

Chapter 2

Background on Viscosity and Lubrication

This chapter introduces the basic concepts of viscosity and lubrication. The optimization of the lubricant viscosity plays a crucial role in preventing wear mechanisms in engineering components. The chapter starts with the basic definition of viscosity and the relation of viscosity with temperature, load and shear rate. This is followed by an overview of the current conventional methods to measure viscosity and on the composition of engine lubricant. Subsequently the different lubrication regimes are described with particular attention to the analysis of different lubrication stages in a journal bearing, the case study in this research.

2.1 Definition of Viscosity

Figure 2.1 schematically shows the contact of two solid components in relative motion separated by a fluid layer. For this situation, the relation between force F applied to the “Component 2”, the relative velocity gradient u and fluid film thickness h is:

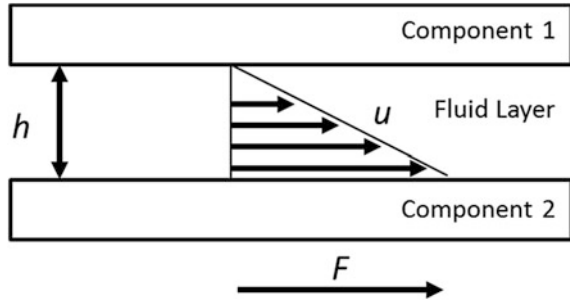
$$F = \eta A \frac{u}{h} \quad (2.1)$$

Where A is the wet surface area and the proportionality constant η is called viscosity.

Shear viscosity therefore defines the resistance offered by a fluid to an applied shear stress. Rearranging Eq. (2.1) it is possible to obtain the following definition for the viscosity:

$$\eta = \frac{F}{A} \bigg/ \frac{u}{h} \quad (2.2)$$

Fig. 2.1 Schematic representation of a lubricated contact with two components in relative motion



and consequently:

$$\eta = \frac{\sigma_y}{\dot{\gamma}} \quad (2.3)$$

where σ_y is the shear stress and $\dot{\gamma}$ is the shear rate. When expressed in Pas, the shear viscosity is referred to as dynamic viscosity, measured in P (Poiseuille) where $1\text{P} = 0.1\text{ Pas}$. The viscosity can be also calculated as kinematic viscosity when the dynamic viscosity is divided by the density of the oil:

$$\nu = \frac{\eta}{\rho} \quad (2.4)$$

where ν is the kinematic viscosity. The kinematic viscosity is measured in St (Stokes). Kinematic viscosity is used to quantify the resistance to the fluid flow. Fluid viscosity is sensitive to thermodynamic and stress changes in the lubricated system and especially to temperature, pressure and shear rate variations.

2.1.1 Viscosity Relation with Temperature

The lubricant viscosity decreases with an increment in temperature because the particles dispersed in the fluid will increase their oscillatory speed and atomic bonds become weaker thus offering an overall less resistance to motion. The relation between viscosity and temperature is described by several empirical laws, the most precise of which being the Vogel equation (Crouch and Cameron 1961):

$$\eta = ae^{\frac{b}{T-c}} \quad (2.5)$$

In this equation a , b and c are constants defined by knowing three values of viscosity at three known temperatures and T is the temperature at which the viscosity is measured.

2.1.2 Viscosity Index

The viscosity index describes how oil viscosity changes with the temperature. This is an empirical parameter that was introduced by Dean and Davis (1929) and is measured as:

$$VI = \frac{L - U}{L - H} \quad (2.6)$$

Where L is the viscosity at 40 °C of an oil of 0 VI having the same viscosity at 100 °C of the oil whose VI has to be calculated, H is the viscosity at 40 °C of an oil of 100 VI having the same viscosity at 100 °C as the oil whose VI has to be calculated, U is the viscosity at 40°C of the oil whose VI has to be calculated. The values of L and H are tabled in the ASTM standard D2270 (1998). The viscosity index is one of the most important in the classification of lubricants. Figure 2.2 compares the viscosity index for base oils and polymer oils with high and low VI.

It can be noticed that high VI guarantees less changes in lubricant viscosity with temperature thus incrementing the stability of the oil in high temperature environments. This is the optimum situation for protection and durability of engine parts, while a lower VI is characteristic of a less stable oil ideal for less friction and higher performances.

2.1.3 Viscosity and Pressure

An increment in pressure produces an increment in viscosity because the fluid particles tend to pack-up increasing atomic bond strength and offering a higher resistance to motion. The pressure-viscosity relation is described by the Barus equation (Barus 1893; Cameron 1981):

Fig. 2.2 Viscosity index for different oil types

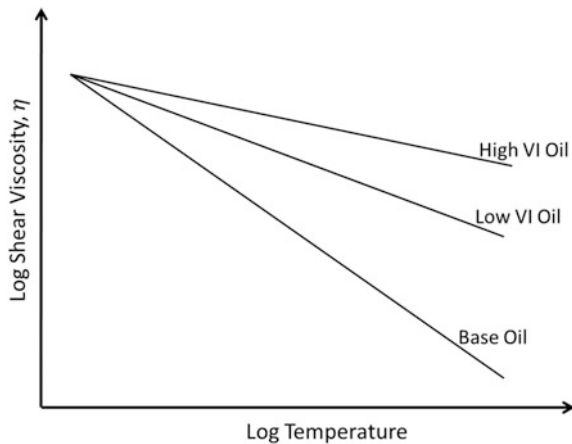


Table 2.1 Pressure-viscosity coefficient for different oils

Lubricant	Pressure-viscosity coefficient α at 30 °C (GPa ⁻¹)
Water	0.8
Castor oil	360
Cylinder oil	810
Heavy machine oil	170
Spindle oil	30.7

$$\eta = \eta_0 e^{\alpha P} \quad (2.7)$$

where η is the viscosity at the pressure P , η_0 is the viscosity at ambient pressure and α is the pressure viscosity coefficient. This value varies with the oil formulation. Table 2.1 reports common values of Pressure viscosity coefficient for different type of fluid and lubricants (as reported by Stachowiack and Batchelor 2001).

Equation (2.7) shows that when the pressure increases the viscosity increments exponentially because the fluid particles tend to pack-up offering a much higher resistance to motion. In conformal contacts the contact area are large and the increment in viscosity due to pressure increment is low, as in the case of journal bearings. In non-conformal contacts, on the other hand, the contact areas are small and the pressure can rise up to 1–3 GPa bringing a significant increment in viscosity and to localised fluid solidification.

2.1.4 Viscosity and Shear Rate

In perfectly Newtonian fluids the shear stress is proportional to the shear rate through the proportionality constant shear viscosity, as shown in Eq. (2.3). This means that for purely Newtonian oils viscosity is not dependent on shear rate. In practice perfectly Newtonian fluids do not exist at the very high shear rates encountered in engine machinery, and oils with high additive concentrations or complex molecular structure show Non-Newtonian behaviour. For these lubricants the relation between stress and viscosity is non-linear. The relation of Non-Newtonian oils viscosity with shear rate is quite complex and various mechanisms are involved. The main categories of Non-Newtonian fluids are the pseudoplastic, or shear thinning, and dilatant, or shear thickening. For the first category, the viscosity decreases as the shear rate increases because the random positioned molecules tend to align thus offering less resistance to the shear, as shown in Fig. 2.3a. On the other hand, shear thickening oils have suspensions of polymer particles with high solid content. When there is no shear applied the solid particle will have a closed-up formation. When shear is applied the voids between the solid particles expand in way that not enough liquid can fill them. This results in an

