Chapter – I

Introduction

1.1 Historical perspective and development of tribology

The word “Tribology” was coined in 1966 by a UK committee headed by Professor Duncon Dowson. It was also reported by Peter Jost in 1966 in the well known report of the British Department of Education and science. Tribology is derived from the Greek word ‘tribos’ meaning rubbing or sliding and from the suffix ‘ology’ means ‘the study of’. Therefore, tribology is the study of rubbing or the study of things that rub. In short, it is the science and practices of interacting surfaces in relative motion. Human interest in the constituent parts of tribology probably started with the aim of reducing their physical labour in moving heavy loads, today’s science of tribology took its first rolling with the invention of wheel. The aim of the tribology is either to find the optimum film material for a given application, or to predict the sequence of events when a sliding contact is to left to generate its own intervening film. This means it comprises the study of characteristics of films of intervening material between contacting bodies and the consequence of either film failure or absence of a film which are usually manifested by severe friction and wear. In fact, tribology is the study of friction, wear and lubrication. Friction is the principal cause of wear and energy dissipation. Considerable saving can be made by improved friction control. In other words, friction is the resistance to motion whenever one solid body moves over another with which it is in contact. The friction force or the coefficient of friction is dependent not only on the material but also on the factors that are independent of the material, such as sliding speed, resting time and the environmental conditions. Therefore, friction is not only a material property but a system response. Friction was used to generate fire. The use of sledges and rollers to reduce friction for the transport of large loads provide generally well-known examples.

The search for effective lubricants and lubrication technologies has colourful past going back as far as the recorded history of humankind. As early as 3500 B.C., the Chinese made use of the lubricating properties of water. Around 1400 B.C., the Egyptians used animal’s fates or olive oil mixed with lime powder for their chariots. By 780 B.C., the Chinese had discovered the friction-reducing properties of a mixture consisting of vegetable oils and lead, and more than 100 years ago, the first discoveries that air can serve as a suitable lubricant were made. In second half of nineteenth century the value of mineral oils meet the requirements; Viz, viscosity, mechanical stability, good thermal stability and resistance to oxidation, admirably these are now used widely in industry. The first scientific development of tribology began during the Renaissance (1450–1600 AD). The most important studies of friction and wear being conducted by Leonardo da Vinci (1452–1519), the great painter, architect and engineer, whose manuscripts contained over 5000 pages. The most important mathematical result of Leonardo.s studies was that the force of friction is not only directly proportional to the applied load, but also independent of the apparent area of contact.

Wear is erosion or sideways displacement of material from its ‘derivatives’ and original position on a solid surface performed by the action of another surface. Wear is related to interactions between surfaces and more specially, the removal and deformation of material on a surface, as a result of mechanical action of the opposite surface.

1.2 Types of lubrication

Lubrication is the process or technique employed to reduce wear of one or both surfaces in close proximity and moving relative to each other, by interposing a substance called lubricant between the surfaces to carry or to help carry the load between the opposite surfaces. The lubricant film should be satisfying two requirements namely

- It should have low shear strength to obtain low friction.
- It should be strong enough to carry the entire load in the direction perpendicular to the surface, to prevent direct contact between surfaces.

The purpose of lubricants is to replace dry friction with either thin film or fluid- film friction, depending on the load, speed or intermittent action of the moving parts. Thin film lubrication, in which there is some contact between the moving parts, usually is specified where heavy loads are a factor. In thick film lubrication, a pressure film is formed between moving surfaces and keeps them completely apart. This type of lubrication cannot easily be maintained in high speed machinery and therefore is used where reciprocating or oscillating conditions are moderate. There are four types of lubrications Viz; hydrodynamic, aerodynamic, elasto-hydrodynamic and boundary lubrication.

1.2.1 Hydrodynamic lubrication

This type of lubrication deals with lubrication of rigid body contacts, and it was first researched by Osborne Reynolds (1842-1912), when a lubricant is applied to shaft and Reynold found that a rotating shaft pulled by a converging wedge of lubricant between the shafts and the bearing. He also noted that, as shaft gained velocity, the liquid flowed between the two surfaces at greater rate. This is because the lubricant is viscous, produces a liquid pressure in the lubricant wedge that is sufficient to keep the two surfaces separated. The principle of hydrodynamic lubrication can also be applied to a more practical example related to thrust bearings used in the hydropower industry. The essence of hydrodynamic lubrication was first clarified experimentally by British Rail road Engineer Beauchamp Tower in 1883. Based on Tower's Experiments, Osborn Reynolds, the physicist, formulated a theory of lubrication in 1886. Since then, Reynolds theory has been foundation of the theory of hydrodynamic lubrication. Following are the assumptions made in the theory of hydrodynamic lubrication.

- Bearing pressure is predominantly developed as the result of viscous forces in lubricants.
- Body forces are neglected, i.e there are no external field of forces (e.g. gravitational force) acting on the fluid.
- The pressure is constant throughout the thickness of the film. As the film is only one or two thousandths of a centimetre (10^{-3} cm), in many practical cases, it is always true.
- There is no slip at the boundaries i.e., the velocity of the lubricant layer adjacent to the boundary is the same as that of the boundary.
- Fluid inertia is neglected.

