Chapter One

General Introduction
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1.0 Introduction:

Tribology has been in existence since the beginning of recorded history. There are many well documented examples of how early civilizations developed bearings and low friction surfaces. The scientific study of Tribology has a long history. Many of the basic laws of friction have been developed by Leonardo da Vinci in 1590. However, the understanding of friction and wear languished for several centuries with concepts to explain the underlying mechanisms. In 1699, Amonton proposed that surfaces were covered by small spheres and that the friction coefficient was a result of the angle of contact between spheres of contacting surfaces.

Tribology was gained until 1886 with the publication of Osborne Reynolds' classical paper on hydrodynamic lubrication. Reynolds proved that hydrodynamic pressure of liquid entrained between sliding surfaces was sufficient to prevent contact between surfaces even at very low sliding speeds. His research had immediate practical application and lead to the removal of an oil hole from the load line of railway axle bearings. The oil, instead of being drained away by the hole, was now able to generate a hydrodynamic film and much lower friction resulted. The work of Reynolds initiated countless other research efforts aimed at improving the interaction between two contacting surfaces, and which continue to this
day. As a result journal bearings are now designed to high levels of sophistication.

Wear and the fundamentals of friction are far more complex problems, the experimental investigation of which is dependent on advanced instrumentation such as scanning electron microscopy and atomic force microscopy. Therefore, it has been possible to study these processes on a microscopic scale where a true understanding of their nature can be found. Tribology is therefore a very new field of science, most of the knowledge being gained after the Second World War. In comparison many basic engineering subjects, e.g. thermodynamics, mechanics and plasticity are relatively old and well established. The basic terms are as follows:

1.1 Bearing:

Bearing is a structural device that transmits loads while facilitating translation and/or rotation. In other words, it can be defined as a support or guide which at the same time allows relative movement to take place between two bodies.

Bearings may be fixed or movable as required for the bridge design. Movable bearings may include guides to control the direction of translation. Fixed and guided bearings shall have lateral strength adequate to resist all applied loads and restrain unwanted translation. Combinations of different types of fixed or moveable bearings should not be used at the
same expansion joint, bent or pier unless the effects of differing deflection and rotational characteristics on the bearings and structure are accounted for in the design.

Bearing design requires an interdisciplinary background. It involves calculations that are based on the principles of fluid mechanics, solid mechanics, and material science.

In engine, the principal bearings are those which allow rotation of the crankshaft about its own longitudinal axis in the main engine entablature, and those between the connecting rod and crankshaft.

In modern engine terminology in the context of a bearing has come to mean the component fitted between the journal and either the main bearing housing or the connecting rod, the shaft being supported in the bearing via an oil or magnetic fluid film [cf. Hamrock [1994]].

1.1.1 Bearing Classifications:

Machines could not operate at high speed without some means of reducing friction and the wear of moving parts. Some important engineering inventions made it possible to successfully operate heavily loaded shafts at high speed, including the rolling-element bearing and hydrodynamic, hydrostatic, and magnetic bearings.

In most applications, the sliding surfaces of the bearing are lubricated. However, bearings with dry surfaces are used in unique situations where lubrication is not desirable. Examples are in the food and...
pharmaceutical industries, where the risk of contamination by the lubricant forbids its application [cf. Majumdar [1985]].

1.2 Wear:

Wear is defined as the progressive damage resulting in material loss due to relative contacts between adjacent working parts. Although, some wear is to be expected during normal operation of equipment, excessive friction causes premature wear and this inflicts significant economic costs due to equipment failure, cost for replacement parts, and downtime. Friction and wear also generate heat which represents wasted energy that is not recoverable. In other words, wear is also responsible for overall loss in system efficiency.

In many applications, materials in contact are subjected to relative motion. In some cases this motion is intentional like in rotating plain bearings, pistons sliding in cylinders, automotive brake disks interacting with brake pads or in the processing of material by machining and many more. It may also be unintentional, e.g., in the small cyclic displacements known as fretting, which may cause wear in some structural joints under oscillating loading. It must be noted that wear resistance is not a material's property and that the mechanism of wear of a particular material and the associated rate of wear depend critically on the precise conditions to which it is subjected.
It is usual to classify wear in terms of four different categories: adhesive wear, abrasive wear, fatigue wear and Tribochemical (Erosive) wear.

1.2.1 Adhesive wear:

It is characterized by the appearance of junctions between the surfaces that are subjected to friction. When these junctions are weak, shear occurs at the interface of the two surfaces and there is no wear. However, when junctions are strong, the softer material is subjected to shearing and, as a consequence, is transferred onto the harder material.

1.2.2 Abrasive wear:

It occurs when a hard material is put into contact with a soft material. This type of wear can cause scratches, wear grooves and lead to material removal.

1.2.3 Fatigue wear:

Surface fatigue wear occurs when a material is subjected to cyclical stresses. Due to strains introduced in the superficial layers of the material, cracks that are parallel to the surface develop within the material. When they reach a critical size, they generate flake-like debris. This phenomenon is also referred to as delamination wear.

1.2.4 Erosive wear:

Erosive wear can be describe as an extremely short sliding motion and is executed within a short time interval. Erosive wear is caused by the
impact of particles of solid or liquid against the surface of an object. Normal wear is inevitable whenever there is relative motion between surfaces. However, wear can be reduced by appropriate machinery design, precision machining, material selection, and proper maintenance including lubrication [cf. Szeri [1980]].

1.3 Friction:

Friction is a force that resists relative motion between two surfaces in contact. Friction is considered to be desirable or undesirable, depending on the application. Some applications, such as tire traction on pavement and braking or when feet are firmly planted to move a heavy object, rely on the beneficial effects of friction for their effectiveness. In other applications, such as operation of engines or equipment with bearings and gears, friction is undesirable because it causes wear and generates heat which frequently leads to premature failure. There are following types of friction:

1.3.1 Dry friction:

It resists relative lateral motion of two solid surfaces in contact and is subdivided into:

- Static friction: It is friction between non-moving surfaces.
- Sliding friction: The frictional force opposes any attempt to move a stationary object along a surface.
1.3.2 **Fluid friction:**

It describes the friction between layers within a viscous fluid that are moving relative to each other.

- **Lubricated friction:** It is a case of fluid friction where a fluid separates two solid surfaces.

- **Skin friction:** It is a component of drag, the force resisting the motion of a fluid across the surface of a body.

1.3.3 **Internal friction:**

It is the force resisting motion between the elements making up a solid material while it undergoes deformation.

- **Rolling friction:** It is experienced when a body rolls over another body.

1.4 **Laws of friction:**

There are three empirical laws for friction discovered in 15th to 18th centuries as follows:

- **Amontons’ First Law:** The force of friction is directly proportional to the applied load.

- **Amontons’ Second Law:** The force of friction is independent of the apparent area of contact.

- **Coulomb’s Law:** Sliding friction is independent of the sliding velocity.
Amontons laws of friction were among the first for quantitative description of a Tribology process. Thus, these laws have been central to the development of Tribology.

1.5 Lubrication:

The purpose of lubrication is to extend the lifespan of those parts subjected to friction so that the parts do not sustain major damage as well as that of the mechanisms and machine parts in which they are involved. Without lubrication, most machines would run for only a short time. Basically, three types of lubricants are used in any machine, namely, oils, greases and anti-friction materials. Lubricants perform the key functions, namely, keep moving parts apart, reduce friction, transfer heat, carry away contaminants & debris, transmit power, protect against wear, prevent corrosion, seal for gases, stop the risk of smoke and fire of objects and prevent rust [cf. Ertel [1939]].

Lubrication is the process 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. In view of principal laws of friction the lubricant film should satisfy two requirements:
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.

