

Make sure you're using the right product by following a systematic approach based on objective, repeatable and widely recognized engineering practices and principles.

## Article highlights:

- Guidelines for selecting lubricants for plain and element bearings, gears and hydraulic systems.
- Analyzing the additive mix for each component.
- Making technically accurate baseline lubricant selection decisions without exacting mathematical expressions.

Lubricant selection is a pivotal starting point in the pursuit of precision lubrication practices. All the effort applied to clean delivery and handling, filtration, dehydration, alignment, balancing, etc., is lost if the lubricant selected for the application cannot support the machine's demands. Many criteria must be considered when selecting a lubricant for a set of machines or machine components.

Lubricant chemistry has an influence on the final decision. The basestock (synthetic or mineral) and the additive systems in use (EP, AW, R&O) exert tremendous influence on the performance qualities of the lubricant. It is common to have multiple choices for a given grade and type of lubricant within a product line (i.e., ISO 220EP/SAE 90EP gear oil), with each example designed to perform effectively for a set of given conditions.

The machines themselves have an influence on final lubricant selections. OEMs design and build machines for general types of service. A gear drive manufacturer could not possibly consider each and every type of application for a given make and model when it begins the design process. Designers can build

sufficient durability into the basic design that allows customers to use those products for similar uses within widely varying production environments. The nature of the production environment (wet, dry, hot, cold, abrasive dirt, harsh chemical exposure, steady state or intermittent operation, etc.) influences the degree of effectiveness for a given lubricant type and grade.

Plant maintenance strategy has an influence on lubricant selection as well. Where management is particularly forward thinking and willing to invest in modifications that improve lubricant management effectiveness (filter connections, continuous filtration, embedded sample ports, bearing isolators, etc.), the company is positioned to maximize the superior value that can be achieved through the use of high-performance lubricants, both mineral- and synthetic oil-based.

With the variety of factors that can impact lubricant film formation and effectiveness, it is to the practitioner's benefit to follow a lubricant selection process that is objective, repeatable and based on widely recognized engineering practices and principles. This article addresses a formal lubricant selection process that could be used to make technically accurate baseline lubricant selection decisions without using exacting mathematical expressions.

### Parsing the machine

In the January TLT I proposed cataloging all lubricated components within a machine for evaluation and specification of a lubricant. The purpose of the process is to consolidate the collection of machine components into a concise set of types and catalog which components rotate, slide, pivot or have other dynamically interacting surfaces. There are relatively few unique types of components, but there are many permutations of each of the few unique options. Let's address these components by general type that represent most industrial machinery, including:

- Plain bearings
- Element bearings
- Gears
- Hydraulic systems

### Plain bearing lubricant selection

Probably the most common manifestation of a plain bearing is a round steel journal riding on a conforming one- or two-part brass or babbitt bushing. There are plenty other machine component types that could be categorized similarly, including machine tool gibbs (slideways), brass bushing, pivot pins, ball screws, worm gears, etc. These bushings are similar to plain bearings in composition, form (shape), and film characteristics.

Regardless of shape and form, all plain bearing components have a common requirement: a full-fluid hydrodynamic film to sustain component life cycles. Two types of decisions must be made:

1. Viscosity grade according to the machine's operating profile.
2. Lubricant chemistry/additive type (R&O, EP, AW, Compounded, Tackified).

The viscosity grade considerations for sliding surface interaction are rooted in a common set of physical realities. Total available surface area, linear surface speed, surface unit loading, lubricant viscosity grade and lubricant replacement rate all have an influence on the formation of a hydrodynamic oil film.

STLE member Bob Scott, manager of LubeWorks, Ltd., in Calgary, offers a method of lubricant selection for plain bearing applications. According to Scott, the minimum information required for the determination of the proper ISO grade for journal bearings includes:

1. Shaft RPM.
2. Temperature of the oil in the bearing.
3. Approximate unit loading pressure (PSI or Newtons / m<sup>2</sup>).

Scott adds that it would be best to also have additional information, including:

- Bearing length, number of bearings and rotor weight (to calculate the bearing pressure to verify the load).
- Shaft diameter to calculate the shaft surface speed.
- Driving horsepower.
- Knowing whether shock loading is present.
- Knowing if the unit is in a warm or cold environment.
- Knowing if cooling water is being applied.

