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Boost Performance and Reduce Costs by Selecting the Optimum Viscosity Grade of Hydraulic Fluid

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Abstract:

Hard working hydraulic equipment must frequently operate at temperature extremes, which can impact system response and reduce equipment life. Low temperature start-up with high viscosity fluid can lead to delayed work schedules, sluggish operation, hydro-mechanical losses, and pump cavitation. At peak operating temperature the fluid viscosity is often too low, resulting in poor pump efficiency, inadequate oil flow, and system overheating.

Mechanical solutions such as increased cooling capacity, or larger pumps and oil reservoirs are often not practical or too expensive. Selection of the proper viscosity grade of hydraulic fluid is a cost effective technique that will allow equipment to start smoothly at low temperatures, and also deliver adequate oil flow rates needed for efficient operation at high temperatures.

Several techniques will be discussed that enable the equipment user to identify the practical operating limits of a hydraulic fluid. This information can be used to determine the "Temperature Operating Window" of a given fluid in a pump. A newly published National Fluid Power Association (NFPA) recommended practice has established a nomenclature for describing fluid viscosity grades, which simplifies the oil selection process for the equipment operator. Multigrade hydraulic fluids can be used to meet OEM warranty requirements, while saving money on maintenance, fuel/electricity, and auxiliary equipment.

Key Words: Hydraulic Fluid, Pump Efficiency, Viscosity Grade, NFPA, TOW, ALTOW

Introduction:

Hydraulic systems are widely used in the manufacturing, construction, forestry, mining, and transportation industries. Over the years, systems for the transmission and distribution of power have become increasingly sophisticated, their applications more numerous and their operating conditions more demanding. Selection of an appropriate hydraulic fluid has become a critical task for the operator. It is important to consider the maximum operating pressure, the minimum and maximum operating temperatures, and the viscosity recommendations provided by the pump manufacturer.

Viscosity is one of the most important criteria in the selection of a hydraulic fluid. At low temperature, excessive viscosity may result in poor mechanical efficiency, difficulty in starting, and wear. As oil temperature increases, viscosity decreases, resulting in lower volumetric efficiency, overheating, and wear. Pump and motor manufacturers often provide in their documentation hydraulic fluid recommendations covering:

- The maximum start-up viscosity under load
- The range of optimum operating viscosity
- The maximum and minimum operating viscosity

Selection of the optimum fluid viscosity grade will provide the most efficient pump performance at standard operating temperatures, which will minimize lost time and energy/fuel costs for the operator.

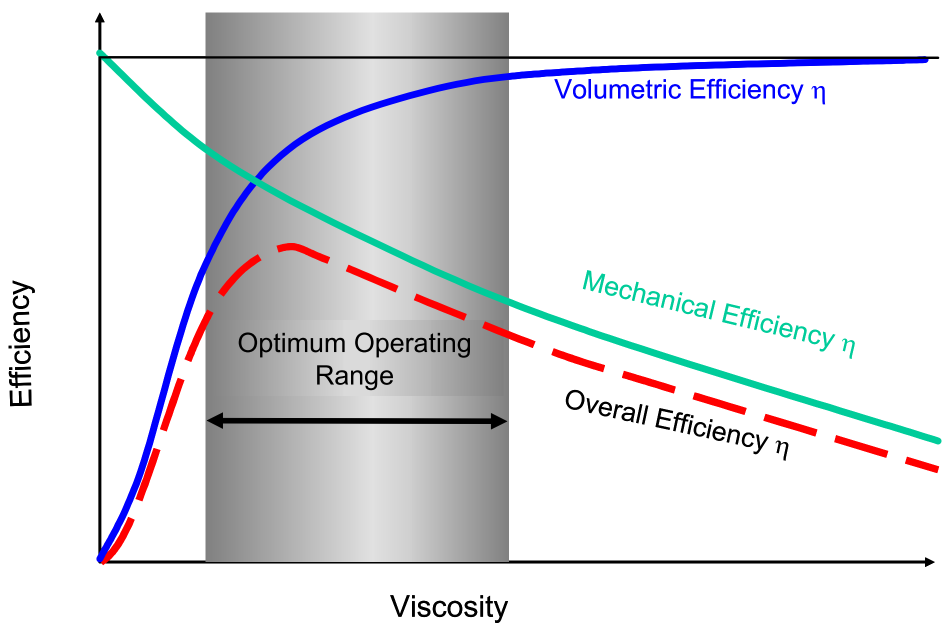
Pump Efficiency

The performance of hydraulic pumps and motors is a critical factor in overall hydraulic system reliability. There are two elements of hydraulic efficiency: volumetric efficiency and hydro-mechanical efficiency. Hydro-mechanical efficiency relates to the frictional losses within a hydraulic component and the amount of energy required to generate fluid flow. Volumetric efficiency relates to the flow losses within a hydraulic component and the degree to which internal leakage occurs. Both of these properties are highly dependent on viscosity.