Lubricants today are classified into two major groups: Automotive lubricants and Industrial lubricants. According to their physical characteristics lubricants are further classified as follows:

- **Solid Lubricants**: It is Layered and non-layered lattice solids.

- **Greases**: It is a mixture of lubricating oils and thickeners obtained by adding alkali and fatty acid to oil.

- **Mineral Oils**: It is a complex mix byproduct of fractional distillation of crude oil.

- **Synthetic Oils**: It is produced by polymerization of low molecular weight hydrocarbons.

- **Boundary Lubricants**: Molecules with strong affinity towards the surface being lubricated.

A good lubricant possesses characteristics like, high boiling point, low freezing point, high viscosity index, thermal stability, hydraulic stability, demulsibility, corrosion, prevention and high resistance to oxidation.
In fluid film hydrodynamics lubrication both mineral and synthetic oils are commonly used. A key property of these lubricants which must be considered in engineering design includes Viscosity [Dynamic and Kinetic].

Fluid films are formed in three ways:

- Hydrodynamic and elastohydrodynamic films are formed by motion of lubricated surfaces through a convergent zone such that sufficient pressure is developed in the film to maintain separation of the surfaces.

- Hydrostatic film is formed by pumping fluid under pressure between surfaces that may or may not be moving with respect to each other.

- Squeeze films are formed by movement of lubricated surfaces toward each other [cf. Hamrock [1994]].

Also, a review on the performance of fluid film bearing was provided by Ghosal [2010].

1.6 Porosity:

If we go back in history, porosity stems from the Greek word *poros* for "pore," which means "passage." So something with porosity lets things through. Most pants have porosity, as when we spill soup on them, the soup goes right through. On the other hand, a raincoat has no
porosity because it's waterproof. If a roof is leaking during the rain, it has some porosity, which is a problem.

A porous medium usually consists of a large number of interconnected pores each of which is saturated with the fluid. However, the exact form of the structure is highly complicated and differs from medium to medium. The driving force necessary to move a specific volume of fluid at a certain speed through a porous medium is in equilibrium with the resistance force generated by internal friction between the fluid and the pore structure. This resistance force is characterized by Darcy's semi-empirical law [cf. DeWiest [1969]]. Let us remember that porous media was introduced in the bearing systems primarily to reduce the effect of friction and wear.

One may remember porosity by imagining liquids pouring through things. In general, porosity is the ratio of pore volume to its total volume. It is controlled by rock type, grain size, pore distribution, cementation, digenetic history and composition. Porosity is the quality of being porous or full of tiny holes.

Flow through porous media has been the subject of considerable research activity in recent years because of its several important applications in the flow of oil through porous rock, the evaluation of the capability of heat removal from particulate nuclear fuel debris that may result from a hypothetical accident in a nuclear reactor, the filtration of
solids from liquids, flow of liquids through ion-exchange beds, drug permeation through human skin, chemical reactor for economical separation or purification of mixtures and many more.

1.7 Magnetics fluids:

The applications of magnetic fluid as a lubricant were subjected to investigation in 1965 in United States. Almost at the same time the research on magnetic fluid was started in Japan. Towards in 1986, the quality and standards of magnetic fluid in application point of view was recognized.

It is well known that various operations of machines and mechanisms are not possible without using friction units [e.g. rollers and sliding bearings, and different pinion drives]. However, the active use of these causes rapid wear of their surfaces which leads to essential energy consumption for overcoming the friction forces. Hence, to provide the operation of friction units, use of different lubricants were tested. The main advantage of a magnetic fluid as a lubricant lies in the possibility of supplying a lubricating medium only to the friction zone and it's positioning in this zone with the aid of a magnetic field. This requires a specific design of a magnetic system. One of the most important properties of the magnetic fluid is that they can be retained at a desired location by an external magnetic field. The magnetic fluid can be made to move with the help of a magnetic field gradient, even in the regions where there is no gravity. This makes the magnetic fluid amenable for the use in space.
vehicles. Lubricating properties of fluids basically are specified by two mechanisms:

- when two solid surfaces wetted with a fluid come in contact
- forcing out of a gap between two solid surfaces.

Removal of a viscous fluid from a thin gap requires large pressure gradients, and therefore a wedging pressure occurring in the fluid prevents the convergence of surfaces. A value of this pressure and though the efficiency of a hydrodynamic lubrication mechanism are proportional to the viscosity of the lubricating fluid. Thus, Ferrofluids have been successfully employed as lubricants in various hydrodynamically lubricated bearings. This motivated many researchers to investigate the theoretical aspects of Ferrofluid lubrication. It may be noted that the presence of magnetic fluid gives rise to nonlinear coupled partial differential equations; therefore, simplified assumptions have been incorporated in various studies.

1.7.1 Preparation of Ferrofluid:

Recently, considerable attention has been paid to the use of magnetic fluid as a lubricant modifying the performance of bearing system. Ferrofluids/ magnetic fluids are a special category of smart nonmaterials, in particular, magnetically controllable nano fluids. These types of nanofluids are colloids of magnetic nanoparticles such as Fe₃O₄, γ-Fe₃O₄, CoFe₂O₄, Co, Fe or Fe-C, stably dispersed in a carrier liquid. The colloid and
thermal stabilities, crucial to many applications, are greatly influenced by the choice of the surfactant. A Ferrofluid may contain approximately by volume 85% carrier, 10% surfactant and 5% magnetic solid. On applying an external magnetic field, this fluid can be confined, positioned, shaped and controlled at desired places. Consequently, these nonmaterials manifest simultaneously magnetic fluids and magnetic properties. The load capacity of a lubricant film of Ferrofluid can also be increased with an appropriate magnetic field. These fluids find their good use in engineering and biomedical applications. In fact, in any discussion of the fluid which has magnetic properties can be divided into the following categories (1) Ferrofluids; (2) Magnetorehological fluids; (3) Dispersions of micron-sized particles, and (4) Fluids containing paramagnetic particles. The magnetic particles generally have to be of approximately 5-12 nm in diameter. Particles of this size, whether they are ferrite or metal, possess a single magnetic domain only, that is, the individual particles are in a permanent state of saturation magnetization. Therefore, a strong long-range magneto static attraction exists between individual particles. Ferrofluids are stable because of dispersion of nano size particles in a carrier liquid when subjected to a magnetic field/gravitational field.

The tribological experiments of Ferrofluid under magnetic fluids conducted under cylindrical tribometer suggested that Ferrofluids registered a good friction reduction performance and this reduction is far more as compared with the carrier liquid. This experimental investigation
stipulated that reduction of friction was necessary for enhancing the performance of the bearing systems.

1.7.2 Magnetic behavior of Ferrofluid

The magnetic moments of the particles are randomly distributed when a magnetic field is absent and the fluid has no net magnetization. But when a magnetic field is applied to a Ferrofluid, the magnetic moments of the particles orient instantly along the field lines. The magnetization of the Ferrofluid responds immediately to the changes in the applied magnetic field and when the applied field is removed, the moments randomize quickly.

In a gradient field the whole fluid responds as a homogeneous magnetic liquid which moves to the region of highest flux. This means that Ferrofluids can be precisely positioned and controlled by an external magnetic field. The forces holding the magnetic fluid in place are proportional to the gradient of the external field and the magnetization value of the fluid. As a result, the retention force of a Ferrofluid can be adjusted by changing either the magnetization of the fluid or the magnetic field in the region.

1.7.3 Ferrofluid properties and their application:

The choice of Ferrofluid depends on many factors such as environments, thermal conductivity, operating life etc. There are many
different combinations of saturation magnetization and viscosity resulting in a Ferrofluid suitable for various applications.

