A Major Project-II

on

Tribological Investigations of Mild Steel Pins of Different Cross-Sectional Area with Cast Iron Plate

Submitted in partial fulfillment for the award of the degree

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Master of Technology

in

Production Engineering

by

Ranjeet Kumar Thakur

(2K11/PIE/24)

Under The Guidance

of

Dr. R. C. Singh	**Dr. Roop Lal**
Professor	**Assistant Professor**

Department of Mechanical Engineering

Delhi Technological University

Bawana Road, Delhi-110042

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Declaration

I, **Ranjeet Kumar Thakur, Roll No.-2K11/PIE/24** hereby declare that the Major project-II titled "**Tribological Investigations of Mild Steel Pins of Different Cross-Sectional Area with Cast Iron Plate"** submitted to the Department of Mechanical Engineering, Delhi Technological University, Delhi in partial fulfillment of the requirement for the award of the **Master of Technology** in **Production Engineering** is original work and not copied from any source without proper citation. This is further to declare that the work embodied in this report has not been submitted for the award of any Degree, Diploma, or Certificate for any other institution or university.

Place: Delhi **Name: Ranjeet Kumar Thakur**

Date: 13.12.2019 **(2K11/PIE/24)**

CERTIFICATE

We hereby certify that the above declaration made by **Mr. Ranjeet Kumar Thakur, Roll No-2K11/PIE/24** is true to the best of our knowledge and belief.

Dr. R. C Singh **Dr. Roop Lal**

Professor Assistant Professor

Department of Mechanical Engineering
Delhi Technological University
Bawana Road, Delhi - 110042

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Name: Ranjeet Kumar Thakur
Roll No.- 2K11/PIE/24

Abstract

Mechanical, electromechanical and biological systems are used in every engineering application and it has been noted that the substantial amount of energy is consumed in overcoming friction between the mating surfaces of these systems. Wear of parts or components results in resource wastage and unnecessary heat generation. A huge amount has been spent till date to cover for friction and wear losses.

The study of tribological properties (i.e., friction, wear and lubricants) has gained significance in the recent years as an effort for the development of new material to reduce the frictional losses. Tribological study has of utmost importance for many systems which depends on wear and friction values. There could be enormous amount savings by improved tribological practices over the system.

The experimental study of tribological characteristics contains study of friction and rate of wear of tribopair for which experiment was carried in lubricated conditions at different temperatures. The high temperature Rotary Tribometer was applied for determination of coefficient of friction and rate of wear at interface of tribopair.

From the results so obtained after experiment, it was found that the wear rate initially had high value of 7×10^{-6} mm^3/Nm and then decreased to a stable value of 2×10^{-6} mm^3/Nm.

The coefficient of friction initially had value of 0.14 which then decreased to a stable value of 0.04 after having attained constant relative speed of tribopair

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Chapter 1

Introduction

1. Tribology

Tribology science is one of the fundamental concept of engineering, in particular engineering design, is the scientific name of the study of mechanism relative motion of interacting type surfaces.

The word of tribology is taken from Greek dictionary word “tribos”, which relates to that rubbing and adding the entire interacting surface in relative motion, sciences technology.

The core analyses and applications are friction field, wear field and lubrication field. The existence of the term tribology was not there till 1964 .there are many pictures of practical tribology from ancient Egypt, when early workers using the knowledge of tribology, utilized oil to make large status sliding possible.

Tribology field is necessarily very interdisciplinary and use skills sets from mechanical engineering, material science and engineering, chemical engineering etc.

Tribology is especially vital in present day’s world because huge amount of energy is lost due to mechanical parts friction. To use little energy, we required to reduce the vital energy which is wasted. a good and important amount of energy is lost due to friction in surfaces which one is sliding, hence finding different ways to decrease friction and overall wear with the help of new practices and technological advancement in tribology is very important to a greener and effective sustainable world.