Hydro-mechanical efficiency drops as fluid viscosity increases due to higher resistance to flow. Conversely, volumetric efficiency increases as fluid viscosity increases because of the reduction of the internal leakage. The overall efficiency of a hydraulic pump is the product of mechanical and volumetric efficiencies [Equation 1], and both factors must be considered together (1). As can be seen in Figure 1, there is a range of hydraulic fluid viscosity that optimizes the overall efficiency (2) .

Overall efficiency = Hydromechanical efficiency * Volumetric efficiency [Equation 1]

Figure 1- Relationship of Viscosity to Pump Efficiency



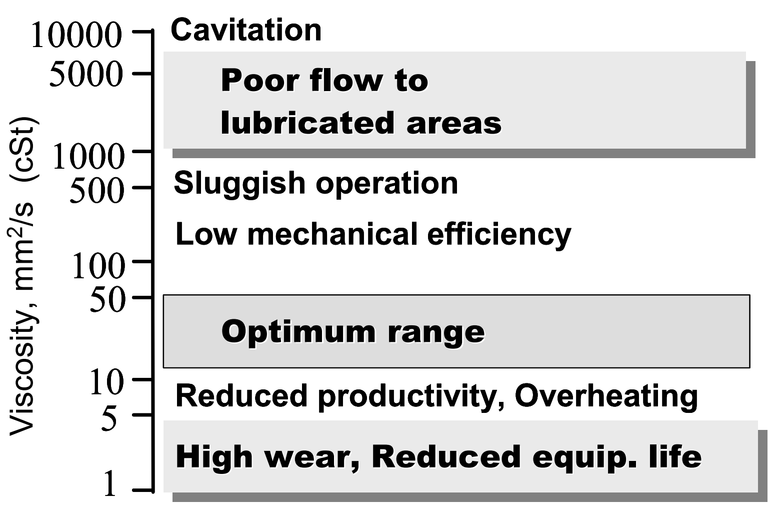
Cavitation, Wear, and System Overheating

At low temperatures, high viscosity has a negative effect on the mechanical efficiency of the hydraulic system and results in reduced system performance, lubricant starvation and cavitation. Viscosity influences cavitation because high viscosity fluids can create excessive pressure drop at the pump inlet. Cavitation causes metal fatigue and spalling that reduces pump life and generates abrasive metal particles in the fluid. Consequently, pump manufacturers specify a maximum fluid viscosity limit at start-up to ensure that cavitation is avoided. Improperly designed or undersized inlets and strainers aggravate the problems associated with high viscosity.

Excessive viscosity under low temperature conditions can also lead to pump starvation that may result in pump failure. Loss of the lubricating film creates high contact temperatures, excessive wear and, ultimately results in pump seizure.

One of the essential functions of a hydraulic fluid is to provide a lubricating film between moving pump parts that reduces wear. The effectiveness of this film depends upon a balance between viscosity, sliding speeds and loads within a hydraulic pump. As temperature increases and the fluid film becomes too thin, the lubricant film ruptures and metal-to-metal contact takes place. This results in wear within the pump and additional fluid heating. While it is intuitive that wear is undesirable, what is less obvious is that it predominantly occurs in locations within a pump that are critical in terms of volumetric efficiency. Loss of volumetric efficiency causes the pump to work harder to produce the required flow to hydraulic actuators. At the same time, high temperatures compromise volumetric efficiency as the result of low viscosity fluid bypassing critical pump clearances. Thus, inadequate viscosity due to high temperatures creates a destructive cycle of rising temperatures, accelerated wear, and increased internal leakage.