1.2.2 Aerodynamic lubrication

Aerodynamic lubrication is an extension of the theory of hydrodynamic lubrication. The theory is the same, in that converging wedge of high pressure is formed between two surfaces, supporting one surface from coming in to contact with another surface, extra attention has to be paid to the system. Instead of using a thin fluid lubricant, a gas of 1000 times less viscosity is used. The distance of separation is minute, requiring close to perfectly smooth surfaces. Besides the surfaces having to be virtually free of defects, using aerodynamic lubrication

1.2.3 Elasto-hydrodynamic lubrication

This type of lubrication deals with the lubrication of elastic contacts. In this the elastic deformation of the contacting bodies and the change of viscosity with pressure play the fundamental roles. In recent years it has been recognized that many loaded contacts of low geometrical conformity such as gears, rolling contacts bearings and cams frequently behave as though they are hydro dynamically lubricated. For some times these observations were at variance with theory, and it is only in the few years that a fairly clear picture of the phenomenon has emerged. The contact conditions in many machines are now being re-examined and power transmission equipment is currently being designed to take advantage of the elasto-hydrodynamic concept. It is observed that some of the assumptions employed in the hydrodynamic lubrication cannot be in Elasto-hydrodynamic lubrication theory.

1.2.4 Boundary lubrication or thin film lubrication

Boundary lubrication occurs whenever any of the essential factors that influence formation of a full fluid film are missing. This type of lubrication is applicable only when hydrodynamic and elasto-hydrodynamic lubrication fails. In boundary lubrication, the bearing surfaces are not separated by the lubricant and the load is carried by the asperity contacts. When a complete fluid film does not develop between potentially rubbing surfaces, the film thickness may be reduced to permit momentary dry contact between wear surface with high points or asperities. This condition is characteristic of boundary lubrication. Boundary lubrication is also defined as that regime in which the load is carried by the surface asperities rather than by the lubricant. The boundary lubrication happens in the following situations.

- When shaft starts moving from rest.
- When the speed is very low.
- When the load is very high.
- When the Viscosity of the lubricant is too low.

Following are the examples for boundary lubrication.

1) Guide and guide shoe in two stroke engine.
2) Lubrication of the journal bearing in diesel engines (mainly during starting and stopping of engine).

1.3 Classification of fluids

On the basis of relationship between stress and rate of strain, fluids are generally classified as Newtonian and non-Newtonian fluids.

1.3.1 Newtonian fluids

Fluids which obey the Newton's laws of viscosity are known as Newtonian fluids. For example natural fluids like water, air, mercury etc. Are Newtonian fluids. In such fluids viscosity does not vary with rate of strain. For Newtonian fluid shear is linearly proportional to the rate of strain i.e.

$$\tau = \mu\dot{\gamma} \tag{1.3.1.1}$$

where τ is the shear stress, μ is the viscosity of the fluid and $\dot{\gamma}$ is the rate of shearing strain.

1.3.2 Non-Newtonian fluids

For many fluids a plot of shear stress against shear rate does not give a straight line. These are so called "Non-Newtonian fluids" Plots of shear stress against shear rate are experimentally determined using viscometer. Non-Newtonian fluids are complex mixtures such as ketchup, blood, pastes gels, gravy, pie fillings, mud, printer ink etc. The non-Newtonian fluids are mainly classified in to Time independent fluids, and Time-independent fluids.

Time independent fluids

In time-independent fluids the apparent viscosity depends only on the rate of shear at any particular movement and not on the time for which the shear rate is applied. For non-Newtonian fluids the relationship between shear stress and shear rate is more complex. The rheological description of the fluids whose properties are independent of the time is given by the equation of the form

$$\frac{d\gamma}{dt} = f(\tau) \tag{1.3.2.1}$$

where $\frac{d\gamma}{dt}$ is the rate of shear and τ is the shear stress. Equation (1.3.2.1) implies that, the rate of shear at any point in the fluid is a simple function of shear stress at that point. Such types of fluids are called as non-Newtonian viscous fluids. These fluids are further classified in to three distinct types depending on the nature of the function $f(\tau)$ in the right hand side of equation (1.3.2.1) and are; Bingham plastics (it requires τ_0 for initial flow), pseudoplastic fluids and dilatant fluids.

