# Gear Oil Classification and Selection

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### Introduction

Today gear drive operators have several options when selecting the proper lubricant for their gearboxes. As in the past, the primary lubricant used for gearbox lubrication is mineral oil. But with the advances in technology, synthetic hydrocarbons (PAOs) and polyglycols show very specific advantages in certain applications. With gear drives becoming more and more precise, it is now also to the benefit of the gear operator to verify that he or she has the proper additive package and viscosity in the lubricant selected. Fig. 1 shows that a gear oil is a combination of a base oil and specific additives. The base oils can be either mineral oil, a synthetic or even in some cases a combination of the two.

### Mineral Oils

Mineral oils are a mixture of hydrocarbons and typically divide into

- Paraffinic oils having a paraffin base of more than 75%,
- Naphthenic oils having naphthene base of more than 75%,
- Aromatic oils having an aromatic base of more than 50%.

Sometimes the oil combination does not fit one of these classifications and is then termed a "mixed base mineral oil." Gear oils are made almost exclusively from paraffinic oils. Apart from the chemical characteristics, the physical values, such as density, viscosity, flow behavior, temperature dependency and other properties, are important. Mineral oils account for approximately 90% of the demand for lubricating oils.

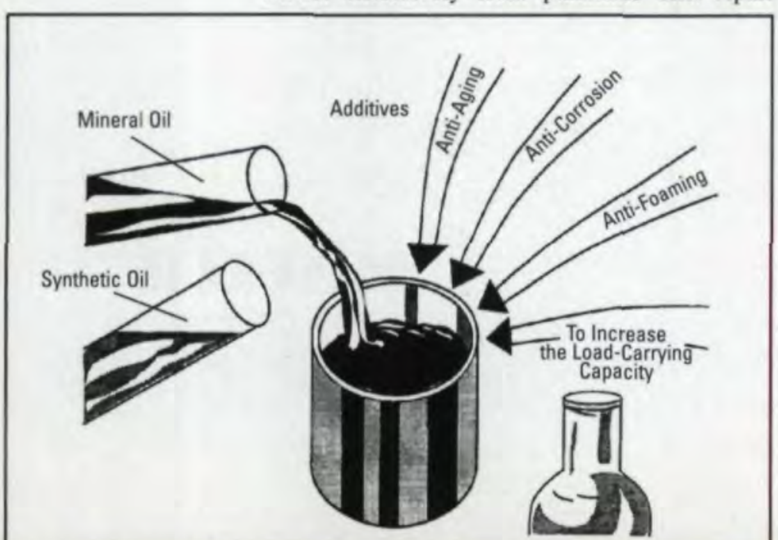



**Fig. 1 — Lubricating oils.**

### Synthetic Oils

Synthetic oils are artificial fluids that can be used for lubricating purposes. These synthetic liquids have some characteristics that are superior to those of mineral oil lubricants, at least for certain types of applications. In general, the advantages of synthetic oils with respect to lubrication are their thermal and oxidation stability, their favorable viscosity-temperature behavior, high flash point and good low temperature behavior. For gear oils, polyalphaolefins (PAOs) and polyglycols also provide lower frictional losses in the gear train. Fig. 2 shows the average values of the most important properties of mineral oil compared to polyglycols and PAOs.

### Additives

The second component of a lubricating gear oil is its additive package. Additives are put into lubricating oils to enhance some of the natural properties of the lubricating oil or to provide properties that are not present in the base oil. Fig. 3 shows a list of possible additives. Not all of these additives would be used in a single formulation, but all could be used in various different products, depending on the primary use of the oil.

One of the primary additives put in gear oils is an extreme pressure (EP) additive. This additive is needed to prevent microscopic welding between metal surfaces under high pressure or temperature, thereby protecting the gear tooth surface from scoring and premature fatigue failure. Various dynamic test machines can measure extreme pressure performance of a lubricating oil. Fig. 4 shows five of the common tests. Of these tests, the FZG procedure most resembles the actual loading conditions experienced by a gear system. If the lubricant can pass the twelfth load step without exceeding the specified wear rate or maximum weight loss, it is considered an extreme pressure oil. Such an oil will meet the needs of almost all gearing systems, provided that it meets the other parameters of the application, such as required viscosity, thermal resistance and oxidation stability.