Figure 2- Fluid Viscosity vs. Performance



Multigrade Fluids Offer Improvement

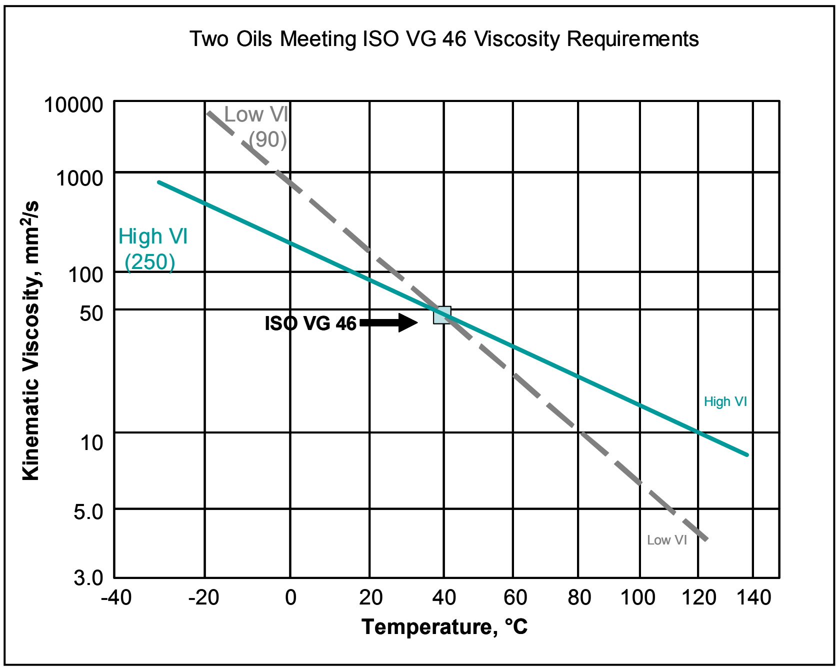
Multigrade hydraulic fluids are often recommended for equipment where the operating temperatures can vary widely [3]. Multigrade fluids enable efficient equipment operation over a wider temperature range than straight-graded oils. This results from their improved viscosity-temperature relationship that is measured by the viscosity index (VI) of the fluid. Multigrade hydraulic oils are also recommended to eliminate seasonal oil changes, since a properly formulated multigrade performs adequately in both winter and summer temperatures [3,4] .

Multigrade hydraulic fluids are good for cold weather start-up because, at low temperatures, their viscosity is lower than a monograde oil having the same ISO viscosity grade at 40 °C. This allows the hydraulic fluid to flow faster, avoiding pump cavitation and starvation. The result is smoother operation, and improved productivity.

Infrequently considered, but just as important, is the multigrade's effectiveness in maintaining pumping efficiency at high temperatures. As temperature in a hydraulic system rises, pumping efficiency drops because the increased temperature reduces fluid viscosity, leading to increased internal leakage. The viscosity of a multigrade fluid decreases less than that of monograde fluid with increasing temperature, thus retaining an acceptable pumping efficiency at a higher temperature.

The relationship of viscosity to temperature, and the viscometric advantages of high VI multigrade hydraulic fluids are shown in Figure 3.

Figure 3: Viscosity-Temperature Relationship for Low and High VI Oils



Fluid Selection Techniques

A viscosity grade selection system aimed at supporting equipment users has been published by the NFPA, based on the recommendations of leading hydraulic pump manufacturers [(5)]. Optimum viscosity grades are selected based on the concept of Temperature Operating Window (TOW), which corresponds to the range of temperature where the oil viscosity provides acceptable performance in the pump (typically 13 to 860 mm²/s) [(6)]. Details on the use of the ALTOW system are given in NFPA Standard Practice T2.13.13-2002, available through the National Fluid Power website at www.nfpa.com.

Performance Advantage of High VI Oils

The most commonly used and widely available viscosity grades are ISO 32, 46, and 68. The following sections will compare the performance of monograde (low viscosity index) and multigrade (high viscosity index) versions of these three fluids. It is important to recognize that shear stable fluids must be used in high pressure hydraulic systems in order to achieve desirable performance. Fluids with low shear stability are commercially available, and are typically intended for use in low pressure systems or for other applications such as automatic transmissions (ATF). The multigrade fluids selected for comparison in this work are intended for high pressure hydraulic system service and have good shear stability. A description of these fluids can be found in Table 1. It is recommended that the fluid supplier be consulted for guidance on the shear stability of their products, in order to achieve the maximum performance advantage for the application.