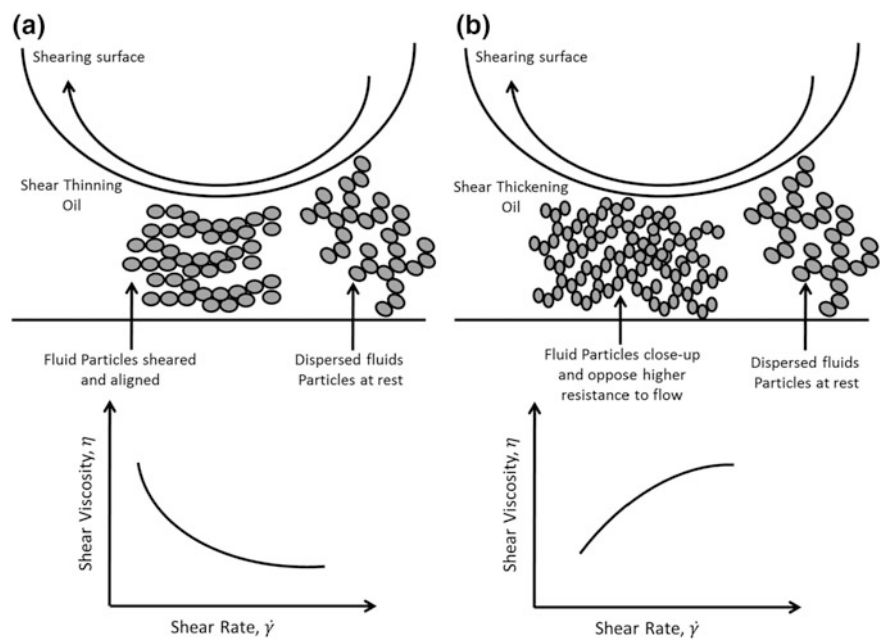


Fig. 2.3 a Shear thinning oil, b shear thickening oil

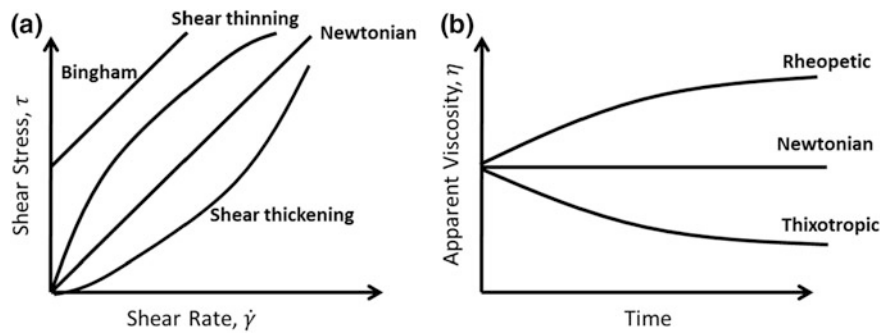


Fig. 2.4 a viscosity categories of shear dependent fluids, b viscosity categories of time dependent fluids

increment to the resistance to shear applied as shown in Fig. 2.3b. Another category of shear dependent fluid is constituted by the Bingham fluid. These fluids “mimic” solid behaviour at low shear and start flowing only after a sufficient stress is applied. The minimum stress at which Bingham fluids flow is called the yield stress. Figure 2.4a shows the different shear rate-shear stress curves for the different strain-dependent fluid type described in this chapter.

Figure 2.4a shows that the shear thinning and shear thickening behaviour can depend on the duration of the applied shear stress. The rupture of the molecular bonds, if time dependent, brings to different measured apparent viscosities. The time dependent shear thinning behaviour is called thixotropic behaviour, on the other hand, shear duration dependent thickening is known as rheopectic behaviour.

In this work shear thinning fluids, not duration dependent, are studied. The most common shear dependent viscosity models are the power law and the Carreau-Yasuda model (Yasuda 1981). The power-law model is the simplest Non-Newtonian model:

$$\eta = K|\dot{\gamma}|^{n-1} \quad (2.8)$$

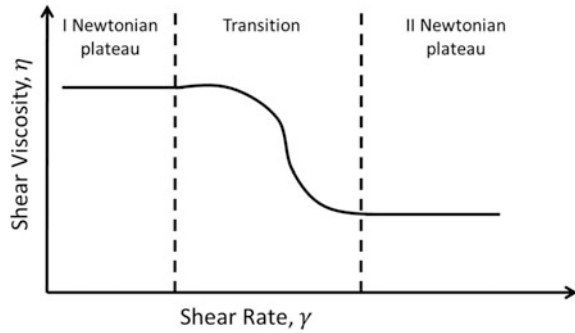
where K is the flow consistency index, n is a natural integer and $\dot{\gamma}$ is the shear rate. It is possible to notice that when $n = 1$ the model is equivalent to the one for Newtonian fluid. When $n > 1$ the model describes shear thickening fluids, while when $n < 1$ the model describes shear thinning fluids. The model presented by Carreau (1972) is given by:

$$\eta = \eta_{\infty} + (\eta_0 - \eta_{\infty}) \left(1 + (K|\dot{\gamma}|)^2 \right)^{\frac{n-1}{2}} \quad (2.9)$$

where η_{∞} is the viscosity at infinite shear rate, η_0 is the viscosity at zero shear rate and a is a constant that for most of the shear thinning fluid is around 2. This shear rate dependent model is more precise than the simple power-law model for the way it fits the data at low and high shear. In practice measurements of η_{∞} are very difficult so this model is used in numerical flow simulations rather than as an analytical tool.

Engine lubricating oils behave commonly in a manner as shown in Fig. 2.3 a. It should be noticed that the oils do not shear thin until the viscosity drop to zero, which is physically impossible. In fact the shear thinning behaviour in many lubricants is isolated to a certain range of shear rates. Figure 2.5 reports the shear rate-viscosity graph that is typical of an engine oil. Three regions can be identified. At low shear rates the oil behaves like a Newtonian fluid, so the viscosity is constant even with a change in shear rate. In the second region the viscosity is dependent on the shear rate and the oil shear thins. Finally in the third region, at very high shear rates, the viscosity is an independent function of shear rate. This region is called the second Newtonian plateau or second Newtonian viscosity. It is important to take into consideration the possibility of existence of a second Newtonian plateau when designing engine oils because the second and third regions in Fig. 2.5 are the operating regions of lubricants in engines. For instance, two lubricants that show the same viscosity at low shear might provide very different performances at the engine operative conditions. A technique to measure oil viscosity at very high shear rate is presented in this work in Chap. 7.

Fig. 2.5 Typical shear rate-shear viscosity curve for engine oils



2.2 Viscosity Measurement

Instruments that measure viscosity are called viscometers or rheometers depending if one or more flow conditions can be analysed. Viscometers can be mechanical, when a body applies a shear to the fluid or vibrational when the fluid is sheared by a vibration. When a mechanical shear is applied, viscosity is commonly measured with rotational viscometers. Another measurement principle is constituted by the vibrational viscometers. These viscometers are usually made of a sensor that vibrates thus displacing shearing the fluid at the interface. Another viscometer type is Electromagnetic Spinning Sphere that correlates the magnetic field necessary to displace a ball immersed in a fluid to the fluid viscosity. Figure 2.6 shows schematically the main viscometers type.

A wide number of viscometers exist and it would be impossible, and out of the purpose of this research, to enumerate and describe them all in this chapter. In this section the main mechanical shear type viscometers are analysed to give the reader the perspective of some of the options the market offers to measure viscosity with conventional systems (Sects. 2.2.1–2.2.3). Along with this, the main commercial type of vibrational viscometers (Sect. 2.2.4) are described. In this research a vibrational viscometer is developed and the results are compared with the reading from a conventional mechanical shear rheometer. Finally, high pressure and high shear viscometry is described in relation to tribological and automotive applicability.

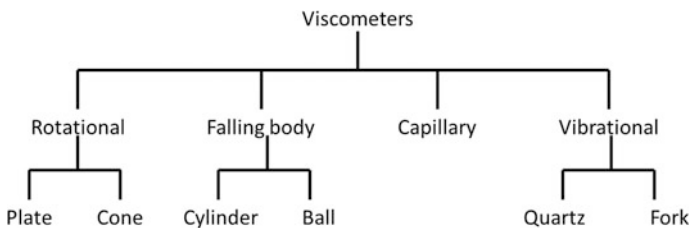


Fig. 2.6 Different viscometers type

2.2.1 Capillary Viscometers

Capillary viscometers measure viscosity by correlating the time a fluid sample takes to travel through a capillary with the kinematic viscosity of the sample. The higher the fluid viscosity, the longer the time needed by the sample to travel through the capillary. Figure 2.7 shows the scheme of capillary viscometers.

Kinematic viscosity in capillary viscometers is measured as (Webster 1999):

$$\eta = \left(\frac{4n(\Delta P D^4 \pi)}{(3n + 1)(128QL)} \right) \quad (2.10)$$

where ΔP is the difference in pressure at the inlet and outlet of the viscometer, D is the diameter of the capillary, Q is the fluid volume flow rate and L is the length of the capillary.

