage as a function of pressure. **Figure 2** represents the actual flow rate measured for the tested fluids as a function of the ratio pressure by kinematic viscosity after 40 minute shear (at test temperature).



Figure 2: Correlation of flow rate and viscosity after shear (ASTM D5621)

Several studies were also conducted using pumps operating at higher pressure (vane and pistons pumps). The results obtained in a Parker Denison TC6CM vane pump operating up to a pressure of 250 bars are summarized here.

In this study, a broader amount of fluids were considered, three monograde (HM) fluids of viscosity classes ISO VG 46, 68 and 100 and a set of 15 multigrade (HV) fluids of viscosity classes ISO VG 32 and 46. The performance of the fluids was evaluated at a pump inlet temperature of 80°C and six different steps of pressure (15, 50, 100, 150, 200 and 250 bar). Like in the works mentioned above, viscosity values after various types of shear were examined in order to determine an adequate shear test to estimate the volumetric efficiency of the pump. **Figure 3** summarizes the results obtained by considering the HTHS viscosity of the fluid at 80°C (extrapolated from the HTHS measured at 100 and 150°C according to the ASTM D 5481 test method).



Figure 3: Volumetric efficiency as a function of pressure and viscosity

It was found that either HTHS viscosity or viscosity after 40 minute sonic shear (ASTM D 5621) can be used to predict efficiency in this vane pump.

# 2.3. Hydromechanical losses and mechanical efficiency: influence of oil viscosity

According to textbooks /Tot00/ hydromechanical power losses (or frictional losses) are proportional to viscosity, speed and pressure (5)

$$\mathsf{P}_{\mathsf{H}\mathsf{M}} = \alpha \cdot \eta \cdot \mathsf{n} + \beta \cdot \Delta \mathsf{P}$$

(5)

with  $\alpha$ ,  $\beta$  geometrical factors, n the rotational speed,  $\Delta P$  the pressure, and  $\eta$  the dynamic viscosity of the fluid. Some work (/Her11/ and /Ali08/) was conducted in the last years to investigate the effect of the viscosity on these losses (hence on the mechanical efficiency) and the relative proportion of volumetric and hydromechanical power losses.

In /Her11/ the performances of a monograde ISO VG 46 fluid were evaluated in a Parker Denison T6CM vane pump operating at 250 bars from a range of pump inlet temperature of 30°C to 95°C. The actual mechanical power (i.e. the power delivered at the shaft of the pump), the theoretical power needed to drive the pump (i.e. the nominal power) and the actual hydraulic power delivered to the pump are reported in **Figure 4**.



Figure 4: Nominal and actual power of a Parker Denison T6CM vane pump as a function of pump inlet temperature

The difference between the power delivered by the shaft and the nominal power represents the hydromechanical power losses which correspond to additional energy consumption. The difference between the actual hydraulic power and the nominal power is a measure of the volumetric losses due to internal leakage. This is reflected in a loss of pump productivity.

The hydromechanical losses show a limited dependence on temperature (and on oil viscosity), and these losses are significantly smaller than leakage losses.

Similar work /Ali08/ was conducted in the same test rig but this time with three multigrade hydraulic fluids having significant different viscosities, a SAE 10W hydraulic fluid, a monograde ISO VG 46 fluid and a very shear stable ISO VG 46 multigrade hydraulic fluid of a viscosity index of 200. The hydromechanical power losses and the volumetric power losses of each of these fluids as a function of the pump inlet temperature are shown in **Figure 5**.

It can be observed from this figure that the use of a more viscous fluid (ISO 46 VI 200) allows reducing the volumetric losses at a given operating temperature significantly without generating major additional hydromechanical losses.





# 3. Pump efficiency improvements through high VI shear-stable hydraulic fluids

The previous paragraph showed that it is possible to improve the volumetric efficiency of a hydraulic pump, and consequently the overall efficiency by using a fluid having a higher viscosity. However, to maintain this higher performance level over the operating time it is necessary to use a shear stable high viscosity index fluid. Indeed such a fluid always shows a higher viscosity than a monograde fluid of the same ISO viscosity grade over the entire operating temperature range, and also offers better low temperature performance. Some work was conducted in vane and piston pumps to establish the minimum viscometric characteristics for high viscosity index fluids that are needed to significantly improve the pump efficiency.