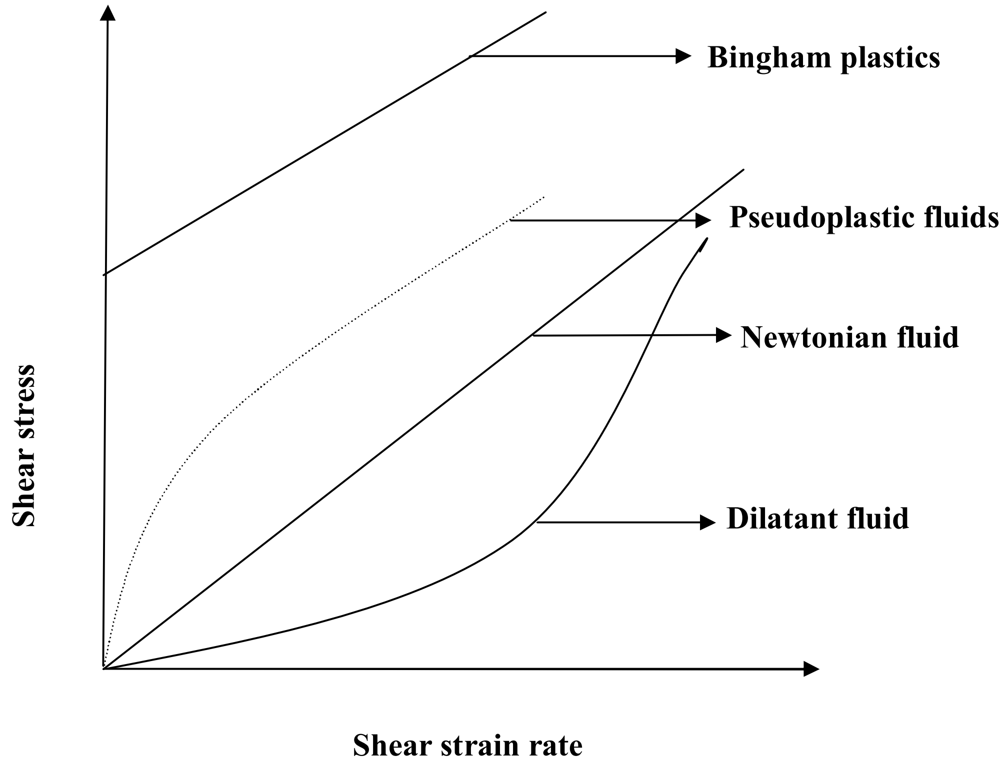

Figure 1.1 Shear stress strain rate relationship for non-Newtonian fluids

Time-dependent non-Newtonian fluids

In time dependent non-Newtonian fluids the viscosity of the fluid varies with the shear time as well as shear rate. These are the fluids where the strain rate is a function of time and are very complex in nature. Such type of fluids are classified in to three types

Thixotropic fluids: Fluids which exhibit a reversible decrease in shear stress and apparent viscosity with time at a constant shear rate for example: Paints

Rheopectic fluids: Fluids which exhibit a reversible increase in shear stress and apparent viscosity with time at a constant shear rate e.g. Gypsum suspensions, bentonite clay.

Visco-elastic fluids: A visco-elastic material is one which possesses both elastic and viscous properties, i.e. although the material might be viscous, it exhibits a certain elasticity of shape

1.4 Rabinowitsch fluid

In recent years, the increasing use of fluids containing micro-structures such as suspensions, additives, long-chained polymers has received great attention. Because, it is found that, the Newtonian fluid constitutive approximation cannot satisfy engineering demands. Hence, the expectation of engineers is that lubricant must provide a protective film which allows for two surfaces to be separated and smoothened, thus lessening the friction between them and correspondingly less heat generation in the machine, thereby keeping the working temperature of machine parts within safe operating limits. The flow behaviour of a Newtonian lubricant blended with various additives cannot be accurately described by the classical continuum theory. Because, the viscosity of these lubricants exhibits a non-linear relationship between the shear stress and the shear strain rate. Rabinowitsch (1929) developed a model which analyzes the behaviour of non-Newtonian fluids. In this fluid model, the non-linear relationship between shear stress and the shear strain rate can be described by a cubic equation as

$$\tau_{xy} + \kappa\tau_{xy}^{3} = \mu\frac{\partial u}{\partial y}. \tag{1.4.1}$$

By using this fluid model, one can analyze the characteristics of pseudoplastic fluids ($\kappa > 0$) and dilatant fluids ($\kappa < 0$) as compared to the Newtonian fluids ($\kappa = 0$). Where μ denotes the zero shear rate viscosity and is equivalent to the viscosity of Newtonian fluids, and κ represents a non-linear factor responsible for non-Newtonian effects.

The field equations governing the motion of an incompressible non-Newtonian fluid - Rabinowitsch fluid model used by Wada and Hayashi (1971) are

$$\frac{\partial u}{\partial x}+\frac{\partial u}{\partial y}=0, \tag{1.4.2}$$

$$\frac{\partial p}{\partial x}+\frac{\partial \tau_{xy}}{\partial y}=0, \tag{1.4.3}$$

$$\frac{\partial p}{\partial x}=0. \tag{1.4.4}$$

Many researchers used this model to analyze the pseudoplastic and dilatant nature of the fluid. Hsu and Saibel (1965) analyzed slider bearing performance with a non-Newtonian lubricant. They concluded that, in the case of pseudoplastic fluids the pressure distribution is lowered over the Newtonian case, and consequently the load carrying capacity is reduced. Wada and Hayashi (1971) presented the hydrodynamic lubrication of journal bearings by pseudoplastic lubricants. They found that, the pressure distribution of pseudoplastic fluids decreases below that of Newtonian fluids whose viscosity is equal to the initial viscosity of pseudoplastic fluids and the load capacity and the frictional force also decrease. The non-Newtonian effects on the static characteristics of one-dimensional slider bearings were