### Viscosity

Viscosity is the most important property of a lubrication oil. It is a measure of internal friction, describing the resistance to relative motion between the molecules under sheer stress. Viscosity depends on pressure and temperature. Since viscosity is dependent on temperature, it is typically measured at 40°C and 100°C. Fig. 5 shows the ISO viscosity grades for lubrication oils, the equivalent AGMA viscosity grades, approximate viscosity values at various temperatures and the equivalent SAE viscosity grades.

Since the lubricating oil's viscosity changes with temperature, the rate of change is an important property identified by the viscosity index (VI). Most mineral oil gear oils will have a viscosity index of 95. A lower viscosity index indicates that the oil changes viscosity faster with temperature change than the specified mineral oil at 95 VI. Conversely, a higher viscosity index indicates a much slower rate of change in viscosity as temperature changes. An oil with a high viscosity index will tend not to thicken as much at lower temperatures as a lower viscosity index product. At higher temperatures, the oil will tend not to thin as much. The ability of the oil to maintain a small viscosity differential over the operating range of the gearbox provides a more consistent lubricating film to the gears and more predictable wear performance. Fig. 6 shows the viscosity temperature curves for a mineral oil, a PAO and a polyglycol. Each of these products has an ISO viscosity grade of 460, or AGMA viscosity number of 7.

### Mineral Oil Vs. Synthetic Oil

Synthetic oils have specific advantages in certain applications. Because of high oxidation resistance, synthetic oils can provide a much longer lubricant life in a gearbox. This will lengthen relubrication intervals and reduce overall oil consumption, as well as waste disposal.

Specific research has determined whether synthetic lubricants will provide less lost energy than

| Lubricating Oils Properties | Mineral Oils | PAOs | Polyglycol Oils |
|---|---|---|---|
| Density (g/ml) at 20°C | 0.9 | 0.85 | 0.9...1.1 |
| Viscosity Index (VI) | 80...100 | 130...160 | 150...270 |
| Pour Point (°C) | -40...-10 | -50...-30 | -56...-23 |
| Flash Point (°C) | < 250 | >200 | 150...300 |
| Oxidation Resistance | Moderate | Good | Good |
| Thermal Stability | Moderate | Good | Good |
| Lubricity | Good | Good | Very Good |
| Compatibility with Elastomers, Coatings, etc. | Good | Good | Insufficient to Good |
| Price Relation | 1 | 5-10 | 6-10 |

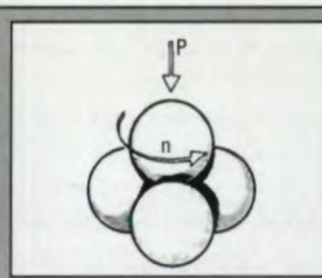
**Fig. 2 — Properties of some lubricating oils (average values).**

