INTRODUCTION CHAPTER 1

1.1 INTERNAL COMBUSTION ENGINE

The invention of internal combustion engine creates great revolution to mankind in the world. There are different types of internal combustion engines which were built and tested in the second half of the 19th century. These engines operated with various variables in different mechanical systems and engine cycles. The first engine was invented by J.J.E. Lenoir (1822-1900) about 1860. The internal combustion engine is a device that converts chemical energy of fuel into useful mechanical energy, usually made available for rotating output shaft. Chemical energy of the fuel is converted to thermal energy by means of combustion or oxidation with air inside the engine. This thermal energy raises the temperature and high pressure of the gases within the engine, it made to expand against the mechanical mechanisms of the engine. This expansion of reciprocating motion is converted by the mechanical linkages of the engine to a rotating crankshaft, which is the output of the engine. The rotation of crankshaft, in turn, is connected to a transmission and/or power train to transmit the rotating mechanical energy to the desired final use. Most internal combustion engines are reciprocating engines having pistons that reciprocate back and forth in cylinders internally within the engine. Reciprocating engines can have one cylinder or up to 20 or more. The cylinders can be arranged in many different geometric configurations and various sizes have range from small model airplane engines with output of the order of 100 watts to large multi-cylinder stationary engines that produce thousands of kilowatts per cylinder [1].

1.1.1 ENGINE CLASSIFICATION

Internal combustion engines can be classified in a number of different ways:

1. Types of Ignition

(a) Spark Ignition (SI). An SI engine starts the combustion process in each cycle by use of a spark plug. The spark plug gives a high-voltage electrical discharge between two electrodes which ignites the air-fuel mixture in the combustion chamber surrounding the plug. In early engine development, before the invention of the electric spark plug, many forms of torch holes were used to initiate combustion from an external flame.

(b) Compression Ignition (CI). The combustion process in a CI engine starts when the air-fuel mixture self-ignites due to high temperature in the combustion chamber caused by high compression.

2. Engine Cycle

(a) Four-Stroke Cycle. A four-stroke cycle experiences four piston movements over two engine revolutions for each cycle.

(b) Two-Stroke Cycle. A two-stroke cycle has two piston movements over one revolution for each cycle.

3. **Basic Design**

(a) Reciprocating. Engine has one or more cylinders in which pistons reciprocate back and forth. The combustion chamber is located in the closed end of each cylinder. Power is delivered to a rotating output crankshaft by mechanical linkage with the pistons.

(b) Rotary. Engine is made of a block (stator) built around a large non-concentric rotor and crankshaft. The combustion chambers are built into the non-rotating block.

1.1.2 ENGINE COMPONENTS

Block Body of engine containing the cylinders, made of cast iron or aluminum. In many older engines, the valves and valve ports were contained in the block. The block of water-cooled engines includes a water jacket cast around the cylinders. On air-cooled engines, the exterior surface of the block has cooling fins.

Camshaft Rotating shaft used to push open valves at the proper time in the engine cycle, either directly or through mechanical or hydraulic linkage (push rods, rocker arms, tappets). Most modern automobile engines have one or more camshafts mounted in the engine head (overhead cam). Most older engines had camshafts in the crankcase. Camshafts are generally made of forged steel or cast iron and are driven off the crankshaft by means of a belt or chain (timing chain). To reduce weight, some cams are made from a hollow shaft with the cam lobes press-fit on. In four-stroke cycle engines, the camshaft rotates at half engine speed.

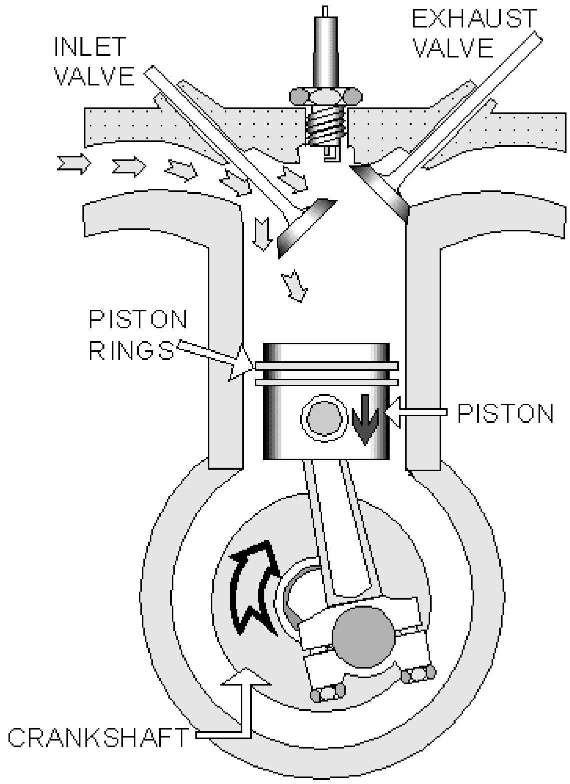


Fig. 1. Components of an I.C. engine

Catalytic converter Chamber mounted in exhaust flow containing catalytic material that promotes reduction of emissions by chemical reaction.

Combustion chamber The end of the cylinder between the head and the piston face where combustion occurs. The size of the combustion chamber continuously changes from a minimum volume when the piston is at TDC to a maximum when the piston is at BDC. The term "cylinder" is sometimes synonymous with "combustion chamber" (e.g., "the engine was firing on all cylinders"). Some engines have open combustion chambers which consist of one chamber for each cylinder. Other engines have divided chambers which consist of dual chambers on each cylinder connected by an orifice passage.

Connecting rod Rod connecting the piston with the rotating crankshaft, usually made of steel or alloy forging in most engines but may be aluminum in some small engines.

Connecting rod bearing: Bearing where connecting rod fastens to crankshaft.

Cooling fins Metal fins on the outside surfaces of cylinders and head of an air-cooled engine. These extended surfaces cool the cylinders by conduction and convection.

Crankcase Part of the engine block surrounding the rotating crankshaft. In many engines, the oil pan makes up part of the crankcase housing.

Crankshaft Rotating shaft through which engine work output is supplied to external systems. The crankshaft is connected to the engine block with the main bearings. It is rotated by the reciprocating pistons through connecting rods connected to the crankshaft, offset from the axis of rotation. This offset is sometimes called crank throw or crank radius. Most crankshafts are made of forged steel, while some are made of cast iron.

Cylinders The circular cylinders in the engine block in which the pistons reciprocate back and forth. The walls of the cylinder have highly polished hard surfaces. Cylinders may be machined directly in the engine block, or a hard metal (drawn steel) sleeve may be pressed into the softer metal block. Sleeves may be dry sleeves, which do not contact the liquid in the water jacket, or wet sleeves, which form part of the water jacket. In a few engines, the cylinder walls are given a knurled surface to help hold a lubricant film on the walls. In some very rare cases, the cross section of the cylinder is not round.

Exhaust manifold Piping system which carries exhaust gases away from the engine cylinders, usually made of cast iron.

Exhaust system Flow system for removing exhaust gases from the cylinders, treating them, and exhausting them to the surroundings. It consists of an exhaust manifold which carries the exhaust gases away from the engine, a thermal or catalytic converter to reduce emissions, a muffler to reduce engine noise, and a tailpipe to carry the exhaust gases away from the passenger compartment.

Fan Most engines have an engine-driven fan to increase air flow through the radiator and through the engine compartment, which increases waste heat removal from the engine. Fans can be driven mechanically or electrically, and can run continuously or be used only when needed.

Flywheel Rotating mass with a large moment of inertia connected to the crankshaft of the engine. The purpose of the flywheel is to store energy and furnish a large angular momentum that keeps the engine rotating between power strokes and smoothes out engine operation. On some aircraft engines the propeller serves as the flywheel, as does the rotating blade on many lawn mowers [7].

1.1.3 PISTON-RINGS

In the early steam engines no piston rings were used. The temperatures and the steam pressures were not as high as the corresponding parameters in today's internal combustion engines, and the need for considering thermal expansions and clearances was smaller. Increasing power demands required higher temperatures, which caused stronger heat expansion of the piston material. This made it necessary to use a sealant between the piston and the cylinder liner to allow a decrease in the clearance in cold conditions, i.e. when the clearances were at their maximum.

Keeping the clearance between the piston and liner wall at a minimum considerably reduces the combustion gas flow from the combustion chamber past the piston. The first piston rings used in an engine had the sole task of sealing off the combustion chamber, thus preventing the combustion gases from trailing down into the crankcase. This development increased the effective pressure on the piston. Ramsbottom and Miller were among the pioneers to investigate the behavior of the piston rings in steam engines. Ramsbottom, in 1854, constructed a single-piece, metallic piston ring. The free diameter of the ring was 10 per cent larger than the diameter of the cylinder bore. When fitted in a groove in a piston, the ring was pressed against the cylinder bore by its own elasticity. Previous piston rings had consisted of multiple pieces and with springs to provide an adequate sealing force against the cylinder bore. Miller, in 1862, introduced a modification to the Ramsbottom ring. This modification consisted of allowing the steam pressure to act on the backside of the ring, hence providing a higher sealing force. This new solution enabled the use of more flexible rings, which conformed better to the cylinder bore (Priest and Taylor, 2000). In the early days, the ring pack was lubricated solely by splash lubrication; i.e. lubrication by the splashing of the rotating crankshaft into the crankcase oil surface [6].

Subsequently, when the combustion conditions became even more demanding, i.e. with higher temperatures, pressures and piston speeds, oil control rings were introduced. A proper lubricant film on the piston, piston rings and liner wall was required in order to prevent damage. The oil control rings were, and are, especially designed to appropriately distribute the oil on the cylinder liner and to scrape off surplus oil to be returned to the crankcase.

Some other functions of rings:-

Some of the combustion chamber heat energy is transferred through the piston to the piston boundaries, i.e. the piston skirt and rings, from which heat transfers to the liner wall. Furthermore, the piston rings prevent excess lubrication oil from moving into the combustion chamber by scraping the oil from the liner wall during the down stroke. The piston rings support the piston and thus reduce the slapping motion of the piston, especially during cold starts where the clearance is greater than in running conditions [5].

Compression rings:-

The cylinder gas pressure acts on the back-side of the ring, especially on the top ring, pressing it against the liner. The ring force distribution depends on the face form. With a rectangular face profile the force is higher than with a barrel-shaped face, as the compression pressure is able to act on the face-side of the barrel-shaped ring and thus counteract some of the force owing to ring pre-tension. Plain compression rings, with a rectangular cross-section, satisfactorily meet the sealing demands of ordinary running conditions and this type of compression ring is the most common one .The tapered face profile enables the compression gas pressure to act on the face-side as well and thus relieve the pressure against the liner wall, which reduces the wear rate during running-in. A tapered face profile has a good oil-scraping ability, and the ring can be used as an oil-scraping ring as well as a compression ring. The bevelled profile causes the ring to twist in the ring groove during engine operation. In running conditions the bevelled ring is pressed flat against the liner wall owing to the gas pressure, which causes an additional stress on the ring. The wedge-type profile or (half) keystone profile is used in order to prevent the ring from seizing in the groove. High temperature may cause the lubricant in the groove to carbonize.

The wedge form makes ring´s axial clearance greater at increasing radial groove clearance. Scraper rings, which are usually used as the second compression rings, can simultaneously be used as oil-scraper rings [5].

Oil control ring:-

The appearance of the oil control ring differs from that of the compression ring. The oil control ring is perforated by slots in the peripheral direction which provides a way for the excess oil to leave the ring pack area. The scraped oil is collected in the oil control ring groove and transported through the piston back to the crankcase. The scraped oil may run through the possible gap between the liner wall and the piston skirt. For this oil is forced in front of the oil control ring. The oil control rings may have a coil spring inserted, as the pre-tension of the ring is not sufficient in all instances.[5]

1.2. WEAR

1.2.1 INTRODUCTION: Wear is a major cause of material wastage, so any reduction of wear can affect considerable savings. The friction will cause the displacement and removal of surface material, so wear can cause the failure of the element, which is in 70 to 80 percent of the total failure. Wear is the progressive damage, which occurs on the surface of a component as a result of its motion, relative to the adjacent working parts. It is defined as removal of material from the surface of component. Wear is one of the main causes that lead to replace the important components of any running system. It depends on many variables such as Hardness, Impact strength, Toughness, Modulus of elasticity, Corrosion resistance, Fatigue resistance, Surface finish, Lubrication, Load, Speed, Temperature, Properties of opposing surface etc [4].

1.2.2 THE HARM CAUSED BY WEAR

Influence the performance quality of machine ; e.g. the wear of the gear tooth surface will destroy the involute surface, and then cause the impact and vibration. The wear of the machinery main shaft bearing will influence the machining precision of element. Impaired the efficiency of machine e.g. the wear of cylinder sleeve of the diesel engines will cause inadequate functioning of power. Reduced the reliability of machine e.g. snaggletooth , wheel track.

1.2.3 THE WEAR PROCESS

Three Phases of Machine Wear:

1. Running-in

Time: the initial stages of wear

Characteristic: high rate of wear (the contact area is small, so the contact stress is large.)

The function of running-in: The material of high spots will be deformed, so the contact area becomes sufficiently large to support the load. It is a process that increases gradually to load, gradually to speed.

Attention to running-in : Load from light load to heavy load should be slow. The lubricant should prevent from pollution; should replace after running-in.

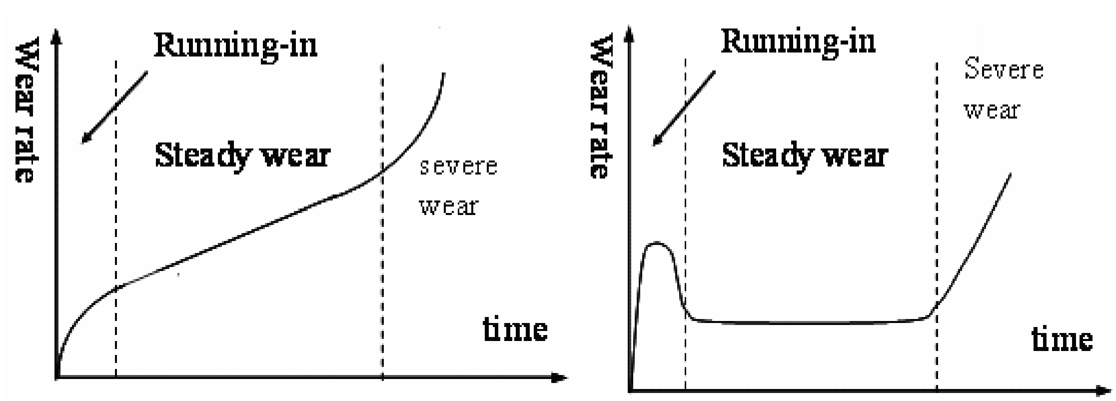


Fig.2. wear rate vs time plot

a) **Steady Wear Phase:**

   Time : late stage of running-in, long-lasting; in the working stage of machine (stand for the useful time)

Characteristic: the speed of wear of slow-motion and reposeful e.g.：the cylinder sleeve of locomotive and diesel engine replace every period

b) Severe Wear Phase:

Time：late stage of normal wear (late stage of life) characteristic：rate of wear will speed up, so the failure of surface element will occur or the clearance of kinematic pair will augment [4].

1.2.4 CLASSIFICATION OF WEAR

A. Adhesive wear:

The asperity of friction surface after cold welding the material will transfer to another surface when there is relative sliding, that is called adhesive wear. Adhesive Wear occurs when two metal surfaces come in contact allowing particles to break away from the components. Insufficient lubrication or lubricant contamination normally causes this. Ensuring the proper viscosity grade lubricant is used can reduce adhesive wear. Reducing contamination in the oil will also help eliminate adhesive wear. Adhesive wear is produced by the formation and subsequent shearing of welded junction between two sliding surfaces. For adhesive wear it is necessary for the surfaces to be in intimate contact with each other. Surfaces, which are held apart by lubricating films, oxide films etc. reduce the tendency for adhesion to occur [4].

Archard's equation for adhesive wear is

$$V/L = K\,W/H$$

Where V = Wear volume

L = Sliding length

W = Normal load

H = Hardness

K = Wear coefficient.

B.Abrasive wear:

The wear is because of the dissociated rigid particles or the peak of asperity. Abrasive Wear is the results of hard particles coming in contact with internal components. Such particles include dirt and a variety of wear metals. Introducing a filtration process can reduce abrasive wear. It is also important to ensure vents, breathers, and seals are working properly. The harder asperities trapped at the interface, when moving relative to the surface, cause abrasion of the surface and the resulting damage is called abrasive wear. The abrasive wear mechanism is basically same as machining, grinding, polishing or lapping that are used for shaping materials. The body abrasive wear occurs when one surface (usually harder than the second) although this mechanism very often changes to three body abrasion as wear debris then acts as an abrasive between two surfaces [4].

Abrasive can act as in grinding where the abrasive is fixed relative to one surface of as in lapping where the abrasive tumble producing a series of indentations as opposed to scratch. Archard has proposed an equation

$$V/L=K_{ab}\,W/H$$

In this equation K_{ab} represents the non-dimensional abrasive wear coefficient. The coefficient depends on the nature of the abrasive and extent of cutting action. One important parameter is the relative hardness of the abrasive to that of the metal. When this ratio exceeds 1.4 strong abrasive action is expected. In general the adhesive wear coefficients tend to be higher for metallic material in comparison to adhesive wear. It is hence important to have efficient filtration system to remove abrasive contaminants.Influenced factor：

(1)The harder the material is, the higher the staying quality is.

(2)The wearing capacity is increasing along with the augment of the particles' size.

