INTER PLANT STANDARDIZATION – STEEL INDUSTRY		
IPSS	**CODE OF PRACTICE FOR SELECTION OF INDUSTRIAL LUBRICANTS FOR STEEL PLANT APPLICATIONS** *(FIRST REVISION)*	**IPSS:1-09-012-11**
	CORRESPONDING IS DOES NOT EXIST	Formerly **IPSS:1-09-012-00**

0. FOREWORD

0.1 This Inter Plant Standard has been prepared by the Standards Committee on Oils & Lubricants, IPSS 1:9 with the active participation of the representatives of the steel plants major consultancy organizations and established manufacturers of industrial lubricants and was adopted in January 2011.

0.2 Inter Plant Standards for steel industry primarily aim at achieving rationalization and unification of parts and assemblies used in steel plant equipment and accessories, and provide guidance in indenting stores or equipment (or while placing orders for additional requirements) by individual steel plants. For exercising effective control on inventories, it is advisable to select a fewer number of sizes/types from among those mentioned in this standard, for the purpose of company standards of individual steel plants. It is not desirable to make deviations in technical requirements.

0.3 An integrated steel plant comprises four major groups (1) iron making (2) steel making (3) rolling and finishing and (4) auxiliary. The equipment and the working environment in each of these groups differ significantly and hence requirements of the lubricants used are also different.

0.3.1 The iron making group includes plants like Raw material bedding & blending yard, ore beneficiatiory plant, palletising plant, sinter plant, coke ovens and blast furnaces etc. The equipment in general are required to work under high temperature dusty (abrasive) and corrosive environment. The equipment used for crushing and screening applications are subjected to high impact as well. The speeds are lower.

0.3.2 The steelmaking group includes plants like Open Hearth and LD furnaces, Continuous Caster as well as Lime Calcining Plant, etc. The equipment are subjected to very high temperature, dusty (abrasive) and corrosive environment. The other working conditions in this group are similar to those in the iron making group.

0.3.3 The equipment in rolling and finishing group are subjected to high impact loads, water and scale ingress, high temperature near reheating furnaces and higher speeds and unit loads.

0.3.4 The auxiliary group comprises Engineering shops, Power plants, Oxygen plants, Loco and Mobile equipment and other facilities. The conditions in this group are generally moderate in nature.

Therefore, while selecting lubricant for steel plant equipment these factors must be taken into consideration. “Based on design, speed and load parameters of the equipment, OEM’S normally recommend the type and grade of the lubricant and cover the performance guarantee. The given code of practice applies for selection of lubricant in the absence of OEM’s recommendation or in case the recommended product is not performing satisfactorily”.

1. SCOPE

1.1 This code of practice provides guidelines for selecting proper lubricant to suit the application. For this purpose most commonly used tribology systems and the components like antifriction bearings, sleeve bearings and gears have been considered.

2. GENERAL

While selecting lubricant for an application two very important aspects, namely, the factors concerning engineering requirements and the factors concerning requirements pertaining to service condition must be considered. The majority of engineering requirements are met by selecting proper viscosity of the oil (called lubricant grade) while the service condition requirements are met by selecting proper additive package for the lubricant (called lubricant type).

The factors concerning engineering requirements include load, speed, surface finish of the interacting surfaces and rise in temperature etc. The viscosity of the lubricant at the working temperature, load and the relative velocity of the interacting surfaces have predominant effect in formation of thick film between them which is required to prevent direct contact between the surfaces. Hence, the first step for selection of lubricant is to determine the minimum viscosity at the working temperature of the system for the prevalent load and the relative velocity between the interacting surfaces. Then based on the viscosity index, the lubricant viscosity at the reference temperature of 40^{o}C is found out.

The problems posed by the service condition of the application is addressed by selecting a suitable additive system and the appropriate base oil for the lubricant. Different additive combinations are used for providing protection against rusting, oxidation, scuffing and corrosion, etc. They can also facilitate quick separation of water, formation of uniform and stable emulsion as well as facilitate collapse of foams whenever required. Suitable additive packages can provide between boundary lubrication, antiwear, friction, modification, detergency and discrepancy characteristics and ensure flowability of the fluid at sub-zero temperature.

In case the selected lubricant is intended for use where already some lubricant is in use, the compatibility of the two lubricants must be ensured. In case of doubt, old lubricant must be removed prior to using new lubricant.

Based on application and the additive system the steel plant lubricants are generally categorised under the following types which are used in different viscosity grades.