2.2.2 Rotational Viscometers

Rotational viscometers are used to measure dynamic viscosity. In this type of viscometer the fluid is placed between two surfaces, one stationary, while the second one rotates. The viscosity measurement is either done by measuring the change in rotational speed given a fixed torque or by measuring the torque change given a constant rotational speed. The main rotational viscometers are cylinder or

Fig. 2.7 Schematic representation of a capillary viscometer

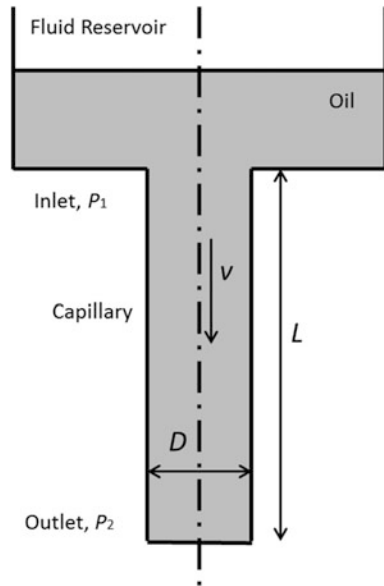
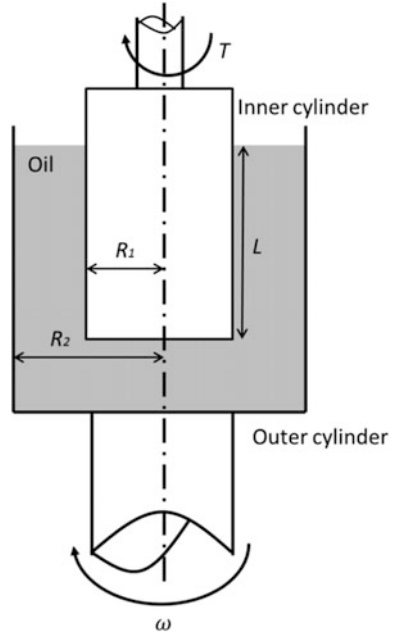


Fig. 2.8 Schematic representation of a cylinder rotational viscometer



cone-on-plate. Figure 2.8 reports the typical setup for a rotating cylinder viscometer. Here the gap between two concentric cylinders is filled with the fluid sample. The inner cylinder rotates thus shearing the contact interface between solid and liquid.

The viscosity is measured as follows (Webster 1999):

$$\eta = \left(\frac{\beta^2 - 1}{4\pi R_2^2 L} \right) \frac{T}{\omega} \quad (2.11)$$

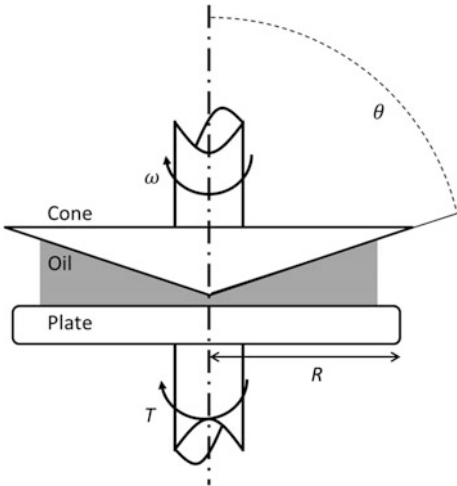
where β is a parameter dependent on the cylinder geometry, ω is the rotational speed, T is the applied torque and L is the effective length of the cylinder at which the torque is measured. Figure 2.9 reports the setup for a cone-to-plate viscometer. The operating principle is the same as in the rotating cylinder, but in this case the rotating part is cone shaped. This type of viscometer is the most used on the market as just a small drop of fluid is needed and very thin films can be analysed.

In cone type viscometers the viscosity is measured as:

$$\eta = \frac{3T\theta_0}{2\pi\omega R^3} \quad (2.12)$$

where θ_0 is the cone angle of incidence and R is the plate radius. The shear rate generated in rotational viscometer varies from $10^0 - 10^4 s^{-1}$ in conventional viscometers to $10^6 - 10^7 s^{-1}$ in recently developed ultra-high shear viscometers

Fig. 2.9 Schematic representation of a cone-on-plate viscometer



(UHSV). The same measurement principle is adopted in other viscometers such as the plate-to-plate viscometer.

2.2.3 Falling Body Viscometers

Falling body viscometers measure viscosity by correlating the time a body takes to travel through a capillary with the resistance offered by the fluid and so to the viscosity. Figure 2.10 reports two examples of falling body viscometers.

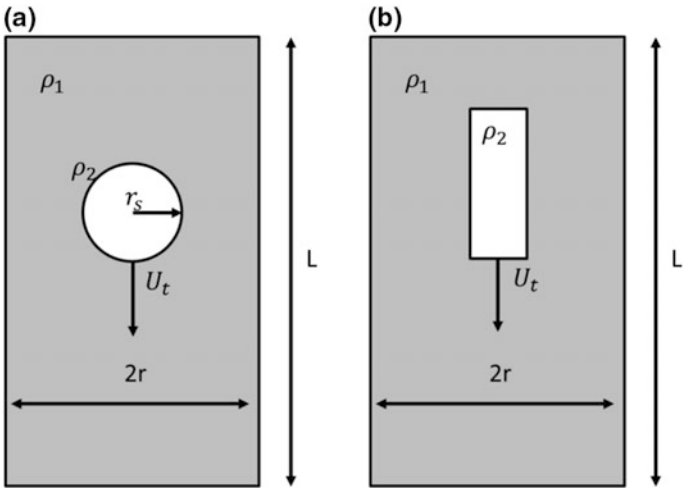


Fig. 2.10 **a** Falling sphere viscometer, **b** falling cylinder viscometer

Figure 2.8a shows schematically a falling ball viscometer, Fig. 2.8b shows a falling cylinder viscometers. In this section only the case of the falling body is analysed, but similar consideration can be made to obtain the value of viscosity from the falling cylinder viscometer type. The restraining force F resulting from the viscous drag in a fluid for a spherical body is calculated from the Stokes law as:

$$F = 6\pi\eta r_s U_t \quad (2.13)$$

where r_s is the radius of the sphere and U_t is the terminal velocity of the falling body. By balancing Eq. (2.13) with the buoyancy force exerted on a ball then viscosity can be calculated as:

$$\eta = 2gr_s^2 \frac{(\rho_2 - \rho_1)}{9U_t} \quad (2.14)$$

More accurate equations can be obtained by taking into consideration the wall effect and the finite length of the capillary.

2.2.4 Vibrational Viscometers

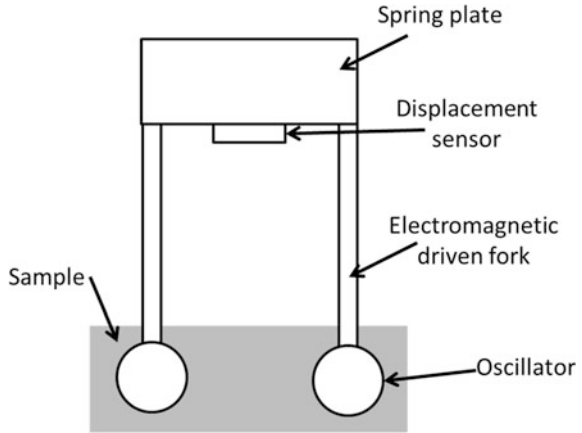
Different types of vibrational viscometers exist. The rotational and the turning-fork viscometer measure the viscosity by determining the amount of power necessary to displace the fluid of a certain quantity. The quartz resonators measure the change in resonant driving frequency when an oscillating quartz crystal is submerged in a liquid. The sonic and ultrasonic viscometer measures the reflection of sound at a solid-liquid interface and correlate the amount of reflected energy to the viscosity of the fluid samples. In this section the rotational and the turning fork viscometer are described, while the quartz resonator is briefly described in Sect. 4.1. The ultrasound viscometer is described in depth in chapter 3, 4 and 5 and the applications of this novel technique are described in Chaps. 6–8.

Figure 2.11 shows schematically a turning fork viscometer.

The rotational viscometer operates similarly, but in that case only a cantilever beam is present. The measuring system is driven by electromagnetic power and the natural frequency of the system is determined by the mass and spring constant of the measuring system. The energy consumed by the measuring system will only be the viscous term of the liquid because the inertial force and the restorative force of the spring balance each other. The motion equation of the system is:

$$F = m \frac{d^2x}{dt^2} + c \frac{dx}{dt} + Kx \quad (2.15)$$

Fig. 2.11 Schematic representation of a vibrational fork viscometer



where F is the excitation force, m is the mass of the system c is the viscosity-density coefficient, K is the spring constant and x is the displacement of the system. After integration, Eq. (2.15) gives:

$$x = \frac{F}{c\omega_n} \quad (2.16)$$

where ω_n is the natural vibrational frequency of the system. The coefficient c is calculated by setting x and ω_n as constant.