(3)The wearing capacity is increasing along with the augment of the hardness of the particles.

C. Fatigue wears (pitting)

It occurs when there is a slight vibratory movement among loaded surfaces in contact and which manifests itself by the pitting of the surfaces and accumulation of oxidized debris. The debris occupies a greater volume than that of the metal destroyed and in a limited space. Fatigue Wear results when cracks develop in the component surface allowing the generation and removal of particles. Fatigue wear can occur in non-conforming machine elements such as rolling element bearing and gas in the form of rolling contact fatigue and pitting. It occurs when the contact stresses approaches the elastic limit. The number of stress cycles necessary to cause failure decreases with increasing stress. Leading causes of fatigue wear include insufficient lubrication, lubricant contamination, and component fatigue [4].

D. Corrosive Wear

The acid and fuel from the air combusting little inorganic acid (like sulphuric acid), and electrochemical action of water together cause the lessening of the surface of material. Corrosive Wear is caused by a chemical reaction that actually removes material from a component surface. Corrosion can be a direct result of acidic oxidation. A random electrical current can also cause corrosion. Electrical current corrosion results in welding and pitting of the wear surface. The presence of water or combustion products can promote corrosive wear.

E. Cutting Wear

Cutting Wear can be caused when an abrasive particle has imbedded itself in a soft surface. Equipment imbalance or misalignment can contribute to cutting wear. Proper filtration and equipment maintenance is imperative to reducing cutting wear.

F. Fretting wear

Fretting wear is a small amplitude oscillatory motion, usually between two solid surfaces in contact. Fretting wear occurs when repeated loading and unloading causes cyclic stresses, which induces surface or subsurface break-up and loss of material. Vibration is a common cause of fretting wear [8].

G. Sliding Wear

Sliding Wear is caused by equipment stress. Subjecting equipment to excessive speeds or loads can result in sliding wear. The excess heat in an overload situation weakens the lubricant and can result in metal-to-metal contact. When a moving part comes in contact with a stationary part sliding wear becomes an issue.

H. Erosive wear

Erosive wear is caused by a gas or liquid, which may or may not carry entrained solid particles, impinging on a surface. When the angle of impingement is small, the wear produced is closely analogous to abrasive. When the angle of impingement is normal to the surface, material is displaced by plastic flow or is dislodged by brittle failure [8]

1.3 FRICTION

1.3.1 INTRODUCTION

Friction is the force resisting the relative lateral (tangential) motion of solid surfaces, fluid layers, or material elements in contact. It is usually subdivided into several varieties: Dry friction resists relative lateral motion of two solid surfaces in contact. Dry friction is also subdivided into static friction between non-moving surfaces, and kinetic friction (sometimes called sliding friction or dynamic friction) between moving surfaces. Lubricated friction or fluid friction resists relative lateral motion of two solid surfaces separated by a layer of gas or liquid. Fluid friction is also used to describe the friction between layers within a fluid that are moving relative to each other. Skin friction is a component of drag, the force resisting the motion of a solid body through a fluid. Internal friction is the force resisting motion between the elements making up a solid material while it undergoes deformation. Several famous scientists and engineers contributed to our understanding of friction. They include Leonardo da Vinci, Guillaume Amontons, John Theophilus Desaguliers, Leonard Euler, and Charles-Augustin de Coulomb. Their findings are codified into these laws:

1. The force of friction is directly proportional to the applied load.
2. The force of friction is independent of the apparent area of contact. . (Amontons' 2nd Law) (Amontons' 2nd Law does not work for elastic, deformable materials. For example, wider tires on cars provide more traction than narrow tires for a given vehicle mass because of surface deformation of the tire)

1.3.2Types of friction

1. Static friction

Static friction is independent of the sliding velocity. (Coulomb's Law of Friction) Friction is not a fundamental force, as it is derived from electromagnetic force between charged particles, including electrons, protons, atoms, and molecules, and so cannot be calculated from first principles, but instead must be found empirically. When contacting surfaces move relative to each other, the friction between the two surfaces converts kinetic energy into thermal energy, or heat. Contrary to earlier explanations, static friction is now understood not to be caused by surface roughness but by chemical bonding between the surfaces. Surface roughness and contact area, however, do affect kinetic friction for micro- and nano-scale objects where surface area forces dominate inertial forces.

2. Coulomb friction

It is named after Charles-Augustin de Coulomb, is a model used to calculate the force of dry friction. It is governed by the equation:

$F_f \leq \mu\ F_{n,}$ Where

F_f is the force exerted by friction (in the case of equality, the maximum possible magnitude of this force).

μ is the coefficient of friction, which is an empirical property of the contacting materials,

F_n is the normal force exerted between the surfaces.

Surfaces at rest relative to each other $\mu = \mu_s$, where μ_s is the coefficient of static friction. This is usually larger than its kinetic counterpart. The Coulomb friction F_f may take any value from zero up to μF_n , and the direction of the frictional force against a surface is opposite to the motion that surface would experience in the absence of friction. Thus, in the static case, the frictional force is exactly what it must be in order to prevent motion between the surfaces; it balances the net force tending to cause such motion. In this case, rather than providing an estimate of the actual frictional force, the Coulomb approximation provides a threshold value for this force, above which motion would commence.

For surfaces in relative motion $\mu = \mu_k$, where μ_k is the coefficient of kinetic friction. The Coulomb friction is equal to F_f, and the frictional force on each surface is exerted in the direction opposite to its motion relative to the other surface. This approximation mathematically follows from the assumptions that surfaces are in atomically close contact only over a small fraction of their overall area, that this contact area is proportional to the normal force (until saturation, which takes place when all area is in atomic contact), and that frictional force is proportional to the applied normal force, independently of the contact area (you can see the experiments on friction from Leonardo Da Vinci). Such reasoning aside, however, the approximation is fundamentally an empirical construction. It is a rule of thumb describing the approximate outcome of an extremely complicated physical interaction. The strength of the approximation is its simplicity and versatility – though in general the relationship between normal force and frictional force is not exactly linear (and so the frictional force is not entirely independent of the contact area of the surfaces), the Coulomb approximation is an adequate representation of friction for the analysis of many physical systems.

3.Kinetic friction

Kinetic (or dynamic) friction occurs when two objects are moving relative to each other and rub together (like a sled on the ground). The coefficient of kinetic friction is typically denoted as μ_k, and is usually less than the coefficient of static friction for the same materials. In fact, Richard Feynman reports that "with dry metals it is very hard to show any difference." Finally, new models are beginning to show how kinetic friction can be greater than static friction. Examples of kinetic friction: Kinetic friction is when two objects are rubbing against each other. The resistance felt when pushing a book across a desk is an example of kinetic friction. Fluid friction is the interaction between a solid object and a fluid (liquid or gas), as the object moves through the fluid. The skin friction of air on an airplane and of water on a swimmer are two examples of fluid friction. This kind of friction is not only due to rubbing, which generates a force tangent to the surface of the object (such as sliding friction). It is also due to forces that are orthogonal to the surface of the object. These orthogonal forces significantly (and mainly, if relative velocity is high enough) contribute to fluid friction.. Since rubbing is not its only cause, in modern fluid dynamics the same force is typically referred to as drag or fluid resistance, while the force component due to rubbing is called skin friction. Notice that a fluid can in some cases exert, together with drag, a force orthogonal to the direction of the relative motion of the object (lift).

1.3.3 Coefficient of friction

The coefficient of friction (COF), also known as a frictional coefficient or friction coefficient and symbolized by the Greek letter μ, is a dimensionless scalar value which describes the ratio of the force of friction between two bodies and the force pressing them together. The coefficient of friction depends on the materials used; for example, ice on steel has a low coefficient of friction, while rubber on pavement has a high coefficient of friction. Coefficients of friction range from near zero to greater than one – under good conditions.

When the surfaces are conjoined, Coulomb friction becomes a very poor approximation (for example, adhesive tape resists sliding even when there is no normal force, or a negative normal force). In this case, the frictional force may depend strongly on the area of contact. Some drag racing tires are adhesive in this way. However, despite the complexity of the fundamental physics behind friction, the relationships are accurate enough to be useful in many applications.

The force of friction is always exerted in a direction that opposes movement (for kinetic friction) or potential movement (for static friction) between the two surfaces. For example, a curling stone sliding along the ice experiences a kinetic force slowing it down. For an example of potential movement, the drive wheels of an accelerating car experience a frictional force pointing forward; if they did not, the wheels would spin, and the rubber would slide backwards along the pavement. Note that it is not the direction of movement of the vehicle they oppose; it is the direction of (potential) sliding between tire and road.The coefficient of friction is an empirical measurement – it has to be measured experimentally, and cannot be found through calculations. Rougher surfaces tend to have higher effective values. Most dry materials in combination have friction coefficient values between 0.3 and 0.6. Values outside this range are rarer, but Teflon, for example, can have a coefficient as low as 0.04. A value of zero would mean no friction at all, an elusive property – even magnetic levitation vehicles have drag. Rubber in contact with other surfaces can yield friction coefficients from 1 to 2. Occasionally it is maintained that μ is always < 1, but this is not true. While in most relevant applications $\mu < 1$, a value above 1 merely implies that the force required to slide an object along the surface is greater than the normal force of the surface on the object. For ex, silicone rubber or acrylic rubber-coated surfaces have a coefficient of friction that can be substantially larger than one. Both static and kinetic coefficients of friction depend on the pair of surfaces in contact; their values are usually approximately determined experimentally. For a given pair of surfaces, the coefficient of static friction is usually larger than that of kinetic friction; in some sets the two coefficients are equal, such as Teflon-on-Teflon. In the case of kinetic friction, the direction of the friction force may or may not match the direction of motion: a block sliding atop a table with rectilinear motion is subject to friction directed along the line of motion; an automobile making a turn is subject to friction acting perpendicular to the line of motion (in which case it is said to be 'normal' to it). The direction of the static friction force can be visualized as directly opposed to the force that would otherwise cause motion, were it not for the static friction preventing motion. In this case, the friction force exactly cancels the applied force, so the net force given by the vector sum, equals zero. It is important to note that in all cases, Newton's first law of motion holds. While it is often stated that the COF is a "material property," it is better categorized as a "system property." Unlike true material properties (such as conductivity, dielectric constant, yield strength), the COF for any two materials depends on system variables like temperature, velocity, atmosphere and also what are now popularly described as aging and de-aging times; as well as on geometric properties of the interface between the materials. For example, a copper pin sliding against a thick copper plate can have a COF that varies from 0.6 at low speeds (metal sliding against metal) to below 0.2 at high speeds when the copper surface begins to melt due to frictional heating. The latter speed, of course, does not determine the COF uniquely; if the pin diameter is increased so that the frictional heating is removed rapidly, the temperature drops, the pin remains solid and the COF rises to that of a 'low speed' test.

1.3.4The normal force

The normal force is defined as the net force compressing two parallel surfaces together; and its direction is perpendicular to the surfaces. In the simple case of a mass resting on a horizontal surface, the only component of the normal force is the force due to gravity, where **N=mg**.

In this case, the magnitude of the friction force is the product of the mass of the object, the acceleration due to gravity, and the coefficient of friction. However, the coefficient of friction is not a function of mass or volume; it depends only on the material. For instance, a large aluminum block has the same coefficient of friction as a small aluminum block. However, the magnitude of the friction force itself depends on the normal force, and hence the mass of the block. If an object is on a level surface and the force tending to cause it to slide is horizontal, the normal force N between the object and the surface is just its weight, which is equal to its mass multiplied by the acceleration due to earth's gravity, *g*. If the object is on a tilted surface such as an inclined plane, the normal force is less, because less of the force of gravity is perpendicular to the face of the plane. Therefore, the normal force, and ultimately the frictional force, is determined using vector analysis, usually via a free body diagram. Depending on the situation, the calculation of the normal force may include forces other than gravity.Static friction is friction between two solid objects that are not moving relative to each other. For example, static friction can prevent an object from sliding down a sloped surface. The coefficient of static friction, typically denoted as μ_s, is usually higher than the coefficient of kinetic friction. The static friction force must be overcome by an applied force before an object can move. The maximum possible friction force between two surfaces before sliding begins is the product of the coefficient of static friction and the normal force: $f= \mu_s * F_n$.

When there is no sliding occurring, the friction force can have any value from zero up to F_{max}. Any force smaller than F_{max} attempting to slide one surface over the other is opposed by a frictional force of equal magnitude and opposite direction. Any force larger than F_{max} overcomes the force of static friction and causes sliding to occur. The instant sliding occurs, static friction is no longer applicable and kinetic friction becomes applicable.An example of static friction is the force that prevents a car wheel from slipping as it rolls on the ground. Even though the wheel is in motion, the patch of the tire in contact with the ground is stationary relative to the ground, so it is static rather than kinetic friction.

The maximum value of static friction, when motion is impending, is sometimes referred to as limiting friction, although this term is not used universally.

1.3.5Angle of friction

For the friction angle between granular materials, see repose. For certain applications it is more useful to define static friction in terms of the maximum angle before which one of the items will begin sliding. This is called the angle of friction or friction angle. It is defined as:

$$\tan \theta = \mu$$

Where θ is the angle from horizontal and μ is the static coefficient of friction between the objects. This formula can also be used to calculate μ from empirical measurements of the friction angle.

Rolling resistance

Rolling resistance is the force that resists the rolling of a wheel or other circular object along a surface caused by deformations in the object and/or surface. Generally the force of rolling resistance is less than that associated with kinetic friction. Typical values for the coefficient of rolling resistance are 0.001. One of the most common examples of rolling resistance is the movement of motor vehicle tires on a road, a process which generates heat and sound as by-products.

1.3.6 Energy of friction

According to the law of conservation of energy, no energy is destroyed due to friction, though it may be lost to the system of concern. Energy is transformed from other forms into heat. A sliding hockey puck comes to rest because friction converts its kinetic energy into heat. Since heat quickly dissipates, many early philosophers, including Aristotle, wrongly concluded that moving objects lose energy without a driving force.

When an object is pushed along a surface, the energy converted to heat is given by: $E_{th} = \mu_k \int F_n (x)\, dx$

This is proved by mass balancing on a differential control volume and a flow chart is developed for solving Reynold's equation and also to find film thickness and other properties for given boundary conditions.

1.3.7 Work of friction

In the reference frame of the interface between two surfaces, static friction does no work, because there is never displacement between the surfaces. In the same reference frame, kinetic friction is always in the direction opposite the motion, and does negative work. However, friction can do positive work in certain frames of reference. One can see this by placing a heavy box on a rug, then pulling on the rug quickly. In this case, the box slides backwards relative to the rug, but moves forward relative to the frame of reference in which the floor is stationary. Thus, the kinetic friction between the box and rug accelerates the box in the same direction that the box moves, doing positive work. The work done by friction can translate into deformation, wear, and heat that can affect the contact surface properties (even the coefficient of friction between the surfaces). This can be beneficial as in polishing. The work of friction is used to mix and join materials such as in the process of friction welding. Excessive erosion or wear of mating surfaces occur when work due frictional forces rise to unacceptable levels. Harder corrosion particles caught between mating surfaces (fretting) exacerbates wear of frictional forces. Bearing seizure or failure may result from excessive wear due to work of friction. As surfaces are worn by work due to friction, fit and surface finish of an object may degrade until it no longer functions properly.

Approximate coefficients of friction

Materials		Static friction, μ_s	
		Dry & clean	Lubricated
Aluminum	Steel	0.61	
Copper	Steel	0.53	
Brass	Steel	0.51	
Cast iron	Copper	1.05	
Cast iron	Zinc	0.85	
Concrete (wet)	Rubber	0.30	
Concrete (dry)	Rubber	1.0	
Concrete	Wood	0.62	
Copper	Glass	0.68	
Glass	Glass	0.94	
Metal	Wood	0.2-0.6	0.2 (wet)
Polythene	Steel	0.2	0.2
Steel	Steel	0.80	0.16
Steel	Teflon	0.04	0.04
Teflon	Teflon	0.04	0.04
Wood	Wood	0.25-0.5	0.2 (wet)

Table 1

1.3.8Surface Roughness

Surface roughness, more commonly shortened to roughness, is a measure of the finely spaced surface irregularities. In engineering, this is what is usually meant by "surface finish".

Waviness: It is the measure of surface irregularities with a spacing greater than that of surface roughness. These usually occur due to warping, vibrations, or defection during machining

Lay: It is a measure of the direction of the predominant machining pattern. A lay pattern is a repetitive impression created on the surface of a part. It is often representative of a specific manufacturing operation.