# 3.1. Vane pump results

The efficiency improvement that can be achieved in a vane pump by using high VI shear-stable hydraulic fluids was investigated in the same vane pump described before (Parker Denison T6CM). A comprehensive study based on the use of seven mineral oil based hydraulic fluids of viscosity grade ISO VG 46 but varying viscometric properties was published in /Ali06/. One monograde (HM) fluid was compared to six different multigrade (HV) fluids with a VI of 150 and 200, respectively. Those six multigrade fluids were blended with three different PAMA-based VI improvers of different shear stability. The viscometric properties of these fluids are reported in **Table 2**.

Viscosity	Viscosity	Kinematic	Kinematic Viscosity at	Kinematic
100	-	6.57	6.57	0%
150	А	8.05	7.52	7.8%
150	В	8.14	7.24	11.0%
150	С	7.94	6.79	14.6%
200	А	9.57	8.10	15.4%
200	В	9.63	7.29	22.1%
200	C	9.86	6.76	31.4%

Table 2: Shear stability of seven fluids evaluated in the efficiency test

The overall efficiency of these seven fluids was measured at 80°C and at three pressures, 150, 200 and 250 bars, respectively. The percentage of improvement in terms of overall efficiency compared to the monograde ISO VG 46 fluid is reported in **Figure 6**.

This figure shows:

- Improvements in efficiency increase with increasing operating pressure.
- Improvements in efficiency decrease with increasing viscosity losses (caused by a lack of shear stability of the fluid) for a given viscosity index.
- Improvements in overall efficiency increase with increasing viscosity indexes for a shear stable viscosity index improver (polymer A)
- Significant improvement of overall efficiency of about 5% can be obtained by proper selection of the fluid.



Figure 6: Percentage of overall efficiency improvement compared to the monograde ISO VG 46 fluid

To determine the minimum viscosity after 40 minutes sonic shear offering significant improvement in efficiency in the most severe conditions of operation for the pump (80°C and 250 bars), a correlation between the overall efficiency gains and the kinematic viscosity at 100°C after shear was conducted (**Figure 7**).compared to viscosity after shear.





This study shows that a good linear correlation is established within this pump and that using a hydraulic fluid with an after-shear kinematic viscosity higher than 7.5 mm<sup>2</sup>/s will provide 5% better efficiency than an ISO 46 monograde fluid.

#### 3.2. Piston pump results

In the same publication similar results obtained with a Komatsu HP 35+35 dual piston pump are reported. Furthermore, the effect of increasing the viscosity index for a set of

fluids having a maximum viscosity loss of 10% after 40 minutes sonic shear was also examined.

The viscometric properties of the considered fluids are reported in **Table 3**. The matrix of fluids consisted of 5 mineral oil based fluids, all ISO VG 46 viscosity grade. Of these fluids one was a monograde (HM) fluid and four were multigrade (HV) fluids. The viscosity index of these HV fluids was set to 120, 140, 160 and 200, respectively. Therefore poly(alkylmethacrylate) (PAMA) viscosity index improvers were chosen in a way that the viscosity loss of the hydraulic fluids after sonic shear testing was in a comparable range and did not exceed 10%. The operating pressure of the pump was 350 bar and the operating temperature was set at 100°C.

Viscosity	Viscosity	Kinematic	Kinematic Viscosity at	Kinematic
100	-	6.81	6.81	0%
120	D	7.20	6.87	4.7%
140	В	7.89	7.18	8.9%
160	А	8.57	7.79	9.0%
200	E	9.66	9.01	6.7%

Table 3: Shear stability of seven fluids evaluated in the efficiency test

The overall efficiency of these fluids is compared in Figure 8. This figure shows that even shear stable multigrade fluids can yield a lower performance than the corresponding monograde fluid if the viscosity index is too low as can be seen for the fluid having a viscosity index of 120. The lower efficiency is a consequence of the shear thinning of the fluid. Indeed although this fluid has a high shear stability of only 5% viscosity loss, its viscosity after shear is of the same level than that of the monograde fluid, and its actual viscosity in the pump is probably even lower. However, the benefit of a high viscosity index overruns this effect above a VI of 140.