The performance and operating life of a product which uses Ferrofluid can be significantly affected by the characteristics of the Ferrofluid. From Ferrofluids with low evaporation rate or vapor pressure to Ferrofluids with viscosity-optimized products, the characteristics of Ferrofluid can dramatically shape the capabilities of the end product.

Thermal conductivity of a Ferrofluid depends linearly on the loading. Fluorocarbon based Ferrofluids have the lowest thermal conductivity of all commercial Ferrofluids; therefore they are the least desirable materials for heat transfer applications.

In different devices, Ferrofluids come in contact with a wide variety of materials. It is necessary to ensure that Ferrofluids are chemically compatible with these materials. The fluids may be exposed to hostile gases, such as in the semiconductor and laser industries; to liquid sprays in machine tool and aircraft industries; to lubricant vapors in the computer industry; and to various adhesives in the speaker industry. Furthermore, Ferrofluids may be in contact with various types of plastics and plating materials. The surface morphology can also affect the behavior of the fluid. The selection of Ferrofluid is carefully engineered to meet application requirements.
Additionally, Ferrofluids may be expected to perform at temperature approximately at the temperature of 150°C continuously or 200°C intermittently, in winter conditions (-20)°C and space environments (-55)°C. They may also be required to withstand nuclear radiation without breakdown.

1.7.4 Characteristics of Ferrofluid:

The thermal stability of a Ferrofluid is related to particle density. The particles appear to behave like a catalyst and produce free radicals, which lead to cross-linking of molecular chains and eventual congealing of the fluid. Catalytic activity is higher at elevated temperatures and, therefore, Ferrofluids congeal more rapidly at these temperatures.

High magnetization Ferrofluids are of interest as they produce volumetric efficiencies of magnetic circuit designs leading to lightweight and low cost products. They can also be used to reduce reluctance of magnetic circuits and fringing field thus increasing useful flux density in the air gap.

Ferrofluids are a unique class of material. Ferrofluid technology is well established and capable of solving a wide variety of technical problems. There are many successful applications of this engineered material, which awaits an immense future potential.

In many applications, Ferrofluid is an active component that contributes towards the enhanced performance of the device. These
actual contacts are made only by the asperities of the two surfaces, specifically along areas over which the atoms of one asperity surface are within the repulsive field of the other. In general, the friction increases with average roughness. The value of roughness is also dependent on the hardness of the material. Roughness parameters are important in applications such as automobile brake linings, floor surfaces, noise and vibration controls bioengineering, tires, paint cracking and performance of ships. The effect of roughness on lubrication has been studied to find its impact on issues regarding lubrication of sliding surfaces, compliant surfaces, and roller bearing fatigue.

1.8.1 Mechanical state of a surface

The mechanical state of a surface can be characterized in terms of four quantities:

- Hardness: which describes the resistance of materials to plastic deformation;

- Young's modulus and the elasticity limit (or plastic flow threshold): which characterizes a material's elastic properties;

- Toughness: which accounts for the relative brittleness of a material;
positive pressure is built up during that interval and the lubricant film then supports the load [cf. Moore [1965]].

### 1.10 Rotational Inertia:

The word 'inertia' is derived from the Latin word 'iners', which means a state of idleness. In Physics, the word 'inertia' refers to a state of idleness, which is rest. In Dynamics, the state of uniform motion \( i.e., \) motion with constant velocity) is considered to be at 'dynamical' rest and thus, the concept of inertia also applies to motions with constant velocity. The term 'inertia' has its root in Newton's first law of motion which states that an object continues to be in the state of rest or continues to be in the state of uniform motion in a straight line unless it is acted upon by an external force. This tendency of the object to 'maintain status' is defined as the 'inertia' of the object. Hence, the tendency of an object to resist any change in its state of rest or of uniform motion is called 'inertia'. The Inertia of an object is a measure of its resistance to motion. All motions need not be in a straight line. Many of the motions in practice are rotational motions. Hence, in case of rotational motions, the measure of resistance of an object to change the rotation is called rotational inertia [cf. Wu [1978]].
1.11 **Elastic Deformation of the Bearing:**

When a sufficient load is applied to a metal or other structural material, it will cause the material to change shape. This change in shape is called deformation. A temporary shape change that is self-reversing after the force is removed, thus the object returns to its original shape is called elastic deformation. In other words, elastic deformation is a change in shape of a material at low stress that is recoverable after the stress is removed. This type of deformation involves stretching of the bonds but the atoms do not slip past each other.

The interacting surfaces of the bearing and the shaft will deform elastically under load. It is very difficult to prevent elastic deformation and the hydrodynamic pressure field is inevitably affected by the imposed changes in film geometry. Elastic deformation of the surfaces affects the lubricant film geometry which, in turn, influences all the other bearing parameters such as pressure distribution, load capacity, friction losses and lubricant flow rate. The effect of elastic deformation on the hydrodynamic pressure field is to reduce the peak pressure and generate a more widely distributed pressure profile.

Elastic deformation can improve the vibrational stability of a bearing so that there is no particular need to minimize deformation during the design of a bearing. Elastic deformation prevents removal of material due to rubbing of rough and hard surfaces. Embedding of the abrasive
particles in the sleeve is possible by means of elastic deformation. The
elastic deformation allows the abrasive particles to roll out and leave the
bearing. A study of the dynamics of fluid flows involves determination of
the forces on an element, which depend on the amount and nature of its
deformation, or strain. Elastohydrodynamic lubrication (EHL) is a form of
hydrodynamic lubrication where elastic deformation of the lubricated
surfaces becomes significant. Non-kinematic systems must rely on elastic
deformation of the components or careful design to balance thermal
expansion.

When a magnetic field is applied to magnetic nanofluid subjected to
a shear deformation, the magnetic particles in the fluid remain rigidly
aligned in the direction of the magnetic field. As a result, larger gradients
in the velocity surrounding a particle are expected and viscous dissipation
increases in the fluid [cf. Kudish and Covitch [2010]].

1.12 Load carrying capacity:

The load carrying capacity is described in terms of the basic
dynamic load ratings. The basic dynamic load ratings for bearings are
matched to empirically proven performance standards and the fatigue
behaviour of the material. The magnitude of load rating is always
influenced by the running speed. For every bearing number its maximum
load carrying capacity is specified corresponding to various running
speeds. The load carrying capacity is considered to be valid only for
bearing having normal dimensional and running accuracy, proper method of lubrication.

The size of the bearing to be used for an application is selected on the basis of its load carrying capacity, which can be related with the loads to be carried and the requirements of life and reliability. In addition, the permissible load for a bearing is determined by the permanent deformation caused by the load at the race way or rolling element. The permanent deformations in the bearing can lead to vibration in the bearing which increases friction. The extent to which these changes are detrimental to bearing performance depends on the demands placed on the bearing in a particular application.

1.13 Review of related literature:

For many years a lot of developments both in terms of analysis, research and developments of bearings have taken place and the branch of Tribology has gained a noticeable independent status. These developments have been documented in a number of books. Some of these have become very famous and are used for reference texts, namely, Fuller [1956], Pinkus and Sternlitcht [1961], Cameron [1966], Szeri [1980], Majumdar [1985], Hamrock [1994], Bharat Bhushan [1999] and Basu et al [2005].