Scott explains there are several different charts that recommend ISO viscosity grades for journal bearings, and most are based on oil temperature and shaft RPM. Figure 1 provides an ISO viscosity

CONTINUED ON PAGE 20

Figure 1. Viscosity selection chart for 'medium load' applications (150–200 psi)

Shaft speed	Operating oil temperature			
	20 C–50 C	60 C	75 C	90 C
800 rpm	ISO 68	ISO 100	ISO 150	
1,200 rpm	ISO 46	ISO 68	ISO 100-150	
1,800 rpm	ISO 32	ISO 46	ISO 68-100	ISO 150
3,600 rpm	ISO 32	ISO 32	ISO 46-68	ISO 68-100
10,000 rpm	ISO 32	ISO 32	ISO 32	ISO 32-46



Figure 2. Generally accepted minimum and optimum viscosity grades

Suggested Minimum Allowable Viscosities

- 7- 8 cSt Process turbines
- 13 cSt General guidance

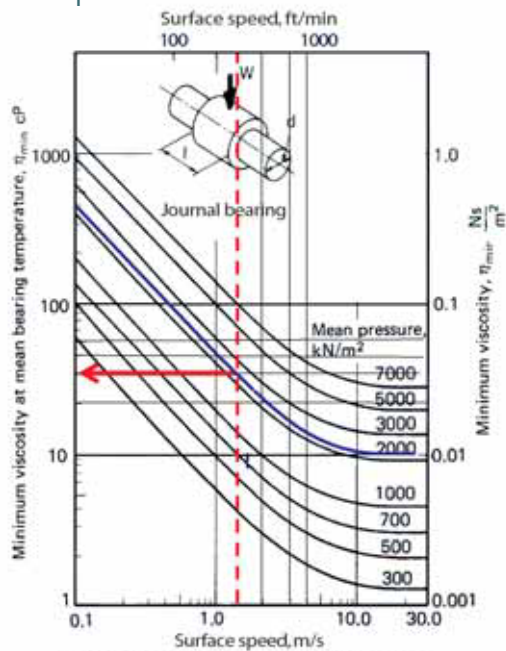
Generally Accepted Optimum Viscosities

- 20-22 cSt (for 3,600 RPM, No Shock Loading)
- 35 cSt (for 1,800 RPM, No Shock Loading)
- 50 cSt (for 1,800 RPM, Heavy Load & Shock Loads)
- 72 cSt (for 500 RPM, No Shock Loading)
- 95 cSt (for 500 RPM, Heavy Load & Shock Loads)

All viscosity units are centistokes at operating temperature

(Courtesy of Lubeworks, Inc.)

Figure 3. Viscosity estimation chart based on actual speed and unit loads



(Courtesy of The Tribology Handbook, Second Edition, Michael J. Neale)

Figure 4. Calculations for plain bearing surface speed and unit load

Bearing Linear Speed Calculation, u

$$u = \pi * d * n$$

Bearing Mean Pressure, p

$$p = \frac{W}{l * d}$$

- Where:
- n = shaft speed
  - p = pressure, kN/m<sup>2</sup>
  - l = bearing length, meters
  - d = shaft diameter, meters
  - π = 3.1415
  - W = load, kN (Kilo-newtons)

grade selection based on oil temperature and RPM for moderate shaft loads (150-200 psi). While this is a common representation, it is the shaft surface speed, not RPM, which should be used to determine the correct viscosity grade.

For higher loads (~300 psi), raise the viscosity by 2 ISO grades. For lower loads (~100 psi), lower the viscosity by 1 or 2 ISO grades.

According to Scott, temperature estimation is critical. Oil temperature at the bearing, shaft surface speed and load should be taken into account. It is the temperature of the oil in the bearing itself that must be considered. When provided with the temperature of a piece of equipment be sure to know, and account for, how and where the temperature was taken. It is the oil temperature in the bearing that is desired. The temperature of the bearing housing is likely 5 C-10 C below the oil temperature and the actual bearing metal temperature may be 15 C higher than the oil temperature. Reservoir temperatures could be 20 C below the oil temperature in the bearing.

Additionally, after settling on a target ISO viscosity grade, it is essential to check the product selection against the *minimum* and *optimum* recommended viscosity for the actual oil operating temperature. Most journal bearings require a minimum of 10 to 13 cSt at operating temperature. Some process turbines may only require 7 or 8 cSt as a minimum, but the 13 cSt minimum is a good rule of thumb that provides some margin for error. If low temperature startups are involved, the pour point and Brookfield viscosity data at low temperatures will also need to be investigated. Figure 2 provides a set of generally accepted minimum and optimum viscosities.

*The Tribology Handbook*<sup>1</sup> provides a useful chart (see Figure 3) that can be used to plot speed against load to arrive at a more narrowly selected lubricant grade.