2.2.5 High Pressure Viscometers

The viscometers described in Sects. 2.2.1–2.2.4 can be modified to work as pressure viscometers to study the pressure-viscosity characteristic of engine oils. Basically, the viscometer is enclosed in a pressurized vessel where the oil to be study is brought to the desired pressure normally using pressure intensifiers. Bair (2007) enumerated and described the characteristic of the most common high pressure viscometers:

- **High pressure capillary viscometer.** the first attempt of pressurizing the capillary viscometers was made by Warburg and Sachs (1884), but the maximum performance (at the current state of the art) were achieved by Novak and Winer (1968) who managed to obtain viscosity at 0.6 GPa at the temperature of 150 °C. Figure 2.12 shows the viscometer.

The high pressure is obtained with two pressure intensifiers. The capillary of the viscometer is placed in between the pressure intensifiers and the pressure is measured with strain gage transducers.

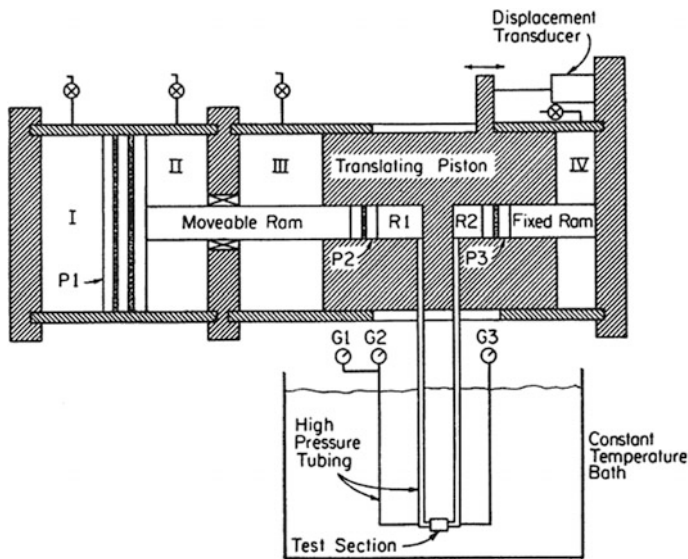
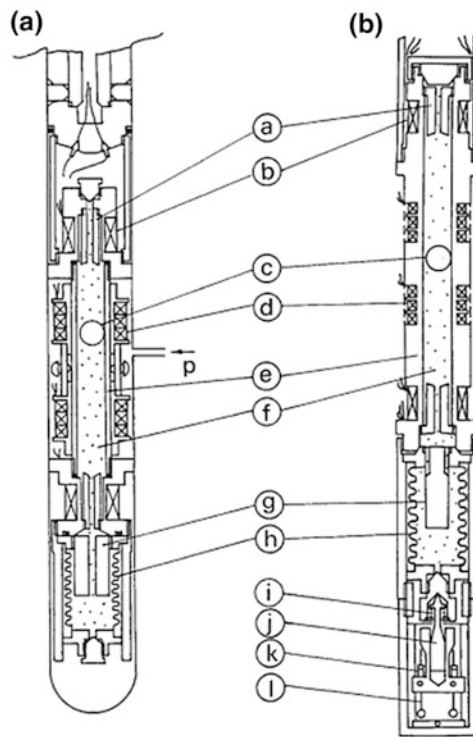


Fig. 2.12 The Novak and Winer high pressure capillary viscometer. R1 and R2 are the translating pistons the displace the fluid and P1 is the piston that determine the pressure at which the fluid is measured (Bair 2007)

- **High pressure dropping body viscometer.** These are divided in falling ball and falling cylinder viscometers. The high pressure falling ball viscometer consists of a falling ball capillary inserted in a pressure vessel. The vessel pressurizes the tube externally and internally. Figure 2.13 shows the high pressure ball viscometer by Sawamura et al. (1990).

In this set-up, the light from a lamp is refracted from the glass ball across a pair of sapphire windows. This allows measurement of the falling time of the ball. This type of viscometer, however, presents some limitations such as complex flow and a limited range of viscosity that can be studied. The falling cylinder viscometer by Irving and Barlow (1971) provided considerable improvement because the sinker does not have to travel along all the length of the capillary. This is because the sinker position is detected monitoring the inductance of different coils that are positioned around the viscometer. This allows using any desirable length of the capillary for the measurement. Figure 2.14 shows the high pressure falling cylinder viscometer. The operative principle is very similar to the falling body viscometer, but for the falling cylinder the falling time is measured by detecting the making and breaking of an electric contact once the sinker touches an electric pin at the bottom of the capillary.

Fig. 2.13 Sawamura high pressure falling body viscometer (Bair 2007)



Viscometer cells of original design (a) and of improved design assembled with the potentiometer (b); a permalloy cylinder, b retaining coil, c ball, d differential transformer, e tube, f sample liquid, g spacer, h bellows, i coil spring, j connecting stem, k leaf spring contact, l manganin wire.

2.2.6 High Shear Viscometers

Common viscometers can measure viscosity at shear rates up to $10,000 \text{ s}^{-1}$. The shear rate in engine journal bearings can exceed $10,000,000 \text{ s}^{-1}$ (Selby and Miller 1995). Starting from the 60s, novel high shear viscometers were designed to measure oil properties at the operating engine conditions and to fulfil the requirements of the ASTM standards. The operative principle of the high shear viscometer is the same as the rotating cylinder viscometer, but the gaps between the cylinder are very thin and the rotational speed of the cylinder is much higher. The first high shear viscometer was the tapered bearing shear viscometer, patented by the Savant Inc. (1979), and its design has been used for decades. Figure 2.15 shows this viscometer schematically.

In this geometry, the stator and the rotor are separated by a gap of just a few micrometres. This means that if the rotor spins at high rotational speed (usually $\omega > 2000 \text{ rpm}$) then high shear rates are obtained. This measurement from this

Fig. 2.14 Irving and Barlow high pressure falling body viscometer. The falling body (9) descends through the capillary (1). The electrical coils (10) allow the identification of the position of the falling body (Bair 2007)

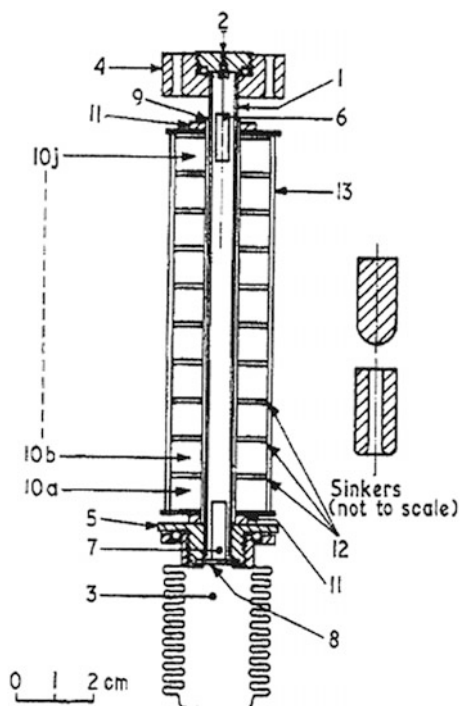
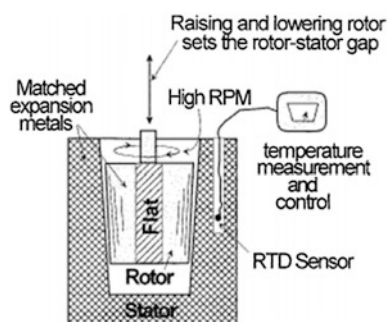


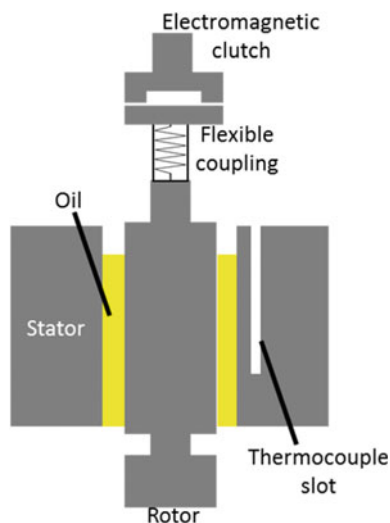
Fig. 2.15 Tapered bearing high shear viscometer (Selby and Miller 1995)



viscometer is affected by the shear heating that occurs at the contact area. This causes distortions in the film shape that affect the accuracy of the measurement at shear rates higher than $3,000,000 \text{ s}^{-1}$. These limitations have been recently overcome by the USV ultra high shear viscometer patented by the PCS instruments. Figure 2.16 shows schematically the rotor and stator of the USV.