Types and grades of steel plant lubricants

- Hydraulic fluids, ISO-VG-32, 46, 68, 100, 150, 220, 320, 460
- Industrial gear oils, ISO-VG-100, 150, 220, 320, 460, 680, 1000
- Compressor oils, ISO-VG-32, 46, 68, 100, 150, 220, 320
- Film bearing oils, ISO-VG-32, 46, 68, 100, 150, 220, 320
- Compounded oils, ISO-VG-220, 320, 460, 680, 1000
- High demulsibility oils (with or without EP), ISO-VG-100, 150, 220, 320, 460, 680
- Turbine oil, ISO-VG-32, 46, 68
- Metal working fluids, mineral based, semi-synthetics and synthetics (heat type & emulsion type)
- Wire rope and open gear compounds
- Automotive crankcase oils, SAE-30, 40, 50, 15 W-40, 20 W-40
- Automotive transmission and hydraulic fluids, SAE-10W, 20, 30
- Automotive gear oils, SAE-90, 140
- Process oils, cold and hot rolling oils
- Mill greases, NLGI-00, 0, 1, 2, 3
- High temperature greases, NLGI-0,1, 2
- Multipurpose greases, NLGI-1,2,3
- Speciality lubricants, M_OS_2 or graphite based
- Preferably for fire prone area (Heat zone) – bio-degradable fire resistant hydraulic fluid HFDU-46 be used

3. SELECTION OF LUBRICANT FOR ANTIFRICTION BEARINGS

The antifriction bearings are generally lubricated with grease. However, for many critical applications as well as for the installations having a common oil circulation system these bearings need to be lubricated with oil. The guidelines for selecting grease or oil as a suitable lubricant for an antifriction bearing are presented in the following paragraphs.

3.1 Guidelines for selection between grease and oil for lubrication

Table-1 provides the guidelines for selection between grease or oil as the lubricant. The table is valid for Calcium, sodium, lithium (with or without EP), lithium complex, calcium complex, calcium-lithium complex and calcium sulfonate complex greases with mineral oil.

TABLE-1 : Grease / Oil selection guideline for antifriction bearings

Bearing type	Bearing Di mm	Limit of $D_m N$ factor for grease lubrication
Ball & cyl. Roller, needle roller brgs. Without inner race	Upto 50	350,000
	Over 50	$350{,}000/(Di/50)^{0.5}$
Tapered & spherical roller, needle roller with inner race	Upto 50	175,000
	Over 50	$175{,}000/(Di/50)^{0.5}$
Full complement cyl. Roller brg, multirow tapered roller and cyl. Bearing, thrust bearing	Upto 50	120,000
	Over 50	$120{,}000/(D_i/50)^{0.5}$

Di = Inner diameter of the bearing (mm), D_o = Outer diameter of the bearing (mm), $D_m = (D_i+D_o)/2$, N = RPM of the bearing

If the $D_m N$ factor is more than that indicated in Table-1, oil should be used for the lubrication of the bearing. However, speciality greases are available for the lubrication of antifriction bearings with higher $D_m N$ values which is beyond the scope of this code of practice.

3.2 Selection of oil viscosity for antifriction bearings.

The process involves determining the oil viscosity (ISO grade) and the additive system (oil type). The speed, load and working temperature of the system determine the oil viscosity for the most reliable operation. The viscosity at the operating temperature influences formation of full elasto-hydrodynamic film between the rolling element and the race way of the bearing. The rotational speed is by far the most important factor for evaluation of correct viscosity at the working temperature. The type of oil to be used is selected based on the requirements of the prevalent service conditions like instances of water ingress in the system, bulk and working temperature, speed and other factors promoting formation of foam, intensity of unit load, etc. The following empirical relationship should be used to evaluate the oil viscosity at the working temperature.

Viscosity at working temperature (cSt) = $K*14.8936*(\text{Max. catalogue speed / Actual speed})^{0.7}$

The value of k should be taken from Table-2.

TABLE – 2 : K factor

Bearing operating condition	Factor k
Normal operation, bearing equivalent load less than18% of the C-Rating	1.00
Normal operation, bearing equivalent load equal to or more than 18% of the C-Rating (heavily loaded bearing)	1.50
Shock and vibration, bearing load less than 18% of the C-Rating	1.50
Shock and vibration, bearing load equal to or more than 18% of the C-Rating	1.75
Very low speed (less than or equal to 10% of the maximum catalogue speed for oil lubrication)	2.00

After determining the oil viscosity at the working temperature, the viscosity at the reference temperature of 40^{o}C should be read from **Appendix-1**. The viscosity at the reference temperature of 40^{o}C should be upgraded to the next higher ISO-grade.