Measurement of surface Finish

Surface finish may be measured in two ways: contact and non-contact methods. Contact methods involve dragging a measurement stylus across the surface; these instruments are called profilometers. Non-contact methods include: interferometry, confocal microscopy, focus variation, structured light, electrical capacitance, electron microscopy, and photogrametry.The most common method is to use a diamond stylus profilometer. The stylus is run perpendicular to the lay of the surface. The probe usually traces along a straight line on a flat surface or in a circular arc around a cylindrical surface. The length of the path that it traces is called the measurement length. The wavelength of the lowest frequency filter that will be used to analyze the data is usually defined as the sampling length. Most standards recommend that the measurement length should be at least seven times longer than the sampling length, and. The assesmental length or evaluation length is the length of data that will be used for analysis. Commonly one sampling length is discarded from each end of the measurement length. 3D measurements can be made with a profilometer by scanning over a 2D area on the surface.The disadvantage of a profilometer is that it is not accurate when the size of the features of the surface are close to the same size as the stylus. Another disadvantage is that profilometers have difficulty detecting flaws of the same general size as the roughness of the surface. There are also limitations for non-contact instruments. For example, instruments that rely on optical interference cannot resolve features that are less than some fraction of the frequency of their operating wavelength. This limitation can make it difficult to accurately measure roughness even on common objects, since the interesting features may be well below the wavelength of light. The wavelength of red light is about 650 nm,[3] while the R_a of a ground shaft might be 2000 nm.The first step of analysis is to filter the raw data to remove very high frequency data since it can often be attributed to vibrations or debris on the surface. Next, the data is separated into roughness, waviness and form. This can be accomplished using reference lines, envelope methods, digital filters, fractals or other techniques. Finally, the data is summarized using one or more roughness parameters, or a graph. In the past, surface finish was usually analyzed by hand. The roughness trace would be plotted on graph paper, and an experienced machinist decided what data to ignore and where to place the mean line. Today, the measured data is stored on a computer, Many factors contribute to the surface finish in manufacturing. In forming processes, such as molding or metal forming, surface finish of the die determines the surface finish of the work piece. In machining the interaction of the cutting edges and the microstructure of the material being cut both contribute to the final surface finish. In general, the cost of manufacturing a surface increases as the surface finish improves. Just as different manufacturing processes produce parts at various tolerances, they are also capable of different roughness. Generally these two characteristics are linked: manufacturing processes that are dimensionally precise create surfaces with low roughness. In other words, if a process can manufacture parts to a narrow dimensional tolerance, the parts will not be very rough.Due to the abstractness of surface finish parameters, engineers usually use a tool that has a variety of surface roughness created using different manufacturing methods.

1.3.9The Surface Texturisation

The surface texturisation refers to making a very precise shapes cut on a surface. The size of the cut is in Microns and that is why it is called as Texture. In oor setup we have made a surface texture on our discs with laser.The standard process for texturization includes wet chemical etching, which creates specific surface morphologies, depending on the type of silicon and etching solutions used. Laser surface texturisation, which avoids chemicals and is insensitive to local crystal orientation etc, is considered as a promising alternative to these conventional methods. Such laser surface texturisation is usually done using femtosecond lasers and/or in special gas atmosphere. Due to the complexity of the system setup this technology can only be found in laboratories so far.

1.4 LUBRICATION

1.4.1 INTRODUCTION

Lubrication is the most effective means of reducing friction and controlling wear. The lubrication system in an engine serves four major purposes:

1. To prevent seizure in the components,
2. To remove the heat generated by friction,
3. To reduce the friction between components,
4. To reduce the wear of the internal components.

These four byproducts of the lubrication system are achieved by effectively separating the internal components to varying degrees with a layer of oil lubricant. A common way to reduce friction is by using a lubricant, such as oil, water, or grease, which is placed between the two surfaces, often dramatically lessening the coefficient of friction. The science of friction and lubrication is called tribology. Lubricant technology is when lubricants are mixed with the application of science, especially to industrial or commercial objectives. The essential properties which a good cylinder lubricant must have are as follows:

It must reduce sliding friction between the rings and the liner to a minimum, thereby minimizing metal to metal contact and frictional wear. It must possess adequate viscosity at high working temperatures and still be sufficiently fluid to spread over the entire working surfaces to form a good adsorbed oil film. It must form an effective seal in conjunction with the piston rings, preventing gas blow by and burning away of the oil film and lack of compression. It must burn cleanly, leaving as little and as soft a deposit as possible. This is especially true of high additive content oils as unsuitable types can form objectionable ash deposits. It must effectively prevent the buildup of deposits in the ring zone and in ports of port exhausted two stroke engines. It must effectively neutralize the corrosive effects of mineral acids formed during combustion of the fuel.

The problem with cylinder lubricating oil:

In first glance it would appear that no lubricant, neither mineral nor synthetic, could withstand all above difficulties to fulfill the above requirements, but significant developments in the lubricating oil field have made it possible.

1. Friction and Wear The surfaces of machinery components appear well-finished to the naked eye. When magnified, however, surface imperfections become readily apparent. These microscopic hills and valleys are called asperities. When dry surfaces move relative to one another, asperities may rub, lock together, and break apart. The resistance generated when these adjacent surfaces come in contact is called friction. The welding together and breaking apart of asperities is a form of adhesive wear. Another form of wear may occur when a hard contaminant particle becomes trapped between two opposing surfaces. When this occurs, the contaminant acts as a miniature lathe, cutting into the softer machinery surface. This process is termed abrasive wear. Another consequence of friction is that the energy created by resistance is converted into heat. The primary functions of a lubricant, then, are the formation of a protective film between adjacent surfaces to reduce wear, and the dissipation of heat generated at these wear surfaces.

2. Corrosion Protection A second role provided by a lubricant is the prevention of system corrosion. In environments where contamination of the system with water is likely, protection of machinery components from corrosion is of the utmost importance. Salt water is considerably more corrosive than fresh water; thus naval machinery must be well protected from this contaminant. Water molecules may also diffuse through the lubricant and enter surface micro cracks, causing hydrogen embrittlement and subsequent surface failure. It is thus imperative that water contamination of machinery systems be minimized. To achieve corrosion protection, lubricants must form a protective barrier on machinery surfaces. Modern-day lubricants often contain corrosion inhibitors which chemically bond to the metallic surfaces of equipment components. Corrosion inhibitors are an example of a class of compounds called additives [9].

1.4.2 KINDS OF LUBRICATION

Lubrication can be further be broken down into three major types:

1. No lubrication, 2. Boundary layer lubrication and 3. Full lubrication.

When there is no lubrication, the surfaces of the interacting components physically interact with each other, most commonly in sliding friction when there is dynamic movement. Under these circumstances, friction is the greatest under static loads, and lowers during dynamic movement. It is also important to notice that as the speed of interaction between the two surfaces increases, the generated heat also increases because of the energy released from the surface reactions. Boundary layer lubrication occurs when there is provided a layer of lubricant to partially separate the interaction components. Under these conditions, the lubricant can significantly reduce the sliding friction between the components, as well as have the added benefit of cooling the components by absorbing the heat generated from the partial interaction as well as the shear force in the lubricant. Components such as cams operate under this type of lubrication. Full lubrication occurs when there is no interaction between the machine elements because of a think layer of lubrication. The advantage of this type of lubrication is that it effectively stops wear between the machine elements because there is only an interaction between the lubricant and the element, but unfortunately, wear still occurs. This type of lubrication takes place in mechanisms such as the valves in the cylinder heads. In applications such as the valves and cylinders, it is also important to take into consideration the prominent effect of viscosity, because as the lubricant's temperature increases, the viscosity of the lubrication decreases. So it must be taken into consideration that the lubricant is viscous enough under operating conditions, but also not be too viscous that the engine cannot turn over in the ignition sequence [10].

COMMON LUBRICANTS

The most common types of oils used in the engine lubrication system are either vegetable oil or mineral oil. Vegetable oil was used in the past for racing applications because of its high film strength, and excellent protection against wear from its high lubricity. But was not widely used in other applications because of its rapid rate of deterioration, which produces gums and lacquers on the machine elements. So mineral oils are more commonly used because they are much more cost effective, readily responsive to additives, can be produced in a wide range of viscosities, as well as deteriorate much less rapidly than vegetable oils. Today, lubricants such as synthetic oils replace natural oils as lubrication for the engine. Besides the higher cost, synthetic oils are much more effective lubricants than mineral oils because they can be chemically developed to have whatever the particular engines specifications require for proper operation.

Other types of lubrication are: Hydrodynamic Lubrication: Formed the oil film by the relative movement of the friction surface.

Hydrostatic Lubrication: Formed the oil film on the friction surface by the external pressure oil.

THE FUNCTION OF LUBRICANT

1. Reduced the friction and wear
2. Rust prevention
3. Cooling effect and thermolysis
4. The oil film is buffering and shock-absorbing
5. Clean the friction surface, gland and dustproof.

1.4.3 TYPES OF LUBRICANT

Introduction The three major types of lubricants are lubricating oils, greases, gases and solid lubricants. The selection of a lubricant type is dependent on the type of machinery to be lubricated, the complexity of the lubricating system allowed by machinery design, and the frequency of lubrication required.

Lubricating Oil

Lubricating oils are used for the majority of applications. They may be classified according to their viscosities and any special properties imparted to them by additives. Oils whose base stocks are derived primarily from crude oil refining are called mineral or petroleum oils. Petroleum oils may be further classified as being paraffinic or naphthenic based on the types of hydrocarbons comprising the base stock. Oils that have been manufactured by chemical synthesis such as polymerization are called synthetic oils. Additives may be blended into the base stock to impart special properties to the finished product.

Reciprocating Internal Combustion Engine (R.I.C.E.) Oil

Reciprocating Internal Combustion Engine (R.I.C.E.) lube oils are formulated with detergent or dispersant additives to keep soot and other combustion by-products from depositing on engine parts. In addition, alkaline additive packages act to neutralize the acidic products of combustion. A third type of additive reduces the wear of internal parts such as cylinder liners, rings, pistons, and bearings. This oil is a single grade SAE 40 oil to be used in all shipboard internal combustion engines operating in ambient temperatures of 0℃ (32℉) or higher.

Solid Lubricants

Solid lubricants are typically used in situations where unusual temperature or environmental conditions preclude the use of conventional fluid lubricants, or when the application of a fluid lubricant is difficult. Solid lubricants form an essentially dry lubricating film between adjacent surfaces. The lubricant may be applied directly in powdered form, or as a colloidal suspension in a vehicle such as isopropanol. Evaporation of the vehicle leaves a thin film of the lubricant on machinery surfaces. The two most commonly used solid lubricants are powdered graphite and molybdenum disulfide ($MoS2$). Other materials such as powdered zinc dust and red lead suspended in petrolatum or mineral oil may also be used. Specific solid lubricant applications are as follows:Dry graphite may be used for the lubrication of such equipment as security locks. Powdered molybdenum disulfide is used primarily as a thread antiseize compound. For the lubrication of threaded steel nuts and bolts, including superheated steam components up to 565℃ (1050℉); high temperature antiseize compound is typically used. This lubricant consists of a mixture of graphite and molybdenum disulfide suspended in mineral oil. For threaded aluminium parts engaged with similar or dissimilar metals, zinc dust-petrolatum antiseize compound shall be used. Additional lubricants for use on threaded fasteners include colloidal graphite in isopropanol and molybdenum disulfide in isopropanol [9].

Special Lubricants

The operating parameters encountered in high pressure (>1500 psi) air, oxygen, and oil-free nitrogen systems require a lubricant that will resist autogenous ignition. For lubrication of these systems, halocarbon oils shall be used.

Classification of Lubricant

1. Natural Oil: Animal oil, seed fat (performed well)
2. Synthetic Oil: Deploy based on the requirement, high cost
3. Mineral Oil : Efficient supplies, stable property, low cost, extensive be applicable.

1.4.4 Characteristics of Lubricating Oils

1. Viscosity

The capability of liquid to resistance to deformation, which is expressed by friction drag of liquid. Technically, the viscosity of an oil is a measure of the oil's resistance to shear. Viscosity is more commonly known as resistance to flow. If lubricating oil is considered as a series of fluid layers superimposed on each other, the viscosity of the oil is a measure of the resistance to flow between the individual layers. A high viscosity implies a high resistance to flow while a low viscosity indicates a low resistance to flow. Viscosity varies inversely with temperature. Viscosity is also affected by pressure; higher pressure causes the viscosity to increase, and subsequently the load-carrying capacity of the oil also increases. This property enables use of thin oils to lubricate heavy machinery. The load carrying capacity also increases as operating speed of the lubricated machinery is increased. Two methods for measuring viscosity are commonly employed: shear and time [10].

When viscosity is determined by directly measuring shear stress and shear rate, it is expressed in centipoise (cP) and is referred to as the absolute or dynamic viscosity. In the oil industry, it is more common to use kinematic viscosity, which is the absolute viscosity divided by the density of the oil being tested. Kinematic viscosity is expressed in centistokes (cSt). Viscosity in centistokes is conventionally given at two standard temperatures: 40 EC and 100 EC (104 EF and 212 EF) [9].

Time. Another method used to determine oil viscosity measures the time required for an oil sample to flow through a standard orifice at a standard temperature. Viscosity is then expressed in SUS (Saybolt Universal Seconds). SUS viscosities are also conventionally given at two standard temperatures: 37 EC and 98 EC (100 EF and 210 EF). As previously noted, the units of viscosity can be expressed as centipoises (cP), centistokes (cST), or Saybolt Universal Seconds (SUS), depending on the actual test method used to measure the viscosity [9].

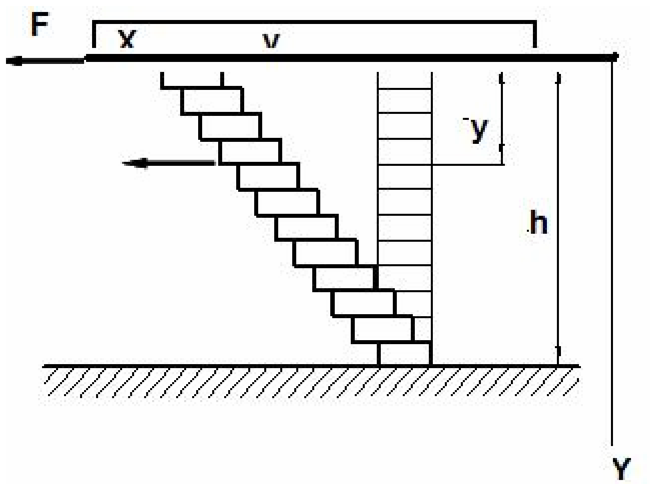


Fig 3 Inertia free flow [10]

laminar flow of oil between parallel plates

Speed of oil layer in the $u = v$ vicinity of active plate

Speed of oil layer in the $u = 0$ vicinity of active plate

Every oil layer moves u in different speed

τ and velocity gradient $\frac{du}{dy}$ is directly proportional.

$\tau = -\eta \frac{du}{dy}$ (Newton's law of viscous)

proportionality constant, namely kinetic viscosity η (use in the calculation of hydrokinetics).

Assuming the liquid of $1 \times 1 \times 1m^3$, if there is relative sliding of the speed of 1m/s between the two plate, when the force is1N in that way the viscosity of liquid is one International System of Units' kinematic viscosity Pa.s

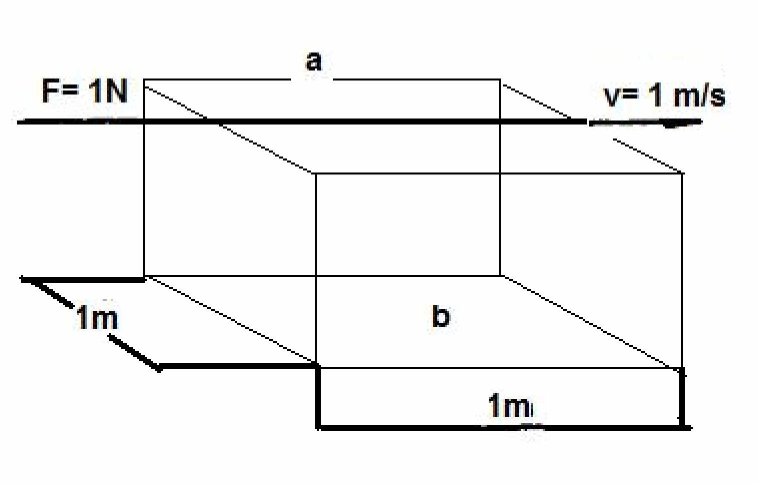


Fig 4. Flow of viscous fluid

kinematic coefficient of viscosity γ (the viscosity of lubricant)

The ratio between kinetic viscosity η and the fluid density ρ at the same temperature [9].

A.Viscosity index

The viscosity index, commonly designated VI, is an arbitrary numbering scale that indicates the changes in oil viscosity with changes in temperature. Viscosity index can be classified as follows: low VI - below 35; medium VI - 35 to 80; high VI - 80 to 110; very high VI - above 110. A high viscosity index indicates small oil viscosity changes with temperature. A low viscosity index indicates high viscosity changes with temperature. Therefore, a fluid that has a high viscosity index can be expected to undergo very little change in viscosity with temperature extremes and is considered to have a stable viscosity. A fluid with a low viscosity index can be expected to undergo a significant change in viscosity as the temperature fluctuates. For a given temperature range, say -18 to 370EC (0 - 100 EF), the viscosity of one oil may change considerably more than another. An oil with a VI of 95 to 100 would change less than one with a VI of 80. Knowing the viscosity index of an oil is crucial when selecting a lubricant for an application, and is especially critical in extremely hot or cold climates. Failure to use an oil with the proper viscosity index when temperature extremes are expected may result in poor lubrication and equipment failure. Typically, paraffinic oils are rated at 38 EC (100 EF) and naphthenic oils are rated at -18 EC (0 EF). Proper selection of petroleum stocks and additives can produce oils with a very good VI [9].

B. Pour point

The pour point is the lowest temperature at which an oil will flow. This property is crucial for oils that must flow at low temperatures. A commonly used rule of thumb when selecting oils is to ensure that the pour point is at least 10 EC (20 EF) lower than the lowest anticipated ambient temperature [9].

C. Cloud point

The cloud point is the temperature at which dissolved solids in the oil, such as paraffin wax, begin to form and separate from the oil. As the temperature drops, wax crystallizes and becomes visible. Certain oils must be maintained at temperatures above the cloud point to prevent clogging of filters.