The USV was designed to limit the shear heating by making the measurements of a very short duration (100 ms). The negligible shear heating allows this viscometer to have a cylindrical, not tapered, geometry. This is also ideal to minimize the calibration procedure because the gap between rotor and stator is constant. This

Fig. 2.16 Ultra high shear viscometer set-up by PCS instrument



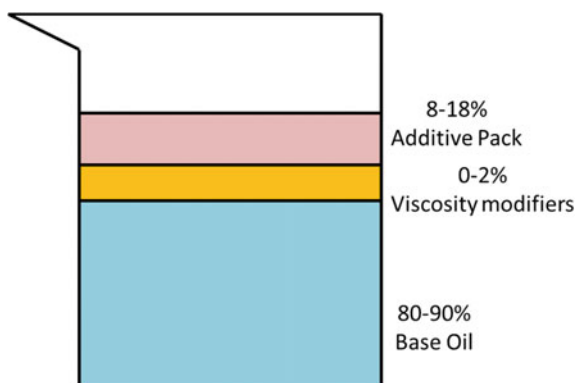
viscometer operates between 40 and 100 °C and the fluid to be tested cannot exceed 50 m Pas in viscosity to avoid inaccurate torque readings.

In this work, the viscosity measurements from a high shear viscometer are compared with the ultrasonic viscometer readings in Sect. 7.5.

2.3 Engine Lubricating Oil Composition

An engine lubricant is a mixture of a base oil and a polymer package chosen to give to the lubricant the desired performances. Figure 2.17 shows the general composition of lubricant.

Fig. 2.17 Engine lubricant composition



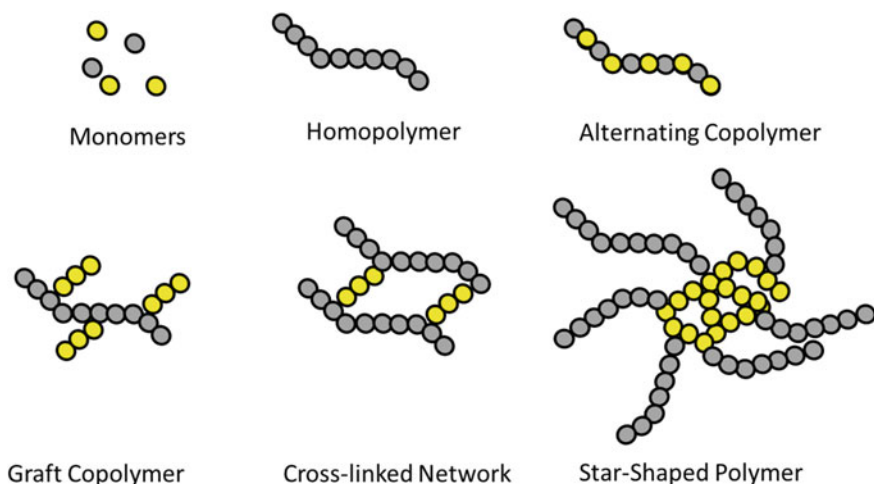


Fig. 2.18 Some polymer architectures

The additive pack in Fig. 2.17 might include, depending on the formulation, polymers with function of dispersants, pour point depressant, detergents, antioxidants, anti-wear and friction modifiers. A polymer is a long chain of repeating monomers arranged in different architectures, as shown in Fig. 2.18.

Figures 2.17 and 2.18 show the complexity involved in the study and design of lubricants. Depending on the polymers nature (geometry and chemical composition) and the number of species present in a lubricant the analysis of contact in engine is complicated by the interaction of these species and their evolution (or deterioration) in new compounds inside the contact. In the following sections the base oil type and viscosity modifier and detergent additives are analysed (Sects. 2.3.1–2.3.2) because the engine lubricant tested in this work are made mainly of these polymers.

2.3.1 Base Oils

Base oils are long chain hydrocarbons refined from crude oil (mineral oils) or they can be synthetically made. Tables 2.2 and 2.3 report respectively the composition of mineral oils and their official API classification.

An “unofficial” classification exists also for synthetic base oils. This classification includes two other groups: Group IV or the synthetic Polyalphaolephin (PAO) group and Group V or Ester group. (Table 2.4)

The 90% of base oil market is constituted of oils from the first three groups, while the synthetic oils constitute only 1% of the overall sales due to the high costs involved in the manufacturing process.

Table 2.2 Mineral oil composition (Arora 2005)

Element	% by weight
Carbon	83–87
Hydrogen	11–14
Sulphur	0–8
Nitrogen	0–1
Oxygen	0.5
Metals	0.02

Table 2.3 API base oil official classification (source API)

API classification	I	II	III
% Saturates	<90	>90	>90
% Sulfur	>0.03	<0.03	<0.03
VI	80–120	80–120	>120

Table 2.4 Unofficial synthetic base oil classification (source API)

API classification	IV-PAO	V-Ester
% Saturates	–	–
% Sulfur	0	0
VI	130	135

2.3.2 Viscosity Modifiers

Viscosity modifiers are polymers that increase the viscosity of the base oil and are among the most important additives. They are designed to increase the viscosity at low temperature as little as they can, while thickening the base at high temperatures, as shown in Fig. 2.19.

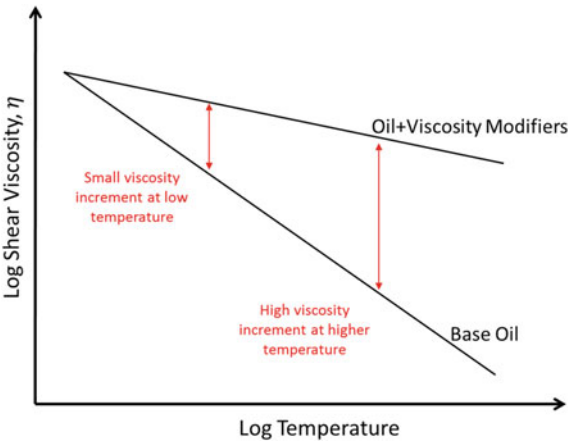


Fig. 2.19 Viscosity-temperature characteristic behaviour with the addition of viscosity modifier

Table 2.5 Main viscosity modifier polymers characteristics (Covitch and Trickett 2015)

Polymer	Advantages	Disadvantages
Olefin copolymer (OCP)	Cost effective High thickening Good solubility	Weak low temperature performances Contribute to piston deposits in engines
Maleic styrene copolymer (MSC)	Cost effective High thickening	Weak low temperature performances Leave high deposits
Poly(alkyl methacrylates) (PMA)	Great low temperature properties Thermal and oxidative stability Excellent engine test performance Used as VM and pour point depressant	Expensive Cannot be made as a solid Relative low thickening

It is possible to notice that the viscosity modifiers increase the VI. The aim of current light duty commercial vehicle market is to increase oil's VI for higher economy and durability as engine components wear is decreased. Another decisive factor in the viscosity modifier design is the polymer shear stability. This is the ability of a fluid to maintain its viscosity after a stress is applied. When a stress is applied to a polymer chain this might break or strain/compress with no rupture. For a viscosity modifier polymer it is desired that the polymer does not break because otherwise the viscosity modifier properties of the polymeric chain deteriorates over time. Table (2.5) reports a list of the main polymers used as viscosity modifiers with benefits and negative effects on the engine parts.

2.3.3 Detergents

Detergents are polymers used in engine oils, gear oils and automatic transmission lubricants. Their major roles are to clean internal engine parts, neutralise acids of combustion and inhibit oxidation. All these functions enhance durability of engine parts. Figure 2.20 shows schematically a detergent polymer particle.

Detergent particles come in different shapes, but Fig. 2.20 gives a good idea of the general detergent polymer structure and behaviour. This polymer is mainly made of a fatty oil soluble tail and a charged polarised head. The detergent polymers head stick to the iron surface at the contact interface with the lubricant oil. This allows inhibiting corrosion and removing undesired polarised particle from the surface. Table 2.6 reports the most commonly used detergent polymer and highlights advantages and disadvantages of using one polymer over the other.