The following Table-2A provides the method of supply-NLGI Grade-Feed Rate :

TABLE – 2A

Method of supply	Recommended NLGI Grade	Feed Rate
Pressure gun or mechanical lubricators	Upto 3	$(4x10^{-2}xd\ cm^{3})/h$ where d = bearing diameter in m
Compression cups	Upto 5	
Centralised supply	Upto 2	
Hot running journals with open wells	6	
Air Pressure	0 to 2	

3.3 Guidelines for selection of type of oil for antifriction bearing

Table-3 provides guidelines for selection of lubricant type for the bearing

Table-3 : Lubricant type for antifriction bearings

Application area	Type of lubricant
Rolling mill equipment (oil bath, air oil system, oil circulation system)	Industrial gear oil with EP
Worm gear boxes (splash or circulation system)	Compounded oil or industrial gear oil with EP
Bevel, Spiral bevel, Spur, Helical gear boxes (splash or circulation system)	Industrial gear oil with EP
Any application with Air Oil System	Any oil up to ISO VG 680
Crushers, grinding, hammer, road and ball mills and similar applications	Industrial gear oil with EP
Compressors	Compressors oil, mineral hydraulic oil

3.4 Estimation of required flow rate

The following empirical relations are used to estimate the oil flow requirement through the bearing :

For spherical roller, angular contact, taper roller and thrust roller bearings:
Oil flow rate, q (litre per minute) = Antilog $((2^{*}\log D_o) - 3.602)$,
For all other bearings, q = Antilog $((2^{*}\log D_o) - 4.4)$
where D_o = Outer diameter of bearing (mm)

3.5 Guidelines for selection of grease type

The selection of grease type for antifriction bearing application depends upon the working temperature, requirement to resist water wash-out, speed of operation, environmental condition and intensity as well as type of load. Table-3 provides the guidelines for selection of grease type.

TABLE - 4 : Grease type (with mineral oil) for anti-friction bearing

Grease type (Thickener)	**Temperature Range**	**Stability for water wash-out**
Calcium soap (with/without EP)	-20 to 50	Very stable
Sodium soap (with/without EP)	-20 to 100	Unstable
Lithium soap (with/without EP)	-20 to 120	Stable upto 90^{o}C
Calcium complex soap	-20 to 130	Very stable
Lithium complex soap	-20 to 150	Stable
Aluminium complex soap	-20 to 150	Stable
Poly urea	-20 to 150	Stable
Calcium sulfonate complex	-20 to 180	Stable
Calcium lithium complex	-20 to 120	Very stable
Clay base grease (with M_oS_2)	-20 to 150	Stable

3.6 Guidelines for selection of base oil viscosity for grease

The base oil viscosity of the grease plays an important role in ensuring proper lubrication of the bearing. The shaft speed, working temperature and the bearing load determine the base oil viscosity of the grease for the best performance. The base oil viscosity for grease is selected on the same guidelines as those for the oil lubrication of the bearing (4.2).

3.7 Guidelines for selection of grease consistency

Table-5 provides the guidelines for selection of grease consistency for an application.

TABLE - 5 : Selection of grease consistency

Factors	**NLGI Grade**		
Bearing type	1	2	3
Roller & Ball bearings d<50mm	-	X	-
Roller & Ball bearings d>50mm	-	X	X
Needle roller bearings	-	X	-
Mounting and operating conditions			
Vertical or inclined position	-	X	X
Low starting torque	(x)	X	-
Good sealing	-	(x)	X
Good transportability	X	X	-
Low running noise	-	x	(x)

X = Suitable, (x) = Limited suitability

3.8 Guidelines for initial fill, relubrication interval and replenishment quality

For initial filling, the space between the rolling elements should be hand packed with clean grease and the bearing housing should be filled as per guidelines in Table-6.