studied by Hashimoto (1994). He observed that, the non-Newtonian effects and fluid inertia effects provide significant influences upon the static characteristics of one-dimensional, high-speed slider bearings. Non-Newtonian flow in infinite-length full journal bearing was analyzed by Hsu (1967). He concluded that the pseudoplaticity and dilatants nature of the fluid provide a significant influence on the pressure distribution, flow rate, frictional force, load carrying capacity, attitude angle, and frictional coefficient. Variation principle for non-Newtonian lubrication: Rabinowitsch fluid model was analyzed by He (2004). Singh et.al (2011) presented the steady performance of hydrostatic thrust bearing: Rabinowitsch fluid model. They found that, the results so obtained are compared and found to be in good agreement with the earlier theoretical and practical results of Dowson (1971) and the effect of viscosity index improver is analyzed. Effects of inertia in the steady state pressurized flow of a non-Newtonian fluid between two curvilinear surfaces of revolution: Rabinowitsch fluid model was studied by Singh *et.al.*(2011). They observed that, the pseudoplastic effect along with the effect of rotational inertia on the pressure distribution, frictional torque and fluid flow rate of externally pressurized flow in narrow clearance between two curvilinear surfaces of revolution. On the performance of pivoted curved slider bearings: Rabinowitsch fluid model was analyzed by Singh *et.al.*(2012). They found that, the steady state bearing performance characteristics i.e., steady state film pressure, load carrying capacity and centre of pressure, as well as dynamic stiffness and damping characteristics vary significantly with the non-Newtonian behaviour of the fluid consistent with the real nature of the problem. Singh *et. al.* (2012) analyzed on the steady state performance of annular hydrostatic thrust bearing: Rabinowitsch fluid model. They observed that, in the limiting case in which there is an absence of pseudoplaticity, the results are compared with the pre-established Newtonian lubricants and are found to be in good agreement. On the squeeze film characteristics between a long cylinder and a flat plate with Rabinowitsch fluid is presented by Singh *et.al* (2013). They concluded that the pressure distribution, load-carrying capacity and time-height relationship under squeezing characteristics shows significant variation with the non-Newtonian pseudoplastic and dilatants behaviour of the fluids. The effect of non-Newtonian Rabinowitsch fluids in wide parallel rectangular squeeze-film plates were studied by Lin *et.al* (2013). They found that, comparing with the Newtonian-lubricant, the effects of non-Newtonian cubic-stress flow rheology provide significant influences on the squeeze film characteristics. It is shown that the non-Newtonian pseudoplastic behaviour reduces the load capacity and the response time; however, the effects of non-Newtonian dilatants lubricant provide an increase in the load-carrying capacity and therefore lengthen the response time of parallel plates. Recently, inclined stepped composite bearings with Rabinowitsch fluids was studied by Naduvinamani *et al*. (2014) and they observed that, the highest load carrying capacity and lowest coefficient of friction is observed for inclined stepped bearing as compared to the other types of bearing systems. The squeeze film lubrication between circular stepped plates with Rabinowitsch fluid model was studied by Naduvinamani *et al*. (2014).They found that, the dilatant fluids significantly enhances the load carrying capacity as compared to the corresponding Newtonian fluids.