1.1. Importance of Tribology

Within last two – three decades, the highly industrialized countries governments have developed in-depth method to reduce energy waste. In 1966, the report of P. Jost indicated that the enormous waste of material and energy that occurred because of the wrong recognition of tribology in the Britain. Similarly, an American government financial report in 1977 suggested that by the proper use of tribology science, 16.25 billion USD might have been saved by the proper use of tribological science. Similar studies and research have also been conducted in different countries including India. for example: related to Germany, W. Bertz carried out research in between 1979 to 1983,where as in 1990, P.Jost outlined potential savings in GDP, which was determined by calculation value of 1.3 – 1.6 % . Here it is important to note that the study of W. Bertz taken under consideration lubricated contacts alone, it recognized the savings possibilities into (j) primary savings because of reduction in friction related to mechanical. (ii) Secondary savings, due to a lesser replacement of machine elements because reduction of wear (iii) tertiary savings in which a replacement material is employed for the manufacture of new parts thus increasing the materials energy content. Presently, the integration of tribology reflects a revolutionized aspect in assessment of life cycle, with the main motion being environmental and best economic performance. In this scenario, the main role of tribology is not only just reducing friction in a machine but also increase the service life of machines.

1.2 Fundamental principle of Tribology

With many years of indepth research and development in tribology, it is concluded from the fact that friction and or properties of wear of a materials of different type are not its internal or intrinsic type properties. Friction and wear properties dependent on many other factors which are mainly related to a definite application. Quantitative measures for frictions and wear in terms of frictions coefficient and rate of wear mentioned in many other engineering text books depends on the below given basic parameter groups:-

(i) The system structure means its elements and their related properties.

(ii) The operation variables means stress/load, motion ,temperature, time

(iii) System's component's mutual interaction

1.3 Tribological Problem Design

1.3.1 Plain Sliding Bearing

Whenever a journal type bearing starts operating within the hydrodynamic film is developed under such conditions, conformed surface are completely separated and an ample amount of flow of lubrication is provided to stop heating. In these conditions of total separation, mechanical type of wear does not occur. Sometimes due to misalignment, inherently due to thermal as well as elastic distortion which may occur due to machine assembly method and or changing nature, could lead to metal to metal type contact. Moreover, contact could occur at moment of beginning ,i.e.- before hydrodynamic film had opportunity to form totally ,the bearing can be over loaded from observed time to observed time and external particles can enter into the fluid film space .

In IC engine case, acids and some other corrosive substance could be generated during combustion and can be carried by the lubricant, thus induce a wear of chemical kind. The continues action and hydrodynamic pressure removal onto the shaft can shift loosely held particles. In further other cases however, foreign matter particles are responsible for maximum wear case in particular situation .mainly, the harder particles are held between the journal and the bearing. Sometimes the particles thus trapped forged in the surface of the comparatively softer material, reliving the situation. However, it is normal for the hard partials to be tapped in the bearing surface which gives increase to wear on the hard and soft surface.

1.3.2 Rolling Contact Bearing

Rolling contact bearing forms the widest category of element of machine which employs hertzian type contact problems. From a most practical and evident point of view, they are normally divided into two main classes, ball bearing and roller bearing. Although contact type and applicable laws of friction and wear behavior are similar to both the cases. In taper roller bearing, contact is although ‘a rolling one’ on a element of sliding is involved. The designers get many type of specialized and specific research papers based on rolling content problems which bwilder him. A designer typically wishes to clear his stored pertaining to relative importance of elasto- hydrodynamic (means. Physical) and boundary (means.physiochemical) phenomenon. Designer needs a frame of reference for the analysis of many type of available materials contact and type of lubricants, and he would surely appreciate information related to what type of application is possible for mechanism of rolling contact and what would be the cost and what is beyond current variables practices. Similar to almost many engineering applications, lubrications of a rolling hertz contact is used for two reasons to reduce the force of friction and to reduce the chance of the contact’s failure. These two things are co equal with sliding elements and control of friction is the most important interest, but control of failure is so far is the most important reason of rolling contact lubrication. It is very true that the lubrication which provides failure free functioning of a rolling contact will also control the friction forces within acceptable limits .Considered as the primary goal of failure control rolling contact type lubrication. The review of lubrication technology of contact type could be based on relation between these lubrication and failure which makes the contact non-functional ,hence for the best value of this treatment process ,effective and effective advances have been made recently in the analyses and understanding of lots of the very important rolling contact failure type. Those times are very near when a failure detected at least in their previous stages will be sufficient to analyze a failure rolling contact and explained, in research, the behaviors of lubrications and behaviors of contact material which has caused to the failure. These many failure methods allow the engineer to develop corrective design changes to this machine and importantly to enhance lubrication so that before life or avoidable rolling contact failure can be controlled .For this a closed correlation between theory of lubrication and the mechanism of failure should be studied because it can provide to verify concept of lubrication of such a level when and where they mostly matter in very practical way.