TABLE - 6 : Initial fill of the bearing housing

Actual RPM / Max permissible RPM with grease	% Fill of housing
<0.2	80
0.2 to <0.5	50
0.5 to <0.8	30
0.8 to 1.0	10

The relubrication interval is governed by the working temperature and the replenishment quantity by the bearing dimension. The evaluation of replenishment interval and quantity is estimated by the following empirical relations :

Relubrication interval, T_1 (hour) = $[10^6 * K/(N * D_i^{0.5}) - c * D_i] * A$
Replenishment quantity, G(gram) = $D_o * B/200$
B (Bearing width) in mm, N in RPM, D_o = outer diameter of bearing in mm, D_i = inner diameter of the bearing in mm

Factors K and C are given in Table-7 and A in Table-8

TABLE-7 : Factors K and C

Bearing type	Series	K	C
Deep groove, self aligning, angular contact ball bearings and cylindrical roller bearings	Light	75	18
	Medium	64	18
	Heavy	53	18
Spherical roller and taper roller bearing Light & Medium	Light & Medium	21	7
Taper roller bearing	Heavy	19	7
Spherical roller bearing	Heavy	16	7
Thrust bearing	All types	21	7

TABLE – 8 : Temperature correction factor

Working temperature, ^oC	A
Upto 70	1.00
>70 to 80	0.70
>80 to 90	0.50
>90 to 100	0.35
>100 to 110	0.28
>110 to 120	0.20
>120 to 130	0.15
>130 to 140	0.10
>140 to 150	0.07

However, the roller bearings of all types with D_i>=300 mm are frequently lubricated with grease, and the feed rate of grease is estimated by the relationship:

G_k = D_o*B*10^{-4}gram/hour, D_o and B (Bearing width) in mm

For the applications prone to dust and water ingress the quantity. Calculated by equation is corrected by multiplying G_k by a factor of 1.75.

4. SELECTION OF LUBRICANT FOR SLEEVE BEARING

The sleeve or journal bearings are mostly lubricated with oil. However, for some applications grease is used which under some specified conditions can provide better reliability. Very low speed and high unit load on the bearing can often create the conditions of boundary or extreme boundary lubrication if oil is used as lubricant. Under such condition grease is found to give better results. The following guidelines are given for selection between oil and grease for the lubrication of journal bearing.

4.1 Selection of grease or oil as sleeve bearing lubricant

If the unit bearing load (P) is less than the limiting unit load (P_1) for the journal speed then grease can be used as lubricant. The first step is to calculate the limiting unit load for the prevalent journal speed which is given by the following empirical relation :

Limiting load for the bearing, $P_1(N/m^2)$ = (Antilog(-0.1761*logV+3.9542))*700
(P1 in N/m^2, V in mm/sec)
V = Surface of journal (mm/sec) = (π *N*D/60)*1000
Actual unit load on the bearing, $P(N/m^2)$ = W/(L*D)
Where W=load on the bearing (Newton)
L = bearing length, metre
D = Inner diameter of the bearing, metre
If $P < P_1$, then grease can be used as lubricant. Otherwise, oil should be used.

4.2 Estimation of oil Viscosity

The first step is to estimate the oil viscosity at working temperature. Then the oil viscosity at the reference temperature of 40^{o}C is evaluated. The evaluation of oil viscosity at the bearing working temperature is done by using Table-9 and the empirical relation given in this respect.

TABLE-9 : L/D ratio and S (Sommerfeld Number)

L/D ratio	0.25	0.5	0.75	1.0	1.5
S	1.0696	0.3161	0.752	0.1242	0.00859

For calculation of viscosity at working temperature the following steps are involved.

Step 1 : Take the value of S from Table-9 for the actual L/D ratio. For intermediate values interpolation should be done.

Step 2 : First approximation of viscosity (Z_t) = 60P*(C/D)^2*S/N
C is diametral clearnace of the bearing (metre)
N is journal RPM

Step 3 : First approximation of minimum film thickness
Minimum film thickness, h_{o1} (metre) = (a*C*(L/D)^{b1}*S$^{b2+b3L/D}$)*0.5
The values of a, b_1, b_2 and b_3 for a particular value of S is taken from Table-10.

TABLE-10 : Values of a, b1, b2 and b3

S <=0.15					**S >0.15**				
	a	**b_1**	**b_2**	**b_3**		**a**	**b_1**	**b_2**	**b_3**
S≤0.04	2.7258	0.83621	0.7510	0.0811	**S≤1**	0.91473	0.4538	0.6119	-0.2890
S<0.04	1.7176	1.0478	0.4999	0.1868	**S<1**	0.89574	0.3895	0.3076	-0.25370

If h_{o1} <55 x10^{-6} metre
Than take h_o (required minimum film thickness) = 55 x 10^{-6} metre
Otherwise h_o = h_{o1}

Step 4 : Calculate the actual value of S_a
$S_a = h_o*(D/L)^{b1}*(2/C)*(1/a)^{1/(b_2+b_3*L/D)}$
Required viscosity at working temperature,
$Z_t=S_a*P*(60/N)*(C/D)^2$ Pa.s
Viscosity in cSt = Z_t x 10^{-3}/0.9

Oil viscosity calculation is based on the working temperature. To calculate at 40 deg C, appendix-1 i.e. ISO viscosity calculation table can be used.
Then the oil viscosity at the reference temperature of 40^{o}C is estimated from Appendix-1. The viscosity at 40^{o}C should be upgraded to the next higher ISO grade.