Figure 4 shows the formulas that would be used to calculate both bearing linear speed (in meters/second) and unit load (in kilo-Newtons/m<sup>2</sup>). Once known, these values can be placed on a viscosity selection chart similar to Figure 3 to arrive at a target viscosity grade in centipoise.

Two steps remain. First, for both approaches the practitioner should plot the selected product on an ASTM viscosity-temperature graph to find the actual viscosity in centistokes for the expected operating temperature. This step provides a useful profile of the viscosity of the oil across its operating temperature range.

Second, for the approach using Figure 1 (units in centipoise), the practitioner should multiply the

selected oil viscosity in centistokes (at operating temperature) by the typical specific gravity for the product to arrive at the expected viscosity in centipoise.

### Plain bearing additives

Plain bearings traditionally have been lubricated by either compounded type products (fortified with fatty-acid compounds to improve lubricity) for low-speed and heavily loaded sliding surfaces, or rust and oxidation inhibited type products for high-speed sliding surfaces. A strong argument can be made for sticking with these general product categories, particularly in defense of avoiding EP type lubricants.

Given the thermal energy state required to activate EP (sulfur and phosphorous) agents and the low likelihood that the steel-on-soft metals interaction (bronze or babbit) can deliver the heat load sufficient to cause these agents to perform their localized function, there is likely more risk from the use of EP products (accelerated lubricant degradation and increase of acidic oil conditions) than there is a benefit from their use. If high load capability is required, and there is concern that the viscosity cannot deliver the required results, then the reliability engineer should consider the use of solid film (molybdenum disulfide, graphite) fortified lubricant that can provide the added protection.

### Element bearing lubricant selection

There are many types of element bearings, including ball, roller, spherical roller, thrust roller, needle roller and single- and double-row types. Fortunately the selection process for the various types is similar. The following discussion pertains to the common ball, roller, spherical roller and thrust roller type element bearings.

Element bearings operate under a wide array of temperature, speed, load and environmental conditions. The first step is to select an appropriate viscosity grade followed by selection of additive type (R&O, AW, EP).

Bearing manufacturers provide fairly useful charts that assist us with this process. The five steps to this process are:





1. Determine the bearing physical size (width, outer diameter, inner diameter), element type (ball, cylindrical roller, spherical roller, thrust ball or roller) and shaft speed. These details often can be found in the shaft RPM machine construction diagrams and drawings. Example: A 6320 bearing has a bore dimension of 100 mm, an outer diameter dimension of 215 mm and a race width of 47 mm.

Figure 5. Bearing speed limit parameters are calculated using the following equation:

$$nDm = n * \frac{(ID + OD)}{2}$$

Where: nDm = Pitch Line Velocity of the Element  
 N = shaft speed  
 ID = Bore Diameter  
 OD = Outside Diameter

Figure 6. General accepted speed limiting values for industrial bearings<sup>2</sup>

Common Element Bearing Speed Factors, $nd_m$		
Bearing Type <sup>c</sup>	Oil Lubricated <sup>a</sup>	Grease Lubricated
Radial Ball Bearing 	500,000	340,000
Cylindrical Roller Bearing 	500,000	300,000
Spherical Roller Bearing 	290,000	145,000 <sup>b</sup>
Thrust, Ball and Roller 	280,000	140,000

a. Oil lubrication is preferred where heat dissipation is required  
 b. Grease lubrication is not recommended for spherical roller thrust bearings  
 c. Precision bearings and high performance lubricants allow 5 to 50% higher operating ranges.

2. Determine the bearing's limiting speed factor. This value is referred to as the "pitch line velocity" or the "nDm." Figure 5 shows the speed limit calculation for element bearings. Bearing speed limits differ for each bearing, but general rules exist around bearings by bearing type and function.

Figure 6 provides practical speed limit values for four common bearing types. Any given bearing may have a unique value for its given quality, but these table values adequately represent range limits for decisions for selecting either oil or grease lubrication.<sup>2</sup>

Example: The 6320 roller bearing turning at 1,200 rpm has an nDm (speed limiting) value of 120,000. This information is particularly useful to assess whether the bearing should be lubricated by oil or grease. Equipment operators often convert oil-lubricated bearings to grease, even though the operating speed of the bearing is beyond that which is considered safe for grease lubrication. Grease type and relubrication volumes and frequencies should be adjusted to maintain

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Figure 7. Element bearing viscosity selection graph