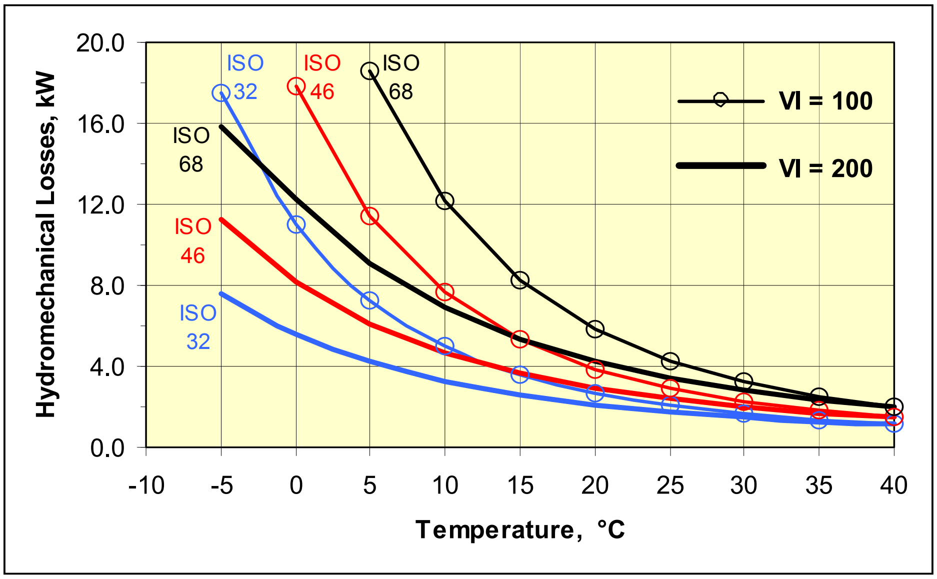
Table 1: Viscometric properties of test oils.

	ISO Grade					
Property	VG 32		VG 46		VG 68	
VI	100	200	100	200	100	200
KV at 100 ºC, mm^2/s	5.36	7.16	6.72	9.53	8.73	13.06
KV at 40 ºC, mm^2/s	32.0	32.0	46.0	46.0	68.0	68.0
Temperature for 860 mm^2/s, ºC	-7	-19	-2	-14	4	-8
KV at 100 ºC after 40 minutes Sonic, mm^2/s, ASTM D 5621	-	6.26	-	8.16	-	10.98
KV at 80 ºC after 40 minutes Sonic, mm^2/s, ASTM D 5621	-	9.07	-	12.19	-	16.84
KV at 40 ºC after 40 minutes Sonic, mm^2/s, ASTM D 5621	-	28.0	-	39.39	-	57.29
VI after 40 minute Sonic, ASTM D 5621	-	184	-	187	-	188
NFPA T2/13.13.2002 Grade	L32-32	L22-46	L46-46	L32-68	L68-68	L32-100

Performance advantage at low temperature.

We have calculated the additional energy, or hydromechanical losses (in kW), required to operate a mobile vane pump having a displacement of 10.8 ml/rev. at 800 rpm and 100 bars, conditions typical of those prevailing at start-up. These data are shown in Figure 4.

Figure 4: Hydromechanical losses as a function of temperature, ISO grade and VI



If we take the example of ISO VG 46 fluids at 0 ºC, the data in Table 4 indicate that the monograde fluid requires 125% more energy (18 kW vs. 8 kW) than the multigrade to overcome the viscous drag in the pump.

The theoretical power input for the pump in this example is only 1.4 kW, assuming no hydromechanical energy losses take place (energy required to turn the rotor with no viscous drag). The additional energy required to overcome the higher viscosity of the 100 VI fluids increases significantly at temperatures below 40 ºC, and dramatically at temperatures below 0 ºC. The volumetric efficiency of this pump will be discussed in the following section (see performance data for "Cartridge A" in Tables 2-5).

Performance advantage at high temperature

We have computed the actual flow rate and the total power requirement for vane pumps based on a given body, using four cartridges of different size. Internal cartridge sets (rotors and vanes) are sized to deliver a specific flow rate by controlling the discharge volume per revolution. Calculations were made at a pressure of 200 bars, a speed of 2000 rpm, and at two temperatures, 80 and 100 ºC.

Flow Rate Advantage – Time Savings

Knowing the actual flow rate Q_a, we can determine the time needed to fill a given linear motor of volume, V. A linear motor is typically a hydraulic cylinder that fills with fluid, displacing a rod that delivers motion under load.

$$\text{Time} = V/Q_a \qquad \text{[Equation 2]}$$

By calculating the ratio of the time required for two oils having the same ISO VG grade but different VI we can estimate the time advantage for the high VI oil to deliver the same volume of fluid. In this work, we always used the viscosity of the high VI oils after the Sonic 40 minute shear test to compute the actual flow rate (see Table 1). This represents a good estimate of the used oil viscosity in a 2000 psi vane or piston pump system.

$$\mathrm{Time}_{(VI=100)}/\mathrm{Time}_{(VI=200)} = Q_{a(VI=200)}/Q_{a(VI=100)} \qquad \text{[Equation 3]}$$

It can be seen that the benefit offered by the high VI oils decreases when increasing the cartridge size and increasing the ISO grade. This results from the fact that the larger the cartridge, the lower the internal leakage relative to the pump flow rate.