D. Flash point and fire point The flash point is the lowest temperature, to which a lubricant must be heated before its vapour, when mixed with air, will ignite but not continue to burn. The fire point is the temperature at which lubricant combustion will be sustained. The flash and fire points are useful in determining a lubricant's volatility and fire resistance. The flash point can be used to determine the transportation and storage temperature requirements for lubricants. Lubricant producers can also use the flash point to detect potential product contamination. A lubricant exhibiting a flash point significantly lower than normal will be suspected of contamination with a volatile product. Products with a flash point less than 38 EC (100 EF) will usually require special precautions for safe handling. The fire point for a lubricant is usually 8 to 10 percent above the flash point. The flash point and fire point should not be confused with the auto-ignition temperature of a lubricant, which is the temperature at which a lubricant will ignite spontaneously without an external ignition source [9].

E. Neutralization Number

The acid or neutralization number is a measure of the amount of potassium hydroxide required to neutralize the acid contained in a lubricant. Acids are formed as oils oxidize with age and service. The acid number for an oil sample is indicative of the age of the oil and can be used to determine when the oil must be changed. As lubricants degrade from oxidation they form a number of acids. These acids are corrosive to Babbitt, yellow metals, carbon steel, cast iron, and if left uncorrected for a period of time will begin a corrosion process and possibly eventual bearing failure. While small increases in the Total Acid Number (TAN) usually indicate oxidation and lubricant degradation, contaminants with acidic constituents can also be a factor. Monitoring the oil's Total Acid Number should be an important part of lubricant maintenance program. Generally when a lubricant's acid number reaches a condemning limit, replacement or sweetening is best option.

1.**Total Acid Number (TAN**) is the standard neutralization number test for industrial lubricating oils. It is performed by titrating a solution of oil and diluents with an alcohol/potassium hydroxide (KOH) solution, a base, until all the acids present are neutralized. The results are reported as milligrams of potassium-hydroxide per gram of sample, or mg/Gm

2.Strong Acid Number (SAN) is similar to TAN, except the 'strong' acids are first extracted from the lubricant. That extract is then titrated with KOH and the SAN reported as mg/gm.

F. Total Base Number (TBN) is a standard test for engine lubricants. It is a measurement of the amount of protection in the lubricant remaining to neutralize acids formed as a result of combustion. A solution of oil and diluents is titrated with an alcohol/Hydrochloric Acid (HCl) solution until all the alkaline or base constituents in the oil are neutralized. Results are reported as milligrams of HCl per gram of sample, or mg/gm. Most lubricating oils have a baseline Acid Number as a result of additives. R&O (rust and oxidation) industrial oils generally have a baseline in the 0.03 to 0.06 mg/gm range. AW (anti-wear) and EP (extreme pressure) industrial oils will have much higher baselines because of the additional additives that give them their AW or EP qualities. Baselines for these lubricants can be over 1.0 mg/gm [9].

1.4.5 EFFECT OF WATER IN THE OIL

Precautions shall be taken to keep water from entering the lubrication system; any water detected shall be removed as soon as possible. Water in the oil increases frictional resistance causes the oil to break down prematurely, corrodes journals and any parts not continuously covered with oil, and may cause corrosion in the entire system. Rusting generally originates on exposed surfaces such as gear casings and upper portions of sump tanks where condensation occurs and, unless remedied, progresses throughout the system. Rusting due to condensation is most likely to occur in cold climates and in installations where portions of the lube oil sump tanks are integral with the skin of the ship. Precautions to be taken include the following:

a), Any drain fitted in the lowest part of a bearing pedestal should be opened and the bearings drained of any water a few hours after securing the lube oil system.

b). Lube oil from bilges, oily waste drain tanks, or miscellaneous tanks shall not be reclaimed nor added to the lubrication system.

c).Because warm, moisture-laden vapour condenses on any cool, exposed surface, any factor tending to cool casings or sump surfaces shall be eliminated. All forced draft ventilation ducts shall be permanently arranged so that no air can blow directly or indirectly on a gear casing.

d).When securing the main propulsion machinery, circulate oil through the turbines and reduction gear lubrication system by means of the ship's oil pumps for at least 1 hour (or until the machinery reaches ambient temperature), to allow machinery component temperatures to equalize. The motor-driven shaft turning gear (if provided) shall be operated during the oil circulation period. If hand jacking only is provided, the propeller shaft shall be jacked intermittently until machinery reaches ambient temperature. The circulation period will vary, depending upon the air and injection temperatures. For ships operating under low air and injection temperature conditions, the period will be somewhat longer.

e).If electrostatic precipitators are installed on turbine and reduction gear and sump vent pipes, the vented air is cleaned of oil and water mist particles only. Moisture, in the form of vapour, will flow through the precipitator unrestricted.

A haze of moist air visible at the electrostatic precipitator vent discharge indicates that excessive moisture is entering the lubrication system elsewhere. Test the purifier operating efficiency and check every possible point at which water might enter the system.

f).If the propulsion reduction gear has a dehumidifier installed, the moisture in the air within the gear case and oil sump shall be maintained at 30-35 percent relative humidity, when the propulsion plant has cooled down and been secured. Prior to starting and warm up of the lubricating oil system, the dehumidifier shall be secured, isolation valves in the dehumidification ducts shall be locked shut, and the electrostatic precipitator activated and its positive closure assembly opened. To secure the machinery, allow the oil temperature during oil circulation to cool down to approximately 10° F above ambient. Then, secure the electrostatic precipitator, close the positive closure assembly, stop the lubricating oil system pumps, unlock and open the dehumidification duct isolation valves and activate the dehumidifier. Verify once per watch that the humidity in the gear case is being maintained within the specified limits. Refer to applicable dehumidifier operational/maintenance manual if the specified gear-case humidity requirements cannot be maintained [9].

INTRODUCTION

The Internal combustion engine efficiency is mainly affected by the friction in Piston-cylinder assembly. The friction in this assembly is approximately 60-70% of the total friction of IC engine. The working of engine is affected based on this friction. This high friction leads to sever wear.

All tribological efforts has been done to reduce this friction and different designs for the piston rings have been suggested based on the boundary conditions and experimental setup done for studies. The friction of piston ring and cylinder liner has been calculated based on many simulation techniques and tried to give the conditions of the real engine .Lubrication is one of the most important features to reduce friction. The method adopted of lubrication needs the kind of lubricant that can take the loads of different sever conditions of friction and wear. The lubricants behave differently based on the time of use and their properties are reduced as they run for the long times.

It was found after extensive investigation of major engine components (piston, piston rings, bearings and cylinder liner) that engine service life appears to be more a function of the acceptable wear of the cylinder liner than of any other engine parts. A method was developed to determine the amount of cylinder liner wear; this involved measuring the depth of scratch marks (using Talysurf model 3 or Surtronic 3 instruments) [11].

Friction loss in an IC engine is most important feature in determining fuel economy and performance. Proper lubrication and surface texture is the key to reducing friction in a piston cylinder system. surface texturing is a new technique used to reduce friction in mechanical components.[12].

Study of tribological characteristics is done through a pin on disc experimental setup to simulate the wear characteristics of an IC engine. It leads to the wear phenomena in 2-strok marine diesel engine. [13].

A test method has been developed to evaluate the friction and wear behaviour of candidate piston ring and cylinder liner materials for heavy-duty diesel engine applications. Oil condition and its effects are important aspects of this test method and are the focus of this work. The test uses actual piston ring segments sliding on flat specimens of liner material to simplify alignment and to multiply the stress to the level normally seen in engine operation. Reciprocating tests were conducted at 10Hz and 10mm stroke at 100°C. Test oils consisted of fully formulated lubricating oils that were conditioned in ASTM standard engine tests. The point contact between the ring segment and flat counter-face, the applied load and elevated temperature, all result in boundary lubrication, which simulates the environment near top-ring-reversal. The oil condition was defined by variables, such as spectroscopic elemental concentrations, soot level, oxidation, and contaminant particle concentration. Compared with engine-measured wear rates, ring wear was magnified by at least an order of magnitude and the liner by about 1.5-2 orders of magnitude as needed for an accelerated test. However, the basic wear mechanism, abrasive wear, was the same as in the engine. The soot concentration also had a strong effect on liner wear but no effect on ring wear. The oil viscosity has a mild effect on the friction at high load in boundary lubrication conditions. The viscosity of the conditioned oils tested here was related to the soot content rather than the oxidation levels [14].

Wear of a cold-running gasoline engine was measured by use of a radioactive piston ring and also by using a neutron activation analysis of oil for iron content. These measurements showed wear to increase greatly when jacket water temperature was reduced, when water was added to the crankcase oil, or when fuel hydrogen content was increased. A tentative explanation of these effects is proposed in which liquid water in the oil is assumed to act as a collector of the acids which cause cold wear. Water is alternately condensed when cylinder pressures are high and re-evaporated when cylinder pressures are low. The average amount of liquid water in the oil, and hence also the wear rate, depends upon the relative magnitudes of condensation and re-evaporation [15].

The present work is an attempt to determine the oil film thickness in a medium-speed four-stroke diesel engine. The study was carried out for different parts of the four-stroke working cycle and for different levels of engine power output. The results were compared with the results of computer simulations, carried out using a commercial software package. The conclusions of the study comprise aspects on the formation and development of the oil film between the rings and the liner under a set of load levels together with the periodical fluctuation during different strokes of the working cycle [16].

Computer models of the lubrication of piston-rings in internal combustion engines normally present an estimate of the minimum oil-film thickness between ring and cylinder wall as a function of crank angle. As a consequence, experimental measurements of film thickness play a critical role in the verification of theoretical models of piston-ring lubrication [17].

Piston and ring friction can account for 65% of the mechanical friction in an internal combustion engine. It shows that cylinder liner lubrication is predominantly hydrodynamic with localized contact between ring and liner at TDC firing. The degree of contact may increase during transient conditions. Piston ring friction in the hydrodynamic region is proportional to the square root of the viscosity. The viscosity is affected by temperature and pressure which can reach peak values. Gains in fuel economy through viscosity reductions have been reduced in the last 25 years due to changes in piston and ring design [18].

The Finite Element (FEM) is applied to solve the governing equations of lubrication of the piston rings and to calculate the friction force on each ring. The ring is assumed to have a circular profile in the direction of motion. This profile changes with time because tilting of the ring with the engine cycle is taken into account. In the circumferential direction, the ring is assumed to be a perfect circle and the bore cross-section is assumed elliptic. Mixed lubrication is considered when the oil film thickness becomes smaller than a certain value which depends upon the roughness of the surfaces in contact. The friction coefficient for this lubrication is taken as a function of the oil film thickness and the surface roughness. The predictions of the friction force are compared with experimental friction data for the same engine [19].

The assumption of axisymmetry, employed by most of studies on piston ring lubrication, probably gives a too idealistic model for the real situation. A theoretical model for a nonaxisymmetrical analysis of piston ring lubrication has been established in the present study. When a piston ring with an arbitrary free shape is fitted into the cylinder bore, the determination of ring deflection and contact load has been modelled mathematically as a Linear Complementary Problem (LCP). By combining LCP solution with lubrication analysis, the film thickness and contact load distribution over the circumference are obtained, leading to a more realistic simulation for piston ring lubrication. The friction force between piston ring and cylinder bore is predicted by the mixed lubrication model including the effects of surface roughness and asperity contact. The static distortion of cylinder bore, gas pressure variation, and lubricant starvation are also considered in the simulation. Results show that the contact pattern and film thickness between piston ring and cylinder bore are not exactly axisymmetrical. The main reason for the non-uniform contact is the asymmetry of ring elasticity, the static distortion and dynamic load created by the secondary movement of piston skirt [20].

The governing equations and appropriate numerical solutions are presented for the heat transfer and the flow of a shear thinning fluid confined between a curved and plane surface in a thin-film configuration. One surface undergoes a reciprocating motion and the fluid experiences transverse squeeze action, as in a typical piston ring of an internal combustion engine. The effects of viscosity variations with temperature, in conjunction with the non-Newtonian shear thinning behaviour of multigrade oils, are included in the analysis. Extensive numerical simulations of the performance of the piston are presented and compared to the isothermal Newtonian solutions. Computations show that shear thinning can have a significant effect on parameters such as film thickness, viscous drag force, and power loss. The thermal effects from viscous dissipation in the clearance space along the piston ring also influence these parameters, but to a lesser degree [21].

A bench friction test system for piston ring and liner contact, which has high stroke length and large contact width has been used to verify the analytical mixed lubrication model presented in a companion paper. This test system controls the speed, temperature and lubricant amount and records the friction force, loading force, crank angle signal and contact temperature data simultaneously. The effects of running speed, applied normal load, contact temperature and surface roughness on friction coefficient have been investigated for conventional cast-iron cylinder bores. Friction coefficient predictions are presented as a function of crank angle position and results are compared with bench test data. Analytical results correlated well with bench test results [22].

In some circumstances, the friction of boundary lubricated surfaces can become higher as they become smoother. This effect can be predicted from an elastoplastic model of asperity contact, and the properties of boundary lubricants. It is proposed that the rapid rise in friction which occurs after running a hot, sparsely lubricated contact for sufficient time is due to a combination of polishing and a modest reduction in the coverage of the rubbing surfaces by the boundary film [23].

The performance of a combustion engine is closely related to the friction force and wear between cylinder liner and piston rings. It is believed that this friction force can be significantly reduced by optimizing the surface topography of cylinder liners. Therefore, it is necessary to understand how liner surface topography affects wear, friction and lubricating oil consumption. Several experimental studies have been carried out for evaluating wear and friction in simulated engine conditions using Cameron–Plint wear testers, Pin-on-disk testers, SRV testers, etc. However, these studies do not reflect the true behavior of inside the engine because of stroke length limitations. The wear and surface property behavior were evaluated at several locations in the liner and found that after running-in an engine, surface of cylinder liner exhibits plateau-honed-like characteristic. Energy dispersive analysis (EDS) has been carried out of liner and top ring for evaluating materials transfer. Coeffcient of friction between three di□erent liner segments and ring was evaluated using an SRV wear tester.

Coeffcient of friction in the piston ring–liner interface increases with increasing average surface roughness for liner [24].
Fresh and used samples of these oils were examined. An original test apparatus simulating piston-liner movement was used for the purpose of clarifying the effects of various parameters such as load, speed and oil type. Amonton's law was obeyed up to a certain limit for some fresh oils and to lower or greater limits for others. Furthermore, a pronounced drop in friction coefficient was observed with used oils. Wear experiments showed a decrease in wear with the increase of the duration distance (the distance over which the engine oil was used in the vehicle). The electromotive force activity of the oils was shown to affect the wear phenomenon of these oils Friction loss in an IC engine is most important feature in determining fuel economy and performance. [25].

3.1 PIN ON DISC SIMULATION

Fig.5. Pin on disc setup

The Tribotech Tribometer uses a pin-on-disk system to measure wear. The unit consists of a gimballed arm to which the pin is attached, a fixture which accommodates disks up to 165 mm in diameter & 8 mm thick, an electronic force sensor for measuring the friction force, and a computer software (WINDUCOM) for displaying the parameters, printing, or storing data for analysis. The motor driven turntable produces up to 3000 rpm. Wear is quantified by measuring the wear groove with a profilometer (to be ordered separately) and measuring the amount of material removed. Users simply specify the turntable speed, the load, and any other desired test variables such as friction limit and number of rotations.

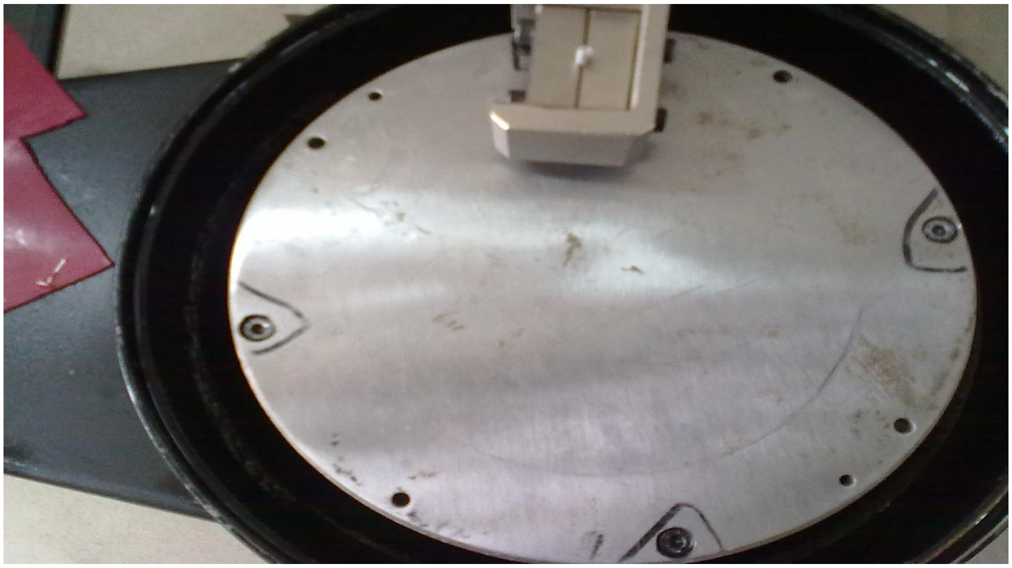


Fig.6. Setup of disc

Designed for unattended use, a user need only place the test material into turntable fixture and specify the test variables. A pre-determined Hertzian pressure is automatically applied to the pin using a system of weights. Rotating the turntable while applying this force to the pin includes sliding wear as well as a friction force. Since pins can be fabricated from a wide range of materials virtually any combination of metal, glass, plastic, composite, or ceramic substrates can be tested.