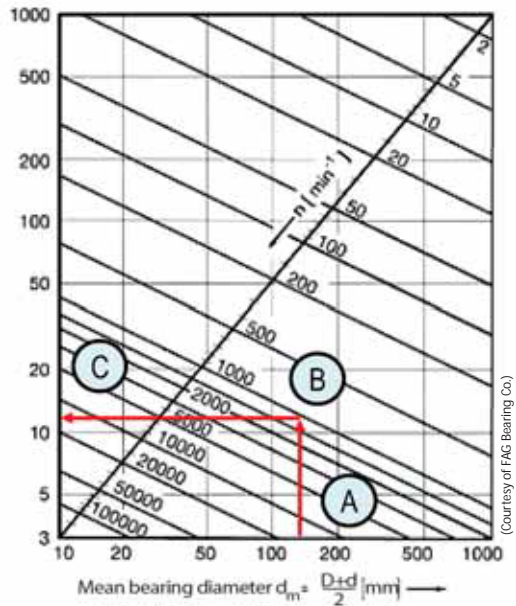
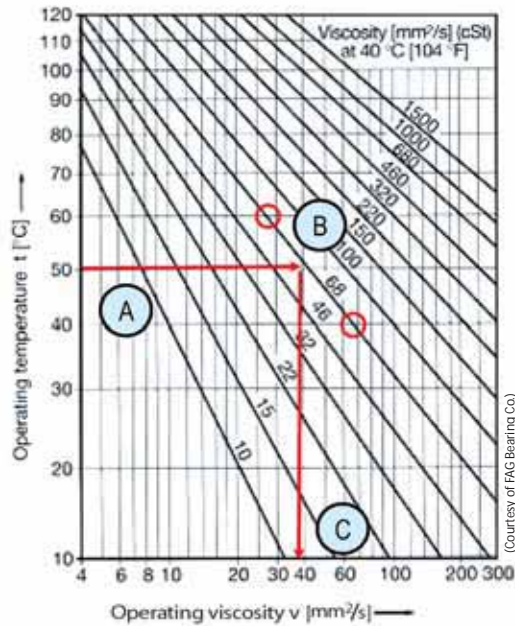


Figure 8. Operating viscosity for mineral oils (with a VI of 100)



bearing reliability under these circumstances.

3. Calculate the Pitch Diameter (PD) of the bearing ((inner diameter + outer diameter) ÷ 2). Example: The 6320 bearing pitch diameter is ((100 mm + 215 mm) / 2 = 157.5 mm). This parameter will be used to identify the minimum viscosity for the bearing at its operating speed.
4. Plot the bearing PD on a bearing specific viscosity limit reference chart. Locate and plot the shaft rotational speed and triangulate to the suggested viscosity for the given running

speed. The result provides the lowest viscosity point at which the bearing would be expected to achieve its projected minimum life cycle. The optimum viscosity is three to five times the minimum recommended viscosity.

Example: This is a multistep process. Begin by locating the pitch diameter value from Step Four at its appropriate location on the X (bottom–horizontal) axis of Figure 7. Next, draw a vertical line from the X axis toward the top of the chart until it intersects with the diagonal line that represents the shaft rotational speed. Next, at that intersecting point, draw a line to the Y axis. Make sure that the final line is horizontal to the X axis. The Y axis represents the bearing minimum suggested viscosity value for a given size and shaft speed.

In this instance, the PD value (157.4) is labeled Point A. The intersection with the diagonal line for shaft speed is labeled Point B, which is the initiating point for the line that terminates at Point C, this bearing’s required viscosity in centistokes. This element bearing would have a minimum oil viscosity requirement of 13 cSt when operating at 1,200 rpm.

Ideally, the operator should maintain three to five times the minimum viscosity in the element at its normal operating temperature. Therefore, for this bearing, the low-end opti-

There are many criteria to consider when selecting a lubricant for a set of machines or machine components.

Figure 9. Types of common industrial gears



mum is (13 cSt\* 3 = 39 centistokes), and high-end optimum is (13 cSt\* 5 = 65 centistokes). If viscosity error must be tolerated it would be best to err on the side of increased oil thickness as long as the overage does not induce heat through fluid friction.

5. Plot the viscosity of the target lubricant on an ASTM viscosity-temperature graph and determine the actual viscosity at the expected operating temperature. Repeat the process until the correct lubricant has been selected.

Example: Figure 8 is a temperature viscosity chart provided by FAG Bearing Co. The bearing operating temperature is located on the Y axis, and is labeled Point A. Draw a straight line from the point that is horizontal to the X axis until it intersects with a red line, representing a standard viscosity grade (32, 46, 68, 100, etc.). An ISO 68 should provide the desired viscosity at the normal operating temperature of the application (50 C). The intersection point with this line for ISO 68 is labeled Point B. Lastly, from this intersection point (B), draw a straight line to the X axis to locate the actual operating viscosity, 38 cSt, which is labeled Point C.