Table 2- Additional Time Required for a 100 VI Fluid to Deliver the Same Volume as a 200 VI Fluid* at 80 ºC, 200 bars and 2000 rpm

		Time Difference, %		
Cartridge Size	Nominal Flow Rate ml/rev.	ISO 32	ISO 46	ISO 68
A	22	20.8%	15.5%	11.1%
B	34	9.2%	7.6%	6.0%
C	46	6.0%	5.2%	4.2%
D	70	3.6%	3.2%	2.6%

* Performance gains are based on used oil viscosity, after shear.

Field studies showed that peak operating temperatures in mobile hydraulic equipment were often in excess of 100 ºC [3,4]. Therefore, we also calculated the flow rates in this series of pumps at this higher temperature. The data in Table 4 indicate that the high VI fluids at 100 ºC deliver between 5 to 30% greater flow rate, allowing a cylinder to fill more quickly.

Comparing the data at 80 ºC to the data at 100 ºC, one can see that high VI fluids show an even greater advantage as fluid temperatures increase above 80 ºC.

Table 3- Additional Time Required for a 100 VI Fluid to Deliver Same Volume as a 200 VI Fluid* at 100 ºC, 200 bars and 2000 rpm

		Time Difference, %		
Cartridge Size	Nominal Flow Rate ml/rev.	ISO 32	ISO 46	ISO 68
A	22	x	x	30.4%
B	34	22.5%	17.9%	13.8%
C	46	13.2%	11.3%	9.2%
D	70	7.2%	6.4%	5.4%

* Performance gains are based on used oil viscosity, after shear.
x Data not reported because the volumetric efficiency for the 100 VI oils ISO 32 and 46 was lower than 50%.

Efficiency Advantage – Cost Savings

Knowing the total power required to deliver the hydraulic power and to overcome the hydromechanical losses, we can determine the energy needed to fill the linear motor of volume V.

$$\text{Energy} = \text{Total Power} * \text{Time} \qquad \text{[Equation 4]}$$

We have calculated the power needed to drive the pump at 80 and 100 ºC, using fluids with different VI. This allowed us to determine the difference in energy required to deliver the same volume of fluid under a given pressure and pump speed.

$$\text{Energy}_{(VI=100)}/\text{Energy}_{(VI=200)} = \text{Power}_{(VI=100)}*Q_{a\,(VI=200)}/(\text{Power}_{(VI=200)}*Q_{a(VI=100)}) \quad \text{[Equation 5]}$$

The data in Table 4 indicate that the high VI multigrade fluid at 80 ºC, 200 bars and 2000 rpm may save between 2 and 20% in energy consumption over the 100 VI fluid.

Table 4- Energy Savings with High VI oils* at 80 ºC to Deliver Same Volume (200 bars, 2000 rpm)

		Energy Savings, %		
Cartridge Size	Nominal Flow Rate ml/rev.	ISO 32	ISO 46	ISO 68
A	22	20.0%	14.6%	9.0%
B	34	8.3%	6.8%	4.7%
C	46	5.7%	4.6%	3.2%
D	70	3.3%	2.7%	2.0%

* Performance gains are based on used oil viscosity, after shear.

Similar to the approach taken in the "Flow Rate Advantage" section, we also calculated energy consumption at 100 ºC, to identify the potential energy savings at a temperature closer to a typical peak operating temperature. The data in Table 5 indicate that the high VI fluid at 100 ºC, 200 bars and 2000 rpm may save between 5 and 28% in energy consumption.

Table 5- Energy Savings with High VI oils* at 100 ºC to Deliver Same Volume

		Energy Savings, %		
Cartridge Size	Nominal Flow Rate ml/rev.	ISO 32	ISO 46	ISO 68
A	22	x	x	27.9%
B	34	22.0%	17.4%	12.9%
C	46	12.8%	11.0%	8.5%
D	70	6.9%	6.2%	5.0%

* Performance gains are based on used oil viscosity, after shear.
x Data not reported because the volumetric efficiency for the 100 VI oils ISO 32 and 46 was lower than 50%.

As fluid temperature increases, the energy savings attributed to the high VI fluids is amplified.