1.5 Surface roughness

The effects of surface roughness on hydrodynamic lubrication of various bearings have attracted the attention of several researchers. This is because, in practice, most of the bearing surfaces are rough. The aspect ratio and the absolute height of the asperities and valleys observed under microscope vary greatly, depending on material properties and on the method of surface preparation. Roughness is one of the most important surface topographic characterizations, which intuitively refers to the unevenness or irregularity of a texture. It gives an idea of how smooth the surface is at a certain length scale. Roughness parameters are, therefore, important in applications such as automobile brake linings, floor surfaces, and tires. The effect of roughness on lubrication has also been studied to determine its impact on issues regarding lubrication of sliding surfaces, compliant surfaces, and roller bearing fatigue.

All smooth surfaces possess some degree of roughness, even if only at the atomic level. For man-made surfaces, this roughness arises from the manufacturing process which may involve chemical deposition, grinding, polishing, etching or several other commonly used techniques. Decreasing the roughness of a surface will usually increase exponentially its manufacturing costs.

. Many researchers have studied different types of bearings with roughness effect Surface roughness effect on slider bearing lubrication was analyzed by Tzeng and Saibel (1967). They found that the effect of roughness of surfaces significantly influences the load-carrying capacity and friction force. Tzeng and Saibel (1967) presented on the effects of surface roughness in the hydrodynamic lubrication theory of a short journal bearing. They concluded that the effect of roughness of surfaces significantly influences the pressure development, load-carrying capacity, attitude and friction. Stochastic models for hydrodynamic lubrication of rough surfaces were studied by Christensen (1969). On the basis of stochastic theory, he developed two different forms of Reynolds-type equation corresponding to two different types of surface roughness. It is shown that the mathematical form of these equations is similar but not identical to the form of the Reynolds equation governing the behaviour of smooth, deterministic bearing surfaces. It is shown that surface roughness may considerably influence the operating characteristics of bearings and that the direction of the influence depends upon the type of roughness assumed. The effects are not, however, critically dependent upon the detailed form of the distribution function of the roughness heights. Christensen (1971) presented some aspects of the functional influence of surface roughness in lubrication. He displayed that surface roughness has a considerable effect on the functional characteristics of a bearing operating in the hydrodynamic, and, especially, in the mixed lubrication regime. A brief outline is given of a new theory of lubrication based upon probabilistic principles, and some of the results of the application of the theory to the analysis of a slider bearing are listed. The effect of surface roughness upon the lubrication of rigid cylindrical rollers was studied by Dowson and Whomes (1971). They observed that this model seeks to reproduce the major features of turned and cylindrically ground surfaces. Values of friction for a rough and a smooth cylinder under the same load are compared, first using an extension of the Martin analysis and then with a more complete numerical solution of the Reynolds equation. It appears that surfaces having the form of roughness considered here generally exhibit a lower coefficient of friction than the corresponding smooth surface. Christensen and Tonder (1971) analyzed the hydrodynamic lubrication of rough bearing surfaces of finite width. They showed that how the effect of surface roughness on the bearing characteristics is closely tied up with features of nominal geometry as well as with operational factors. Waviness and roughness in hydrodynamic lubrication was presented by Tonder and Christensen (1972). They observed that the corrugation wavelength is a major factor, pressure ripples vanishing with increasing corrugation density. It is further shown that at the same time, the load-carrying capacity tends towards that predicted by the authors' statistical roughness theory, the analysis thus constituting a numerical proof of the mathematical soundness of that theory. Tonder and Christensen (1972) studied lubrication of cylindrical rollers with surface corrugations. They found that how the wavelength of corrugations on a cylindrical roller strongly influences its hydrodynamic properties. It is further shown how these properties, as the wavelength decreases, tend towards those obtained by the authors' probabilistic roughness theory. The hydrodynamic lubrication of rough journal bearings was analyzed by Christensen and Tonder (1973). They describe the application of this theory to the analysis of the full journal bearing of finite width. The analysis demonstrates how the roughness influences the characteristics of the bearing and also shows how roughness interacts with features of nominal geometry and operating factors to determine the bearing response. The effects on bearing load-carrying capacity of two-sided striated roughness were studied by Rhow and Elord (1974). They observed that the term representing transient effects is of the same order of magnitude as others retained in the final equation governing the average pressure. As an example, it is shown that with the same overall roughness characteristics, the load-carrying capacity of an infinitely wide slider bearing varies according to how the same roughness is distributed on opposing surfaces. The effect of surface roughness on the average film thickness between lubricated rollers was analyzed by Chow and Cheng (1976). Results were obtained for purely transverse as well as purely longitudinal surface roughness for cases with or without slip. The reduced pressure is shown to decrease slightly by considering longitudinal surface roughness. The transverse surface roughness, on the other hand, has a slight beneficial effect on the average film thickness at the inlet. The same approach was used to study the effect of surface roughness on lubrication between rigid rollers and lubrication of an infinitely-wide slider bearing. Sun and Chen (1977) studied first effects of stokes roughness on hydrodynamic lubrication. They found that the roughness effects can be determined by three parameters; viz., the rms roughness, the correlation length, and the correlation wavelength. The present results are significantly different from those in the literature, where the Reynolds equation is used for analysis. Lubrication of surfaces having area-distributed isotropic roughness was analyzed by Tonder (1977). He displayed that a mathematical treatment of the problem of lubrication of bearing surfaces exhibiting two-dimensionally distributed uniform or isotropic roughness. Gupta and Deheri (1996) studied effect of roughness on the behavior of squeeze film in a spherical bearing. They observed that the performance of the bearing is adversely affected by the composite roughness of the surfaces. A note on squeeze film between rough anisotropic porous rectangular plates was analyzed by Bujurke and Naduvinamani (1998). They found that the loci of maximum load are more sensitive to the anisotropic permeability than the roughness parameter. Zhang and Qiu (1998) presented effects of surface texture on hydrodynamic lubrication of dynamically loaded journal bearings. They concluded that the effects of roughness are closely tied to the roughness texture and structure, features of nominal geometry, and operating factors. Andharia *et.al* (1999) studied effect of transverse surface roughness on the behaviour of squeeze film in a spherical bearing. They observed that the composite roughness of the surfaces affects the performance of the bearing adversely. The effect of roughness parameter on the performance of hydrodynamic journal bearings with rough surfaces was analyzed by Turaga *et.al* (1999). They found that the transverse roughness tends to increase significantly load carrying capacity and stability with roughness values, whereas in the case of other roughness patterns the effect is seen to be very small. Gururajan and Prakash (1999) presented surface roughness effects in infinitely long porous journal bearings. It is shown that the results are significantly different than those for the case of an infinitely long journal bearing. Naduvinamani *et. al* (2002) studied surface roughness effects in a short porous journal bearing with a couple stress fluid. They observed that the effects of surface roughness on the bearing characteristics are more pronounced for couple stress fluids as compared to the Newtonian fluids. Effect of surface roughness on the static characteristics of rotor bearings with couple stress fluids was analyzed by Naduvinamani *et.al.*(2002). They found that the surface roughness considerably influences the static characteristics of the rotor bearing system. The effect of couple stresses is to increase the load carrying capacity and to decrease the coefficient of friction for both types of roughness patterns. The bearings with couple stress fluid operate with smaller attitude angle as compared to the Newtonian case. Further, the effects of couple stresses are more pronounced for the longitudinal roughness pattern as compared to the transverse roughness pattern. Hsu *et. al.* (2003) presented combined effects of couple stresses and surface roughness on the lubrication of short journal bearings. They concluded that compared to the Newtonian-lubricant smooth-bearing case, the couple stress effects and the longitudinal roughness improve the load carrying capacity, and thus decrease the attitude angle and friction parameter, while the effect of transverse roughness is opposite to that of the longitudinal one in the journal-bearing system. Hydrodynamic lubrication of rough slider bearings with couple stress fluids was studied by Naduvinamani *et.al.* (2003). They observed that for all the lubricant film shapes under consideration, the negatively skewed surface roughness increases the load carrying capacity, frictional force and temperature rise, while it reduces the coefficient of friction. On the contrary, the reverse trend is observed for positively skewed surface roughness. Further, these effects are more pronounced for the couple stress fluids.