Software included with this model provides for quick calculation of the Hertzian pressure between the pin and disk. The cup-like (housing) enclosed fixture permits the use of liquid lubricants during a wear test (optionally).

Specifications

Parameter	**Unit**	**Min**	**Max**
Pin Size	Mm	3	12
Ball Diameter	Mm	10	12.7
Disc Size	Mm	165 x 8 mm Thick	
Sliding Speed	M/s	0.05	10
Disc Rotation	RPM	200	2000

Normal Load	N	0	200
Frictional Force	N	0	200
Wear	Mm	0	2
Track Radius	Mm	to be set manually	

Table 2

Data Acquisition

The friction coefficient signal is displayed in real time on a PC Screen. Data can be viewed as it is logged for the entire specified test duration, which can be recalled later for detailed analysis. The software allows 9 different logged test files for on-line analysis / mapping The software displays the test time, turn count, linear velocity, and user-defined test parameters. This data can be stored and printed along with the friction traces.

Purpose

Records friction and wear in sliding contact in dry, lubricated, controlled environment and partial vacuum.

Application

Fundamental wear studies. Wear map ping and PV diagrams. Friction and wear testing of metals, ceramics, soft and hard coatings, plastics, polymers and composites, lubricants, cutting fluids, heat processed samples.

Features

Displays and records friction, wear and pin temperature (optional).Dry, lubricated, controlled environment and vacuum tests (optional).Wide sliding speed range (continuously variable thru the variable drive)User can program RAMP tests to be specified by the user (available optionally)

Standards

ASTM G-99

Instrumentation and Data Acquistion System for the measurement of:

RPM, Wear, Frictional force, Temperature, Electrical Contact resistance measurement (40 mV Signal)

PC acquires data online and displays it in several ways. Graphs of individual tests can be printed. Results of different tests can be superimposed for comparative viewing. Data can be exported to other software.

3.2 SPECTROSCOPY ANALYSIS

Spectrographic metals analysis is usually the 'heart' of most oil analysis programs. Using a Rotrode Emission 20 or more metals/substances can be simultaneously determined. The metals analyzed for include wear, additive, and contaminant metals and are reported in parts per million (ppm). The instrument is quick and easy to operate and is accurate within acceptable limits.

Fig.7 spectroscopy

The Spectrometer has a particle size detection limitation of between 3µ and 10µ (depending on the particular metal in question and the amount of surface oxidation on the particle surface). Results of the Spectrometer are accurate to about 1 or 2 ppm. The advantage of the Spectrometer is that no dilution of the sample is required.

3.3 DESIGN OF DISC

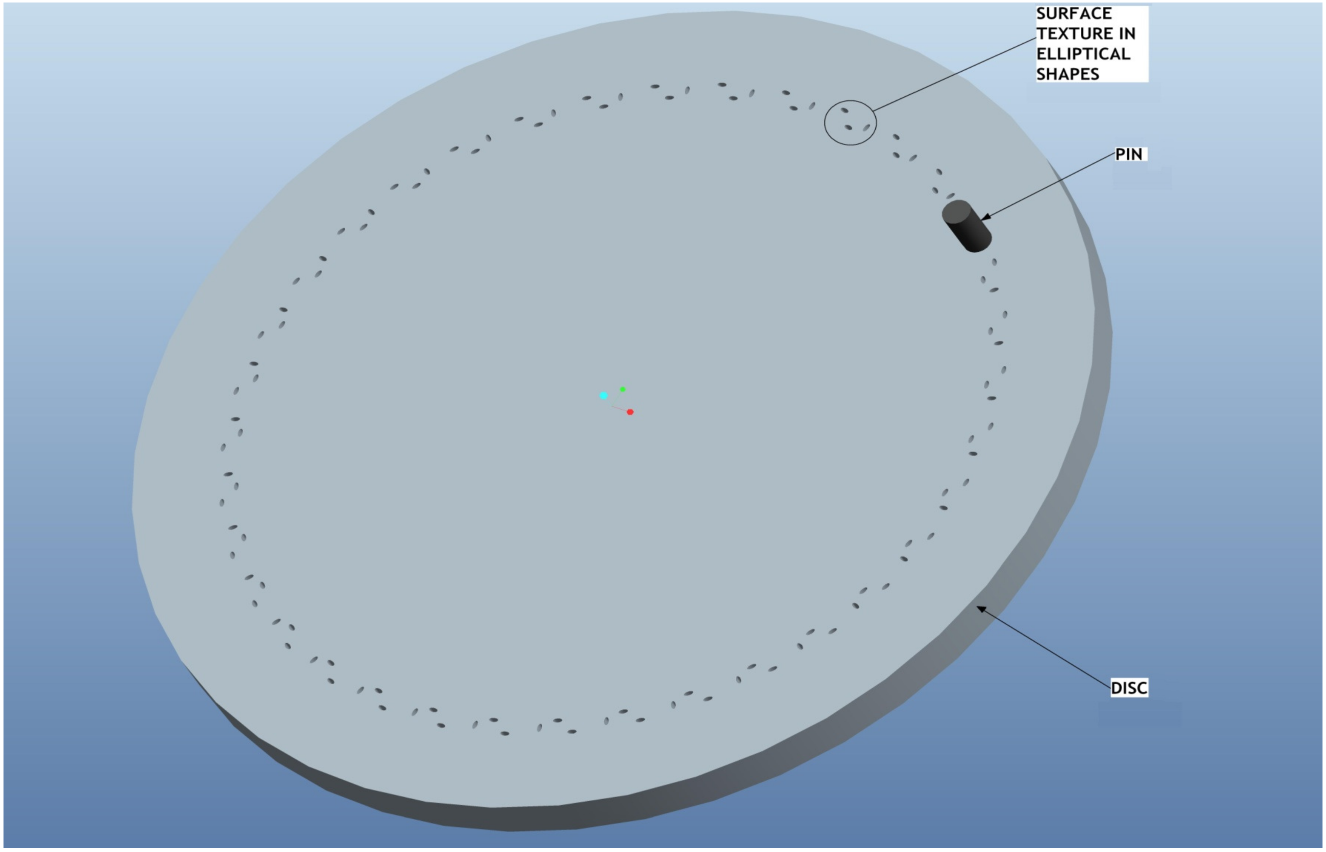


Fig.8 Model of disc [29]

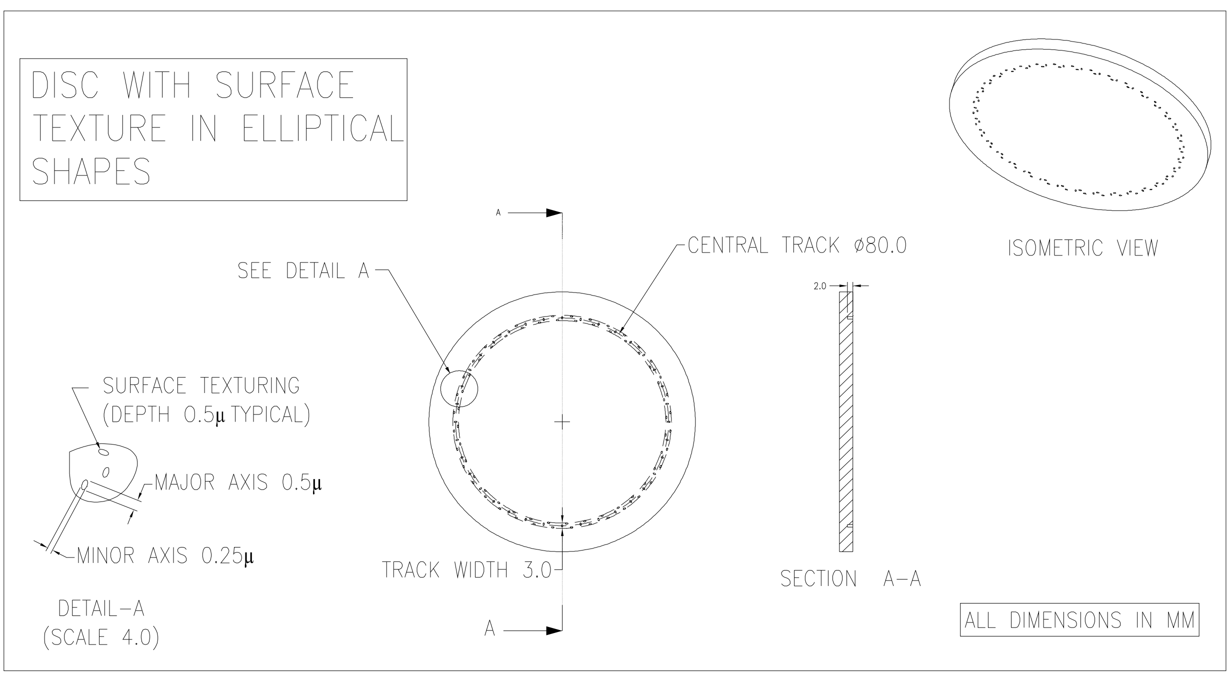


Fig 9 Disc parameters [30]

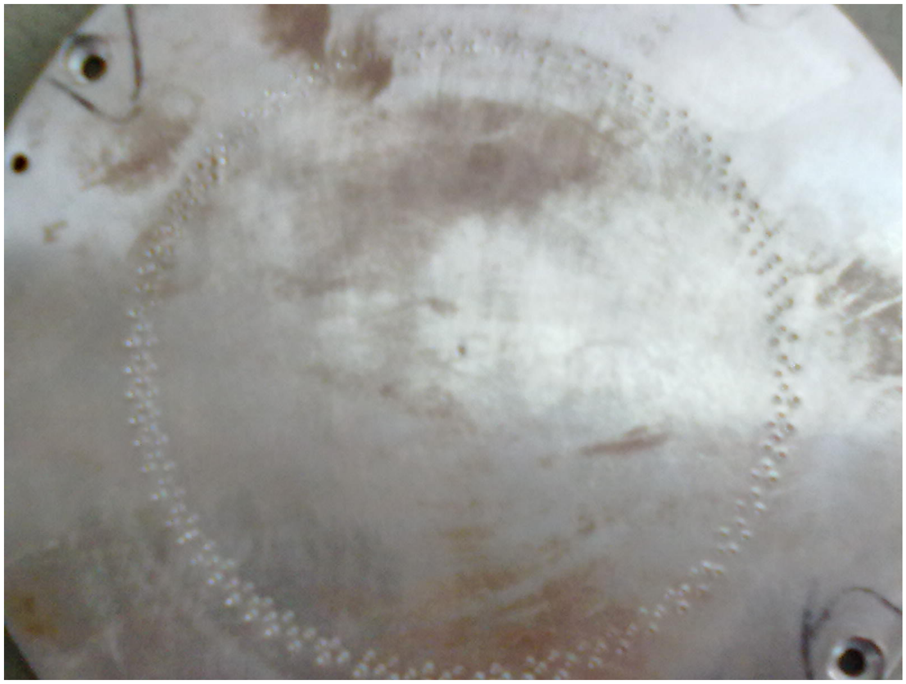


Fig 10 Surface texture of disc [29]

3.4Calculations for texturing (circular case)

The surface texture is created in the elliptical shape. They are done along thee circumference line of the circle at a fixed diameter.
The central diameter is 80 MM.

Perimeter = ¶ * D
= ¶* 80 MM
= 250 MM

Now taking a pitch of 1 MM and dimple diameter as .1mm (100 microns)

Total No. of dimples = 250/ 1
=250 (approx) in inner
Middle
outer

Now taking pitch = 1.5 mm and same dia as .1 mm

Perimeter = ¶ * D
= ¶* 80 MM
= 250 MM
Total No. of dimples = 250/ 1.5
=167 (approx) inner
Middle
outer

Now taking pitch = 2 mm and same dimple dia as .1 mm

Perimeter = ¶ * D
= ¶* 80 MM
= 250 MM
Total No. of dimples = 250/ 2
=125   inner
Middle
outer

so gradually taking same dimple dia and varying pitch and vice versa we ge get the calculations.

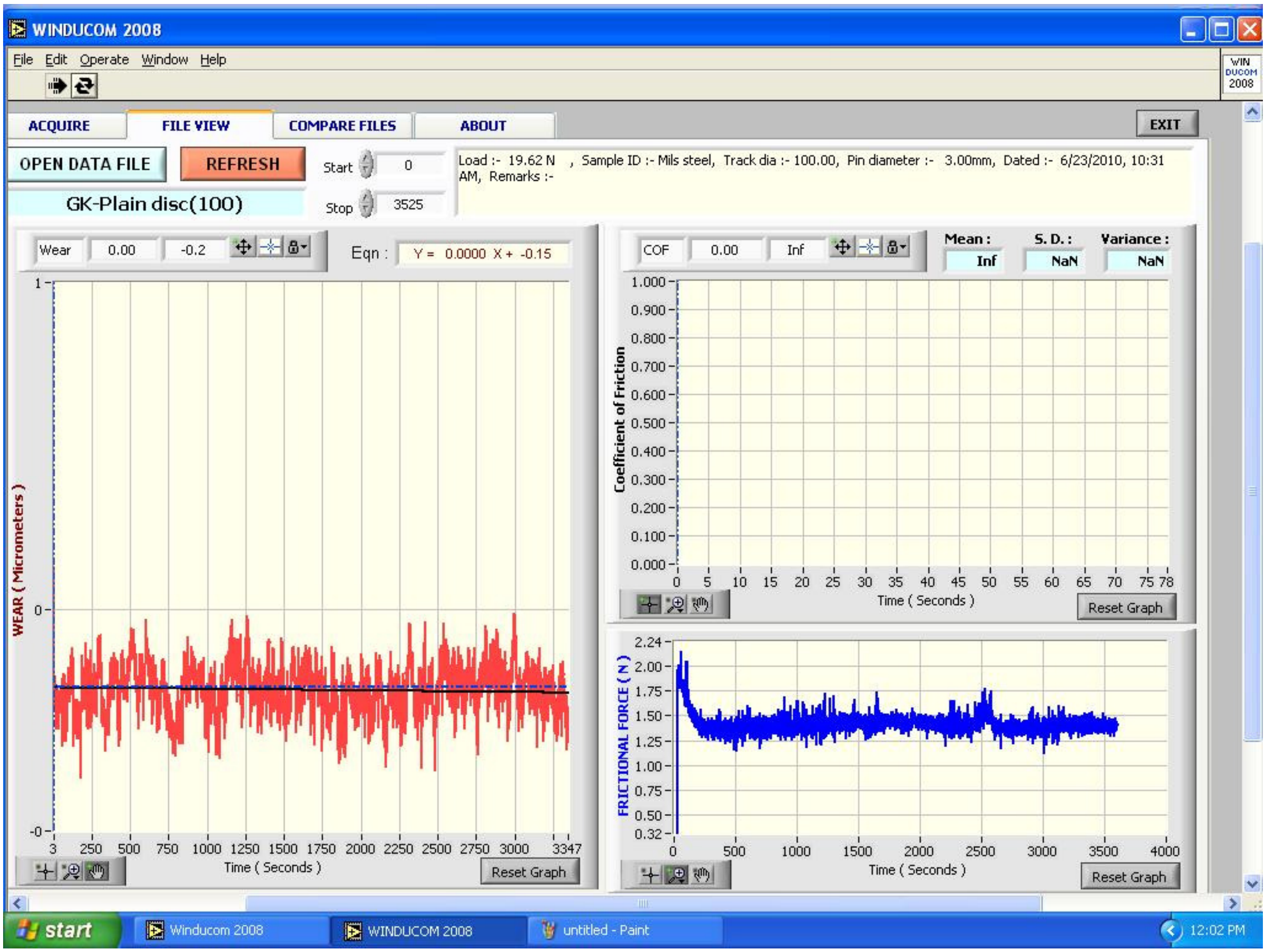
<u>Cases</u>

1. Dimple dia = 0.1 mm (100 micron)   Pitch= 1.0 mm
1.5 mm
2.0 mm
2.5 mm
3.0 mm
2. dimple dia = 0.125 mm (125 microns)   Pitch= 1.0 mm
1.5 mm
2.0 mm
2.5 mm
3.0 mm
3. dimple dia = 0.13 mm (130 microns)   Pitch= 1.0 mm
1.5 mm
2.0 mm
2.5 mm
3.0 mm
4. dimple dia = 0.15 mm (150 microns)   Pitch= 1.0 mm
1.5 mm
2.0 mm
2.5 mm
3.0 mm
5. dimple dia = 0.2 mm (200 microns)   Pitch= 1.0 mm
1.5 mm
2.0 mm
2.5 mm
3.0 mm

4.1 Friction & Wear analysis

Test on the disc without texture with the lubricating oil (HPMILC 30W 40)
Wear & friction force calculation

Fig 12 Wear ,friction calculation



Test on the disc with texture with the lubricating oil (HPMILC 30W 40)
Wear and friction force