Relative Operational Cost Comparisons

The relative energy saving data presented above can be used to generate cost saving estimates for specific applications. Consider the case of a single mobile vane pump running at typical mobile construction equipment operating conditions of 200 bar, 2000 rpm, and 80°C. Depending on the particular pump size, the potential diesel fuel savings are on the order of 200-300 gallons per year. Cost saving calculations can be made using the following formula:

Total Fuel Consumption (liters) =
Pump Power Requirement (kW) *
Hours of pump operation (hours) *
Diesel fuel consumption rate (0.22 kg/kWh) *
Density of diesel fuel (1.19 liters/kg) [Equation 6]

Fuel Savings (liters) =
Total Fuel Consumption *
Relative Energy Savings (% from Tables 4 and 5) [Equation 7]

Cost Savings = Fuel Savings * Local cost of diesel fuel [Equation 8]

Fuel and cost savings calculations for an ISO 46 hydraulic fluid in a single pump and a medium size construction equipment fleet are presented in Tables 6 and 7.

Table 6 – Fuel and Cost Savings- Single Mobile Vane Pump at 200 bar, 2000 rpm, 80°C

Pump	A	B	C	D
kW	15.5	24.0	31.9	48.1
Gallons of Diesel Fuel Used	2,141	3,316	4,407	6,645
Gallons of Diesel Fuel Saved	313	224	201	182
Annual $ Saved	$ 469	$ 336	$ 301	$ 272

Assumptions: 8 Hours/day, 250 Days/year, Diesel Fuel in USA @ $1.50/gallon

Table 7 – Fuel and Cost Savings- Construction Equipment Fleet- 100 Units [200 Mobile Vane Pumps at 200 bar, 2000 rpm, 80°C]

Pump	A	B	C	D
kW	15.5	24.0	31.9	48.1
Gallons of Diesel Saved	37,557	26,888	24,108	21,797
Annual $ Saved	$ 56,335	$ 40,332	$ 36,162	$ 32,695

Assumptions: 8 Hours/day, 150 Days/year, Diesel Fuel in USA @ $1.50/gallon

Conclusions

The comparison of the performance at low and high temperature of six hydraulic fluids with three different ISO grades (VG 32, 46 and 68) and two different Viscosity Indices (100 and 200) showed that:

The high VI oils contributed to significantly lower hydromechanical losses at temperatures lower than 40 °C. The gain in hydromechanical efficiency can exceed 50% at start-up temperature, resulting in lower energy consumption, shorter warm-up times and reduced wear.

At temperatures of 80 and 100 °C, calculations made for a series of vane pumps showed that the high VI oils deliver a higher flow rate and a better overall efficiency. This translates into higher equipment productivity, as well as significantly lower operating costs for the equipment user due to lower fuel consumption. Energy/fuel savings in the range up to 20% can be expected under standard operating conditions when high VI multigrade oils are used. Higher productivity gains and savings can be achieved at peak operating temperatures.

The cost savings associated with the use of multigrade hydraulic fluids in a single vane pump are approximately $345 per year per pump. This advantage could be expected to result in approximately $41,000 savings annually for a medium sized equipment fleet.

References

(1) P.W. Michael, S.N. Herzog, T.E. Marougy, "Fluid Viscosity Selection Criteria for Hydraulic Pumps and Motors". NCFP paper I00-9.12 presented at the International Exposition for Power Transmission and Technical Conference. 4-6 April 2000, Chicago, IL, USA.

(2) G.E. Totten, Handbook of Hydraulic Fluid Technology, Marcel Dekker, New York, 2000, p. 27.

(3) I. Makkonen, "Performance of Seasonal and Year-Round Hydraulic Oils in Forestry Machines", FERIC Technical Note TN-251, Forest Engineering Technical Research Institute of Canada, 12/96.

(4) D.G. Placek and C.W. Hyndman, "Cost and Performance Advantages of Multigrade Hydraulic Fluids", Proceedings of the 7th Annual Fuels & Lubes Asia Conference, 2/01, Bangkok, Thailand.

(5) NFPA Recommended Practice T2.13.13-2002. "Fluid Viscosity Selection Criteria for Hydraulic Motors and Pumps". 2002. www.nfpa.com

(6) S.N. Herzog, C.D. Neveu, D.G. Placek, "Predicting the Pump Efficiency of Hydraulic Fluids to Maximize System Performance". NCFP I02-10.8/SAE OH 2002-01-1430 presented at the IFPE / SAE Off-Highway Meeting, March 19-23, 2002 Las Vegas, NV, USA.