It is evident from this exercise that this bearing, operating at a running speed of 1,200 rpm at 50 C temperature requires an ISO 68 viscosity grade (with a VI of 100).

### Element bearing additives

Bearing manufacturers suggest that as long as the operating viscosity is above the minimum it would be best to lubricate without “doped” lubricants. Doped pertains to lubricants with AW or EP agents that form physico-chemical barrier films following surface contacts. AW and EP agents may over time influence the microcrystalline structure of the bear-

Figure 10. The shaft characteristics of different gear design types

Gear design type	Shaft characteristic
Spur, helical, herringbone	Non-intersecting and non-crossing shafts
Bevel (straight and spiral tooth)	Intersecting shafts (any angle)
Worm and hypoid gears	Non-intersecting shafts

ing metals in a negative way, even if the additive agent is not fully engaged in surface protection.<sup>3</sup>

If shock loading is expected, then plan to use EP/AW-fortified lubricants to provide a margin of error. If not, assuming that the operating viscosity is in an ideal range, use an R&O lubricant. If the operating conditions include momentary high-temperature excursions that bring the operating viscosity below the allowable minimum, then EP/AW type lubricants are in order.

### Industrial gearing lubricant selection

Gears are used to increase or decrease transmitted speed and load and change the direction of rotation from one shaft to another. As shown in Figure 9, gears come in a variety of configurations and tooth shapes.

These types of gears can be identified by observing the shaft characteristics (*see Figure 10*).

The majority of industrial gears are spur, helical, worm and bevel types.

Gears are designed to optimize energy input with speed or load output. Large loads require large gear teeth and heavy construction. Conversely, small loads can be accomplished with small teeth and light construction.

CONTINUED ON PAGE 24



Gear drives can be lubricated with either grease or oil. Gear type, size and function are the leading factors in this decision. While the great majority of industrial gear applications operate in an oil bath, some gears operate without one. These gears, called 'open' gears, are lubricated with specialized greases that are sprayed onto the gear set on an intermittent basis or applied manually (grease gun, brush, spray can). Open gear lubrication will be addressed later in this series.

Once again, viscosity is the central feature of the lubricant that must be accurately identified and selected in order for components to achieve their expected life cycles. As is the case with the previously evaluated components, the oil must possess viscometric properties that cover cold-flow requirements at low starting temperatures and provide sufficient oil thickness at normal operating temperatures to resist the squeezing effect of the meshing gears.

The low temperature flow requirement is dependent on a variety of application-specific factors, including weather, degree of shelter (indoor, outdoors), presence of thermal controls and likelihood that the unit will be started at extreme (cold) conditions. A safe pour-point limit for gear oils is 10 F/5.2 C below the coldest likely starting temperature.<sup>4</sup>

Most modern gear OEMs provide lubricant

selection guidelines that conform to the current AGMA standard for industrial gear lubricant, Standard 9005-EO2. In most instances gear drive housings include selection advice on the gear case itself. Figure 11 shows how the information is displayed on a common gear drive.

Over time reliability engineers may have to revisit the lubricant selection following a change from the original operating state. In these instances, there needs to be a relatively uncomplicated and accurate way to estimate the target lubricant type and grade. AGMA standard 9005-EO2 for lubricant selection is a helpful document to support effective lubricant selection and will be referenced for the balance of this selection process.

AGMA promotes viscosity selection based on the pitch line velocities of each gear reduction. Pitch line velocity (PLV) is calculated based on the following formula:

$$PLV, \text{ meters/second} = (\text{Gear pitch diameter in millimeters} \times \text{Gear rpm}) \div 19098$$

The gear pitch circle dimensions and shaft speeds must be known in order to make these calculations. These values can be provided by the drive manufacturer or derived from other tooth form and ratio data.

Once known, the PLV can be correlated to tables from 9005-EO2, Appendix B, to reference the required viscosity by ISO grade for a given sump temperature and for the referenced gear tooth forms.

Consider the following example for a drive on a wire-drawing machine, as seen in Figure 12:

**Step 1.**

Determine the dimensions of the bevel gear set:

- pinion pitch diameter of .032 meters
- bull (driven) gear pitch diameter of .093 meters.

**Step 2.**

Measure or calculate the gear shaft speed (pinion = 3750, bull= 350 rpm).