4.3 Guidelines for selection of oil type for sleeve bearing

Table – 11 provides the guidelines for selection of oil for an application of sleeve bearing.

Table-11: Selection of oil type for sleeve bearing

Application	Oil type
Worm gear boxes	Industrial oil with EP, Compounded oil
Hypoid gear boxes	Automotive gear oil
Other gear boxes	Industrial oil with EP, R&O type oil
Fans and blowers	R&O type oil, hydraulic fluid, turbine oil
Turbine	Turbine oil
Automotive engine	Automotive crank case oil
Compressor	Compressor oil
Morgan bearing	High demulsibility oil with or without EP

5. GUIDELINES FOR SELECTION OF LUBRICANTS FOR THRUST BEARINGS

5.1 Plain Thrust Bearings

Plain thrust bearings are simple and occupy little axial space. Thrust bearings with radial grooves to encourage hydrodynamic action are suitable for light loads upto 0.5 MN/m^2 (75 lbf/in^2)

5.2 Profiled pad thrust bearings

The bearing comprises a ring of sector-shaped pads similar to plain thrust bearings. However, each pad is profiled so as to provide a convergent lubricant profile, which is necessary for the hydrodynamic generation of pressure within the film. Lubricants access to feed the pads is provided by oil ways which separate the individual pads. Rotation of the thrust runner in the direction of decreasing film thickness establishes the load carrying film. For bi-directional operation a convergent divergent profile must be used.

ISO VG 32, 46, 68, 100 oils are normally chosen so that with a lubricant inlet temperature of 50 degree C, the oil temperature rise across the bearing is 20 degrees C (max).

5.3 Tilting pad thrust bearings (Michell)

The tilting pad bearing is able to accommodate a large range of speed, load and viscosity conditions because the pads are pivotally supported and able to assume a small angle, relative to the removing collar surface. This enables a full hydrodynamic fluid film to be maintained between the surface of pad and collar.

Each pad must receive an adequate supply of oil at its entry edge to provide a continuous film and this is usually achieved by immersing the bearing in a flooded chamber. The oil is supplied at a pressure of 0.035 to 0.15 MN/m^2(5-22 lbf/in^2) and the outlet is restricted to control the flow.

Oil is circulated through the bearing to provide lubrication and to remove the heat resulting from the power loss. It is usual to supply oil at about 50 degrees C and to allow for a temperature rise through the bearing of about 17 degrees C. Complex design procedures have been established by the tilting pad thrust manufacturers to ensure optimum selection with minimum power loss.

Appropriate quality of ISO VG 32, 46, 68 and 100 oils are commonly applied.

There is some latitude in the choice of oil flow and temperature rise, but large deviations from these figures will affect the performance of the bearing.

5.4 Viscosity selection for thrust bearings

The minimum allowable viscosity at operating temperature for tilting pad or tapered land thrust bearings whose pads or lands have approximately the same radial width as their circumferential length, i.e. are approximately square, may be calculated as follows:

cP = 0.8 (P/U) (D/B)

Where cP = is the minimum viscosity at operating temperature

P is the mean bearing pressure in kg/cm^2
U is the mean circumferential speed of the thrust collar in m/sec.
D is the mean diameter of the assembly of pads or lands of the bearing in m
B is the radial dimension of the pads or lands in m.

An operating viscosity above the minimum is normally required and a minimum factor of safety of 2 is suggested. Moreover, in many cases the load on the bearing depends on the speed of operation. For example, with blowers and with shops' propellers the load increases approximately with the square of the speed.

Note 1: The nominal area of the load carrying lands or pads should not be more than 80% of the annular area (i.e. DB). This limitation is to allow sufficient space between the pads or lands for unrestricted flow of oil to their leading edges and to allow proper cooling.