It was Tower [1883] who recorded first the existence of a pressurized lubricant film. After this, Reynolds [1886] developed a theory
to explain fluid film formation. He extended the *Navier-Stokes* equations assuming small film thickness relative to the contact length, non-varying pressure across the film thickness and the dominance of certain viscous terms. The equation thus obtained relates fluid pressure to the rate of gap convergence, surface velocities and lubricant viscosity, which is referred to as the *Reynolds' Equation*. The generalized form of *Reynolds' equation* is derived by several investigators. *Cameron* and *Robertson* [1962] derived the full *Reynolds Equation* in three dimensions by three different methods while each of which differed by the continuity condition applied on it. *Dowson* [1962] developed a generalized *Reynolds' equation* for fluid film lubrication from the fundamental equations of hydrodynamics with minimum assumptions which could be used to reduce most of the forms employed in the analysis of fluid film bearings. In fact, this generalized form of *Reynolds' equation* permitted the variation of several relevant quantities across as well as along the lubricant film.

*Theyse* [1964] discussed fundamentals of hydrodynamic lubrication and their consequences in design engineering with spread reference to covering full fluid film lubrication. This study concluded that the absolute viscosity of the lubricant governing the fluid flow was predominantly important. *Moore* [1965] presented fundamental equations including the effects of inertia, variable viscosity, non-Newtonian fluids, surface tension dynamic loading and surface roughness. At the same time, the development of squeeze film from the *Reynolds-Stefan equation* was
proved. There are some important research articles and survey papers which include Christensen and Tonder [1969a], Saibel and Macken [1973] and Wu [1978].

Neale [1967-68] studied various applications to assess what bearing loads and rotational speeds would be expected in machines of different sizes and also presented the information to assist designers for bearing selection. Allen and Mckillop [1970] investigated the squeeze film performance between rotating plane annuli incorporating the inertia effect due to centrifugal effect. Dowson [1973] depicted the early history of Tribology. Christensen and Shukla [1975] derived a generalized form of Reynolds' equation applicable to rough bearings with the assumption that the flux was represented in terms of power series of a stochastic film thickness function. It was shown that the load capacity and friction force increased for decreasing values of roughness parameter in case of a step slider bearing and same trend was noticed in case of hydrostatic bearing.

In 1978, Bartz published a review article on Tribology, lubricants and lubrication engineering in which more than 200 papers were evaluated. Indeed, these papers dealt with the fundamentals of friction, wear and lubrications, including the state of friction, lubricating materials. Bartz discussed questions concerning the testing and evaluation of lubricants for machining processes and metalworking. Toshio Sakurai [1984] outlined researches on Tribology in Japan. Several problems are as follows:
The effect of surface irregularities on the lubricating performance and increase in friction in high speed bearings due to turbulent flow in fluid film lubrication.

Elastohydrodynamic lubrication and its related problems.

The correlation between the lubricating properties and the chemical reactivity of lubricant additives.

It was emphasized that the rheology of lubricants at high pressure was one of the most important properties governing lubricant concentrated contacts. Further, the deformation behaviour of materials and structures were investigated by employing the Finite Element Method.

Spikes [2001] discussed some of many challenges facing fundamental research in Tribology over the first half of the new century. Burbidge Adam and Colin [2004] described the evolution of the flow of a sample film sandwiched between a pair of lubricant films and a pair of plates. They observed three clear flow regimes: super-lubricated, apparent slip, and lubrication failure in a particular experiment which depend on the viscosity ratio, relative thickness of the films and the applied stress or strain respectively. The difference between this investigation and the earlier published work was that here the flow fields were derived from the radially symmetric strain function solution in context to previously published approaches involving the lubricant approximation. The
possibility of pumping an additional amount of lubricant fluid through the top plate in order to ensure pure bi-axial stretching of the sample film was subjected to investigation.

Lee-Prudhoe et al [2006] reported an experimental and theoretical approaches to thin film Lubrication problems and limitations of the existing experimental and modeling capabilities and associated new challenges to be examined. Abbasov and Memmedov [2007] analyzed the numerical and approximate analytical analysis of the effects of electromagnetic field on the squeeze film behavior between two parallel disks with an electrically conducting fluid. It was found that the electromagnetic correction factor in the magneto-hydrodynamic load carrying capacity was more pronounced with large Hartmann numbers. Ciulli [2009] reviewed developments related to basic study on friction and lubrication at atomic- scale level, surface topography, contact mechanics aspects, coatings and treatments, wear behaviour of different materials for applications in the field of Bio-tribology, the practical applications such as sliding bearings, gas and magnetic bearings, gears, sealing systems, automotive and rail components. Oladeinde and Akpobi [2010a] embarked on a comparative study on pressure distribution and load carrying capacity of infinitely wide parabolic and inclined slider bearing. Panday et al [2012] numerically studied the unsteady analysis and thin film lubrication of a journal bearing in turbulence effect.
To maintain the fluidity of the bearing surfaces various methods were tried:

DeWiest [1969] analyzed the flow through porous media. The porosity was modified suitably in view of the Darcy's law. The effect of porosity was discussed in details. Prakash and Vij [1973] analyzed the squeeze films between porous plates of various shapes such as circular, annular, elliptic and rectangular and conical. The combined effect of the porosity and shapes were discussed. Murti [1973] dealt with the squeeze film behaviour between two circular disks with one disk having a porous facing. It was concluded that with a suitable value of permeability for the porous facing the pressure distribution became more even leading to uniform wear of the entire facing. Murti [1974] investigated an infinitely wide porous slider bearing press-fitted into the porous housing and working with a full film of lubricant. It was concluded that the permeability of the bearing material reduced the load carrying capacity and increased the friction in the bearing. In 1975, Murti studied the squeeze film performance in curved circular plates describing the film thickness by an exponential expression. This investigation was based on the assumption that the central film thickness was constant.

Srinivasan [1977] analyzed the performance of squeeze films between double layered porous plates of various shapes like annular, circular, elliptic, rectangular plates. Here the expressions for pressure
distribution, load carrying capacity and time-height relation in closed form, were derived. Wu [1978] reviewed the squeeze films performance between two fully lubricated plates, one of which had a porous facing. Fundamental equations, results and effects were discussed. Verma et al [1979] studied a porous inclined slider bearing lubricated with a micropolar fluid and it was reported that a porous inclined slider bearing could support more load when the lubricant is a micropolar fluid as compared to a Newtonian fluid.

Zaheeruddin [1980] analyzed the behaviour of squeeze film narrow porous journal bearings using a lubricant containing solid particle additives and characterized as a micropolar fluid. It was concluded that the load capacity and time of approach increased as the micropolar parameter characterizing the concentration of the microstructure imparted to the lubricant by the additives increased and the load capacity and time of approach decreased as the permeability and/or the thickness of the porous wall increased. Bhat and Patel [1981] discussed the behaviour of a squeeze film in an inclined porous slider bearing. It was shown that the centre of pressure the coefficient of friction decreased in the presence of squeeze. Puri and Patel [1981] studied the squeeze film behaviour in a porous composite slider bearing. It was established that the pressure, load capacity, the friction increased due to squeeze and the position of the centre of pressure moved slightly towards the inlet face. Moreover, the response time for a composite slider bearing was more than that of an
inclined slider bearing. Zaheeruddin [1981] investigated the pressure distribution for a micropolar lubricant in a dynamically loaded porous journal bearing and applied to one dimensional squeeze film journal bearings operating under a cyclic load. The microstructure in the lubricant, the permeability of the bearing material and the bearing wall thickness influenced significantly, the operating eccentricity ratio.

Gupta [1981] analyzed the effect of axial current induced pinch on the squeeze film characterizes between two circular plates. The application of an axial current increased the pressure and consequently, the load carrying capacity. Patel and Gupta [1983] studied an inclined porous slider bearing with slip velocity at the porous boundary. It was noticed that the slip parameter played a crucial role to increase the load carrying capacity. Puri et al [1983] discussed the performance of a pivoted porous slider bearing considering the upper surface with a convex pad and concluded that the load carrying capacity, centre of pressure, friction and coefficient of friction of the bearing with convex pad surface are more than those with a flat surface load and friction.