| Type of Agent | Chemical Compound | Purpose | Mechanisms of Action |
|---|---|---|---|
| Oxidation Inhibitors | Hindered phenols, amines, organic sulphides, zinc phosphorodithloates | Minimize polymerization to form resin, varnish, sludge, acids or polymerizates. | Stop the chain reaction of oxidation by reducing organic peroxides. Decrease acid formation by reduced oxygen absorption of the oil. Inhibit catalytic reactions. |
| Corrosion Inhibitors | Zinc phosphorodithloates, sulphurized terpenes, phosphosulphurized terpenes, sulphurized olefines | Protect bearing and other metal surfaces against rust. | Have the effect of anti-catalysts, film formation on metal surfaces to protect them against the attack of acids and peroxides. |
| Rust Inhibitors | Aminophosphates, sodium, calcium & magnesium sulphonates, alkyl succinic acids, fatty acids | Protect ferrous metal surfaces against rust. | Polar molecules are primarily adsorbed on metal surfaces and serve as a barrier against water. Neutralize acids. |
| Metal Deactivators | Triarylphosphites, sulphur combinations, diamines, dimerkaptane thiadiazole derivates | Eliminate catalytic influences on oxidation and corrosion. | Adsorption of a protective film on metal surfaces, which prevents the contact between the base metal and the corrosive substances. |
| Anti-Wear Additives | Zinc phosphoro-dialkyl-dithioates, tricresylic phosphates | Reduce excessive wear between metal surfaces. | The reaction with metal surfaces leads to the formation of layers which undergo a plastic deformation and improve the contact pattern. |
| Extreme Pressure Additives | Sulphurized greases and olefines, chlorinated hydrocarbons, lead salts of organic acids, aminophosphates | Prevent microscopic welding between metal surfaces under high pressure or temperature. | The reaction with metal surfaces leads to new compounds with a lower shear stability than the base metal. A continuous process of shearing-off and rebuilding. |
| Friction Modifiers | Fatty acids, fatty amines, solid lubricants | Reduce friction between metal surfaces. | Molecules with a high polarity are adsorbed on metal surfaces and separate the surfaces. Solid lubricants form a friction-reducing film on the surface. |
| Pour Point Depressants | Paraffin alkylation of naphthalenes and phenoles, polymethacrylates | Lower the pour point of the oil. | Prevent the agglomeration of paraffin crystals by covering them. |
| Viscosity Index Improvers | Polyisobutylenes, polymethacrylates, polyacrylates, polyethylene-propylene, stryrene maleic acid ester copolymers, hydrogenated butadiene-styrene copolymers | Reduce the dependency of viscosity on temperature. | Polymer molecules have a high tendency to coiling in unsuitable solvents (cold oil), whereas they uncoil in suitable solvents (warm oil) and, consequently, become larger in volume. This leads to a relative thickening of the oil. |
| Anti-Foam Agents | Silicone polymers, tributylphosphate | Prevent a stable foam formation. | Attack the oil film surrounding each air bubble, thus reducing interfacial tension. This leads to the agglomeration of small bubbles into larger bubbles which then rise to the surface. |
| Tackifiers | Soaps, polyisobutylenes and polyacrylate polymerizates | Improve the oil's adhesiveness. | Increase of viscosity. Agents are thick and sticky. |

**Fig. 3 — Important types of additives.**

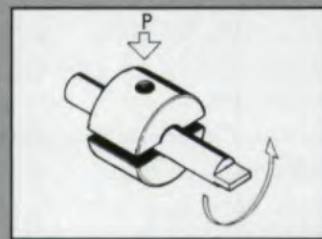
**FOUR-BALL EP AND WEAR TESTER (DIN 51 350, ASTM D 2596)**

| | |
|---|---|
| Purpose: | Determine extreme pressure properties of oils, dispersions, greases and pastes |
| Test Piece | 4 balls @ 12.2 mm |
| Sliding Velocity | 0.55 m•s-1 |
| RPM | 1420 |
| Load | 0.6-12 kN (57 load steps) |
| Test Period | 1 min per load step |
| Type of Contact | 3 stationery balls w/ 4th rotating on top |
| Type of Friction | Sliding friction |
| Value Measured | Welding load, O.K.load |




**ALMEN-WIELAND EP/WEAR TESTER**

| | |
|---|---|
| Purpose: | Determine extreme pressure and wear properties of oils, dispersions, greases and pastes |
| Test Piece | 1 steel shaft, 2 steel bearing shells |
| Sliding Velocity | 0.066 m•s-1 |
| RPM | 200 |
| Load | 0-20 kN |
| Type of Contact | Linear (shaft against bearing shells) |
| Type of Friction | Sliding friction |
| Value Measured | Stress load, abrasion, temperature, friction force |

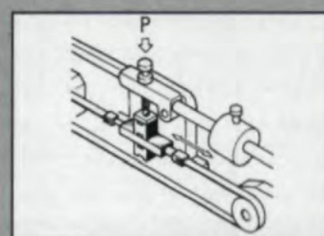



**FALEX EP/WEAR TESTER**

| | |
|---|---|
| Purpose: | Determine extreme pressure properties of oils, dispersions, greases, pastes and bonded coatings |
| Test Piece | 1 steel shaft, 2 v-shaped steel blocks |
| Sliding Velocity | 0.055 m•s-1 |
| RPM | 290 |
| Load | Up to 20 kN |
| Type of Contact | Linear |
| Type of Friction | Sliding friction |
| Value Measured | O.K. load, wear |