Note 2: High speed bearings normally run completely immersed in an oil bath. It is necessary that the oil pressure should be sufficient to prevent the formation of vortices and accumulation of air bubbles between the pads or lands.

6. SELECTION OF LUBRICANT FOR GEARS

The type of gearing and enclosure, operating speed, load and temperature as well as the method of application determine the choice of lubricant for reliable service. The most commonly used lubricant is oil. However, grease, open gear compound and solid lubricants are also used in the applications where oil cannot be used.

The first step towards selection of a suitable lubricant is evaluation of oil viscosity which will ensure formation of the full elastohydrodynamic film between the mating pairs of the gear teeth. Then, the type of lubricant to suit the service condition is selected.

6.1 Selection of lubricant viscosity

The pitch line velocity of the gears is a good index of the required viscosity. The following empirical relation is used to determine the required viscosity :

Viscosity at $40^{o}C$ (cSt) = $498.92/V^{0.5}$, where V=pitch line velocity of gear or pinion in m/s.

6.2 **Application of lubricant -** Table-12 provides the guidelines for lubricant application.

TABLE-12 : Application of lubricant to gear train

Pitch line velocity (m/s)	Method of application	Remarks
Upto 16	Splash lubrication	Gear to dip in oil upto twice the tooth depth
More than 16 upto 26	Splash lubrication	Use baffles and oil pans to reduce churning
More than 26 upto 36	Pressure fed	Oil jets before the mesh approach
More than 36 upto 72	Pressure fed	Oil jets after the gears leave the mesh
More than 72	Pressure fed	Oil jets at both sides of the mesh, 2/3 rd of oil to be supplied to the outgoing side and 1/3 rd to the incoming side.

6.3 Lubricant flow rate

The following empirical relation gives the required flow rate of lubricant to the gear train :

q = pxc (litre/min) where p = transmitted power (kW), c is the factor given in Table-13.

TABLE –13 : Factor c

Application	lpm per kW	Flow condition
Heavy duty	0.0250	Copious
Moderate duty	0.0170	Adequate
Light duty	0.0125	Boundary lubrication

6.4 Selecting type of lubricant

Table-14 provides the guidelines for selecting the type of lubricant.

TABLE-14 : Selection of lubricant type

Application	Load condition	Lubricant type
Spur, helical, straight and spiral bevel gears	Light	R&O mineral oil
Spur, helical, straight and spiral bevel gears	Moderate & heavy	Industrial gear oilwith EP/Antiwear Hydraulic oils
Worm gears	Light & moderate	Compounded mineral oil with 3 to 10% fatty oil
Worm gears	Heavy	Industrial gear oil with EP provided there are no phospho-Bronze Components
Hypoid gears	Heavy	Automotive gear oil with active EP also Industrial EP Oils. VG 220/320/460

7. CONCLUSION

In conclusion, these guidelines are intended to provide help in selecting proper grade and type of lubricant for steel plant applications. The effectiveness of using these guidelines depends upon accurate evaluation of the basic engineering data and the operating parameters of the tribo-elements like bearings and gears, etc. It is intended that these guidelines should provide at least the initial means of selection of lubricant in absence of adequate experience or information. However, a note of caution is made for monitoring the actual field performance of the system if the lubricant selection is done based on these guidelines. It is suggested that necessary adjustments in the grade and the type of lubricant be made if required.

APPENDIX-1

ISO VISCOSITY CLASSIFICATION

ISO Viscosity Grade	Mid point Viscosity cSt at 40^oC	Kinematic viscosity limits cSt at 40^oC	
		Minimum	Maximum
ISO VG 2	2.2	1.98	2.42
ISO VG 3	3.2	2.88	3.52
ISO VG 5	4.6	4.14	5.06
ISO VG 7	6.8	6.12	7.48
ISO VG 10	10	9.00	11.00
ISO VG 15	15	13.50	16.50
ISO VG 22	22	19.80	24.20
ISO VG 32	32	28.80	35.20
ISO VG 46	46	41.40	50.60
ISO VG 68	68	61.20	74.80
ISO VG 100	100	90.00	110
ISO VG 150	150	135	165
ISO VG 220	220	198	242
ISO VG 320	320	288	352
ISO VG 460	460	414	506
ISO VG 680	680	612	748
ISO VG 1000	1000	900	1100
ISO VG 1500	1500	1350	1650