Sanni and Ayomidele [1991] investigated the lubricating characteristics of a plane porous slider and found new ranges of some parameters for a simplified form of Reynolds’ equations to obtain a satisfactory performance. Lin [1996] studied the viscous shear effects on the squeeze film performance between porous circular disks with respect
to Brinkman model (BM). The viscous shear effects of the BM provided an enhancement in the load carrying capacity increased the response time of the squeeze film behavior; but these trends were reversed as compared to that derived by using the DM. Ahmad et al [1998] launched a numerical and experimental investigation on to the porous squeeze film in bearing order to apply it for the printing process. It was found that the inclusion of slip velocity affected the film thickness decay, but only by a small amount, when this model was applied to the parameters of actual printing process. It was investigated that the volume flow rate of the ink dot penetrating into the paper spreading on it could be obtained from the model while the penetrated component was negligible.

In the above discussions slip velocity was neglected. However, to get exact information, the effect of slip velocity is investigated by many researchers. Shah and Bhat [2003] analyzed an exponential slider bearing, stator with a porous facing of uniform thickness using a ferrofluid lubricant and considering slip velocity. It was seen that load carrying capacity and friction decreased and the coefficient of friction increased when the permeability parameter increased. Singh and Ahmad [2011] studied a theoretical model for a porous inclined slider bearing lubricated with magnetic fluid together with slip velocity boundary condition. Recently, Shah et al [2012b] discussed the inclined slider bearing with porous layer [plates or matrix] attached to slider as well as stator including effects of slip velocity and squeeze velocity. It was concluded
that the load capacity was increased by 1.5% in the case of small thicknesses of both the porous plates.

A metal or other structural material change shape when sufficient amount of load is applied. The phenomenon of change in shape is called deformation. When the temporary shape change that self reversing after the force is removed, as a result the object returns to its original shape, which is known as elastic deformation. This deformation of the bearing surfaces has a strong influence on the performance of the bearing systems. The effect of deformation was studied by many researchers:

In 1939, Ertel extended the analysis of Hertz [1881] and Barus [1892] considering the effects of elastic deformation and viscosity increase due to the high pressure and provided an elegant semi analytic solution to the problem. Archard [1957] examined whether the hypothesis of elastic deformation of surface protuberances was consistent with Amonton's law, that, the friction is proportional to the two-third power of the load. It was observed experimentally that the friction was proportional to the true area of contact whether the Amonton's law was obeyed upon the surface topography or not. Osterle and Saibel [1958] analyzed the slider bearing with the bearing elastic deformation for load carrying capacity. It was concluded that the load carrying capacity was reduced by the deflection of the bearing under load. O’Donoghue et al [1967] established a solution to the problem of hydrodynamic lubrication of journal bearing taking elastic
distortions of the shaft and the bearing into consideration and also evaluated the elastic deformation for a given pressure distribution for the bearing. Allan [1971] presented the effect of deformation on the behaviour of hydrodynamic journal bearings. Ramanaiah and Sundarammal [1982a] analyzed the influence of bearing deformation on squeeze films between two infinitely long rectangular plates. It was concluded that under high loads the bearing surfaces deformed producing a wedge effect in the lubricant film increasing the squeeze. Ramanaiah and Sundarammal [1982b] presented an integral equation for the determination of film thickness in a slider bearing with a deformable surface. Also, a numerical solution was found for a step slider bearing. Singha [1993] analyzed the pressure distribution in the lubricant film of a long hydrodynamic journal bearing. The pressure distribution on the bearing was represented by Fourier series and their coefficient was calculated by Simpson's composite formula and the deformation of the inner surface of the bearing was evaluated. In 1999, Okamoto et al studied the effects of changing both the housing stiffness and the bearing length on bearing performance using the elastohydrodynamic lubrication theory on connecting rod bearing for diesel engines.

Piffeteau et al [2000] developed a numerical procedure for the analysis of transient thermoelastohydrodynamic (TEHD) behavior of connecting-rod bearings under dynamic loading. Fangli and Yunlong [2000] calculated numerically the effect of elastic deformation on the
lubrication performance of finite journal bearing using the software MARC and also, presented the results of the film thickness, the film pressure and load capacity. *Schwarz* [2003] presented a new method to calculate the elastic deformation of a sphere on a flat surface considering the influence of short-range and long-range attractive forces both inside and outside the actual contact area. *Lingshan et al* [2003] discussed a method to obtain the isothermal elastohydrodynamic lubrication equation and influence of the elastic deformation on the lubricating films of dynamically loaded journal bearings. *Muhsin* [2009] dealt with the effect of elastic deformation of bearing surfaces and analyzed the steady state performance of offset halves bearing. It was shown that elastic deformation had a considerable influence on the performance especially, for larger eccentricity values. *Gultekin* [2010] studied the effects of the total bearing deformation on the performance of hydrodynamic thrust bearings. It was found that maximum load capacity loss occurred in the steel runner-bronze pad pair as 3.03 per cent. Also, for a fixed load, when bearing having small dimensions and with large deformations, the capacity loss due to runner deformations was nearly of the same order as those caused by pad deformations. *Muhsin* and *Hussein* [2011] investigated the effects of the elastic deformation and thermal distortion on the performance characteristics of the conventional hydrodynamic journal bearing. It was found that the effect of thermal distortion was greater than for elastic deformation at high speed range 1000-10000 rpm of journal. *Benasciutti et al* [2012] analyzed numerically
the effect of shaft and housing elastic deformation on the pressure distribution within oil film and it was shown that elastic deformation of bearing components had more influence on oil pressure distribution, as compared with results for ideally rigid components obtained by Raimondi and Boyd solution.

Recently, Bode et al [2013] used a model developed in MATHCAD for studying the impact of the deformation field of bearing material on the field pressure and hydrodynamic characteristics of hydrodynamic bearings.

It was strongly pointed out that the deformation of the bearing as a phenomenon could not be neglected in the study of static and dynamic behavior of a rotating system as a whole.

Wu [1971] analyzed the squeeze film behavior between rotating porous annular disks and showed that rotation caused reduced load carrying capacity and response time. Sarker [1981] discussed large deflection analysis of circular plates whose edge was elastically restrained against rotation. In this paper, Berger’s method had been adopted with Galerkin procedure to obtain the large deformation of the plate. Gupta et al [1982] considered the effect of rotational inertia on the squeeze film load between porous annular curved plates. It was shown that the load carrying capacity decreased when the speed of rotation of upper disc increased up to certain value of the curvature parameter. Gupta and Benarjee [1985]
studied a generalized Reynolds' equation by taking into account the effect of rotation in lubrication problems. Mike et al [1993] reported a study on the flow of thin viscous lubricant films on the surface of rotating silicon disks. Hsu et al [2008] presented the study of squeeze film characteristics between rotating circular discs with an electrically conducting lubricant in the presence of a transverse magnetic field. It was shown that the squeeze film characteristics of rotating circular discs were improved by the use of an electrically conducting fluid in the presence of a suitable transverse magnetic field. Shah et al [2002] studied magnetic fluid based squeeze film between porous annular curved plates with the effect of rotational inertia. Shah and Bhat [2003] analyzed the squeeze film behaviour between rotating annular discs, when the upper disc with a porous facing approached the parallel lower disc. It was observed that pressure, load carrying capacity and response time increased with increase in magnetization while these characterizations decreased with increase in speed of rotation of the upper disc.

Especially, after having some run-in and wear the bearing surfaces develop roughness. Some times the chemical degradation of the surfaces contributes to roughness. The roughness of the surfaces offers resistance to the motion of the lubricant thereby affecting system adversely.