Naduvinamani *et. al* (2003) analyzed effect of surface roughness on characteristics of couple stress squeeze film between anisotropic porous rectangular plates. They found that found that the surface roughness effects are more pronounced for couple stress fluids as compared to the Newtonian fluids. On the squeeze effect of lubricants with additives between rough porous rectangular plates was presented by Naduvinamani *et.al* (2004). They concluded that there exists a critical value of the aspect ratio above which the roughness effects are pronounced due to the size effects of the lubricant additives. Further, this critical value is dependent on the microstructure size of additives. Naduvinamani and Siddanagouda (2007) presented effect of surface roughness on the hydrodynamic lubrication of porous step-slider bearings with couple stress fluids. They observed that the negatively skewed surface roughness pattern increases the load carrying capacity and decreases the coefficient of friction whereas the adverse effects were found for the positively skewed surface roughness pattern. Effect of surface roughness on the squeeze film lubrication between curved annular plates was studied by Bujurke *et.al* (2007). They found that the effect of radial (circumferential) roughness pattern is to shift the point of maximum pressure towards the inlet (outlet) edge and also observe that the mean load-carrying capacity increases (decreases) for the circumferential (radial) roughness pattern compared with the corresponding smooth case, for both concave and convex pad geometries. Naduvinamani and Patil (2007) analyzed the negatively skewed surface roughness and couple stresses improve the performance of the porous journal bearings as compared to the smooth journal bearings with Newtonian lubricants. However, the presence of positively skewed surface roughness on the bearing surface affects its performance. Combined effects of surface roughness and couple stresses on squeeze film lubrication between porous circular stepped plates were presented by Naduvinamani and Siddangouda (2007).They concluded that the squeeze film characteristics are improved for the azimuthal surface roughness pattern. However, the performance of the squeeze film bearing suffers due to the presence of radial roughness pattern. Deheri and Abhangi (2008) studied magnetic fluid based squeeze film between curved rough circular plates. They observed that by properly choosing the curvature parameters of both the plates and the magnetization parameter the performance of the bearing system can be enhanced considerably in the case of negatively skewed roughness, especially, when the negative variance is involved. Therefore, this study makes it mandatory that the roughness must be accounted for while designing the bearing system. Numerical solution of finite modified Reynolds equation for couple stress squeeze film lubrication of porous journal bearings was analyzed by Naduvinamani and Patil (2009). Effect of roughness on hydro-magnetic squeeze films between porous rectangular plates was studied by Naduvinamani *et.al.* (2010), they observed that roughness effect enhances pressure, load carrying capacity and squeeze film time. Naduvinamani and Kashinath (2010) analyzed hydrodynamic analysis of rough curved pivoted porous slider bearings with couple stress fluid. They found that the improved performance due to the couple stresses and the presence of negatively skewed surface roughness. However, the presence of porous facing and positively skewed surface roughness affects the performance of the pivoted porous slider bearing. Effect of surface roughness on magneto-hydrodynamic squeeze film characteristics between finite rectangular plates was presented by Bujurke *et. al.*(2011). They concluded that the performance of squeeze film suffers due to positively skewed surface roughness pattern.

Squeeze film lubrication between rough poroelastic rectangular plates with micropolar fluid: A special reference to the study of Synovial joint lubrication was studied by Naduvinamani and Savitramma (2013). Effects of viscosity variation and surface roughness with micropolar fluid as lubricant of journal bearings were studied by Naduvinamani *et.al* (2014). They observed that, the transverse surface roughness improves the squeeze film characteristics where as adverse effects are observed for the longitudinal roughness pattern. Naduvinamani and Savitramma (2014) analyzed the micropolar fluid poroelastic squeeze film lubrication between a cylinder and rough flat plate with special reference to the synovial joint lubrication. They found that, the transverse roughness pattern is more pronounced than the longitudinal roughness pattern and improves the squeeze film performances of bearings. Siddangouda *et.al.* (2014) analyzed the effects of micro polarity and surface roughness of slider bearings and have observed that, the negatively skewed surface roughness increases the load capacity frictional force. Effect of surface roughness on the squeeze film characteristics of circular plates with couple stress fluid and transverse magnetic field studied by Fathima *et.al.* (2015), they observed that, the surface roughness effects are more pronounced for couple stress fluid as compared to the Newtonian fluid. Lin *et.al.* (2015) analyzed the effect of squeeze film characteristics and radial surface roughness in a couple stress magnetic field lubricants.

1.6 Squeeze films

When two surfaces containing lubricant in between them approach each other, then the fluid is squeezed resulting in the build up of pressure which helps in avoiding the possible contact of surfaces. This is termed as squeeze film lubrication. The fluid film so formed is known as squeeze film. The relevant literature related to squeeze film lubrication can be found in Moore (1968) and Archibald (1956). As the viscous fluid is squeezed out between the surfaces, it flows towards less constrained surroundings. The squeeze film investigations focus upon the film pressure, the load carrying capacity and the minimum film thickness. Squeeze film mechanism is commonly observed in many areas of engineering and applied sciences, such as bearings in automotive and aircraft engines, turbo machinery, squeeze film dampers, etc. As the pressure increases between approaching surfaces, leads to fluid layer to more towards less constrained surroundings. As the fluid layer is very thin, the viscous forces become dominant and offer high resistance to such a fluid which in turn reduces the wear and tear of the parts. The classical continuum theory focuses on the use of Newtonian lubricant in various squeeze film mechanisms. Archibald (1956), Jones (1975), Pinkus (1961) and Tichy (1970).

The squeeze film lubrication between two infinitely long parallel plates is studied by Cameron (1981). The flow of an incompressible fluid between two parallel plates due to the normal motion of the plates is investigated by Bujurke *et. al.* (1995). The squeeze film with Newtonian lubricant has been studied by several investigators, Jackson (1963), Burbidge (2004) and Gupta and Gupta (1977). The non-Newtonian characteristics of the lubricants become important, when the lubricant contains additives with the large quantity of high molecular weight polymers as viscosity index improvers. Grease, emulsion, liquid crystals and the body fluids like blood and synovial are examples of such lubricants. The Rayleigh step-bearing with non-Newtonian fluids has been studied by many researchers, Huges (1963), Bujurke *et. al.*(1987), and Maiti (1973). Squeeze films and thrust bearings lubricated by fluids with couple stress was analyzed by