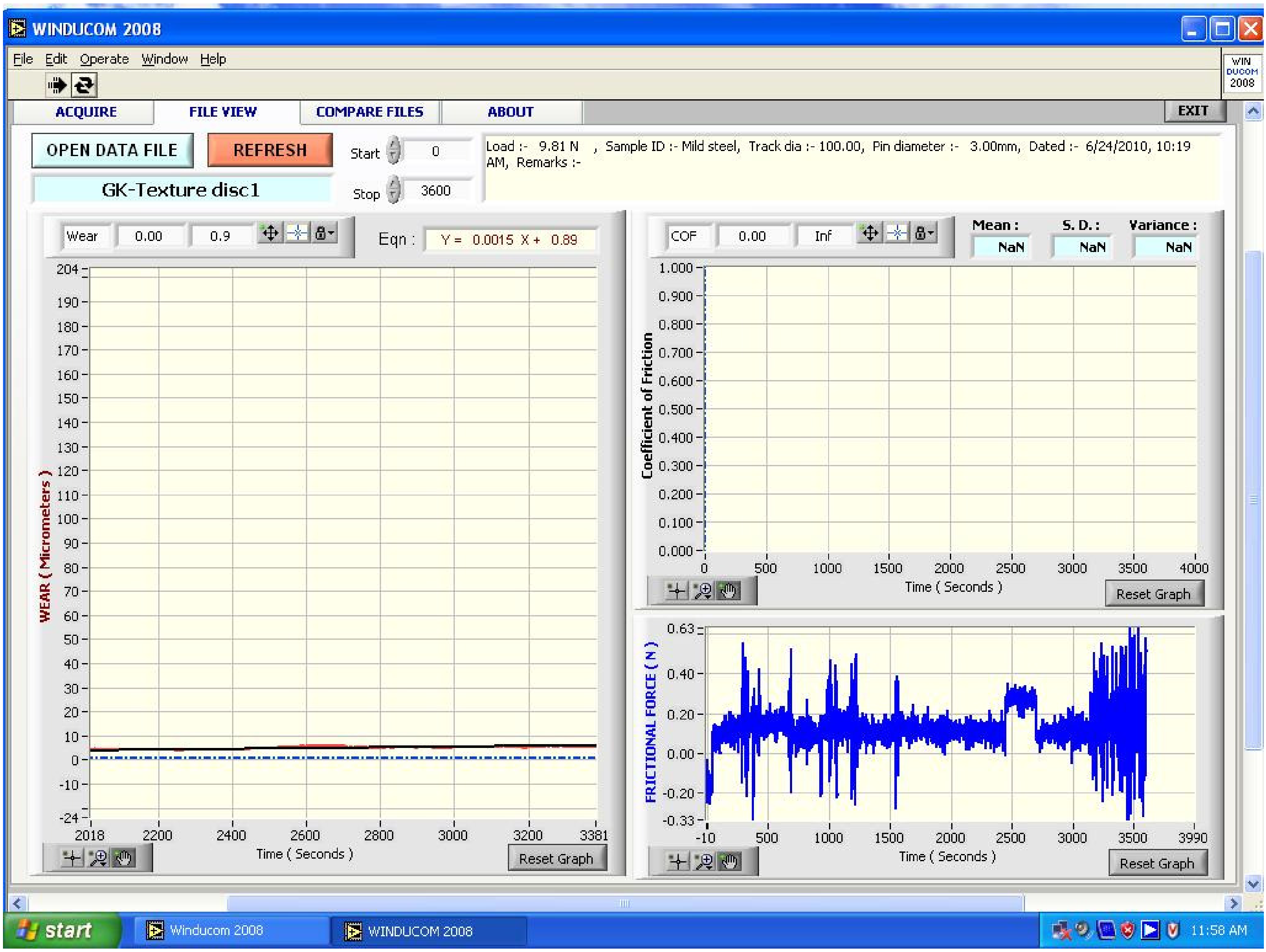


Fig 13 Wear ,friction force calculation

Comparison of discs without texture to the disc with texture for same load &rpm

1. Wear comparison

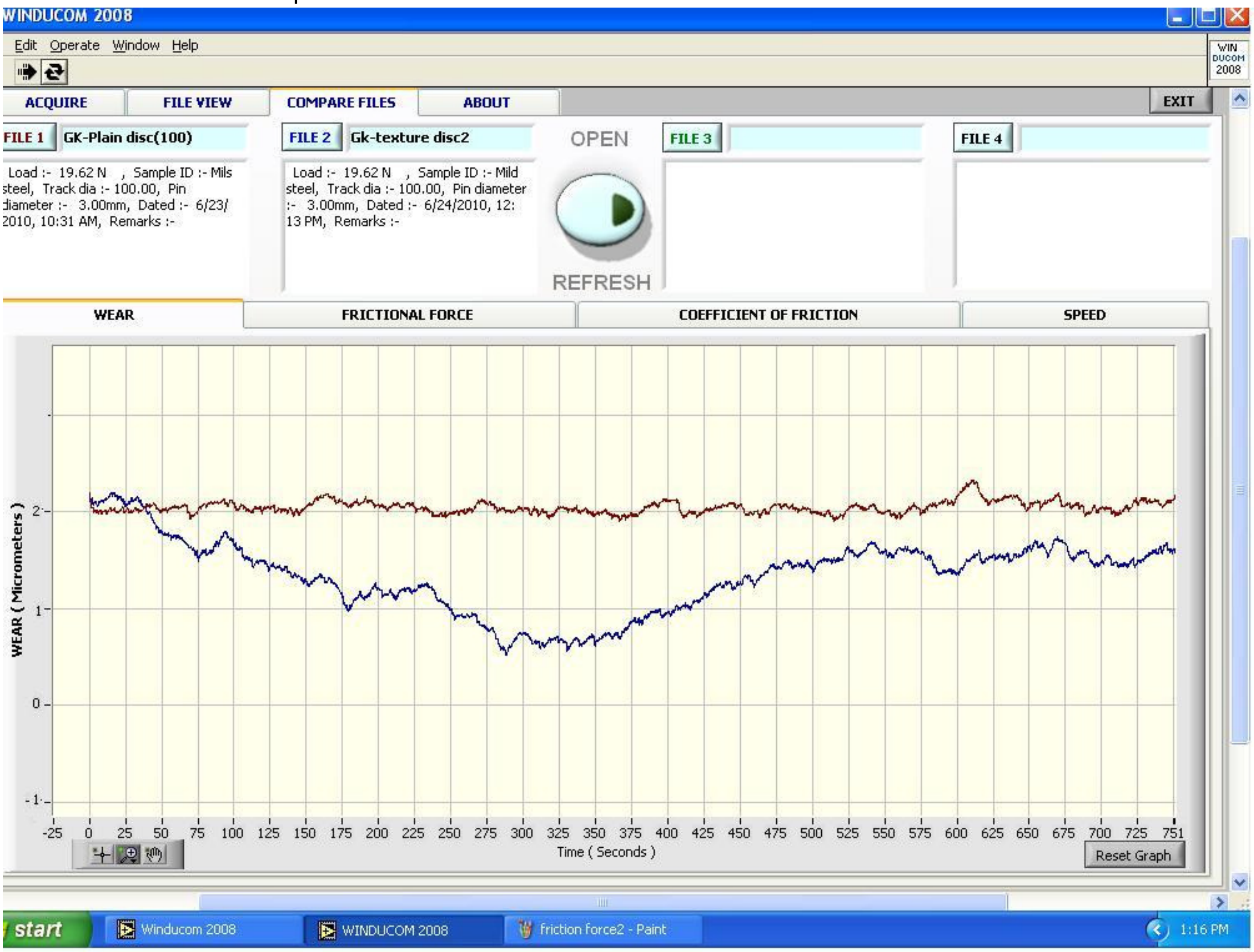


Fig 14 Wear & friction comparision

The wear in the surface textured disc got decreased as shown. The effect is very much visible through the screen of the software "WIDUCOM". The blue line stands for disc with texture. The set up ran for the same conditions.

- Load: 2 kg
- Rpm: 500
- Time : 3600 seconds

2. friction force comparision

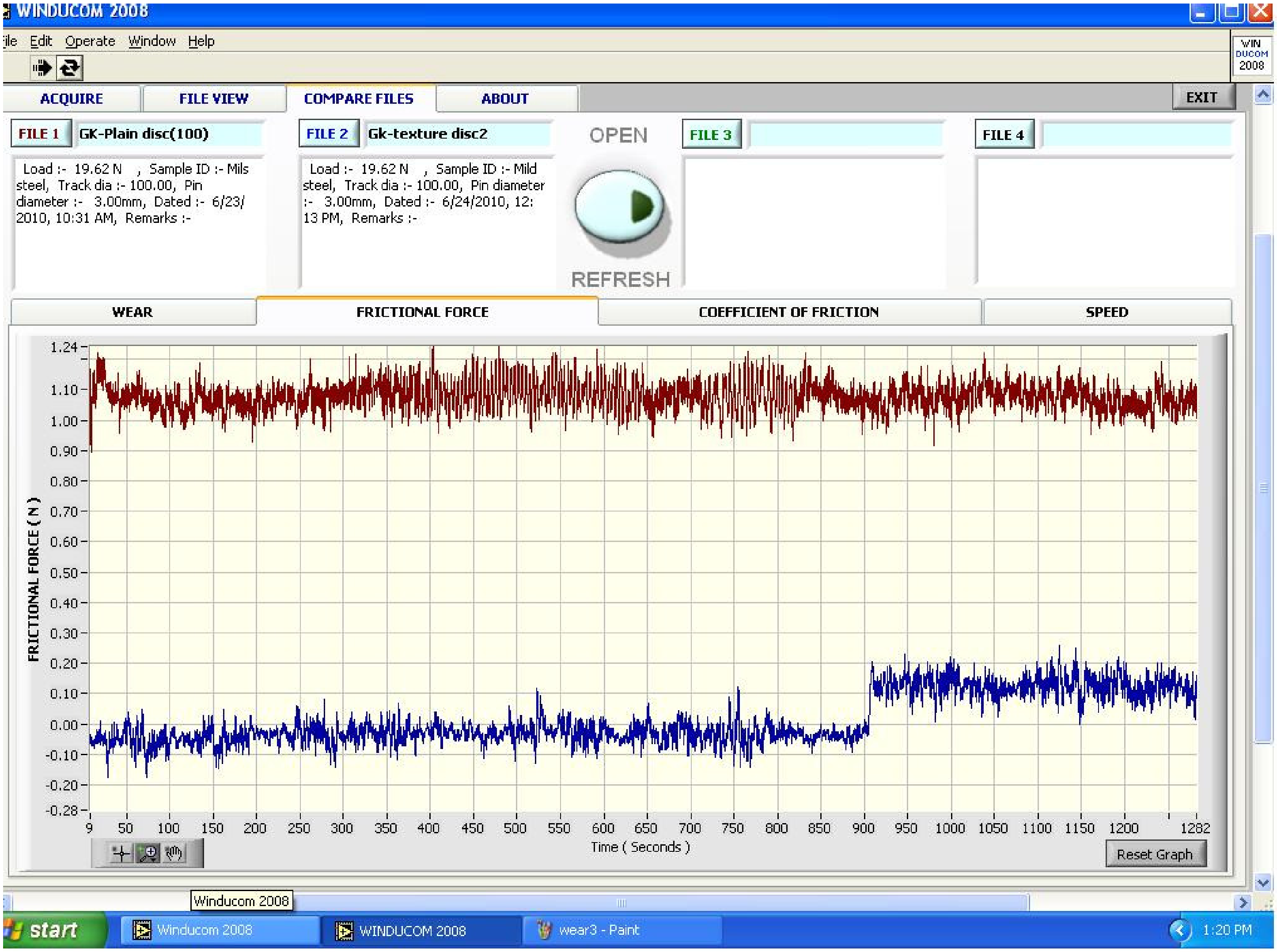


Fig 15 Friction force comparision

The friction force in the surface textured disc is less as shown. This is very much visible through the screen of the software “WIDUCOM”. The blue line indicates the structured disc. The set up ran for the same conditions as.

- Load: 2 kg
- Rpm: 500
- Time : 3600 sec

3. **coefficient of friction comparison**

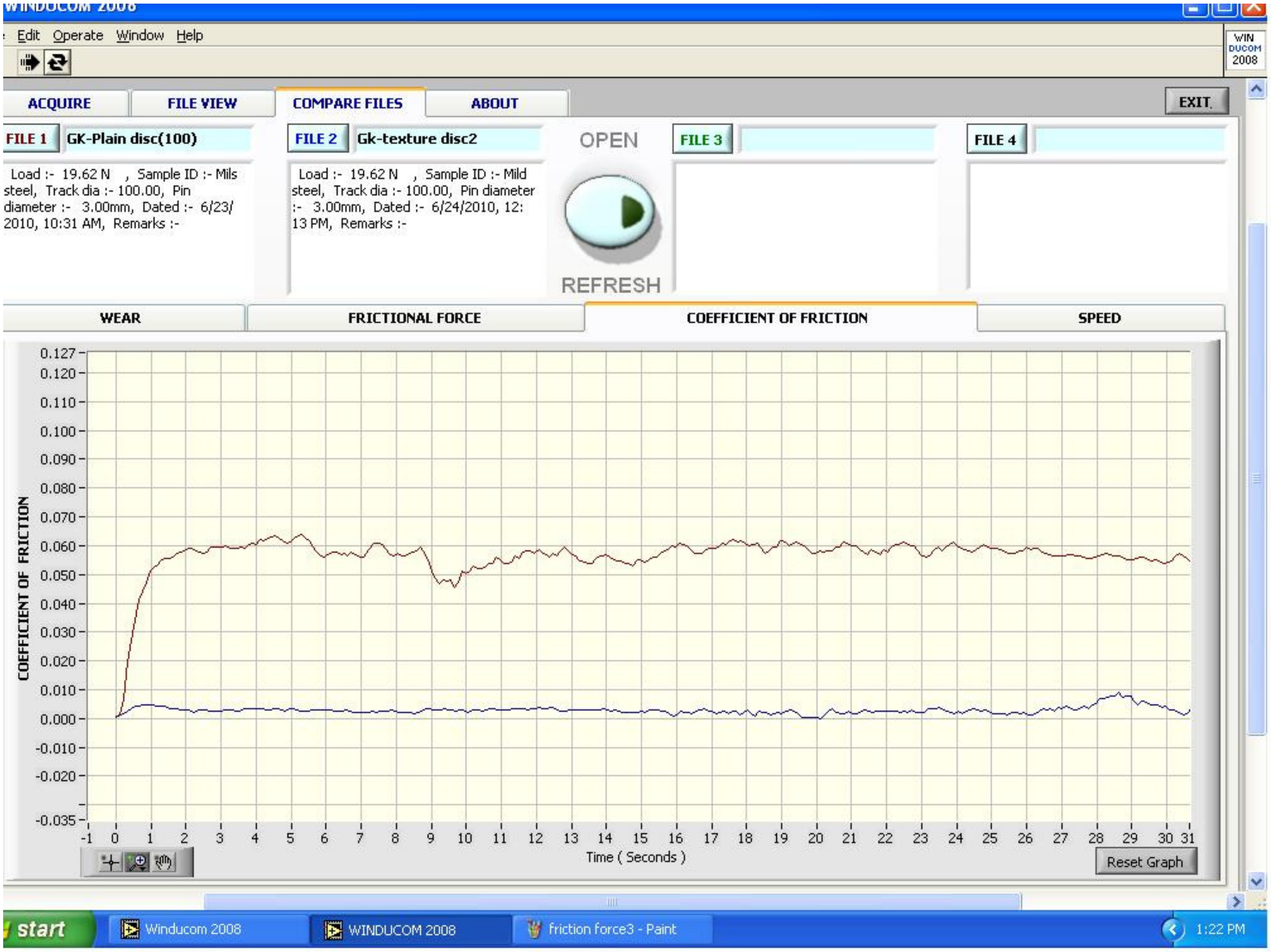


Fig 16 Coefficient Of friction comparison

The coefficient of friction in surface structured disc is less as shown. This is very much visible through the screen of the software “WIDUCOM”. The blue line indicates the structured disc. The set up ran for the same conditions as.

- Load: 2 kg
- Rpm: 500
- Time : 3600 sec

4.2 LUBRICATING OIL ANALYIS

Different test has been done on Lubricating oil to check the contamination of the oil (used in tests).The fluid scan analysis and some other analysis.