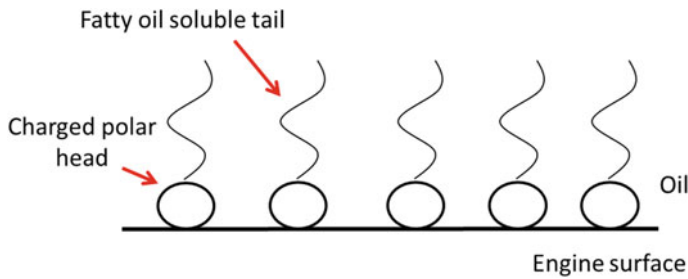


Fig. 2.20 Schematic representation of detergent polymers in contact to an engine surface

Table 2.6 Unofficial synthetic base oil classification (Mortier and Orszulik 2010)

Detergent type	Advantages	Disadvantages
Sulfonates	Excellent detergency Cost effective	No antioxidancy
Phenates	Good detergency Antioxidant performance	Expensive
Salicylates	Excellent antioxidant Good detergency Sulphur free	Expensive

2.4 Oil Classification by Viscosity

Oil viscosity in combustion engines is classified by the SAE J-300 standard. This standard establishes different oil grades and is updated every two years to meet the latest industrial standards. Table 2.7 reports the oil grades viscosity as stated in the SAE J-300 standard of April 2013.

In this table the letter W means “winter” and characterize the so called multi-grade oils. This category of lubricants have good cold start capabilities, but tend to shear thin at high shear rates that occur in components such as journal bearing. The oils that appear in Table 2.7 without the letter W are called monograde because they meet only one SAE grade. It has to be highlighted that in the SAE J-300 the extreme grades are the ones more exposed to change. For example the grade SAE 16 has been recently added to lower the limit of HSHT (high shear high temperature) applications. Also, the upgrades are made to meet the pollution emission regulations; engines that operate with oils outside the limits of this table are considered harmful and such engines need to be upgraded or dismissed.

Table 2.7 SAE J-300, oil classification by viscosity (*source* SAE standards)

SAE viscosity grade	Low-temperature (°C) cranking viscosity, mPas	Low-temperature (°C) pumping viscosity, mPas	Low-shear rate kinematic viscosity (mm ² /s) at 100 °C Min	Low-shear rate kinematic viscosity (mm ² /s) at 100 °C Max	High shear rate viscosity (mPas) at 150 °C Min
0 W	6200 at -35 °C	60000 at -40 °C	3.8		
5 W	6600 at -30 °C	60000 at -35 °C	3.8		
10 W	7000 at -25 °C	60000 at -30 °C	4.1		
15 W	7000 at -20 °C	60000 at -25 °C	5.6		
20 W	9500 at -15 °C	60000 at -20 °C	5.6		
25 W	13000 at -10 °C	60000 at -15 °C	9.3		
16			6.1	<8.2	2.3
20			6.9	<9.3	2.6
30			9.3	<12.5	2.9
40			12.5	<16.3	3.5
40			12.5	<16.3	3.7
50			16.3	<21.9	3.7
60			21.9	<26.1	3.7

2.5 Lubrication Principles in Mechanical Components

2.5.1 The Stribeck Curve

Separation of sliding surfaces by means of a lubricant layer is necessary to reduce friction and wear in mechanical components assemblies. Two sliding components in contact operate in ideal condition when this separation is achieved. In practice three different lubrication regimes occur among mechanical components in contact: boundary lubrication, mixed lubrication and hydrodynamic (HD). The Stribeck curve, Fig. 2.21, identifies these regimes in a graph where the non-dimensional product of viscosity η , rotational speed U and the load in the contact area W is plotted against the friction μ .

The first region of the graph is called boundary lubrication because the lubrication film is not formed and contact between asperities occurs. In this region of the Stribeck curve the load is completely carried by the solid asperities in contact because there is absence of the hydrodynamic lift. Because of this, the boundary lubrication region is characterized by high values of friction coefficient. This is the case for most of the start-up conditions of mechanical components.

Once the components are in relative motion the contact area between asperities reduces and a lubricant film starts forming. This causes a drastic drop in friction that is characteristic of the mixed lubrication region. In this lubrication region the lubricant film starts forming and the load is partially supported by the solid

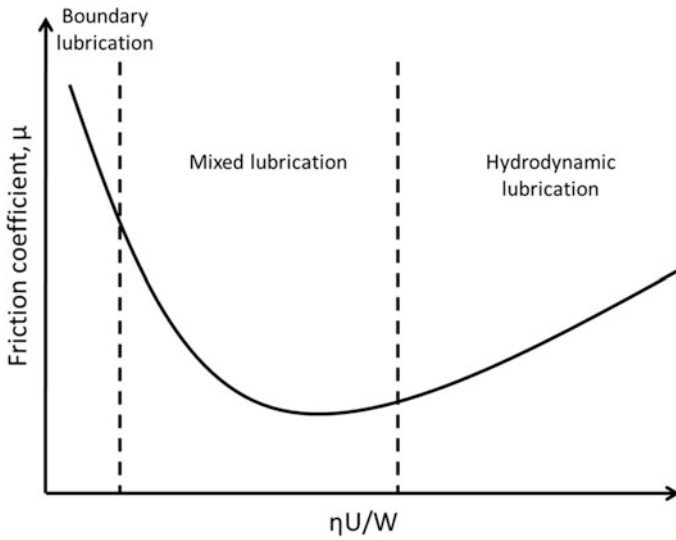
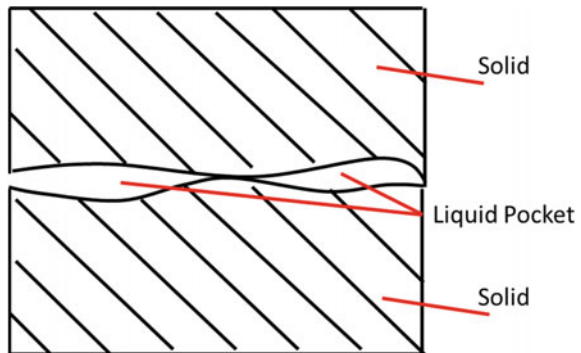


Fig. 2.21 The Stribeck curve

Fig. 2.22 Schematic representation of mixed lubrication contact



asperities and by the forming lubricant film. Figure 2.22 shows schematically a mixed lubrication regime situation.

When mixed lubrication occurs, the hydrodynamic lift of the lubricating layer is still not sufficient to form a full lubricated layer. Consequently, the contact region is characterized by pockets of oil and asperities in contact. The mixed lubrication region physical nature is still a matter of debate. Spikes (Spikes and Olver 2003) states: “*The key feature of mixed lubrication is considered to be that it contains a mixture, i.e., the presence of two distinctly different regimes of lubrication within one contact, one being conventional fluid-film lubrication ... and the other something else. This begs the question as to what is the ‘something else’.*” That ‘something else’ can be arise from complex solid-solid asperities interaction,

formation of micro-EHD (elasto-hydrodynamic) regions, and/or formation of thin layer of lubricant that behaves differently compared to the bulk case.

As a full lubrication film is formed, the value of load per unit area decreases drastically and the product of viscosity and rotational speed increases. Consequently the friction increases linearly because of the increment in viscous drag forces. The region characterized by the absence of asperity contact is referred to as hydrodynamic lubrication zone. HD lubrication was first studied by Tower (1883) and Reynolds (1886) developed the governing equation describing fluid film separating two wedge shaped components:

$$\frac{d}{dx} \left(\frac{h^3}{\eta} \frac{dp}{dx} \right) = 6U \frac{dh}{dx} \quad (2.17)$$

where h is the film thickness and U is the velocity gradient. This equation has been developed to suit the analysis of different components geometries such as plain journal bearings.

2.5.2 Journal Bearing Lubrication

Journal bearings are widely used in automotive engines and their main purpose is to support and align the load from a rotating shaft, for instance the crankshaft. Depending on the load direction the journal bearing can be axial or radial. In this work the case of radial journal bearing is studied. In contrast to ball or cylindrical roller bearings, the surface of the journal bearing is in direct contact with the shaft to support. Figure 2.23 shows the main components constituting the journal bearing assembly. This is made of a shaft or journal in direct contact with the bearing or bush. The separation of journal and bush in automotive engines is usually obtained through a layer of liquid lubricant.