Prakash and Christensen [1978] investigated the effects of various types of surface roughness on the response of a squeeze film between two
finite rectangular plates. It was shown that the nominal geometry had a profound effect on the system. Kumar and Sachidananda [1979] analyzed the combined effect of viscosity variation and surface roughness in short journal bearings. The relative increase in load carrying capacity due to roughness was found to be more significant than the corresponding increase in friction, for a fixed viscosity. Phan-Thien [1982] discussed the hydrodynamic lubrication of a squeeze film bearing with a sinusoidal or an isotropic surface roughness. Sinha et al [1983] analyzed the dynamically loaded rough bearings using Christensen's stochastic model. It was concluded that transverse roughness decreased and longitudinal roughness increased the velocity of the journal centre. Prakash and Tiwari [1983] studied the effect of surface roughness on the response of a squeeze film between two circular plates when one plate had a porous facing. An exact solution was found for the film pressure and pressure in the bearing matrix valid for arbitrary wall thickness. Prakash and Tiwari [1985] analyzed the effect of surface roughness on the response of a squeeze film between two rotating annular discs when one disc had a porous facing. It was found that the results were significant from the practical point of view. Chow and Saibel [1978] analyzed the influence of surface roughness on load carrying capacity of a bearing in hydrodynamic lubrication. It was concluded that the upper bound for absolute mean deviation and root mean squares deviation of a normalized load carrying capacity depended critically on the behavior of the autocorrelation of the
roughness. Jeng and Hamrock [1986] studied theoretically surface roughness effects on the elastohydrodynamically lubricated point contact. Hu and Zheng [1989] studied the film formation for a rough slider bearing. In this paper, the influence on the critical load capacity of the height profile of rough surfaces was demonstrated theoretically and was conformed by experiment. Guha [1993] analyzed the coefficients of surface roughness in the dynamic characteristics of hydrodynamic journal bearings of finite width. It was concluded that stability of bearings degenerated with increasing roughness parameter which indicated higher stability for smoother bearing surfaces. Turaga et al [1999] investigated the effect of roughness parameter on the performance of hydrodynamic journal bearings with rough surfaces. It was observed that the transverse roughness increased significantly the load carrying capacity and stability while in the case of other roughness patterns the effect was seen to be very small. Gururajan and Prakash [1999] observed the effect of surface roughness in an infinitely long porous journal bearing operating under steady conditions. It was shown that the surface roughness considerably influenced the bearing performance and the direction of the influence depends on the roughness type. Lin et al [2002] analyzed the surface roughness effects on the pure squeeze-film behavior of a long partial journal bearing operating under a time-dependent oscillating load. The mean squeeze-film characteristics were found to be significantly affected by the roughness pattern and the height of roughness. The squeeze-film
bearing with longitudinal roughness structure resulted in reversed trends. Gururajan and Prakash [2002] examined the effect of surface roughness in hydrodynamic narrow porous journal bearings operating under steady conditions. It was exhibited that the results were significantly different than those for the case of an infinitely long journal bearing. Naduvinamani et al [2002] studied theoretically the effect of surface roughness in hydrodynamic lubrication of a porous journal bearing with couple stress fluid as the lubricant. It was noticed that, the effects of surface roughness on the bearing characteristics were more significant for couple stress fluids as compared to the Newtonian fluids. Naduvinamani et al [2003] analyzed the combined effects of couple stresses and surface roughness on the performance characteristics of hydrodynamic lubrication of slider bearings with various film shapes. It was concluded that the negatively skewed surface roughness increased the load carrying capacity, frictional force and temperature rise but it reduced the coefficient of friction while, the reverse trend was observed for positively skewed surface roughness and these effects were more pronounced when the couple stress parameter increased. Hsu et al [2003] theoretically examined the performance of the combined effects of couple stresses and surface roughness on the lubrication of journal-bearing systems. It was found that the couple stress effects and the longitudinal roughness improved the load carrying capacity and thus, decreased the attitude angle and friction parameter, while the effect of transverse roughness was opposite to that of
the longitudinal one in the journal-bearing system. *Naduvinamani* and *Gurubasavaraj* [2004] studied the effect of surface roughness on the performance of the squeeze film between circular curved plate and a flat plate. It was established that the circumferential one-dimensional surface roughness pattern increased the mean squeeze film pressure and the load carrying capacity whereas the radial one-dimensional roughness pattern affected the squeeze film characteristics adversely. *Hsiu et al* [2004] studied the combined effects of couple stresses and roughness. The transverse roughness caused the reduction in the attitude angle and friction parameters while the effect of longitudinal roughness remained opposite to that of transverse roughness. *Kung et al* [2004] analyzed the surface roughness effects on the dynamic squeezing behavior of a finite length partial journal bearing under a time dependent oscillating load. It was observed that the roughness pattern and the height of roughness had significant effects upon the mean oscillating squeeze-film characteristics of partial bearings. *Kango* and *Sharma* [2010] studied the influence of roughness on the performance of journal bearings. This paper established that if used appropriately eccentricity, speed and surface texture might help in reducing the friction. *Rushma et al* [2011] dealt with the combined effects of surface roughness and couple stresses on squeeze film lubrication of spherical bearings. It was concluded that, the effect of couple stresses increased the load carrying capacity also, in case of transverse
roughness pattern the load capacity increased while it was decreased for longitudinal roughness pattern.

Various methods were tried for improving the performance characteristics of bearing systems. It was exhibited experimentally magnetization increased the effective viscosity of the lubricant thereby, increasing the pressure distribution in the bearing systems. Consequently, magnetic fluid was employed as a lubricant.