**Step 3.**

Measure the internal oil sump temperature (50 C).

**Step 4.**

Calculate the PLV of each gear.

**Step 5a.**

Using the appropriate table from the 9005-EO2 standard, locate the column for PLV and the row for oil sump temperature. The appropriate oil grade is found where the column and row intersects.

Figure 11. Gear drive faceplate label showing viscosity parameters



(Courtesy of Advanced Machine Reliability Resources)

## Step 5b.

If the lubricant under consideration is a synthetic, use the table that corresponds to the high viscosity index (VI) property. Figure 13 corresponds to 9005-E02, Table B-3 for synthetics oils with VI = 160.

When multi-stage reductions produce PLVs that are at least one or more standard ISO viscosity grades apart, the owner must rationalize the selection based on lubricant application (force feed vs. splash feed), risk imposed by likely load, risk imposed by temperature spikes and risk of cold start-up to arrive at a final selection. In this exercise, where there is a high pinion speed, low heat loads and little risk of load spikes, the preferred mineral oil selection would be an ISO 150, and the preferred synthetic oil selection would be an ISO 100. A synthetic VG 150 also would be acceptable. Bearing viscosity selection should be evaluated for all gears with high operating speeds or temperatures.

The 9005 standard provides additional charts that are useful for selecting lubricant viscosity grades for worm gears and large open gears with caveats and explanations for each type of application.

### Industrial gear drive additives

There are three classifications of oils that apply to geared drives: R&O, EP/Antiscuff and Compounded types. R&O oils are intended where pitch line velocities are high and shock loading is low. EP/Antiscuff oils are intended where gear materials are steel (no brass or bronze) and where PLV values are low and the risk of shock loading is high. Most industrial applications require EP oils. (EP oils must be avoided for gears with internal backstops.)

Lastly, compounded oils are used in drives with high sliding contact surface pressures and soft metal gears—such as with worm gear type applications. Although some lubricant suppliers advocate the use of antiscuff oils for worm gear applications, the 9005 standard does not recommend this practice.

### Hydraulic system lubricant selection

Lubricant selection for hydraulic systems is a topic for an entire book. Let's look at a few central themes at the heart of all hydraulic oil selections that can be addressed:

1. Viscosity grade at operating temperature.
2. Base fluid stock type (mineral oil, synthetic, water-based, biodegradable).
3. Additive type.

Figure 12. Gear oil viscosity calculation and selection

Estimating the gear drive lubricant viscosity requirement for a bevel gear				
PLV = $\Pi \cdot D \cdot N$			Pinion (m)	Crown (m)
	Pi	D	3.1415	3.1415
<b>Step 1</b>	Measure the Gear Pitch Dia (m or in)	D	0.032	0.093
	Measure/calculate the gear speed -	N		
<b>Step 2</b>	RPM	N	3750	375
<b>Step 3</b>	Measure the sump temperature		50°C	50°C
<b>Step 3</b>	Calculate the PLV in minutes		376.9	109.5
<b>Step 4</b>	Calculate the PLV in seconds		6.3	1.8
<b>Step 5(a)</b>	Plot the PLV on AGMA 9005-E02, Table B-1 to estimate the Mineral Oil VG (where V.I. = 100)	VI = 100	ISO 150	ISO 320
<b>Step 5(b)</b>	Plot the PLV on Table B-3 to estimate the Synthetic VG (where V.I. = 160)	VI = 160	ISO 100	ISO 220

(Courtesy of Advanced Machine Reliability Resources)

Figure 13. AGMA Standard 9005-E02 (Appendix B, Table 3.)

Table B-3 - Viscosity grade<sup>1)</sup> at bulk oil operating temperature for oils having a viscosity index of 160<sup>2)</sup>

Temp °C	Pitch line velocity, m/s <sup>3)</sup>							
	1.0 - 2.5	2.5	5.0	10.0	15.0	20.0	25.0	30.0
10	32	32						
15	46	32	32					
20	68	46	32					
25	68	46	32					
30	100	68	46	32				
35	150	100	68	46	32			
40	150	100	68	46	32	32		
45	220	150	100	68	46	46	32	
50	220	150	100	68	46	46	46	32
55	320	220	150	100	68	68	46	32
60	460	220	150	100	68	68	68	46
65	460	320	220	150	100	100	68	46
70	680	460	320	150	100	100	100	68
75	680	460	320	220	150	150	100	68
80	1000	680	320	220	150	150	150	100
85	1500	680	460	320	220	220	150	100
90	1500	1000	680	320	220	220	220	150
95	2200	1500	680	460	320	220	220	150
100	3200	1500	1000	460	320	320	220	150