**TANNERT STICK-SLIP TESTER**

| | |
|---|---|
| Purpose: | Determine sliding properties of lubricants and materials at low speed (e.g. slideway oils, sliding oils, adhesive oils) |
| Test Piece | 2 stationary test blocks, 1 flat sliding member |
| Sliding Velocity | 0-0.243 mm•s-1 |
| Load | Varies from 12.5 to 30 N/cm2 |
| Type of Contact | Surface |
| Type of Friction | Sliding friciton |
| Value Measured | Friction number, stick-slip |




**FOUR-SQUARE GEAR TEST RIG (FZG PROCEDURE) (DIN 51 345)**

| | |
|---|---|
| Purpose: | Determine lubrication properties of gear oils |
| Test Piece | 2 gear wheels |
| Circumferencial Speed | 8.3 m•s-1 ir 16.6 m•s-1 |
| RPM of Pinion | 2170 or 4340 |
| Load | 12 load steps up to a max. pinion torque of 545 Nm |
| Test Period | 15 min. per load step |
| Type of Friction | Rolling friction |
| Type of Motion | Turning wheels |
| Value Measured | Specific wear in mg/kWh |
| Fretting Load | Sudden increase in loss of weight (scuffing occurs) |

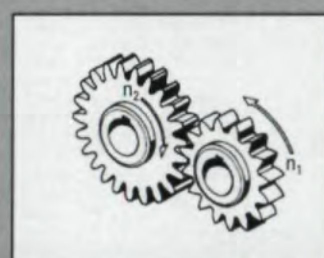



**Fig. 4 — The most common testing machines for lubricating oils.**

| ISO (DIN 51 519) | AGMA Lubricant Viscosity Number | Average Viscosity (40°C) and Approximate Viscosity Values in mm²•s-1 (cSt) at | | | | | Approximate classification of | |
|---|---|---|---|---|---|---|---|---|
| | | 20°C A [mm²•s-1] | 40°C [mm²•s-1] | 50°C B [mm²•s-1] | 50°C [Engler] | 100°C [mm²•s-1] | motor oils SAE | automotive gear oils SAE |
| 5 | | 8 (1.7E) | 4.6 | 4 | 1.3 | 1.5 | | |
| 7 | | 12 (2E) | 6.8 | 5 | 1.4 | 2.0 | | |
| 10 | | 21 (3E) | 10 | 8 | 1.7 | 2.5 | | |
| 15 | | 34 | 15 | 11 | 1.9 | 3.5 | 5W | |
| 22 | | 55 | 22 | 15 | 2.3 | 4.5 | 10W | 70W 75W |
| 32 | | 88 | 32 | 21 | 3 | 5.5 | | |
| 46 | 1 | 137 | 46 | 30 | 4 | 6.5 | 15W 20W 20 | 80W |
| 68 | 2 | 219 | 68 | 43 | 6 | 8.5 | | |
| 100 | 3 | 345 | 100 | 61 | 8 | 11 | 30 | 85W |
| 150 | 4 | 550 | 150 | 90 | 12 | 15 | 40 | 90 |
| 220 | 5 | 865 | 220 | 125 | 16 | 19 | 50 | |
| 320 | 6 | 1340 | 320 | 180 | 24 | 24 | | 140 |
| 460 | 7 | 2060 | 460 | 250 | 33 | 30 | | |
| 680 | 8 | 3270 | 680 | 360 | 47 | 40 | | |
| 1000 | 8A | 5170 | 1000 | 510 | 67 | 50 | | 250 |
| 1500 | | 8400 | 1500 | 740 | 98 | 65 | | |

**Fig. 5 — ISO viscosity grades, DIN 51 519.**

mineral oil lubricants in heavily loaded gearboxes. One study was performed using the FZG gear testing rig to determine the relative friction loss with mineral oil at various viscosities as compared to frictional loss of a PAO and polyglycol. For the same viscosity grade, a PAO and polyglycol gave lower frictional losses than straight mineral oil. With less friction, the synthetics provide less heat, less energy consumption and a higher efficiency rating for the gear drive. Friction modifiers help mineral oils, but at high speeds, synthetic lubricants significantly out-perform the mineral oil lubricants. At the higher speeds, the polyglycol and the PAO have very similar frictional loss.