Fluid-scanTest :

The test conducted on the used oil.1. Test result of 623 HR. used oil

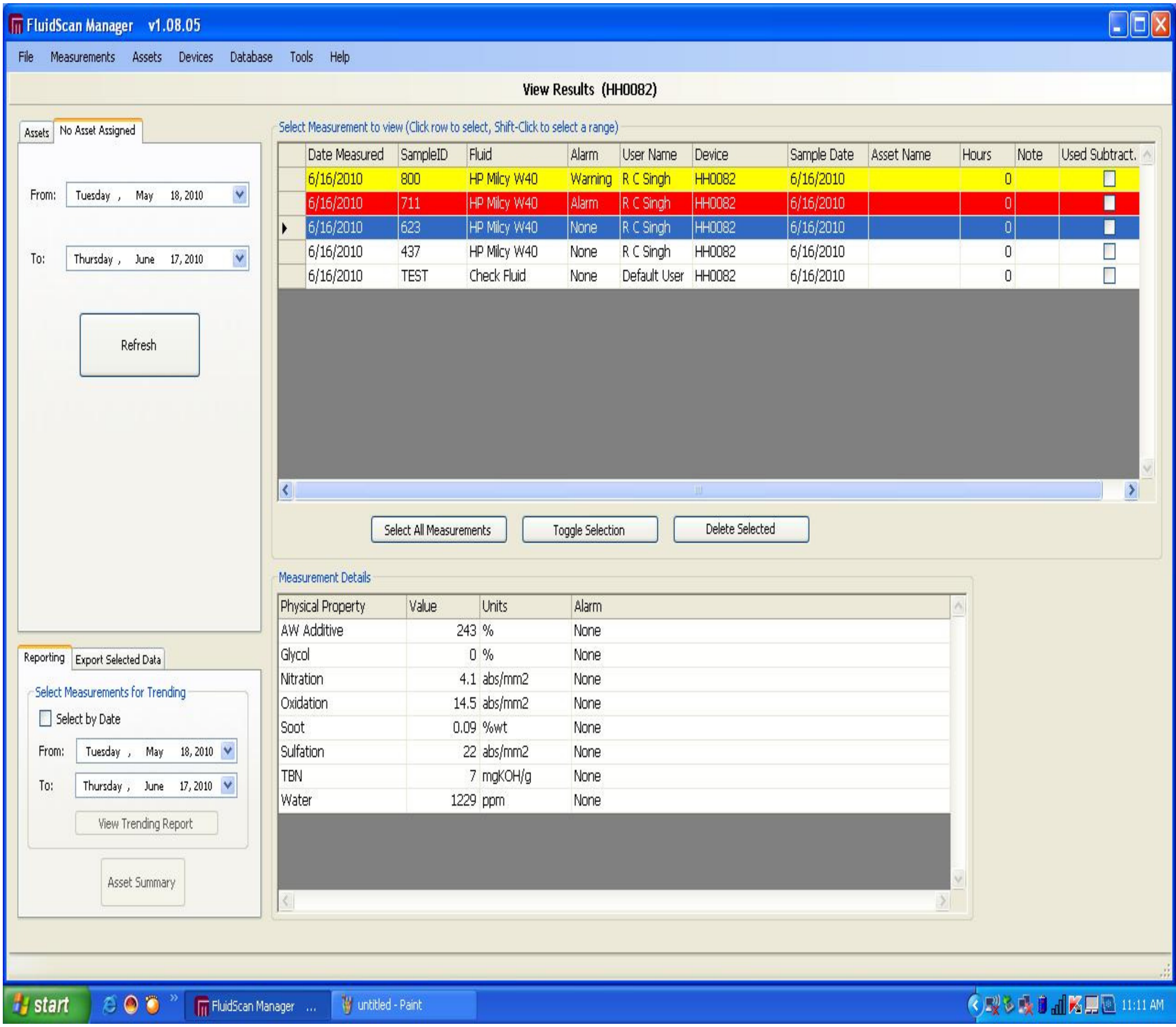


Fig 17 Fluid scan

Test result of 800 HR. used oi

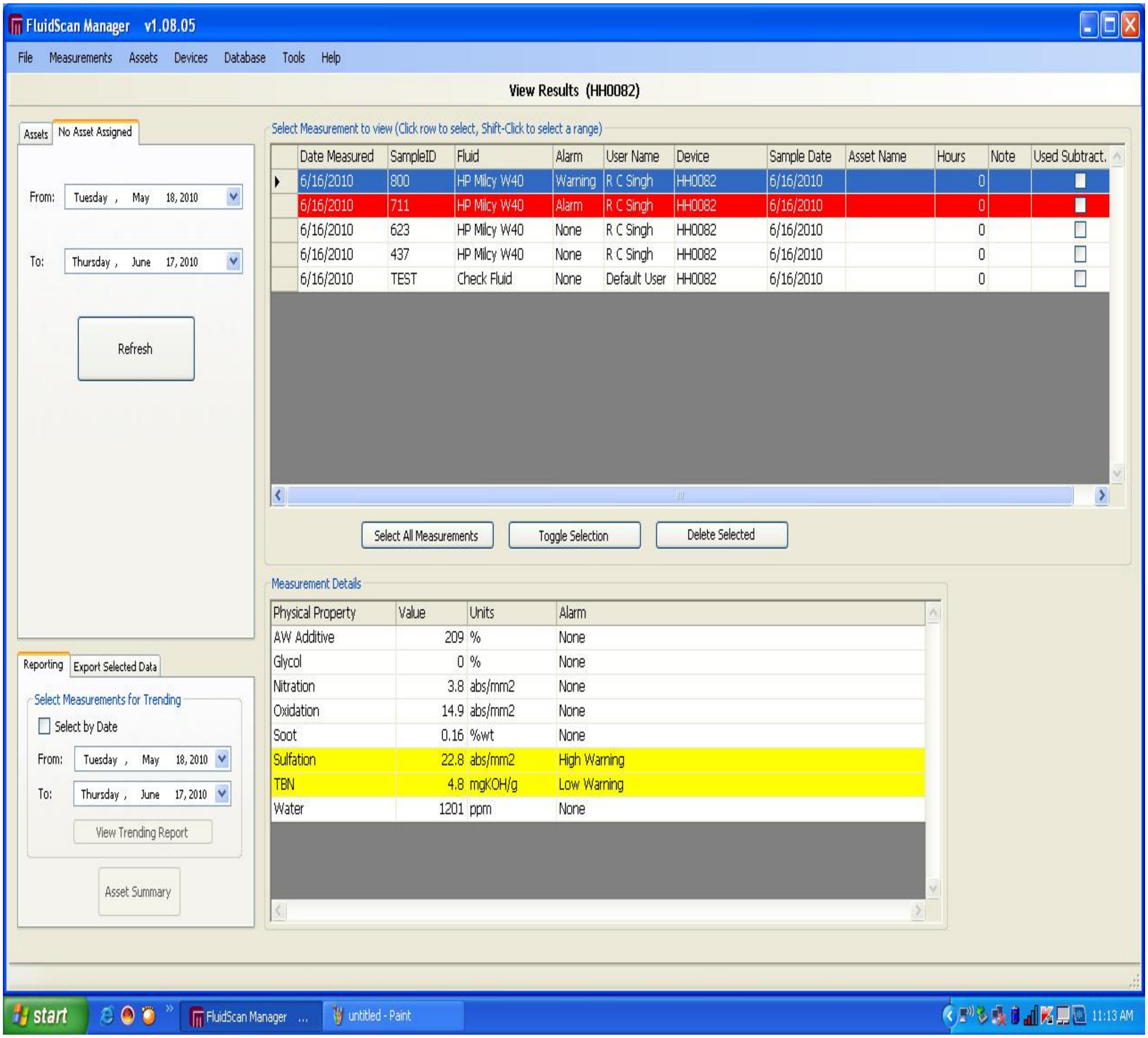


Fig 18 Fluid scan

Test result of 1200 HR. used oil

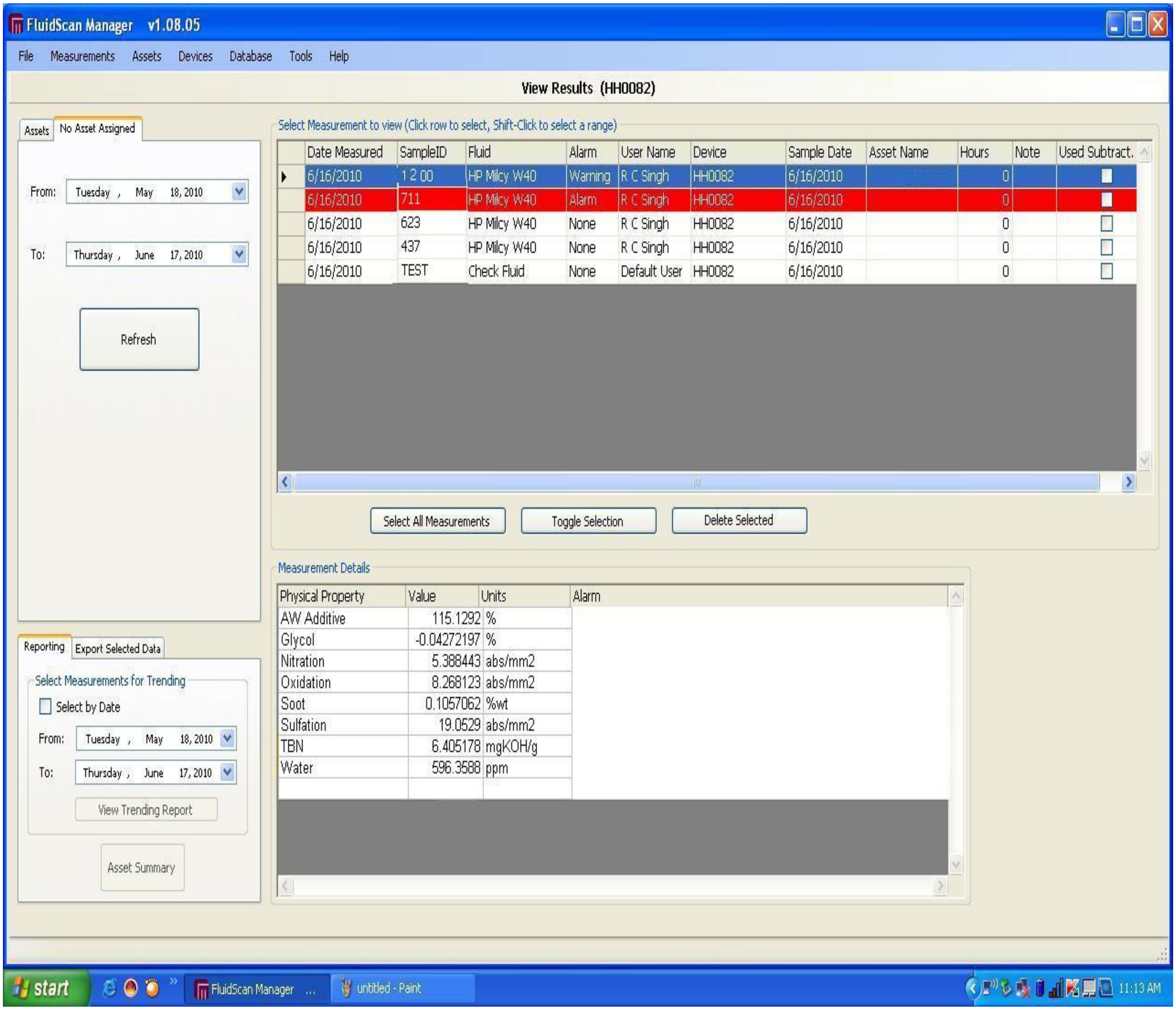


Fig 19 Fluid scan

4.3 Spectroscopy analysis

The spectroscopy analysis of the above oil gave the following results for contamination

Result for 623 HR used oil

6150/08 MAIN PROG PPM 19/04/2010 12:17:14

|623:53 | User Burn Count: 82

	Fe	Cr	Pb	Cu	Sn	Al	Ni	Ag
1	14.48	0.32	6.51	4.66	~0.00	6.04	1.73	0.00
2	14.10	0.35	7.32	4.56	~0.00	6.55	1.57	0.03
AV	14.290	0.332	6.919	4.610	0.000	6.295	1.652	0.017

	Si	B	Na	Mg	Ca	Ba	P	Zn
1	18.52	44.92	71.65	24.98	3342	0.46	1196	1426
2	53.97	44.13	8.87	24.81	3397	0.46	1190	1462
AV	36.243	44.526	40.258	24.897	3369.6	0.456	1192.7	1444.1

	Mo	Ti	V	Mn	Cd
1	11.26	0.13	0.83	1.33	0.17
2	12.48	0.08	1.11	1.52	0.06
AV	11.872	0.105	0.971	1.425	0.113

Table 3

Test result for 800 HR used oil

6150/08 MAIN PROG PPM 29/04/2010 12:31:54

|800:00 | User Burn Count: 86

	Fe	Cr	Pb	Cu	Sn	Al	Ni	Ag
1	16.60	0.41	6.95	4.69	~0.00	6.51	1.78	0.01
2	15.30	0.35	6.71	4.55	~0.00	6.88	1.93	0.04
AV	15.949	0.379	6.830	4.619	0.000	6.694	1.853	0.025

	Si	B	Na	Mg	Ca	Ba	P	Zn
1	15.36	45.92	73.16	24.67	3483	0.46	1272	1478
2	14.06	41.32	70.46	23.66	3511	0.50	1248	1523
AV	14.713	43.619	71.810	24.165	3497.0	0.482	1260.1	1500.6

	Mo	Ti	V	Mn	Cd
1	9.61	0.10	0.99	1.40	0.04
2	13.03	0.09	1.24	1.75	0.01
AV	11.315	0.093	1.112	1.573	0.022

Table 4

Test result for 1200 HR used oil.

6150/08 MAIN PROG PPM 21/05/2009 12:41:19

|1200:53 | User Burn Count: 90

	Fe	Cr	Pb	Cu	Sn	Al	Ni	Ag
1	7.54	0.21	6.21	1.55	~0.00	6.29	1.37	0.01
2	7.44	0.15	5.71	1.49	~0.00	5.99	1.36	0.01
AV	7.490	0.179	5.960	1.523	0.000	6.140	1.362	0.010

	Si	B	Na	Mg	Ca	Ba	P	Zn
1	6.63	2.18	520.13	16.72	3190	0.41	816.78	1164
2	6.68	2.23	512.67	16.07	3064	0.38	799.28	1162
AV	6.653	2.208	516.40	16.395	3127.0	0.396	808.03	1163.1

	Mo	Ti	V	Mn	Cd
1	12.04	~0.00	1.23	1.75	~0.00
2	8.52	~0.00	1.18	1.68	0.09
AV	10.282	0.000	1.203	1.716	0.047

Table 5

CONCLUSION

After observing the graph developed by WINDUCOM on Pin-on Disc simulation process we can conclude that Wear rate, force of friction and coefficient of friction get reduced by the method selected to cut textures on the surface. And it can also be said that Wear has mainly three phases: initial wear, constant rate wear, and rapid wear. And also, frictional force and rate of wear is more in case of contaminated lubricating oil than fresh lubricating oil. Initial wear rate is high as contact area is small, and so the material of high spots will be deformed. In constant wear zone, the speed of wear becomes slow i.e. it remains steady in this period. In rapid or severe wear zone, rate of wear will speed up, so the failure of surface element will occur. And, wear depends on physical properties of two mating surfaces, lubrication, and other given conditions such as load, speed, etc.

The Surface texturing method is a very usable and good process. The method will reduce the force of friction to a countable extent.

The coefficient of friction is also reduced to a good value.

After observing and comparing the results from spectroscopy of fresh lube oil and contaminated lube oil (used for approximately 600 hours, data provided by CASRAE- DCE), we can say:

Wear from abrasive particles and deposits from carbon and oxide insolubles will interfere with efficient combustion in an engine. Valve train wear (cams, valve guides, etc.) can impact timing and valve movement. Wear of rings, pistons and liners influences volumetric compression efficiency and combustion blow-by which results in power loss.

When hard clearance-size particles disrupt oil films, including boundary chemical films, increased friction and wear will occur. There is an extremely high level of sensitivity at the ring-to-cylinder zone of the engine to both oil and air-borne contaminants. Hence, abrasive wear of the ring/cylinder area of the engine translates directly to increased friction, blow-by, compression losses and reduced fuel economy.

Wear particles contribute to oxidative thickening of aged oil. High soot load and/or lack of soot dispersancy can also have a large impact on oil viscosity increases. Viscosity-related internal fluid friction not only increases fuel consumption but also generates more heat that can lead to premature degradation of additives and base oil oxidation.

Deposits in the combustion chamber and valve area can lead to restriction movements in rings and valve control. When hard particle contamination agglomerates with soot and sludge to form adherent deposits between valves and guides, a tenacious interference, called stiction, results. Stiction causes power loss. It causes the timing of the port openings and closings to vary, leading to incomplete combustion and risk of backfiring.The use of optimized lubricants can lead to significantly energy savings (i.e. reduced fuel or electricity costs) and also reduced emissions. Over realistic driving cycles, vehicles fuel consumption can in some circumstances decrease by between 2-5% when fuel economy lubricants are used. For cold-start, short-trip driving, the fuel economy improvement may be even higher still. Hence changing the ISO viscosity grade, or moving to higher Viscosity Index (VI) lubricant could lead to significant energy savings under cold-start conditions. If we assumed that engine friction does scale with the square root of the viscosity, then it would suggest that using a lower viscosity lubricant would lead to friction saving, which would result in lower fuel consumption.

RECOMMENDATIONS FOR FUTURE WORK

The present work has prepared a base for the work to be done for the future. The method can be adopted on the piston ring and a texture can be generated on them to get the friction reduced in the piston cylinder assembly. One of the methods can be adopted as to make the texture on cylinder liner but it is difficult process to be done.The study of lubricating oil indicated that the efficiency of the engine can be increased by proper lubrication and selection of better lube oil. Using a lower viscosity lubricant would lead to friction saving, which would result in lower fuel consumption and better efficiency.

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APENDIX - I

THE REYNOLDS EQUATION

The Reynolds Equation: The x-momentum equation for inertia-free flow provides the basis for the Reynolds equation. The Reynolds equation is used to analyze the flow of viscous fluids through small gaps (i.e., lubrication problems). For example, Figure 3-2 illustrates an inertia-free flow through the gap formed between a stationary surface (at y = H, where H may be a function of x and t) and a surface moving with velocity up (at y = 0).