APPENDIX-2

NLGI STANDARD LUBRICATING GREASE CLASSIFICATION

This classification was developed by the National Lubricating Grease Institute of USA. It identifies a succession of grades defined as ranges of the values of the 60-stroke worked penetration at 25^{o}C as grades determined by ASTM Designation D217, “Cone penetration of lubricating grease”, the original method being applicable to grades 0 to 6 and the alternative method to grades 00 and 000. The series is as follows :

NLGI GRADE NO.	60-STROKE WORKED PENETRATION AT 25^{o}C ASTM D217
000	445-475
00	400-430
0	355-385
1	310-340
2	265-295
3	220-250
4	175-205
5	130-160
6	85-115

APPENDIX-3

ISO VISCOSITY CLASSIFICATION WITH CORRESPONDING KINEMATIC VISCOSITIES AT VARIOUS TEMPERATURES FOR DIFFERING VISCOSITY INDICES

ISO VISCOSITY GRADE	KINEMATIC VISCOSITY RANGE cSt at 40°C	Approximate Kinematic Viscosity at other temperatures for different values of viscosity index								
		Viscosity index = 0			Viscosity index = 50			Viscosity index = 95		
		cSt at 20°C	cSt at 37.8°C	cSt at 50°C	cSt at 20°C	cSt at 37.8°C	cSt at 50°C	cSt at 20°C	Cst at 37.8°C	cSt at 50°C
ISO VG 2	1.98-2.42	(2.82-3.67)	(2.05-2.52)	(1.69-2.03)	(2.87-3.69)	(2.05-2.52)	(1.69-2.03)	(2.92-3.71)	(2.06-2.52)	(1.69-2.03)
ISO VG 3	2.88-3.52	(4.60-5.99)	(3.02-3.71)	(2.37-2.83)	(4.59-5.92)	(3.02-3.70)	(2.38-2.84)	(4.58-5.83)	(3.01-3.69)	(2.39-2.86)
ISO VG 5	4.14-5.06	(7.39-9.60)	(4.38-5.38)	(3.27-3.91)	(7.25-9.35)	(4.37-5.37)	(3.29-3.95)	(7.09-9.03)	(4.36-5.35)	(3.32-3.99)
ISO VG 7	6.12-7.48	(12.30-16.00)	(6.55-8.05)	(4.63-5.52)	(11.90-15.30)	(6.52-8.01)	(4.68-5.61)	(11.40-14.40)	(6.50-7.98)	(4.76-5.72)
ISO VG 10	9.00-11.00	20.20-25.90	9.73-12.00	6.53-7.83	19.10-24.50	9.63-11.90	6.65-7.99	18.10-23.10	9.60-11.80	6.78-8.14
ISO VG 15	13.50-16.50	30.50-43.00	14.70-18.10	9.43-11.3	31.60-40.60	14.70-18.00	9.62-11.50	29.80-38.30	14.60-17.90	9.80-11.80
ISO VG 22	19.80-24.20	54.20-69.80	21.80-26.80	13.30-16.00	51.00-65.80	21.70-26.60	13.60-16.30	48.00-61.70	21.60-26.50	13.90-16.60
ISO VG 32	28.80-35.20	87.70-115	32.00-39.40	18.60-22.20	82.60-108	31.90-39.20	19.00-22.60	76.90-98.70	31.70-38.90	19.40-23.30
ISO VG 46	41.40-50.60	144-189	46.60-57.40	25.50-30.30	133-172	46.30-56.90	26.10-31.30	120-153	45.90-56.30	27.00-32.50
ISO VG 68	61.20-74.80	242-315	69.80-85.80	35.90-42.80	219-283	69.20-85.00	37.10-44.40	193-244	68.40-83.90	38.70-46.60
ISO VG 100	90.00-110.00	402-520	104-127	50.40-60.30	356-454	103-126	52.40-63.00	303-383	101-124	55.30-66.60
ISO VG 150	135-165	672-862	157-194	72.50-86.90	583-743	155-191	75.90-91.20	486-614	153-188	80.60-97.10
ISO VG 220	198-242	1080-1390	233-286	102-123	927-1180	230-282	108-129	761-964	226-277	115-138
ISO VG 320	288-352	1720-2210	341-419	144-172	1460-1870	337-414	151-182	1180-1500	331-406	163-196
ISO VG 460	414-506	2700-3480	495-608	199-239	2290-2930	458-699	210-252	1810-2300	478-587	228-274
ISO VG 680	612-748	4420-5680	739-908	283-339	3700-4740	728-894	300-360	2880-3650	712-874	326-393
ISO VG 1000	900-1100	7170-9230	1100-1350	400-479	5960-7640	1080-1330	425-509	4550-5780	1050-1290	466-560
ISO VG 1500	1350-1650	11900-15400	1600-2040	575-688	9850-12600	1640-2010	613-734	7390-9400	1590-1960	676-812

NOTE: Values in parentheses have been derived by extrapolation and are approximate.