A pioneer work for a foundation of magnetic granulometry—a method of determining of particle size with the aid of magnetic measurements was done by Elmore [1938a]; [1938b]. Stephen Pappell [1960] developed and classified Ferrofluid at NASA as a method for controlling fluids in space and which was used as rotating shaft seals in satellites. Agarwal [1963] discussed the problem of lubrication under the influence of a uniform magnetic field, when lubricant was an incompressible and electrically conducting liquid. The performance of one dimensional thrust bearings was analyzed by deriving the Reynolds’ equation. It was shown that the load carrying capacity increased while the frictional force decreased. Shukla [1963] derived a modified Reynolds’ equation governing the flow of an electrically conducting, incompressible, viscous lubricant in the presence of a magnetic field. This was applied to analyze the infinitely long and very narrow bearing. Neuringer and Rosenweig [1964] found an analytical solution for the problem of source flow with heat addition in
order to display the thermo-magnetic and magneto-mechanical effects attendant to simultaneous heat addition and fluid motion in the presence of a magnetic field. Shukla and Prakash [1966] analyzed theoretically squeeze film between two rectangular and elliptical plates in the presence of electrically conducting fluid. It was shown that load carrying capacity and time of approach increased as the strength of the applied magnetic field increased. Tarapov [1972] studied Ferrofluid lubrication for cylindrical bearings. Shliomis [1972] investigated the effect of a homogeneous magnetic field on the viscosity of a suspension with solid particles possessed intrinsic magnetic moments. It was shown that Brownian motion and hydrodynamic process exerted a disorienting effect on the magnetic moments. And it was established that the viscosity of magnetic suspensions dependent on the field strength. Prakash and Sinha [1976] studied theoretically squeezing flow in half and full journal bearings with micropolar fluid as a lubricant. In this paper a comparison was made between the performance of full and half bearings. Gupta and Bhat [1979] considered a hydromagnetic inclined porous slider bearing with a transverse magnetic field. It was observed that the load carrying capacity and friction increased significantly with the increase in Hartmann number. Patel and Gupta [1979] studied analytically the effect of a transverse magnetic field on the behaviour of a squeeze film between porous plates of different geometries. The effect of applied field was exhibited graphically. Shliomis and Raikher [1980] gave a review of
physical experiment with magnetic fluids. Various methods of enhancing the magnetic susceptibility were discussed. Malik and Singh [1980] derived a generalized Reynolds' equation for a magnetohydrodynamic journal bearing with the magnetic field perpendicular to the bearing axis and the electric current in the axial direction. The solution of the Reynolds' equation was expressed in the form of a double series which was revalidated by some sample results. Salem et al [1980] investigated the velocity distribution in rotating magnetohydrodynamic bearings. It was observed that rotational inertia effect had a significant effect on the radial velocity at small film thicknesses. Further, the results showed that the application of a magnetic field reduced both radial and tangential velocities. Salem et al [1982] studied effects on the performance of radial and rotational lubricant inertia for externally pressurized magnetohydrodynamic bearings. It was demonstrated that reduced radial inertia caused decrease in the load carrying capacity while reduced rotational inertia provided an increase in load carrying capacity. Verma [1986] investigated magnetic fluid-based squeeze film between two surfaces. It was observed that the time for the upper plate to come down was more than that for viscous squeeze film. Agrawal [1986] discussed a magnetic fluid based porous inclined slider bearing. It was shown that the magnetic-fluid-based porous inclined slider bearing registered a performance superior to that of the viscous porous inclined slider bearing. Phan-Thien et al [1987] reported the use of a numerical method to solve
the squeeze-film flow problem for a viscoelastic fluid. A simple explanation for the enhancement suggested that the stress overshoot appeared quickly and took a long time to die away, implying that steady-state viscous behaviour was not very relevant. Deheri and Bhat [1991] investigated the squeeze film behaviour between two annular discs, when the upper disc with a porous facing approached the parallel lower disc. It was observed that the performance of the bearing with the magnetic fluid lubricant was relatively better as compared to the traditional lubricant. Dinesh Kumar et al [1992] analyzed Ferrofluid squeeze film for spherical and conical bearings with a constant external magnetic field applied in the direction transverse to that of fluid flow. The solution was obtained by perturbation method in terms of non-dimensional Brownian relaxation time parameter and studied numerically the effect of magnetic fluid parameters on various bearing characteristics. Childs et al [1994] attempted to model the mechanics of magnetic fluid grinding in order to predict the onset of skidding motion. Prajapati [1995] analyzed the porous squeeze film, lubricated with a magnetic fluid in a bearing of various geometrical shapes like circular, annular, elliptical, infinitely long rectangular, conical, triangular and truncated conical, in the presence of an externally applied oblique magnetic field. The bearing working with magnetic fluid as lubricant had a superior performance with regards to that of an identical bearing working with conventional lubricant. Das [1998] studied slider bearings lubricated with couple stress fluids in magnetic field. A
A comparative study of optimum load carrying capacity for finite and infinite slider bearing was made. It was observed that both the values of maximum load capacity and corresponding inlet-outlet (10) film ratio depended on couple stress and magnetic parameters. Odenbach and Thurm [2002] exploited the advantages of magnetoviscous effects in Ferrofluids in order to apply in damping technology. Lin et al [2004] analyzed the squeeze-film performances between curved annular plates with an electrically conducting fluid in the presence of a transverse magnetic field. It was concluded that the magnetic field significantly enhanced pressure, load carrying capacity and response time. Burcan et al [2004] obtained experimental data for the lubricated bearings of precision. It was concluded that the inclusion of a magnetically active additive to a lubricant decreased motion resistance and improved the dynamics of the bearing system which led to increase in the durability of the whole machine. Odenbach [2004] reviewed the progress in magnetic fluid research and its applications particularly from a rheological point of view. The dependence of structures of magnetic nanoparticles on the magnetic field strengths and shear stress applied to the fluid resulted in strong changes of viscosity. Ochonski [2007] presented some new designs of sliding bearings lubricated with magnetic fluids (Ferrofluids) and the possibility of using them in modern bearing technology, in new computer and audiovisual equipment among others. The paper offered some new designs of compact, low friction and self-contained magnetic fluid sliding bearings and pointed
out their practical applications. *Huang et al* [2009] studied on static supporting capacity and tribological performance of Ferrofluids. It was shown that magnetic fluid under a magnet had a higher supporting capacity compared with the carrier liquid and the supporting capacity increased with the increased magnetization. With an external magnetic field applied, significant improvement in antifriction and wear resistance was obtained by using magnetic fluid with proper magnetization as lubricants. *Wang et al* [2010] analyzed theoretically the static characteristics of the journal bearing lubricated with a magnetic fluid. It was shown that the effect of magnetic field was to increase the load bearing capacity. *Oladeinde* and *Akpobi* [2010b] evaluated the effects of coupe stress lubrication on load carrying capacity of infinitely wide exponentially shaped slider bearing. *Huang et al* [2011] conducted experimental research on lubrication performances of Fe₃O₄ magnetic fluid lubricant. *Kuzhir et al* [2011] determined the free boundary of the lubricant layer of a Ferrofluid bearing under the action of a magnetic field and static load. It was shown that the magnetic field formed free boundaries of the lubricant layer and reduced cavitations' bubble as a result hindered the lubricant flowing out from the bearing. *Huang et al* [2011] studied the Ferrofluid lubrication with an external magnetic field. It was observed experimentally that Ferrofluids had a good friction reduction performance in the presence of an external magnetic field compared with the carrier liquid and that its lifetime of friction could be
greatly improved. Lu and Deng [2011] studied squeeze flow analysis of magnetorheological fluids between two parallel disks. Manivasakam and Sumathi [2011] investigated theoretically, the combined effects of fluid inertia forces, curvature of the disk and non-Newtonian couple stresses on the squeeze film behavior. A significant enhancement in the squeeze film behavior was noticed. These were explained through their effects on the squeeze film pressure and the load carrying capacity of the fluid film as a function of time. Recently, Lin [2012a] derived Ferrofluid lubrication equation of cylindrical squeeze films with convective fluid inertia forces and suggested application to circular disks. Also, Lin [2012] investigated the influence of fluid inertia process on the Ferrofluid squeeze film between a sphere and a plate in the presence of an external magnetic field. According to the results, the effect of fluid inertia process enhanced the load carrying capacity and increased the approaching time. Lin et al [2012a] derived a modified lubrication equation for hydromagnetic non-Newtonian cylindrical squeeze films. It was concluded from the comparison with the case of a non-conducting Newtonian lubricant that the hydromagnetic non-Newtonian effects provided better performance for circular squeeze films. Lin et al [2012] discussed the influences of convective fluid inertia forces in magnetic fluid-based conical squeeze film in the presence of external magnetic fields on the basis of the Shliomis Ferromagnetic fluid model. Gstottenbauer et al [2008] dealt with the experimental and numerical studies of squeeze mode behaviour of
magnetic fluids. Cavitation's effect was influenced by the squeeze mode behaviour. Ravaud et al [2009c] presented three dimensional study of a Ferrofluid seal along with centering effect and static capacity. The magnetic pressure was calculated numerically. Ravaud et al [2009b] considered a three dimensional analytical model for examining the shape and the pressure of Ferrofluid seals in totally ironless cylindrical structures. This approach was based on the computation of the magnetic fluid components created by ring permanent magnets whose polarizations were either radial or axial. Lin and Lu [2010] investigated theoretically the dynamic characteristics of wide exponential shaped slider bearings with an electrically conducting fluid in the presence of a transverse magnetic field. The improvement of bearing dynamic characteristics was more pronounced with increase in Hartmann numbers and decrease in minimum film thickness. Huang et al [2011] studied the Ferrofluid lubrication with an external magnetic field. It was observed experimentally that Ferrofluids turned in a good friction-reduction performance in the presence of an external magnetic field compared with the carrier liquid and that its lifetime of friction could be greatly improved.