Ramanaiah and Sarkar (1978). They observed that the load capacity of a thrust bearing can be increased and the squeeze in the squeeze film can be decreased by the use of a fluid with couple stress as the lubricant. Ramanaiah (1979) studied the squeeze films between finite plates lubricated by fluids with couple stress. The influence of couple stresses in squeeze films was analyzed by Bujurke and Jayaraman (1982). They found that the bearings with couple stress fluid as lubricant provide significant load carrying capacities which increases the bearing life. Bujurke and Naduvinamani (1990) studied the lubrication of lightly loaded cylinders in combined rolling, sliding and normal motion with couple stress fluid. The results show that the load capacity and frictional drag increase as the squeeze velocity increases. Porous slider bearing with couple stress fluid was analyzed by Bujurke *et.al.* (1990). They showed that the load-carrying capacity increases and the coefficient of friction decreases. Bujurke and Naduvinamani (1991) studied on the performance of narrow porous journal bearings lubricated with couple stress fluid. They observed that they bearings with couple stress fluid as lubricant provide significant load carrying capacity and ensure considerable reduction in the coefficient of friction compared with viscous lubricants. Bujurke *et.al.* (1992) analyzed the influence of couple stresses on the dynamic properties of a double layered porous slider bearing. Lin (1997) analyzed static and dynamic behaviours of pure squeeze films in couple stress fluid lubricated short journal bearings. He concluded that under a cyclic load the couple stress effects provide a reduction in the velocity of the journal centre as well as an increase in the minimum permissible height of the squeeze film. Lin (1997) studied squeeze film characteristics of long partial journal bearings lubricated with couple stress fluids. He found that, compared with the Newtonian lubricant case, the couple stress effects provide an enhancement in the load-carrying capacity and lengthen the response time of the squeeze film action. Effects of couple stresses on the lubrication of finite journal bearings were presented by Lin (1997). He concluded that the effects of couple stresses enhance the load-carrying capacity, as well as reduce the friction parameter and the attitude angle. Lin (1998) analyzed squeeze film characteristics of finite journal bearings: couple stress fluid model. He reported that the presence of couple stresses improves the characteristics of finite journal bearings operating under pure squeeze film motion. Static and dynamic characteristics of externally pressurized circular step thrust bearings lubricated with couple stress fluids were presented by Lin (1999). Squeeze film characteristics between a sphere and a flat plate: couple stress fluid model was presented by Lin (2000). He found that, the couple stress effects characterized by the couple stress parameter produce an increase in value of the load-carrying capacity and the response time as compared to the classical Newtonian lubricant case. Naduvinamani et. al (2001) studied squeeze film lubrication of a short porous journal bearing with couple stress fluids. They observed that, the lubricants which sustain the couple stresses yield an increase in the load carrying capacity. Elsharkawy and Guedouar (2001) analyzed an inverse solution for finite journal bearings lubricated with couple stress fluids. They concluded that as the percentage of random error added to the numerically calculated pressure data points increases and the number of iterations required for convergence increases slightly. Derivation of dynamic couple stress Reynold's equation of sliding squeezing surfaces and numerical solution of plane inclined slider bearings was studied by Lin *et.al.* (2003). They observed that the effects of couple stresses provide an improvement on both the steady-state performance and the dynamic stiffness and damping characteristics especially for the bearing with a higher value of aspect ratio. Naduvinamani *et.al* (2003) analyzed couple-stress squeeze films between porous rectangular plates. They reported that a significant increase in the load-carrying capacity and the delayed squeeze-film time are observed for the couple-stress fluids in comparison with Newtonian fluids. Ma *et. al* (2004) analyzed a study of dynamically loaded journal bearings lubricated with non-Newtonian couple stress fluids. They found that the couple stress fluids lubrication improves the bearing performance under dynamic loads. On the squeeze film lubrication of long porous journal bearings with couple stress fluids was studied by Naduvinamani *et. al* (2005). They found that the effect of couple stresses is to increase the load carrying capacity and to lengthen the squeeze film time as compared to the corresponding Newtonian case. The effect of permeability is to reduce the load carrying capacity and to decrease the squeeze film time as compared to the corresponding solid case. Naduvinamani and Patil (2009) presented numerical solution of finite modified Reynolds equation for couple stress squeeze film lubrication of porous journal bearings. They observed that, under a cyclic load, the effect of couple stress is to reduce the velocity of the journal centre and to increase the minimum permissible height of the squeeze film. Squeeze film lubrication between circular stepped plates of couple stress fluids was presented by Naduvinamani and Siddangouda (2009). They observed that the influence of couple stresses enhances the squeeze film pressure, load carrying capacity, and decreases the response time as compared to the classical Newtonian case. The load carrying capacity decreases as step height increases. Lin et. al (2010) studied non-Newtonian couple stress effects on the frictional and flow-rate performances of wide composite slider bearings. They observed that, comparing with the Newtonian lubricant composite slider bearing case, the effects of non-Newtonian couple stresses provide a reduction in values of the friction parameter and the volume flow rate required. These improvements on bearing characteristics are more emphasized with increasing values of the couple stress parameter. Lin and Chou (2013) presented non-Newtonian dynamic characteristics of wide composite slider bearings lubricated with couple stress fluids. They observed that, comparing with the non-Newtonian incline-plane bearing, the non-Newtonian composite bearing provides an improvement in the dynamic stiffness and damping coefficients; better bearing characteristics are achieved for the non-Newtonian composite bearing under specific length-ratio parameters. Non-Newtonian couple stress poroelastic squeeze film was analyzed by Nabhani *et. al* (2013). They found that the numerical results of the simulations show that all these effects have a significant influence on the porous squeeze film performance. Magneto-hydrodynamic couplestress squeeze film lubrication of circular stepped plates was studied by Naduvinamani et.al (2011). They observed that the effect of applied magnetic field on the squeeze film lubrication between circular stepped plates with couplestress fluids is to increase the load-carrying capacity significantly and to delay the time of approach considerably as compared to the corresponding non-magnetic case. Naduvinamani and Rajashekar (2011) analyzed MHD couple stress squeeze film characteristics between sphere and plane surface. They concluded that the influences of couple stresses and the magnetic effects on the squeeze film characteristics are significantly apparent and the MHD couple stress fluids have better lubricating qualities than the corresponding Newtonian case. Naduvinamani *et.al.* (2012) analyzed effect of surface roughness on magneto-hydrodynamic couple-stress squeeze film lubrication between circular stepped plates.