APPENDIX-4
VISCOSITY CONVERSION TABLE
The following table is for the conversion of viscosities in one system to those in another system at the same temperature.

Kinematic viscosity cSt	*Engler degree	Redwood No.1 seconds	Saybolt universal seconds		Kinematic viscosity cSt	*Engler degree	Redwood No.1 seconds	Saybolt universal seconds
1.0	1.00	28.5			20.0	2.9	86	97.5
1.5	1.06	30			20.5	2.95	88	99.6
2.0	1.12	31	32.6		21.0	3.0	90	101.7
2.5	1.17	32	34.4		21.5	3.05	92	103.9
3.0	1.22	33	36.0		22.0	3.1	93	106.0
3.5	1.26	34.5	37.6		22.5	3.15	95	108.2
4.0	1.30	35.5	39.1		23.0	3.2	97	110.3
4.5	1.35	37	40.7		23.5	3.3	99	112.4
5.0	1.40	38	42.3		24.0	3.35	101	114.6
5.5	1.44	39.5	43.9		24.5	3.4	103	116.8
6.0	1.48	41	45.5		25	3.45	105	118.9
6.5	1.52	42	47.1		26	3.6	109	123.2
7.0	1.56	43.5	48.7		27	3.7	113	127.7
7.5	1.60	45	50.3		28	3.85	117	132.1
8.0	1.65	46	52.0		29	3.95	121	136.5
8.5	1.70	47.5	53.7		30	4.1	125	140.9
9.0	1.75	49	55.4		31	4.2	129	145.3
9.5	1.79	50.5	57.1		32	4.35	133	149.7
10.0	1.83	52	58.8		33	4.45	136	154.2
10.2	1.85	52.5	59.5		34	4.6	140	158.7
10.4	1.87	53	60.2		35	4.7	144	163.2
10.6	1.89	53.5	60.9		36	4.85	148	167.7
10.8	1.91	54.5	61.6		37	4.95	152	172.2
11.0	1.93	55	62.3		38	5.1	156	176.7
11.4	1.97	56	63.7		39	5.2	160	181.2
11.8	2.00	57.5	65.2		40	5.35	164	185.7
12.2	2.04	59	66.66		41	5.45	168	190.2
12.6	2.08	60	68.1		42	5.6	172	194.7
13.0	2.12	61	69.6		43	5.75	177	199.2
13.5	2.17	63	71.5		44	5.85	181	203.8
14.0	2.22	64.5	73.4		45	6.0	185	208.4
14.5	2.27	66	75.3		46	6.1	189	213.0
15.0	2.32	68	77.2		47	6.25	193	217.6
15.5	2.38	70	79.2		48	6.45	197	222.2
16.0	2.43	71.5	81.1		49	6.5	201	226.8
16.5	2.50	73	83.1		50	6.65	205	231.4
17.0	2.55	75	85.1		52	6.9	213	240.6
17.5	2.60	77	87.1		54	7.1	221	249.9
18.0	2.65	78.5	89.2		56	7.4	229	259.0
18.5	2.70	80	91.2		58	7.65	237	268.2
19.0	2.75	82	93.3		60	7.9	245	277.4
19.5	2.80	84	95.4		70	9.2	285	323.4

Note: The first part of the table marked with an asterisk (*) should only be used for the conversion of Minematic viscosities into viscosities Engler, Redwood or Saybolt, or for Engler, Redwood and Saybolt between themselves. They should not be used for conversion of Engler, Redwood or Saybolt into Kinematic viscosities.

For higher viscosities the following factors should be used :

Engler = 0.132 Kinematic	Kinematic = 7.58 Engler	Kinematic = 0.247 Redwood	Kinematic = 0.216 Saybolt
Redwood = 4.05 Kinematic	Redwood = 30.7 Engler	Engler = 0.0326 Redwood	Engler = 0.0285 Saybolt
Saybolt = 4.62 Kinematic	Saybolt = 35.11 Engler	Saybolt = 1.14 Redwood	Redwood = 0.887 Saybolt

APPENDIX-5

VISCOSITY TEMPERATURE CHART