1.3.3 Piston, its rings and Cylinder liners.

Piston with cylinder is a very common element of machine which is a part of an engine. The main functioning of a piston assembly has to perform like a seal and to counter balance the fluid action force which acts on the piston head. In the most of the cases, piston rings used as for sealing action, although these are avoided some times in the fast running high pressure hydraulic machine, on the other hand maximum wear occurs proximity of the top dead center. Wear pressure, temperature and velocity and its combination least favorable to the functioning of a hydrodynamic film. In the IC engine, the cylinder condition can be introduce to the concept of tribo-design which is due to the availability of sulphur and or other dangerous elements in fluids and all are very corrosive .Corrosion may be in particularly be dangerous before an engine has warned up and walls of cylinder are very much less than the dew point of the acid solution**.**

1.3.4 Cam and its followers.

Elasto hydro dynamic theory of lubrication may help to know how cam and followers contact function from the lubrication view point but it has not effectively provided the concrete design criterion. In the cam and tappet mechanism, friction is relatively not so important factor which affects its performance and their important effects is to generate unnecessarily heat. Hence the minimum desirable value is required. So far as contacts are concerned, the important design requirement is that surfaces in working should support the applied loads without much wear or some other type of failure of surface. Hence it is understood that the cams and the tappets development is dominated by the requirement to avoid failure of surface. The main design consideration is to ensure an appropriate thickness of film. It is true that if the nose radius of cam reduces, which in turn raises stress and that relative velocity and hence the thickness of oil film. The cam having film of more thickness operates effectively in the service while cam having less thick film fails well before its life.

The limitations of temperature are very important in field of cams which are necessary to operate under intensed conditions and in this case scuffing is the high potential type of failure. The fact of non steadiness of loading conditions of cams should also be taken into account during stage of design.

1.3.5 Friction drive.

Friction type drives are mainly used largely in infinite variables gear are inverse hypoid gears on as far as intended two smooth machines elements must roll without showing sliding together, while being able to transfer circumferential force from one element to another.

Friction drives usually functional in the elasto-hyrodynamic lubrication case. Three principle modes can be identified if friction traction verses sliding speed is plotted. Liner mode is the first mode in which traction and relative velocity of sliding are proportional to each other. Second mode is in transition mode in which a maximum traction is reached and lastly the third zone with falling characteristic. The material region is related to rheological properties of the oil and the main parameter is viscosity. It is surprising that the maximum value is absorbed in second zone. Hence from this it is concluded that lubrication within a, film behaves glass like solid under hertzian contact under high pressure, it has a limiting strength compare to other solids corresponding to traction's maximum value. Related to third zone, decrease in traction is mainly linked to the decrease in their viscosity related with lubricants' operating temperature rise.

1.3.6 Involute Gears.

In the involute gears, teeth rolls one over other without sliding at the moment where contact line passes through the common tangent of the pitch circle. During the left out time of interaction, means whenever the zone of contact lies in between the addendum and the dedendum, a less magnitude of relative sliding observed. That is why pitting surface failure is most likely to occur on the pitch line while scuffing is observed in the addendum, dedendum zones. It is having experimental evidence that with good quality gears of hardened nature, scuffing happens at the point of combination of acceleration and overload where maximum disturbance occurs, but, before scuffing stage, some different type of damage is observed which is near the tip of both gear and pinion teeth. This sort of damage is assumed to be because of abrasion due to hard detached type debris from the wedge tip. Due to hertzian cycle stress surface fatigue is observed. The rise of fatigue nature cracks can be linked to the lubricant trapped in crack development during next cycle. Due to excess factor of safety, many gear systems now a day used are not much influenced by shortage of lubrication. However in linear and compact design which is required to have more reliability degree, at elevated operating stress, temperature or speed .the lubrication truly become an important factor. there are many means of methods have been formulated to know the sufficient lubrication of gears they have given a design purpose in general but in more limits to the gear size and different operating conditions. Gradually with the search keeps on going, the speed range, the loads continue to expand and with this, designers preferring to keep away from structure non mathematical approach. Two different type of sufficient lubrication defining methods have gained little popularity in past recent years. First being least film thickness and the second is criteria of critical temperature .These two concepts have a theoretical type failure mode remains as hypothetical.