Not only do gearboxes experience less friction, but documented studies have shown that just switching to a synthetic oil immediately reduced the shock impulse activity in the gearbox as well as the vibration. Fig. 7 shows the results of this study.

In worm gearboxes with high reduction ratios, polyglycol oils provide a significant advantage over mineral oils in the following performance factors:

- Improved energy efficiency
- Reduced maintenance, improved reliability and longer life
- Increased design ratings.

At 60% of rated power, a polyglycol was found to operate approximately 10°C cooler in a worm drive gear than a mineral oil. At 100% rated power, the polyglycol operates at the same temperature as mineral oil operating a 70% rated power. Less heat means less friction, which consumes less energy and produces less wear (Fig. 8).

### Viscosity Selection

The correct viscosity is the most important parameter in selecting the proper gear oil. The manufacturer of the gearbox or gear system generally makes a viscosity recommendation, and this recommendation should always be followed. If the OEM of the gear system has not provided these recommendations, and the viscosity has not been calculated based on elastohydrodynamic (EHD) theory, it can be selected in accordance with various worksheets. The differing viscosity-temperature and viscosity-pressure behavior of synthetic oils as compared to mineral oils also must be taken into account.

The correct viscosity must be selected independently of any specific gear stage, realizing that a compromise is required for multi-stage gears. The selection of the correct viscosity in accordance with the worksheet is based on the oil's expected operating temperature; i.e., the sump temperature or the temperature of the injected oil. This temperature is calculated by determining the