Figure 2.23 shows, also, in detail the geometrical parameters of a journal bearing. In this scheme C is the clearance between journal and bush, R_i is the internal radius, R_e is the external radius and e is the eccentricity and the film thickness is then equal to:

$$h = C + e * \cos(\theta) \quad (2.18)$$

where θ is called the attitude angle or the angle that determines the misalignment between the ideal centre of the bearing and the actual centre of the journal. Figure 2.17 also shows the pressure distribution given the applied load W . The reason for the journal eccentricity (the centre of the journal bearing O_j is not coincident with the centre of the bearing O), and the position of the maximum load P , misaligned of an angle ϕ with respect to the position of the application of the load, can be explained by analysing the different lubrication stages encountered by the bearing. Figure 2.24 highlights the different lubrication regions as the journal

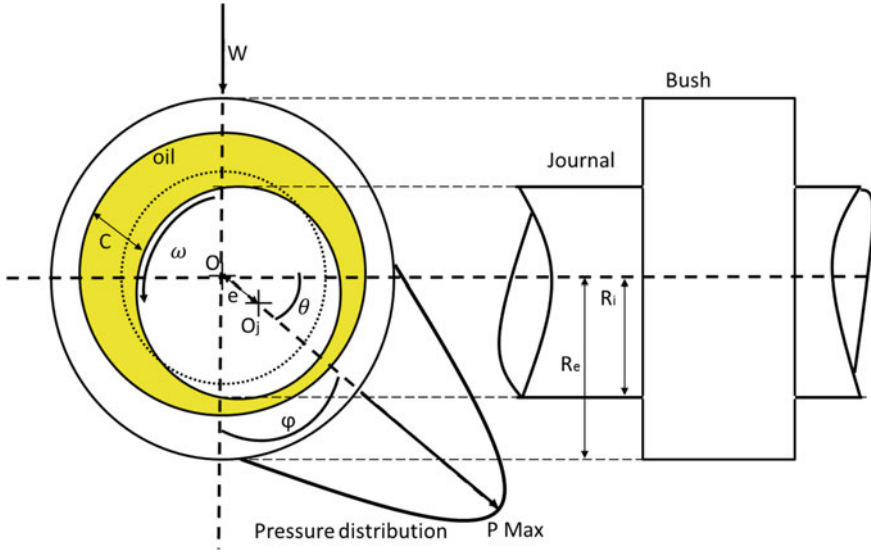


Fig. 2.23 Journal bearing representation with highlighted geometry and load distribution

bearing operates with respect to the Stribeck curve. Figure 2.24a refers to the condition with the journal at rest when solid-to-solid contact occurs because the conditions for the generation of a full lubricant layer do not exist. Figure 2.24b refers to the start-up conditions when the shaft starts rotating. The shaft starts spinning and at first is constrained by the load W , but when the journal rotational speed is high enough the shaft slips on the bush surface towards the right by dragging lubricant. This is the condition at which the journal is not in full solid contact with the bush, but mixed solid-liquid lubrication occurs. Figure 2.24c shows the situation in which the shaft rotational speed is sufficient to drag the oil thus making a lubrication layer that separates the shaft-bush surfaces.

The determination of the maximum eccentricity and the minimum film thickness is complex and dependent on many parameters; the most important of which are the radius of the shaft, the width of the bush, the rotational speed of the shaft, the surface finish, and the applied load. The design of journal bearings starts with the Reynolds equation, Eq. (2.17). For a narrow bearing ($L/D < 1/3$) the Reynolds equation becomes (Cameron 1966):

$$P = \frac{3U\eta\epsilon\sin\theta}{Rc^2(1 + \epsilon\sin\theta)^3} \left(\frac{L^2}{4} - y^2 \right) \quad (2.19)$$

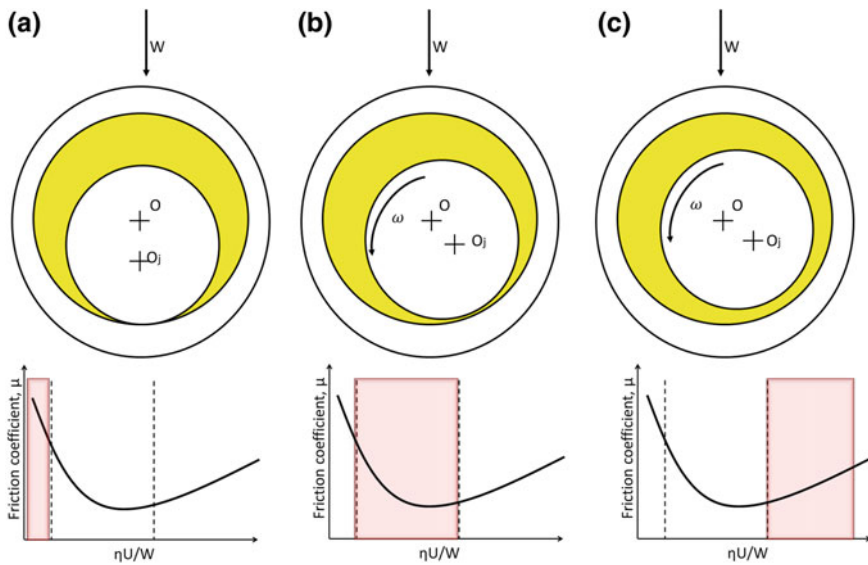


Fig. 2.24 **a** Journal bearing at rest solid-solid contact, **b** rotation of shaft initiate the friction coefficient decreases and the lubrication is mixed, **c** full hydrodynamic lubrication

where y is the length on the axial direction. This solution is very convenient, but in practice $L/D < 1/3$ are not common in engines because the bearing vibrations are not well-damped (Alford 1911). Common values of L/D ratio ranges normally between 0.6 to 1 for bearings used in engines. Because of this the Reynolds equation was solved numerically by Raimondi and Boyd (1958) for bearings with different L/D ratio. Figure 2.25 show the results plotted as function of the non-dimensional Sommerfield parameter. This parameter is very important in bearing design because it correlates the load, the pressure and the geometry parameter to the bearing eccentricity.

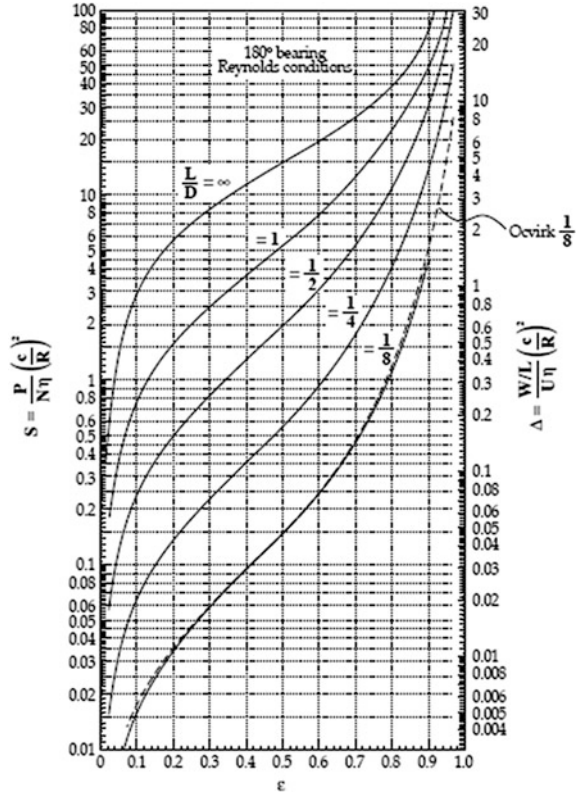
In Fig. 2.25 the Sommerfield number (S) can be written in terms of the load (Δ) and in terms of the pressure (P).

$$\Delta = \eta \omega \frac{LD}{W} \left(\frac{R}{C} \right)^2 \quad (2.20)$$

$$S = \frac{P}{N\eta} \left(\frac{c}{R} \right)^2 \quad (2.21)$$

Figure 2.26 shows an alternative solution of the Reynolds equation. In this case the Sommerfield number is correlated to the minimum film thickness in the bearing.