A magnetic fluid was used to compensate the slip induced adverse affect. Singh and Gupta [2012] presented a theoretical investigation concerning the effect of Ferrofluid lubrication on the dynamic characteristics of a curved slider bearing based on Shliomis model based
magnetic fluid flow. It was observed that the effect of rotation of magnetic particles improved the stiffness and damping capacities of the bearing. Shah and Patel [2012a] presented the mathematical model of Ferrofluid lubricated slider bearing with an inclined pad surface [plate or pad] including combined effects of porosity, anisotropic permeability, slip velocity at both the ends and squeeze velocity under an oblique magnetic field. Various shaped sizes of the porous matrix at both the ends were also discussed for the possible optimization of the bearing characteristics.

The roughness of the surfaces has an adverse affect on the performance of the bearing systems in almost all situations. In order to counter the adverse effect of roughness the use of magnetic fluid was thought to be appropriate.

Salama and Mech [1950] investigated the effect of surface micro-roughness on the performance of a parallel thrust bearing theoretically and experimentally. It was observed that the micro-roughness played an important role in the behavior of the bearing, as it provided passages with both feed the surfaces as the lubricated and allowed to the formation of hydrodynamic films showed that the performance corresponded closer to that for film lubrication.

Hamilton et al [1966] developed a theory of hydrodynamic lubrication between two parallel surfaces with roughness on one or both of the surfaces. It was verified results theoretically and experimentally that
surface roughness helps in the pressure buildup between the two surfaces and maintained the load support which kept the surfaces from collapsing into each other.

Christensen [1969] dealt with hydrodynamic aspects of rough bearing surfaces. Two different forms of Reynolds' type equation corresponding to transverse and longitudinal roughness were developed. It was established that surface roughness considerably influenced the operating characteristics of the bearing and that the direction of the influence depended on the type of roughness pattern. Patel and Hingu [1978] analyzed the effect of a uniform transverse magnetic field on the squeeze film behavior when a circular disk with a porous facing approached another disk with uniform velocity. The pressure distribution, load carrying capacity and film thickness were obtained as functions of time as revealed by the analysis. Verma and Vedan [1979] investigated the steady rotation of a non-conducting sphere in an unbounded electrically non-conducting incompressible paramagnetic fluid. Elshekh et al [1996] obtained a similarity solution for the flow of an electrically conducting viscous fluid film squeezed between two rotating disks. It was found that the axial magnetic force increased the azimuthal velocity of the fluid while the angular velocity of the upper disk dispersed the axial and azimuthal components of the induced magnetic field. Shah and Bhat [2007] investigated the effect of anisotropic permeability and slip velocity on the action of the squeeze film formed when a secant shaped circular upper plat...
with a porous facing approached an impermeable and flat circular lower plate considering rotation of the plates. It was observed that the pressure distribution and the load carrying capacity decreased with the increase in radial permeability, slip parameter and rotational inertia while they were increased with increase in axial permeability and material constant of the Jenkins model. Zueco and Beg [2010] studied numerically the hydromagnetic squeeze film between two rotating disks using the numerical network simulation method. The study found applications in hydromagnetic lubrication of braking devices, slider bearings, rotating machinery, in hydraulic shock absorbers in electromagnetic braking for potential spacecraft in planetary orbits. Bujurke et al [2011] analyzed the effect of surface roughness on the magnetohydrodynamic squeeze film behavior between two rectangular plates. The magnetic effect improved the performance of the squeeze film lubrication as compared to the classical non-conducting lubricant case. Further, the roughness had a significant effect on the squeeze film behaviour. Sukla and Deheri [2011] analyzed the performance of a rough hyperbolic slider bearing under the presence of a magnetic fluid lubricant. It was noticed that positively skewed roughness and variance (positive) decreased the friction.

Wu [1972a] analyzed the squeeze film between porous rectangular plates which was extended to including the effect of velocity at the fluid and porous material interface. It was concluded that the existence of slip
velocity further reduced the load-carrying capacity and the response time of the squeeze film.

Even to compensate the combined effect of slip velocity and porosity, magnetic fluid was adopted as a lubricant.

*Bhat* [1980] derived a general differential equation for the lubrication of an anisotropic porous slider in the presence of a transverse magnetic field. The effect of slip velocity was considered there in.

*Ting* [1975] analyzed the lubricated clutch engagement behavior of two annular disks, one of which was covered with a layer of porous material which is considered to be elastically deformable having rough surfaces. *Prakash* and *Peeken* [1985] studied numerically the combined effect of two-sided roughness and elastic deformation on the hydrodynamic lubrication of a one-dimensional slider bearing problem. It was established that there was a strong interaction between the roughness and elasticity but elasticity acted so as to decrease the roughness effects.

Recently, the limited use of magnetic fluid as a lubricant to minimize the negative effect of porosity and roughness has been investigated.

*Patel et al.* [2011] investigated the performance of a magnetic fluid based squeeze film between rough circular plates while the upper plate had a porous facing of variable porous matrix thickness. It was observed that these performance characteristics increased with the increase in magnetization parameter and suggested that the bearing with magnetic
fluid lubricant had better performance characteristics than that with the conventional lubricant. In addition, it was noticed that with a proper selection of the thickness ratio parameter, a magnetic fluid based squeeze film bearing with variable porous matrix thickness, in the case of negatively skewed roughness could be made to perform considerably better than that of a conventional porous bearing with a uniform porous matrix thickness working with a conventional lubricant. Recently, Shukla and Deheri [2012] investigated the performance of an infinitely long rough porous journal bearing in the presence of a magnetic fluid lubricant. The results suggested that the bearing system registered an enhanced performance as compared to that of a bearing system working with a conventional lubricant.

Shimpi and Deheri [2010a] analyzed the performance of a magnetic fluid-based transversely rough short bearing. It was concluded that the negative effect of standard deviation can be compensated by the positive effect of the magnetization parameter in the case of negatively skewed roughness by suitably choosing the aspect ratio. Shimpi and Deheri [2011a] analyzed the performance of a magnetic fluid based squeeze film between porous infinitely long rough rectangular plates. This investigation indicated that the bearing system registered an improved performance as compared to that of the bearing system working with conventional lubricant. It was revealed that the negative effect of the standard deviation and porosity could be mitigated to some extent by the positive effect of
magnetization parameters in the case of negatively skewed roughness by suitably choosing the aspect ratio. Shimpi and Deheri [2011a] analyzed the effect of deformation on the performance of a short bearing. It was established that the deformation had a considerable adverse effect which could be reduced by choosing a suitable combination of the aspect ratio and the parameters $h_2/L$ and $h_2/B$. Recently, Shimpi and Deheri [2012a] investigated the squeeze film behavior between a curved rough porous circular plate and a flat rough porous circular plate under the presence of a magnetic fluid lubricant. The curved film thickness was described by a secant function. The computed results suggested that the performance of bearing system enhanced considerably as compared to that of a bearing system working with a conventional lubricant as the magnetization increases the effective viscosity of the lubricant. Shimpi and Deheri [2012b] investigated the behaviour of a magnetic fluid based squeeze film between two rotating transversely rough porous circular plates taking bearing deformation into consideration. It was found that the combined effect of rotation and deformation caused significantly reduced load carrying capacity while the adverse effect of porosity, deformation and standard deviation could be compensated up to some extent by the positive effect of magnetic fluid lubricant in the case of negatively skewed roughness by suitably choosing curvature parameters.