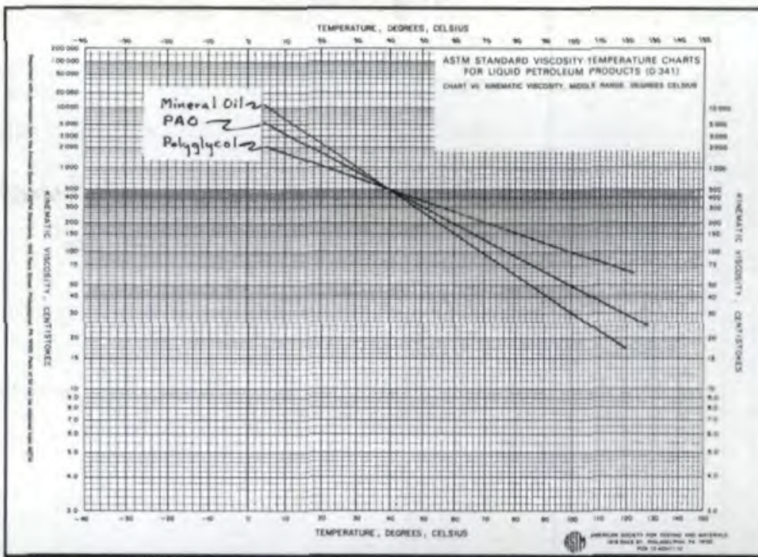



**Fig. 6 — ASTM standard viscosity-temperature charts for liquid petroleum products (D 341).**

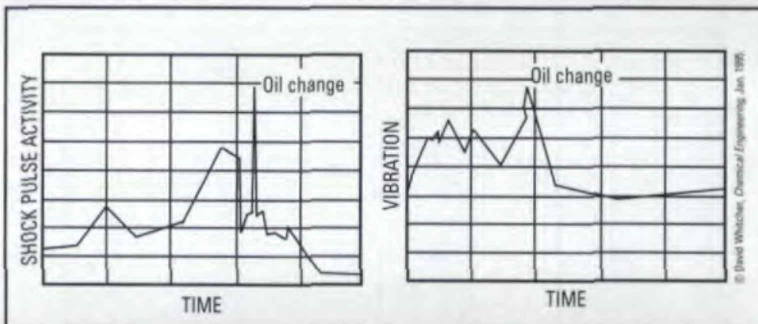



**Fig. 7 — Switching to a synthetic oil immediately reduces shock pulse activity and vibration.**

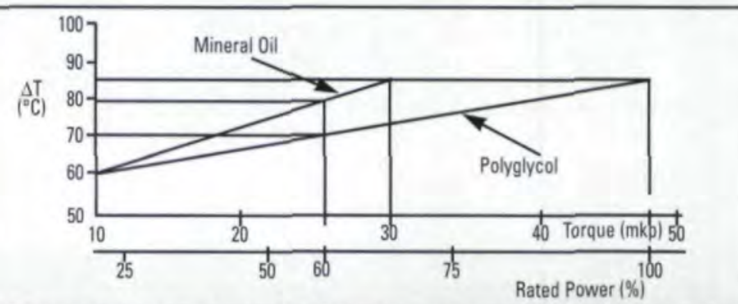



**Fig. 8 — Comparison of polyglycol vs. mineral oil in a worm drive gearbox.**

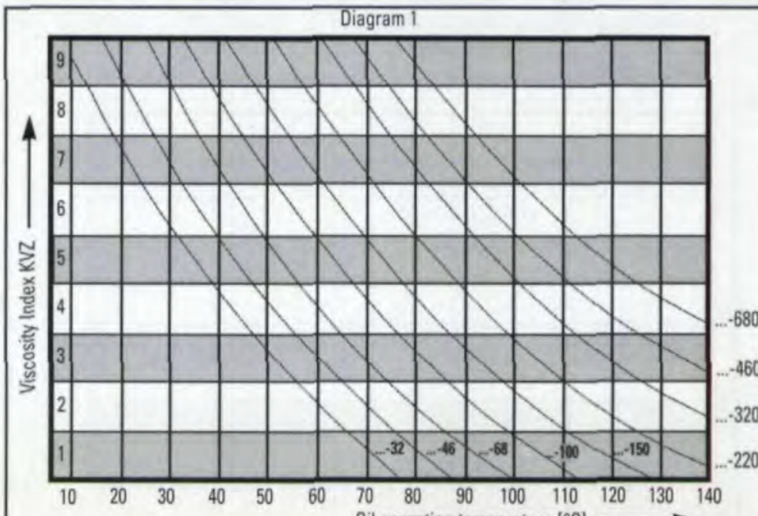



**Fig. 9 — PAO viscosity selection chart.**

**Table I — Determination of the Viscosity Index for a Spur Gear Drive**

| Force-speed factor $K_s/v \left[\frac{MPa \cdot s}{m}\right]$ | Viscosity Index KVZ |
|---|---|
| ≤0.02 | 1 |
| >0.02 to 0.08 | 2 |
| >0.08 to 0.3 | 3 |
| >0.3 to 0.8 | 4 |
| >0.8 to 1.8 | 5 |
| >1.8 to 3.5 | 6 |
| >3.5 to 7.0 | 7 |
| > 7.0 | 8 |

v = Peripheral speed at the reference circle [m/s]
$K_s$ = Rolling pressure acc. to Stribeck [N/mm²]

$$K_s = \frac{F_t}{b \cdot d_1} \cdot \frac{U+1}{U} \cdot Z_H^2 \cdot Z_\varepsilon^2 \cdot K_A \; [N/mm^2 \mathrel{\hat{=}} MPa]$$

$F_t$ = Nominal peripheral force [N]
b = Tooth width [mm]
$d_1$ = Diameter of reference circle [mm]
U = Gear ratio $Z_2/Z_1$
$Z_H$ = Distribution factor *1
$Z_\varepsilon$ = Contact ratio *1
$K_A$ = Application factor *2

*1 Note: Determination of $Z_H$ and $Z_\varepsilon$ according to DIN 3990 Pt 2.
For a rough calculation: $Z_H^2 \cdot Z_\varepsilon^2 = 3$.

*2 Note: Determination of $K_A$ according to DIN 3990 Pt. 1.
For a rough calculation: $K_A = 1$.