NOTES:  
<sup>1)</sup> Consult gear, bearing and lubricant suppliers if a viscosity grade of less than 32 or greater than 3200 is indicated.  
<sup>2)</sup> Review anticipated cold start, peak and operating temperatures, service duty and range of loads when considering these viscosity grades.  
<sup>3)</sup> Select the viscosity grade that is most appropriate for the anticipated stabilized bulk of operating temperature range.  
 Selective stabilized bulk of operating temperature and bearing lubrication requirements.  
<sup>4)</sup> This table assumes that the lubricant retains its viscosity characteristics over the expected oil change interval. Consult the lubricant supplier if this does not apply.  
<sup>5)</sup> Determine pitch line velocity of all gear sets. Select viscosity grade for critical gear set taking into account cold start-up conditions.

Bearing manufacturers suggest that as long as the operating viscosity is above the minimum it would be best to lubricate without 'doped' lubricants.



The hydraulic system, in a simplistic view, accomplishes work by squeezing a small quantity of oil into a pipe very rapidly and then directing the pressurized fluid through a set of valves to be released where specific mechanical force is needed.

There are several common components in a hydraulic system, including:

- **Reservoir.** Provides a supply of oil to the pump.
- **Pump.** Adds flow/pressure to the oil.
- **Piping system.** Contains the pressurized oil.
- **Valve/control system.** Channels energy in the form of pressurized oil.
- **Work components of some type.** Performs the required work.
- **Filters.** Protects critical components from catastrophic damage, cleans the oil.

These various components represent a mixture of frictional properties, including rolling friction (hydraulic motor and input motor bearings), sliding friction (pump vanes and pistons, hydraulic rods and cylinders) and a combination of the two (gear pump tooth contacts, hydraulic motors). All these working components are sensitive to the physical, chemical and cleanliness properties of the fluid.

Flow control systems tend to dictate fluid cleanliness requirements. Cleanliness will be addressed in a future article. Pump type (mechanical interface) and efficiency parameters typically dictate physical viscosity parameters. Fluid type is dictated by plant environmental factors: basestock type and additive performance selection. It is not unusual for a given product selection to have conflicting priorities.

Despite the many differences between systems, they are all intended to operate within an efficiency range. The allowable viscosity for a given system

is established by pump type and the specific pump's efficiency parameters. If the fluid is too thick the pump may cavitate, overheat due to fluid friction, increase mechanical stress on bearing elements and consume excessive energy to fulfill its function. If the fluid is too thin the pump contact surfaces will wear and lose efficiency, and the pump may leak across internal and external seals and clearances and overheat.

System manufacturers offer a window for appropriate viscosity performance. The equipment owner must fit the product into the viscometric window. The following seven steps represent a systematic approach that may be used to arrive at the best overall viscosity selection.

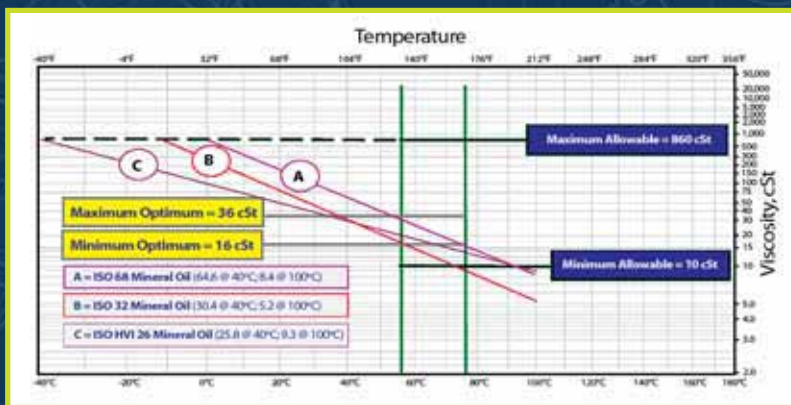
1. Plot the operating temperature range for the system on a standard temperature-viscosity selection chart, as shown in Figure 14. The green lines represent high and low temperature operating ranges. If the system hasn't been commissioned, use the OEM's suggested temperature profile.
2. Hydraulic pump designers provide viscosity limits that include absolute maximum and minimum limits (noted by the blue boxes) and optimum maximum and minimum limits. Next, plot these provided values on the chart.