Figure 8: Dependence of efficiency on viscosity index in a dual piston pump

Therefore to ensure significant improvements in shear stability, it is important to consider not only the shear stability of a fluid through its percentage losses in viscosity after shear but also through its absolute viscosity after shear and its viscosity index.

# 4. Effect of operating time on oil viscosity and pump efficiency

Since multigrade hydraulic fluids undergo shear during use several studies /PIa05/ were conducted to investigate the stability of the efficiency gain brought by the use of high viscosity index shear stable hydraulic fluids over the life time of the fluid.

#### 4.1. Effect of operating time on oil viscosity

The effect of extended operating time on oil viscosity was evaluated using a shearstable high VI hydraulic fluid (ISO VG 46, VI=152) in a Parker Denison T6CM vane pump. The pump was operated under cycling pressure conditions during 300 hours. The pressure cycles were interrupted every 50 hours, and the flow rate was measured at 80°C and 6 different pressures. In addition, samples of the fluid were taken to measure the viscosity of the fluid (**Figure 9**). A slight decrease of kinematic viscosity was observed over time. The viscosity loss after 300 hours was only 5% compared to 11.8% for the same fluid after the KRL 20 hour test. Apparently the KRL test method is too severe and does not correlate with the shear occurring in hydraulic pumps.



Figure 9: Kinematic viscosity as a function of operating time in a vane pump

#### 4.2. Effect of operating time on flow rate

In **Figure 10** the pump flow rate measured at different pressure over time is reported. It can be observed that despite the loss of kinematic viscosity of the fluid during operation

the flow rate remains more or less constant over time even at highest pressure. This leads to the conclusion that the in-service viscosity "seen" by the pump remains unchanged over time.



Figure 10: Flow rate in a vane pump as a function of operating time

# 4.3. "In-service" viscosity seen by the pump

In **Figure 11** the authors give a possible explanation of the constant flow rate, i.e. the "in-service" viscosity seen by the pump. The used oil viscosity of the samples is decreasing over time due to the permanent shear loss of the fluid. However, a viscosity index improver which is a polymer coil sheared over time in two different ways. At the beginning, the polymer coil stretches without breaking. Therefore when measuring the kinematic viscosity of the fluid in a capillary viscometer, the viscosity loss due to this temporary shear cannot be observed since the polymer coil has time to relax under the small shear stress seen in such a viscometer. When the polymer coil is fully stretched, it will eventually break creating permanent viscosity loss over time. The fact the "inservice" viscosity seems to be constant over time could be explained by the following:

- The in-service viscosity is equal to the fresh oil viscosity less the sum of permanent and temporary viscosity losses.
- The two types of shear losses compensate each other over time.



Figure 11: Flow rate in a vane pump as a function of operating time

# 5. Field test results

Beside pump tests, several field tests were conducted to verify the possible efficiency gain in the field using a high viscosity index shear stable hydraulic fluid. An example of the results obtained was reported in /Ali08/.

# 5.1. 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<sup>3</sup> 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) 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.

# 5.2. 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 and a high VI of almost 200.

# 5.3. Test protocol

The work day consisted of the following steps: 15 minutes for machine warm-up followed by refueling and three hours of morning work. Each working cycle consisted of 55 minutes of continuous work and 5 minute break and was repeated three times. After an one hour break and refueling the same procedure was repeated in the afternoon. Then the total fuel consumption was determined. The tests were made under mild climatic conditions. The ambient temperature ranged between 7°C and 18°C (45°F to 65°F). The equipment was operated with the engine running at 90% and 100% throttle. The complete test program was an ABA cycle with fluid and filter changes inbetween.

#### 5.4. Test results

The fuel consumption for the two test fluids and the two throttle settings are detailed in **Table 4**. Fuel consumption increased when going from the 90 to the 100% throttle setting. Results show that the equipment consumed significantly less fuel per hour with fluid B irrespective of the throttle setting. The number of work cycles per hour was recorded and the average values are also shown in **Table 4**. More cycles per hour were completed with fluid B irrespective of the throttle setting.