They observed that the effect of azimuthal (radial) roughness pattern on the bearing surface is to increase (decrease) the mean load-carrying capacity and squeeze film time. The applied magnetic field increases the load-carrying capacity and lengthens the squeezing time. Combined effects of MHD and surface roughness on couple-stress squeeze film lubrication between porous circular stepped plates were analyzed by Naduvinamani *et. al* (2012). They concluded that the effect of applied magnetic field on the squeeze film lubrication between circular stepped plates with couple stress fluids is to increase the load carrying capacity significantly and to delay the time of approach as compared to the corresponding non-magnetic case. Rajashekar Derivation of two-dimensional couple-stress hydromagnetic squeeze film Reynolds equation and application to wide parallel rectangular plates was analyzed by Lin *et.al* (2013). They found that the effects of couple stresses and external magnetic fields provide an increase in the load capacity and the response time as compared to the classical Newtonian hydrodynamic rectangular squeeze-film plates. Wada and Hayashi (1971) investigated theoretically and experimentally the non-Newtonian squeeze film characteristics in different types of journal bearings using the cubic equation model assuming that the inertia forces were absent. All the above investigations revealed the importance of non-Newtonian rheology in squeeze films with and without inertia forces and showed that the non-Newtonian fluids gave better performance in hydrodynamic lubrication than Newtonian fluids.

Ramanaiah and Sarkar (1978) studied the effects of couple stresses in squeeze films considering different geometries and showed that fluids with couple stresses were better lubricants than Newtonian fluids. This study revealed that the load carrying capacity of a thrust bearing could be increased and that the squeeze in the fluids can be slowed down by using couple stress fluid as lubricant.

The effects of couple stresses and surface roughness are investigated in rotor bearings (Naduvinamani *et. al.*2002), slider bearings (Naduvinamani *et. al*. 2003) and between a sphere and a flat plate (Naduvinamani *et. al.* 2005). All these investigations revealed that the rough journal bearings with couple stress fluid as lubricant carry more load as compared to the Newtonian model for different types of roughness patterns. Naduvinamani and Siddangouda (2009) analysed the squeeze film lubrication between circular stepped plates using couple stress fluids and proved that the effects of couple stresses enhanced the load carrying capacity about 62% higher in comparison with the corresponding Newtonian model and also showed that their results agreed well with the experimental observations by Oliver (1988). Recently squeeze film characteristics between parallel stepped plates with Rabinowitsch fluid are investigated by Naduvinamani *et. al.* (2015) and showed that as the squeeze film thickness decreases the non-dimensional response time increases for dilatant lubricants. Effect of surface roughness on squeeze film characteristics between parallel stepped plates with Rabinowitsch fluid are investigated by Naduvinamani *et. al.* (2016) and showed that there is significant increase in load carrying capacity for dilatant fluids as compared to the corresponding Newtonian fluids for both longitudinal and transverse roughness pattern whereas the reverse trend is observed for the pseudoplastic lubricants. The effect of surface roughness on squeeze film lubrication between circular stepped plates are analysed by Naduvinamani *et.al.*(2016). It is observed that the effect of azimuthal (radial) surface roughness pattern on squeeze film lubrication between circular stepped plates with Rabinowitsch fluid is to increase (decrease) the load carrying capacity and squeeze film time significantly as compared to the smooth case and the squeeze film performance improves (suffers) due to the use of dilatant (pseudo plastic) lubricants. Wu (1970, 1972) analysed the squeeze film behaviour of a Newtonian fluid between two porous annular disks and between porous rectangular plates respectively and showed that the effects of porosity reduced the squeeze film pressure and load carrying capacity and accelerated the squeezing process and reduced the gap between the disks. Naduvinamani *et. al* (2001a, 2001b) respectively examined the effects of couple stresses on the behaviour of squeeze film lubrication of porous journal bearings with and without the assumption that the polymer additives present in the lubricant do not percolate into the porous matrix. They showed that the lubricants with polar additives can sustain a larger load for a longer period of time to prevent direct contact of the journal bearing, thus enhancing the bearing life. Naduvinamani and Siddangouda (2007a, 2007b) studied the combined effects of surface roughness and couple stresses on the squeeze film behaviour of porous circular stepped plates and porous circular step slider bearings respectively. Bujurke *et. al.* (2007) investigated a mathematical model for understanding the combined effects of surface roughness and couple stresses on lubrication aspects of synovial joints. All these investigations revealed that couple stress fluids provided increased load carrying capacity and lesser coefficient of friction in porous squeeze films as compared to the Newtonian case.

In this thesis the problems of squeeze film behaviour in parallel stepped plates lubricated with Rabinowitsch fluid (chapter II) and with surface roughness effects (Chapter III) are analyzed. Squeeze film behaviour in circular stepped plates with surface roughness effects lubricated with Rabinowitsch fluid (chapter IV) is studied. It is expected that, the results obtained in the thesis will be useful in many lubrication processes. Further, the results presented in this thesis may be extended to more geometry of practical applications.