Example: a pump manufacturer provides an absolute maximum limit of 1,000 cSt and a minimum of 10 cSt. These lines establish finite limits for operating temperatures (cold start, high operating temperature limits) for the final fluid selection. Additionally, the OEM offers the ideal minimum (16 cSt) and ideal maximum viscosity (36 cSt) values. The gray lines on the chart represent viscosities that support optimum volumetric efficiency.

3. Find the intersecting lines between the operating temperature limits (green lines) and optimum viscosity limits (grey lines). This cross-section represents the target space through which we would like our best product choice to pass.

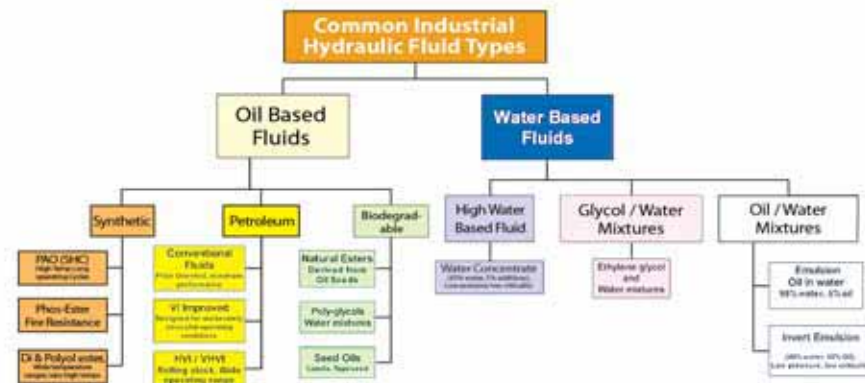
4. Review and plot the viscosity parameters for each of the available fluid choices on the viscosity temperature chart. Example: There are three available options. Option A is an ISO 68 mineral oil

Figure 14. Hydraulic oil viscosity selection chart



Courtesy of Advanced Machine Reliability Resources

Figure 15. Cross-section of types of industrial hydraulic fluids



(Courtesy of Advanced Machine Reliability Resources)

with a VI of 100. Its default viscosity measurements are 64.6 cSt at 40 C, and 8.4 cSt at 100 C. These points are plotted and a line is drawn from the far left of the chart through these points to the far right of the chart. The extreme ends of this line represent temperature extremes that will be considered later. This process is repeated for the other available products (options B and C are provided).

- Evaluate the cold flow temperature limit for each of the plotted lines and eliminate any that do not meet the limits for cold weather operation.

Example: Identify the product that has the longest line segment passing through the optimum area (noted in Step 3). In this case it appears to be option A. Observe the left side of the chart where the 'A' line passes through the OEM's absolute low viscosity limit. Line A crosses this (1,000 cSt) line at about 0 C. This temperature is the lowest possible temperature that the system can be allowed to run with this fluid. If it is likely that the system will be expected to start at temperatures lower than 0 C, then another option must be considered. Repeat the steps for each additional product option.

- Make the final selection based on the best fit for optimum viscosity within the operating temperature range and the least risk of cold start failure.

Example: If the fluid is being selected for a hydraulic system that is bolted to the floor, it is unlikely that the cold start temperatures will ever fall below 0 C. To control this risk,

the OEM might install a thermostatically-controlled heater to assure safe start conditions. If the system happens to be installed in outdoor construction machinery, then it is likely that eventually this machine will have to function below the cold start limit suggested for fluid A, making it undesirable for the application.

- Repeat this evaluation process for each product type until the best option is identified.

### Industrial hydraulic applications

As shown in Figure 15, several categories of hydraulic fluids may be considered. Water-based fluids are preferred for applications where fire is a

CONTINUED ON PAGE 28

Compounded oils are used in drives with high sliding contact surface pressures and soft metal gears—such as with worm gear type applications.



threat, and oil-based fluids are preferred for applications where either there is no risk of fire or where there is both risk of fire and a high degree of component reliability is necessary for other reasons (i.e., safety factors such as fluids used in aircraft hydraulics).

In each instance, regardless of the basestock type, wear-resistance is critical. The primary mechanical component subject to wear is the pump. Pumps operate with significant but low incremental-force mechanical interaction (boundary and mixed film state). AW fluids are appropriate for this type of service. These additives generate a physi-chemical organo-metallic barrier layer on contacting surfaces that can be repeatedly wiped off and replaced many times during normal operation. These chemi-physical films provide ongoing wear resistance without excessive chemical reactivity.

Lubricant manufacturers have many different lubricant formulas that accomplish these results, with varying degrees of longevity, additive stability and overall system friendliness. Standardized wear-resistance performance testing should be included in the final lubricant selection process. **TLT**

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