Fig. 2.25 Raimondi-Boyd chart for journal bearing design (Juvinal 2006)



2.5.3 Considerations for Journal Bearing Design

The charts in Figs. 2.25 and 2.26 are at the basic stage of journal bearing design. The following have to be taken into consideration when designing a journal bearing:

- The bush material. The material chosen for the bush have to resist the operative machinery temperature, guarantee elastic deformability and resistance to the corrosion.
- The maximum operating pressure and the minimum film thickness. The minimum film thickness has to provide full lubrication at regime given the operative pressure. These two parameters can be determined using the Raimondi-Boyd charts (Fig. 2.25 or 2.26).
- The friction coefficient. This is correlated to the film thickness and can be obtained with one of the Raimondi-Boyd solutions.
- Bearing misalignment. The best bearing design can fail if bearing and shaft are misaligned when mounted. Figure 2.27 schematically shows an aligned and a misaligned journal bearing. When the journal and bush are completely aligned

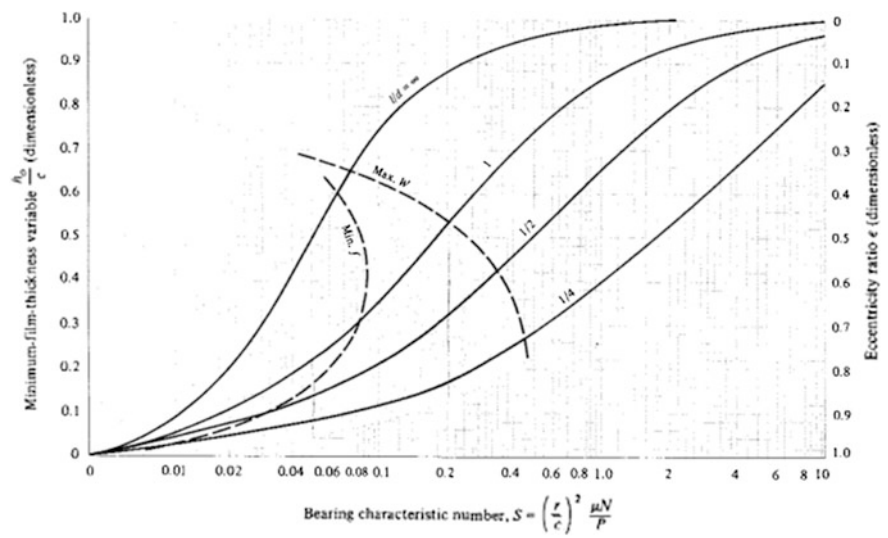


Fig. 2.26 Raimondi-Boyd chart for the determination of the journal bearings minimum film thickness (Juvinall 2006)

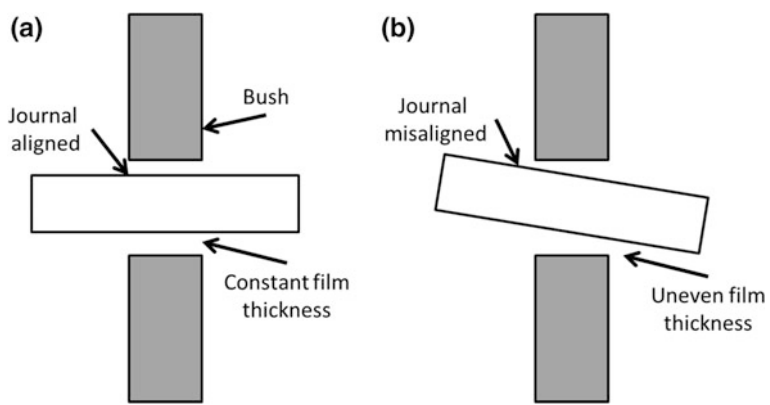


Fig. 2.27 **a** Aligned journal bearing and **b** Misaligned journal bearing

the maximum pressure is obtained at the centre of the bearing. When the bearing is misaligned, the maximum pressure is obtained where the minimum thickness is encountered. The maximum pressure at misalignment is higher than the maximum pressure in the aligned case because the film thickness is less and the friction increases. The bearing, then, operates in non-ideal conditions.

- Oil supply. Figure 2.28 illustrates two common ways to feed oil into the bearing contact interface. The simplest approach uses a simple cylindrical feeding hole. This is ideal when the expected working pressure and load is not severe.

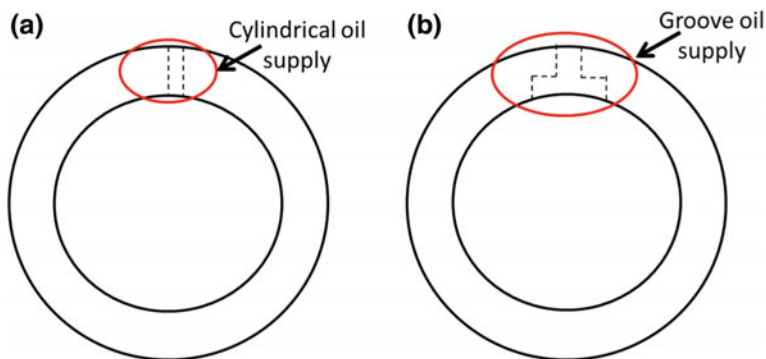


Fig. 2.28 Some possible designs for the oil supply

Another approach consists of supplying oil through a groove. The groove can be localised at the inlet or can be machined along the bush circumference. Insertion of a groove causes the pressure to drop because the film thickness increases, but this can lead to cavitation.

2.6 Conclusions

Machinery elements require lubricants to operate. Viscosity is one of the most important parameters in the design of lubricants and lubrication. This parameter is correlated to the lubricant film thickness, friction and elements power at the interface. The viscosity depends upon the temperature, pressure shear rate in the contact and the composition of the oil. It was shown through the analysis of the Raimondi and Boyd (1958) charts that viscosity is also one of the key parameters to take into account when designing journal bearing. This parameter is commonly measured with rheometers, but in-situ measurement with conventional technique is prohibitive. Development of a novel tool to analyse viscosity in real time can therefore benefit the design of journal bearing and of lubricant additives.

References

- L.P. Alford, *Bearings and Their Lubrication*, 1st edn. (Mc Graw Hill, 1911)
- A. Arora, *Text Book of Inorganic Chemistry*. (Discovery Publishing House, 2005)
- S. Bair, *High Pressure Rheology for Quantitative Elastohydrodynamics*, vol. 54. (Elsevier, 2007)
- C. Barus, Isotherms, isopiestic and isometrics relative to viscosity. *Am. J. Sci.* **45**, 87–96 (1893)
- A. Cameron, *Principles of Lubrication*, 1st edn. (Longmans Green and Co. Ltd, 1966)
- A. Cameron, *Basic Lubrication Theory*, 1st edn. (Ellis Horwood Limited, 1981)

- P.J. Carreau, Rheological equations from molecular network theories. *Trans. Soc. Rheol.* **16**(1), 99–127 (1972)
- M.J. Covitch, K.J. Trickett, How polymers behave as viscosity index improvers in lubricating oils. *Adv. Chem. Eng. Sci.* **5**(02), 134 (2015)
- R.F. Crouch, A. Cameron, Viscosity-temperature equations for lubricants. *J. Inst. Pet.* **47**, 307–313 (1961)
- E.W. Dean, G.H.B. Davis, Viscosity variations of oils with temperature. *Chem. Met. Eng.* **36**, 618–619 (1929)
- M.F. Fox, *Chemistry and technology of lubricants*. Eds. R.M. Mortier, S.T. Orszulik, vol. 107115. (London: Springer, 2010)
- J.B. Irving, A.J. Barlow, An automatic high pressure viscometer. *J. Phys. E.* **4**(3), 232–236 (1971)
- R.C. Juvinall, M.M. Kurt, *Fundamentals of Machine Component Design*, 5th edn. (Wiley, 2006)
- J.D. Novak, W.O. Winer, Some measurements of high pressure lubricant rheology. *ASME J. Lubr. Techn.* **90**(3), 580–591 (1968)
- A.A. Raimondi, J. Boyd, A solution for the finite journal bearing and its application to analysis and design. *ASLE Trans.* **1**, 159–209 (1958)
- O. Reynolds, On the theory of lubrication and its application to Mr. Beauchamp Tower's experiments including an experimental determination of the viscosity of olive oil. *Phil. Trans. Roy. Soc. London* **177**, 157–234 (1886)
- S. Sawamura, Takeuchi et al., High pressure rolling-ball viscometer of a corrosion-resistant type. *Rev. Sci. Instrum.* **61**(2), 871–873 (1990)
- H.A. Spikes, A.V. Olver, Basics of mixed lubrication. *Lubr. Sci.* **16**(1), 1–28 (2003)
- T.W. Selby, G.C. Miller, U.S. Patent No. 5,565,621, Washington, DC: U.S. Patent and Trademark Office, 1995
- G.W. Stachowiack, A.W. Batchelor, *Engineering Tribology*, 2nd edn. (Butterworth Heinemann, 2001)
- B. Tower, First report on friction experiments. *Proc. Inst. Mech. Engrs.* 632–659 (1883)
- S. Warburg, W. Sachs, The effect of density on the viscosity of liquids. *Wiedmanns Annalen.* **22**, 518 (1884)
- J.G. Webster, *Mechanical Variables Measurement-Solid, Fluid, and Thermal* (CRC Press, 1999)
- K.Y. Yasuda, R.C. Armstrong, R.E. Cohen, Shear flow properties of concentrated solutions of linear and star branched polystyrenes. *Rheol. Acta* **20**(2), 163–178 (1981)

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Schirru, M.

2017, XVI, 167 p. 155 illus., 80 illus. in color., Hardcover

ISBN: 978-3-319-53407-7