1.3.7 Hypoid gears.

Hypoid type gears are generally utilized in automobiles with axles and associated right angle drives, in this gear tooth action combine both the rolling actions of spiral level, gears having a degree of sliding and that is why this type of gear is critical for surface loading conditions. For successful operation of hypoid gear, provision of extreme pressure oil is very much required. Extreme pressure oil is oil which contains additives which generate surface protective layer which is very much required at elevated temperature. Lead- soap, active sulphure are few additives which may prevents scuffing which in particular have not been run-in when of gears phosphate have not been done. These additions are not much suitable in maximum torque requirements but these are very effective in raised speed requirements. Tribology in additives in machine design is usually satisfied in high toque, less speed case but exceptionally low so at elevated speeds. In this case, failure modes are scuffing and pitting

1.3.8 Worm gears.

Worm gear has greater degree of conformity which is highest among all gears. That is why it is a special, lower pair’s family of screw pair. In this case of high degree of relative sliding, it becomes critical phosphor bronze along with hardened steel is only suitable option from wear point of view. Accurate right positioning and a good surface finish are also essential. Lubrication are used in warm gear is mixed or boundary lubrication type with lubrications generally contain surface active additives. That is why mild wear and corrosive type is observed due to boundary lubrications. It is clear from above presented discussions that an engineer converted for tribological design must be able to analyze the problem which is in front of him, be it moving parts or bearing, and bring the best solutions.

1.4. Wear

From the engineering point of view, wear is a surface damage which comes out due to mainly relative motion between two surfaces. Wear is a type of damaged and is not just loss of material, however if loss of materials occurs, then it is one way of experiencing wear on surface. Another way it can be defined as movements of material without having loss of materials.

For example plastic deformation of a component and change of shape and size thereof. e.g. repeated hammering.

Third way of defining wear is damage to a surface without mass loss or dimensional changes.

Example is network of cracks on a surface .it is more important in case where optical transparency maintaining is required. Examples are lenses and air craft windows.

1.4.1. Wear Mechanism

In wear mechanism, wear occurs in different engineering field practice have been divided into four main types, namely adhesive type wear, surface fatigue type wear, abrasive type wear and chemical type wear. Wear basically is a loss of material from contacting surfaces which are in relative type motion. Wear is governed by the material properties, operating and environment conditions and contacting bodies' geometry. For some material especially organic polymer relative motion kinematic and zone of contact can also be listed. There are two category of wear mechanism, first is related to mechanical behavior and second is related to chemical nature of material. In every case, it can be identified the possible loading wear mechanism, be it mechanical properties related or chemical solubility of material with the temperature condition within zone of contact and operating conditions.

- **Adhesive wear**

Adhesive type wear is the removal of material during rubbing of two metals together under sufficient force. The removal of material occurs from less wear resistant surface. Adhesive wear depends on physical as well as chemical factors for example material properties, presence of corrosive surrounding or chemicals at the same time it depends on dynamic like velocity and applied load. In this wear phenomenon, it is considered force instead of chemical reaction.