The height of the gap may be a function of position, x, and time, t, and therefore the pressure is a function of position and time as well. Quasi-steady implies that the characteristic time associated with any changes in the flow (e.g., oscillations of the gap or variations in the flow boundary conditions) is much larger than the characteristic time required for momentum to diffuse across the channel

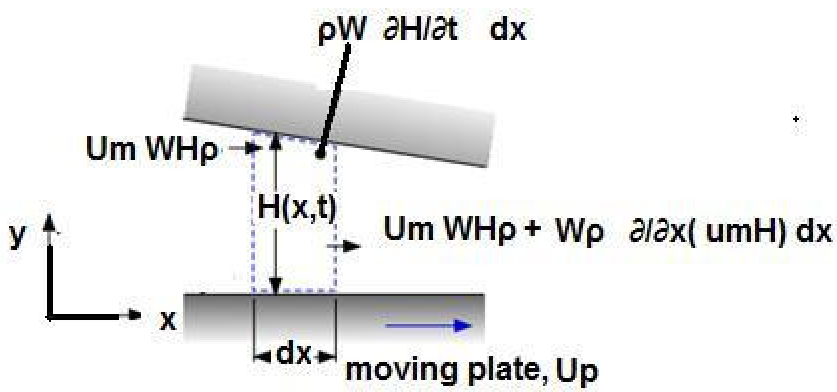


Fig. 11 Flow across plate

$$\frac{\partial p}{\partial x} = \mu \frac{\partial^2 u}{\partial y^2} \qquad \text{(Eq. 3.1)}$$

And integrated twice in y in order to obtain:

$$u = \frac{\partial p}{\partial x} \, {y^2}/{2\mu} + C_1 y + C_2 \quad \text{(Eq. 3.2)}$$

The constants of integration for Eq. (3-2) are obtained by applying the no-slip condition at y = 0 and y = H. At y = 0, the fluid must reach the velocity of the moving surface:

$$u_{y=0} = u_p \rightarrow C_2 = u_p \qquad \text{(Eq. 3.3)}$$

At y = H, the fluid must be stationary:

$$U_{y=H} = 0 \rightarrow \frac{\partial p}{\partial x} \, {H^2}/{2\mu} + C_1 H + u_p = 0 \qquad \text{(Eq. 3.4)}$$

Substituting Eqs. (3-3) and (3-4) into Eq. (3-2) leads to:

$$u = {1}/{2\mu} \frac{\partial p}{\partial x} (y^2 - H^* y) + u_p \left[1 - \frac{y}{H}\right] \quad \left[1 - \frac{y}{H}\right] \qquad \text{(Eq. 3.5)}$$

The mean velocity is obtained by integrating the velocity distribution, Eq. (3-5), across the gap:

$$u_m = \frac{1}{H} \int_0^H u\,dy = {1}/{2\mu H} \frac{\partial p}{\partial x} \int_0^H (y^2 - Hy)\,dy +$$

$$u_p / H \int_0^y \left[1 - \frac{y}{H}\right] dy \qquad \text{(Eq. 3.6)}$$

This leads to: $u_m = - \frac{\partial p}{\partial x} \frac{H^2}{12\mu} + u_p / 2$ (Eq. 3.7)

Equation (3.7) indicates that the mean velocity in the gap will increase with either the plate velocity or the pressure gradient.

A mass balance on a differential control volume (in x) is shown in Figure 3-2:

$$U_m W H \rho = u_m W H \rho + W \rho \frac{\partial}{\partial x} (u_m H)\,dx + \rho W \frac{\partial H}{\partial t} dx \qquad \text{(Eq. 3.8)}$$

where W is the width of the passage (into the page).

Equation (3.8) is simplified to:

$$0 = \frac{\partial}{\partial x}(u_m H)+\frac{\partial H}{\partial t} \qquad \text{(Eq. 3.9)}$$

Equation (3-7) is substituted into Eq. (3-9) in order to obtain a partial differential equation

For the pressure in terms of gap height.

$$0= \frac{\partial}{\partial x}\left[-\frac{\partial p}{\partial x}\,\frac{H^2}{12\mu}+\ up\ /2\ \mathrm{H}\right]+\frac{\partial H}{\partial t} \qquad \text{(Eq. 3.10)}$$

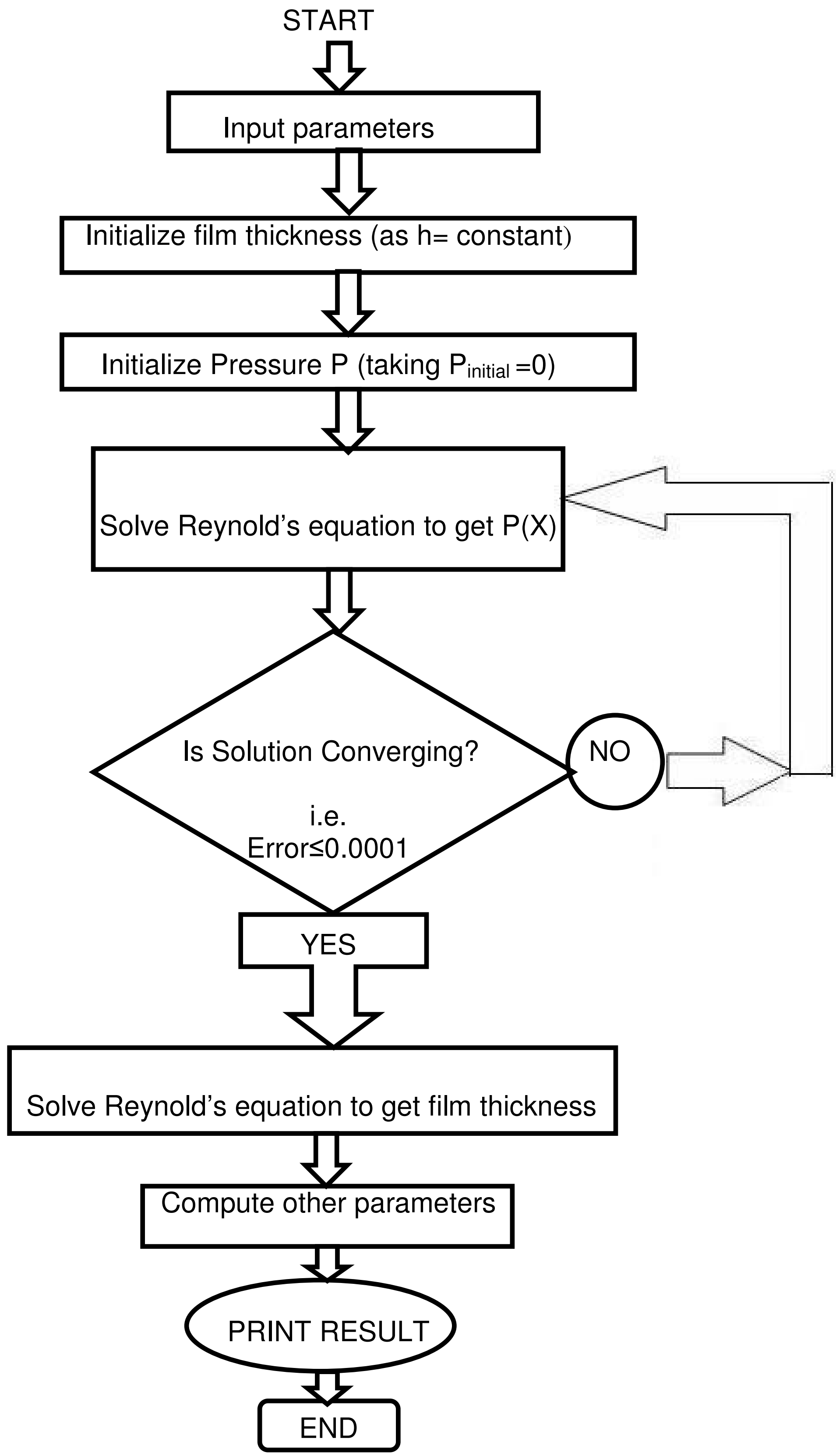
i.e., $$\frac{\partial}{\partial x}\left[\frac{\partial p}{\partial x}\ H^3\right] = \ 6\mu u_p\ \frac{\partial H}{\partial t}\ \ + 12\ \mu\frac{\partial H}{\partial t} \qquad \text{(Eq. 3.11)}$$

Equation (3-11) is one form of the Reynolds equation for lubrication problems. Given an arbitrary description of the gap height as a function of position and time and boundary conditions for the pressure, it is possible to solve Eq. (3-11) either analytically or numerically in order to obtain the pressure variation. The pressure distribution coupled with Eq. (3-6) provides the velocity distribution.

The Reynolds equation can be solved in the practical problems by taking some considerations and some assumptions of boundary conditions. For the basic solution of Reynolds equation, we can assume that initial film thickness (h) to be constant. Now by providing Initial pressure value equal to zero ($P_{initial}= 0$). We compute for the film thickness value , the value is corrected by using iterative mathematical methods. When the value is stabilized we compute other parameters and then based on this value of film thickness we calculate the existing value of pressure.

Based on the method an algorithm has been developed to solve the Reynold's equation and a programme in C++ language has been written based on the algorithm developed.

Also a flow chart has been made to understand the basic steps of solution based on the same algorithm that has been developed. The programme is in the Appendix-II.


START
Input parameters
Initialize film thickness (as h= constant)
Initialize Pressure P (taking P initial =0)
Solve Reynold's equation to get P(X)
Is Solution Converging?
i.e.
Error≤0.0001
NO
YES
Solve Reynold's equation to get film thickness
Compute other parameters
PRINT RESULT
END

APENDIX - II

The C program considers inputs same as those considered by Jeng [26]. The input parameters considered are as given in table along with the values taken from those of Jeng's.

SR.. N.o	Parameter	Value and unit
1	Speed (R.P.M)	2000
2	Viscosity (Pa-s)	6.89E-3
3	Angular speed (rad/s)	209.4
4	Length of connecting rod (m)	141.9E-3
5	Half stroke length (m)	40E-3
6	Thickness of ring (m)	1.475E-3
7	Minimum film thickness (m)	0.37E-6
8	Crown height (m)	14.9E-6

Table 6.

The C program is as follows:

```
#define N  2000;

#define nu          6.89e-3      //viscosity
#define dia         88.9e-3;     //bore dia
#define w           2*pi*N/60    //angular speed
#define l           141.9e-3     //length of connecting rod
#define r1          40e-3        //half stroke
#define b           1.475e-3     //thickness of the ring
#define n           261;
#define m           261          //grid values n-theta , m-x_direction
#define h0          .37e-6               //minimum film thickness composit
surface roughness 1.54E-05
#define nt          8
#define dtheta              4*pi/(n-1)   //discretized theta
#define dx          b/(m-1)              //discretized x T0=35+273;
#define beta        .014         //temp visco coeff
#define Cp          3820         //specific heat
#define rho         970          //density
```

```
#define R               2.6396E-05      //Crown height

for (j=1;j<=m;j++)
{
        for(i=1;i<=(n-1)/2;i++)
        {
                Hx[i,j]= R*((n/2-i)*2/n)^3;
        }
        for(i=(n-1)/2;i<=n;i++)
        {
                Hx[i,j]= R*((i-n/2)*2/n)^3;
        }
}


for (j=1;j<=m;j++ )
{
        for (i=1;i<=n ;i++)
        {
                P1[i,j]=0;
                Pth1[i,j]=0;
                T[i,j]=T0; initializing the temp
                Hm[i,j]=h0; profile matching with the micron size
                nu1[i,j]=nu*exp(-beta*(T[i,j]-T0));
                nu2[i,j]=exp(-beta*(T[i,j]-T0));
                H[i,j]=Hx[i,j];
                if((j<=(n-1)/8+1) && (i==n))
                {
                        P1[i,j]=+1600e3+(+1600e3-160e3)*(j-1)/(1-m/7.5);
                }

                if((j>(n-1)/8+1) && (j<=(n-1)*7/8) && (i==n))
                {
                        P1[i,j]=+160e3;
                }
                if((j>(n-1)*7/8) && (i==n))
                {
                        P1[i,j]=+1600e3-(+1600e3-160e3)*(j-m)/((m-1)*7/8-m);
                }
                if((j<=(n-1)/8+1) && (i==1))
                {
                        P1[i,j]=(+800e3+(+800e3-160e3)*(j-1)/(1-m/7.5));
                }
                if((j>(n-1)/8+1) && (j<=(n-1)*7/8) && (i==1))
                {
```

```
                    P1[i,j]=+160e3;
                }
                if((j>(n-1)*7/8) && (i==1))
                {
                    P1[i,j]=(+800e3-(+800e3-160e3)*(j-m)/((m-1)*7/8-m));
                }
            }
        }
        for (j=2j<=n-1;j++)
         {
            for (i=2;i<=(m-1);i++)
            {
                P1[i,j]=P1[1,j]+i*(P1[n,j]-P1[1,j])/n;
            }
         }
        ------------------ Pressure Loop (Static Condition) ----------------------

        for (j=1;j<=n;j++)
        {
            for (i=1;i<=m ;i++)
            {
                Ht[i,j]=Hx[i,j]+Hm[i,j];
            }
        }

        error1=1;
        itr1=0;
        while (error1>=0.0001 )
        {
            sumP1=0;
            diffP1=0;
            for (j=2;j<=n-1;j++)
            {
                for (i=2;i<=m-1;i++)
                {
                Pold1[i,j]=P1[i,j];
                CC1=(P1[i+1,j]+P1[i-1,j])/(dx^2);
                CC2=(Hx[i+1,j]-Hx[i-1,j])/(2*dx);
                CC3=(P1[i+1,j]-P1[i-1,j])/(2*dx);
                CC4=(Hm[i,j+1]-Hm[i,j-1])/(2*dtheta);
                CC5=Ht[i,j]^3*CC1+3*Ht[i,j]^2*CC2*CC3;
                CC6=6*nu1[i,j]*u(j)*CC2+12*nu1[i,j]*CC4*w;
                P1[i,j]=(CC5-CC6)*dx^2/(2*Ht[i,j]^3);
                if (P1[i,j]<0 )
                {
                    P1[i,j]=0;
```

```
                    P1[i+1,j]=P1[i-1,j];
                }
                P[i,j]= P1[i,j];
                sumP1=abs(sumP1+P1[i,j]);
                diffP1=abs(diffP1+(P1[i,j]-Pold1[i,j]));
                }
        }
        error1=diffP1/sumP1;
        itr1=itr1+1;
        itr1
}
Pt=P';
 k=(m-1)/4;
for (i=1;i<=m-1;i++)
{
        pl[i,1]=P[i,k];
}
%------------------ Minimum Film Thickness (Static Condition) ----------------------

error1=1;
itr2=0;
sumHm1=0;
iffHm1=0;
for (j=2;j<=n-1;j++)
{
        for (i=2;i<=m-1;i++)
        {
                Hmold1[i,j]=Hm[i,j];
                CC1=(P1[i+1,j)-2*P1[i,j]+P1[i-1,j])/(dx^2);
                CC2=(Hx[i+1,j]-Hx[i-1,j])/(2*dx);
                CC3=(P1[i+1,j]-P1[i-1,j])/(2*dx);
                CC7=12*nu1[i,j]*w/dtheta*nt;
                CC8=6*nu1[i,j]*CC2*u(j);
                CC9=Ht[i,j]^3*CC1+3*Ht[i,j]^2*CC2*CC3-CC8;
                Hm[i,j]=Hm[i,j+1)+CC9/CC7;
                if (Hm[i,j]<h0 || i>(m-1)/2)
                {
                        Hm[i,j]=h0;
                }
                sumHm1=abs(sumHm1+Hm[i,j]);
                diffHm1=abs(diffHm1+(Hm[i,j]-Hmold1[i,j]));
        }
}
error1=diffHm1/sumHm1;
```

```
//^^^^^^^^^^^^^^^^^^^^^^^^^^^ LOAD CALCULATION
^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^
for (j=1;j<=m-1;j++)
{
        Wsum1=P1[1,j]+P1[n-1,j];
        Wsum2=0;
        for (i=2;i<=(n-2);i=i+2 )
        {
                Wsum2=Wsum2+P1[i,j];
        }
         Wsum3=0;
        for (i=3;i<=(n-1);i=i+3 )
        {
        Wsum3=Wsum3+P1[i,j];
        }

        Wsum(j)=Wsum1+4*Wsum2+2*Wsum3;
}
Wsum11=Wsum(1)+Wsum(m-1);
Wsum22=0;
for (j=2;j<=(m-2-1) ;j=j+2)
{
        Wsum22=Wsum22+Wsum(j);
}
Wsum33=0;
for (j=3;<=(m-1) ;j=j+2)
{
        Wsum33=Wsum33+Wsum(j);
}
TWSUM=Wsum11+4*Wsum22+2*Wsum33;
LOAD=(dx*dtheta*TWSUM/9)*pi()*b;


//^^^^^^^^^^^^^^^^^^^^ Friction Force Calculation ^^^^^^^^^^^^^^^^^^^^^^^^^
for (i=2;i<=(m-1-1);i++)
{
        for(j=2:1:n-1-1)
        {
                tau[i,j]=nu*u(i)*nt/Ht[i,j]+Ht[i,j]/2*(P1[i+1,j]-P1[i-1,j-1])/2/dx;
        }
}

// tau
for (j=1;j<=(m-1-1 );j++)
{
        Wsum1=tau[1,j]+tau[n-1-1,j];
```

```
        Wsum2=0;
        for (i=2;i<=(n-2);i++ )
        {
                Wsum2=Wsum2+tau[i,j];
        }
        Wsum3=0;
        for (i=3;i<=(n-1);i=i+2)
        {
                 Wsum3=Wsum3+tau[i,j];
        }
        Wsum(J)=Wsum1+4*Wsum2+2*Wsum3;
}
Wsum11=Wsum(1)+Wsum(m-1);
Wsum22=0;
for (j=2;j<=(m-2-1);j=j+2 )
{
        Wsum22=Wsum22+Wsum[J];
}
Wsum33=0;
for (j=3;j<=(m-1);j=j+2 )
{
        Wsum33=Wsum33+Wsum[J];
}
TWSUM=Wsum11+4*Wsum22+2*Wsum33;



frictionf=(dx*dtheta*TWSUM/9)*b;
co_fric=frictionf/LOAD ;
xFF=max(FF);
yFF=min(FF);
xPOW=max(POW);
mHm=(max(Hm));
for (i=1;i<=n;i++ )
{
        xHx[I]=Hx[I,1];
}
```