**Example 1**
*Single-stage spur gear driving a fan.*

| | |
|---|---|
| Drive | Electric Motor |
| Nominal peripheral force | $F_t$ = 3000N |
| Tooth width | b = 25 mm |
| Diameter of reference circle | $d_1$ = 230 mm |
| Gear ratio | U = 2.5 |
| $Z_H^2 \cdot Z_\varepsilon^2$ | ≈ 3 |
| $K_A$ | ≈ 1 |
| Peripheral speed | 4 m/s |
| Expected oil sump temperature | ≈ 90°C |
| Rolling pressure acc. to Stribeck | $K_s$ = 2.2 MPa |
| Force-speed factor | $K_s/v = 0.55 \frac{MPa \cdot s}{m}$ |
| Acc. to Table I, viscosity index | KVZ = 4 |

**Table II — Determination of the Viscosity Index for a Worm Gear Drive**

| Force-speed factor $K_s/v \left[\frac{N \cdot min}{m^2}\right]$ | Viscosity Index KVZ |
|---|---|
| ≤ 60 | 5 |
| >60 to 400 | 6 |
| >400 to 1800 | 7 |
| >1800 to 6000 | 8 |
| > 6000 | 9 |

$$\text{Force-speed factor } K_s/v = \frac{T_2}{n_1 \cdot a^3} \cdot f_1 \left[\frac{N \cdot min}{m^2}\right]$$

$T_2$ = Output moment [Nm] — $n_1$ = Worm speed [min⁻¹]
a = Center distance [m] — $f_1$ = Application factor

Note: The application factor is listed in the manufacturer's instructions.
For rough calculation: $f_1 = 1$.

**Example 2**
*Worm gear stage of a gearmotor driving a circular conveyor.*

| | | | |
|---|---|---|---|
| Output moment | $T_2$ = 250 Nm | Worm speed | $n_1$ = 350 rpm |
| Center distance | a = 0.063m | Application factor | $f_1$ = 1 |

$K_s/v = 2856.6 \frac{N \cdot min}{m^2}$

Expected oil sump temperature ≈ 85°C
Viscosity index acc. to Table II KVZ = 8

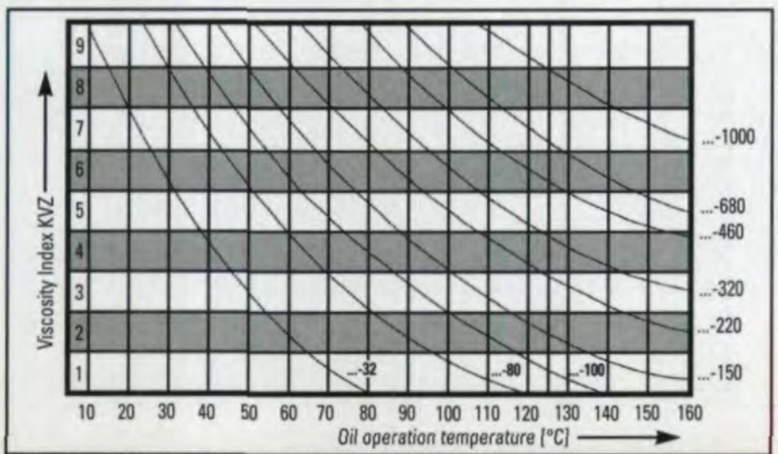



Fig. 10 — Polyglycol viscosity selection chart.

gear's thermal economy, taking into account the frictional losses, or in the case of gears already installed, by measuring the temperature of the sump. A lubricant with a lower viscosity might have to be chosen to assure that oil is supplied during a cold start or at lower ambient temperatures. In every case, it is necessary to check the viscosity at the existing starting temperature, especially in the case of oil circulation systems.

Tables I and II are typical worksheet methods for determining the viscosity for a spur gear drive and a worm gear drive. Table I and Example 1 apply to a typical spur gear drive situation; Table II and Example 2 to a typical worm drive situation. Once the KVZ is determined, Figs. 9 and 10 must be used to determine the correct ISO viscosity grade (VG) depending on the chemistry of the oil. Because of the different viscosity-temperature behavior of different oils, different ISO viscosity grades are selected for the same KVZ. In Example 1, an ISO-VG 220 would be selected for a PAO gear oil, and an ISO-VG 150 would be selected for a polyglycol. Conversely, in Example 2 an ISO-VG 680 PAO would be selected vs. an ISO-VG 460 polyglycol.

This article has only hit upon a few of the highlights about gear oil lubrication. The more informed gear drive operators are, the better decisions they can make concerning lubricant selection. They should locate a knowledgeable and reputable lubricant supplier and use this source for making important decisions that will affect energy consumption, machine life, lubricant consumption and waste oil generation. O

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