There are many steps which lead to the formation of adhesive type wear particles which can be shown as

a) Contacting asperities deformation
b) Surface films removal
c) Adhesive junction formation
d) Junction failure and material transfer
e) Transferred fragment modifications
f) Transferred fragment removal and loose wear particles creations

The removal of volume of material due to adhesive wear can be calculated by below formula as given by 'Archard'

$$V_a = k.\frac{W}{H}.L$$

Where: 'k'= coefficient of wear

'L'= distance of sliding

'H'= softer material hardness

Following parameters influence the adhesive wear characterizing the contact body

(i) structure related to electronic
(ii) structure related to crystal
(iii) orientation of crystal
(iv) cohesive strength

For an example compare to any type body centered cubic (BCC) or face centered cubic (FCC) metals, hexagonal closed packed (HCP) metals are effectively more resistive to adhesive wear in general

- **Abrasive Wear**

Abrasive type wear although being generally common type, is very important and dangerous type of wear. It develops when physical contact of two different interacting surface occurs and in which one of them should be significantly harder compare to other. Under the action of normal load, the softer surface is penetrated by harder surface asperities which results in plastic deformation.

- **Wear due to surface fatigue**

In rolling contact bearing, gears, cams and follower, friction drives like numerous machine elements, there are many sites of relative motion of hertzian contact which are load bearing, non-conforming contact. The surface relative motion consists of many degrees of pure sliding and rolling. Continued load cycling in case of loads being not negligible later on leads to material failure at the surface of contact. The failures are assigned to many reversals contact type stress zone and that is why called as fatigue type failure.

There are many steps which can be indentified for the wear particle generation, these are:

(i) Stress transmission at point of contact
(ii) Per cycle plastic deformation growth
(iii) Crack nucleation as well as surface voids
(iv) Formation of cracks and its propagation
(v) Wear particle creation

Material removal due to fatigue, V_f can be calculated from the below expressions for sliding contact.

$$V_f = C.\frac{\eta.\gamma}{\varepsilon_1^{-2}.H}.W.L$$

Where η = asperity heights distribution,

γ = particle size constant,

ϵ_1 = strain to the failure per loading type cycle

H = observed hardness.

- **Wear due to chemical reactions**

Wear due to chemical reaction which is developed due to friction is affected by surroundings and contact material active type interaction. There is a definite and clear defined series of events which leads to formation of different wear particles.

Initially, the contact surfaces combine with the surroundings which create reaction type products which get accumulated at the surfaces. And the second further step includes the removal of product of reaction which is due to mainly formation of crack and abrasion. Thus original material is once again come to attack of environment. Friction process can further lead to mechanical activation and thermal activation itself of the layers of surface which will bring following changes.

(i) Occurrence of higher reactivity because of higher temperature. Due to this, the reaction product formation is highly in a condition of acceleration.

(ii) Higher brittleness observed due to high properties of work hardening.

A very simple and the effective model of chemical wear may be utilized to calculate the extent of loss of material.

$$V_t = \frac{k.d.W}{\xi^2\rho^2\,H.V}\,.L$$

Where, k = oxidation velocity factor,

d = asperity contact diameter,

p = reaction layer's thickness, (critical thickness)

H = material hardness

1.4.2. Wear model (Archard's law)

It was proposed by Archard that says total volume of wear is directly proportional to actual contact area times distance of sliding. Proportionality constant 'K' which is the coefficient between actual contact area, distance of sliding and the value of wear has been given as,

$$V = K.A_r.l = K.l.\frac{W}{H}$$

Where: V = represents volume of the wear (m^3)

K = represents the constant of proportionality

A_r = represents the real contact area (m^2)

W = represents the load (N).

H = represents the softer surfaces Vicker's hardness (Pa)

l = represents the distance of sliding (m)

The coefficient 'K' is called as Archard coefficient, wear constant or wear coefficient.
The low value of 'K' shows that wear is due to very small portion of small asperity constant

The same terms, coefficient of wear is defined by the below given equation.

$$k = \frac{V}{W * L}$$

Where , k = specific rate of wear (wear rate) [mm^3/N-mm];

V = wear volume [mm^3];

W = normal load [N];

L = sliding distance [mm];

1.4.3. Contact Mechanics

Contact is a complex phenomenon involving deformation and molecular force. Simpler abstractions are used to make sense of it.

There are following type of contact models:

- Zero friction point contact
- Point contact along with coulomb friction.
- Contact of soft finger.