		Fuel consumed		Work cycles	
Throttle	Fluid	kg/hour	improvement	#/hour	improvement
Full	Α	19.5		53.5	
Full	В	16.8	14%	56.6	6%
90%	Α	15.2		40.0	
90%	В	13.9	9%	49.7	24%

Table 4: Fuel consumption and work cycles completed per hour

Combining the fuel consumption per hour and the number of cycles per hour yielded the average fuel consumption per work cycle which is measure for the productivity. The productivity improvements achieved when using fluid B are 26.3% and 18.4% at 90% and 100% throttle respectively.

This impressive achievement is summarized in **Figure 12**. The improvement of fuel consumption, productivity and efficiency when using the high VI shear stable fluid instead of the OEM recommended monograde fluid is shown for the two throttle settings of 90% and 100%.



Figure 12: Percentage improvement of fluid B over fluid A

#### 6. Conclusion

Fundamental studies carried out in gear, vane and piston pumps have shown that pump losses comprise of hydromechanical losses and volumetric losses. Under typical operating conditions hydromechanical losses are small compared to volumetric losses. Therefore the efficiency of hydraulic pumps strongly depends on the viscometric properties of the hydraulic fluids. Since these properties can change over time due to shear in the pump it is not possible to describe the pump efficiency using only the viscometric characteristics of the fresh fluid. However, various experiments have shown that viscosity after shear (40' min sonic shear test according to ASTM D5621) is a suitable measure to predict efficiency of hydraulic fluids. Shear-stable fluids are needed to retain the viscometric performance over the lifetime of the fluid and thus deliver maximum efficiency. The results of comprehensive laboratory studies were confirmed in a number of rig tests and field trials. Significant efficiency gains of up to 10% were seen in these tests.

Shear-stable hydraulic fluids with a high viscosity index help to increase the overall efficiency of hydraulic systems.

#### 7. References

- /Tot00/ Totten G.E., Handbook of Hydraulic Fluid Technology, Marcel Decker New York 2000
- /Her02/ Herzog S.N., Neveu C.D., Placek D.G., Simko R.P., Predicting the Pump Efficiency of Hydraulic Fluids to Maximize System Performance, SAE OH 2002-01-1430, IFPE Las Vegas April 2002

- /Nev02/ Herzog S.N., Neveu C.D., Placek D.G, Influence of oil viscosity and pressure on the internal leakage of a gear pump, STLE 2002
- /Pla05/ Placek, D.G., Herzog, S.N., Neveu, C.D., Effect of Operation Time on Oil Viscosity and Hydraulic Pump Efficiency, IFPE, March 2005.
- /Ali06/ Alibert M., Görlitzer H., Herzog S.N., Neveu C.D., Efficiency Advantages in Vane, Piston and Gear Pumps, IFK 2006, Aachen, Germany
- /Gör06/ Görlitzer H., Alibert M., Herzog S.N., Neveu C.D., Dependence of Pump Flow Rate on the Viscosity of High VI Hydraulics Fluids, Technische Akademie Esslingen, International Symposium Esslingen 2006
- /Ali08/ Alibert M., Neveu C.D., Efficiency Improvements through High VI Shear Stable Hydraulic Fluids, Technische Akademie Esslingen, International Symposium Esslingen 2008
- /Her11/ Herzog S.N., Neveu C.D., Relative Impact of Hydromechanical and Volumetric Losses on Hydraulic Pump Efficiency at High and Low Temperatures, IFPE 2011

#### 8. Nomenclature

ΔP	pressure
PAMA	poly(alkyl methacrylate)
VI	viscosity index
$\eta_V$	volumetric efficiency
$\eta_M$	mechanical efficiency
η	dynamic viscosity
n	rotational speed
Q	pump flow rate
M <sub>theoretical</sub>	theoretical input torque
M <sub>actual</sub>	actual input torque
M <sub>friction</sub>	extra torque needed to overcome the frictional losses
P <sub>HM</sub>	hydromechanical power losses
P <sub>Vol</sub>	volumetric power losses