1.4.3.1. Point Contact

1. Consider rigid objects A and B that make contact over region R
2. Contact pressures $\rho(x) \geq 0$ for all $x \in R$

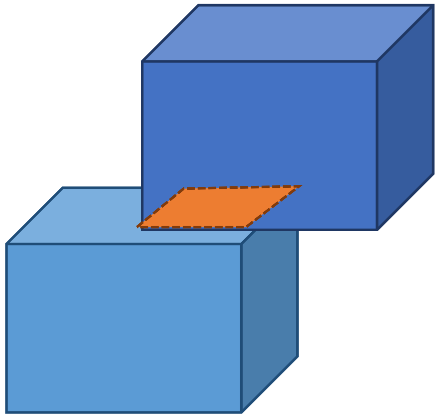

Fig 1.1: Point Contact

1. If R is a planar region, with uniform friction and uniform normal, then all pressure distributions over R are equivalent to
    a. A combination of forces on convex type hull of R
    b. If R is polygonal, a combination of forces on the vertices of the convex type hull of R as shown in Fig. 1.1

1.4.3.2. Frictionless contact points:

- Contact point $C_{i,}$ normal n_i =1…..N
- Non penetration constraint on objects motion: $n_i^T c_i > 0$
- Here c_i is measured w.r.t the motion of the object
- Unilateral constraint as shown in Fig.1.2

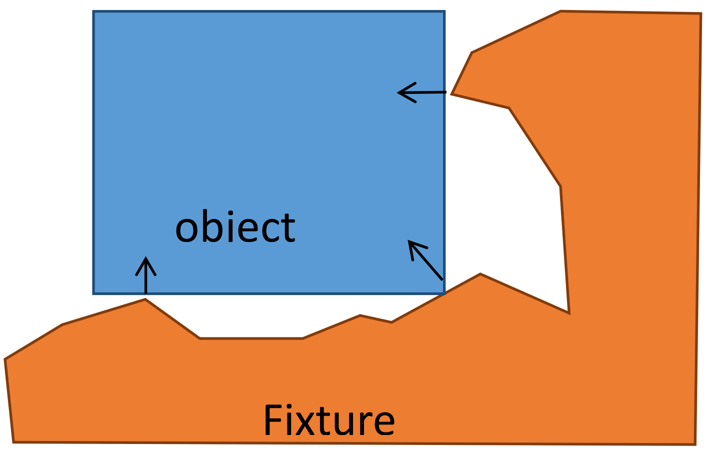

Fig 1.2: Frictionless Point Contact

1.4.3.3 Frictionless Dynamics

Assumptions of the Frictionless Dynamics that depicts according to the Fig.1.3

- Assume body at the rest.
- Consider pre-contact acceleration a, angular acceleration ω.
- Non penetration $n_i^T c_i \geq 0$ must be satisfied post-contact.
- Solve for non-negative contact forces f_i that alter acceleration to satisfy constraints.

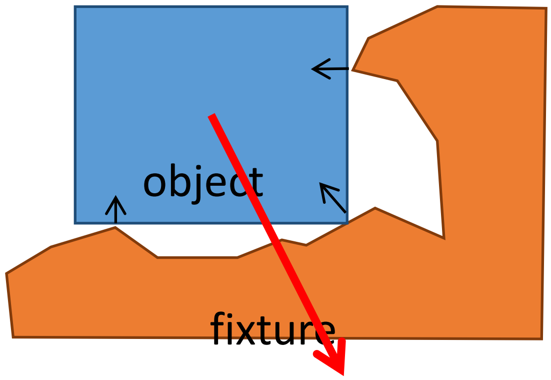

Fig 1.3: Frictionless Dynamics

1.5 FRICTION

Friction may be defined as any forces which resists a relative motion between surface in contact. Due to inter-surfaces adhesive combination dry friction arises. Apart from this ,surface roughness, surface deformation and contamination of surfaces are few factors for dry friction development. Due to the complexity of those interactions surface, the calculation of friction from first principles becomes impractical.

Thus we need to use methods which should be empirical for the analyses along with development of theory.

By any object, work done observed against friction is depended on path. In the case of friction in form of heat some energy is lost.

Wear represented as empirical laws of three numbers

- Amonton's friction first law :- The friction force is directly proportional to load applied
- Amonton's friction second law:- the friction force is not depended on apparent contact area.

- Coulomb's friction law: - Kinetic nature of friction is not dependent on the velocity of sliding.

Dry friction is a force that resists the sliding of one surface over the other i.e. it shows resistance to relative transverse motion between two solid surfaces which are in contact. Dry type friction is of two types: the first type is the static friction which exists between surfaces of non-moving, and second is the kinetic friction, also known as sliding type friction and also dynamic type friction, which exists between moving type surfaces.

The possible maximum friction force which exist between two given surfaces before sliding beginning is the product of static type friction coefficient and normal force of friction: $F_{max} = \mu_s$ x N .When it is no relative motion or sliding between the two contact surfaces , frictional force can have any value between 0 to F_{max} . When any other force lesser than F_{max} attempts to slide over the other, it is opposed by a force of friction of same magnitude but in the opposite direction. Any force, greater than f_{max} which overcomes the friction force and causes the surfaces to slide over each other. Static friction acting on the body becomes inapplicable as soon as the sliding motion starts – the friction acting when the motion has started is said to be kinetic friction.

The force of friction acting in between two surfaces post sliding starts is the product of kinetic type friction coefficient and normal force: i.e. $Fk = \mu_k$ x N.

Chapter 2

Literature Review

Bortoleto et al. [1] presented experimental as well as mathematical analysis of dry nature contact in the test of pin-on-disc. This work shows a computer based study on the linear wear law of Archard and important concept of finite element modeling (FEM). In the analysis of un-lubricated sliding wear found on pin on disc type tests. Those modeling was analyzed and developed using finite element computer software 'ABAQUSS' with three dimensional deformable geometries and behavior of material for contact surface as elastic plastic type into 'FORTRAN' users subroutine (UMESH MOTION), Archard's model of wear was applied. From the experimental process, value of global wear coefficient was obtained and modeling of debris and formation of oxides were done.

Kennedy et al [2] carried out a research on During pin on disc tribometer testing machine , the contact temperature and its effects on wear was carried out as a research. This research presents most useful mathematical analysis and numerical method which can be used to find surface temperature increase in dry as well as boundary lubrication pin-on- disc tribometer. The mathematical expressions has been developed which is simple and accurate and also easy to use which can be applied to find contact temperatures in pin on disc contact of sliding ceramic (zirconia), metal (stainless steel) and polymer (polyethylene's) wear type tests were conducted on pin-on-disc tribometer.

Xinmin et al. [3] uses of pin-on-disc, they studied different tribological characteristics of sintered in comparison to standard gear materials of steel and they performed simulation of part of gear tooth which is sliding contact in two different regions of boundary and lubricated regions of mixed type there after comparing tribological properties of two numbers sintered gear materials verses standard gear materials. In different configuration like standard verses standard gear material, sintered versus sintered gear material and sintered verses standard gear materials. The comparison were done between damage mechanism, friction and wear between these materials.
From results, it is clear that same material pair has lower friction coefficient. For PM as well as RS combinations, with RS disc material, PM pins have low coefficient of friction than the same

with RS pins and PM disc materials. At less and more speeds, coefficient of wear, RS pins exhibit more resistance of wear than same of AQ or MO pins, reason being more hardness and due to microstructure of compacted type.. For RS-PM sets, MO pins exhibits more wear resistance then AQ pins, due to large and many more pores which enable sufficient lubrication. In MO-RS combination type pins, the wear resistance is observed highest, mainly due to MO disc pores hold lubricant, which lubricate contact surface and which prevent adhesive wear. For RS pin in combination of MO-RS combination alongwith the pin AQ in RS-AQ, the observed wear is adhesive type and scuffing. In the PM-PM, or RS –PM, or AQ-RS AND RS –RS pins combination, the scuffing type adhesive wear is observed as damage mechanism.

Verma et al [4], brake system tribology phenomena are not only important and interesting for its own system , but for other respect also. Over the years, the issue gaining importance is wear debris particle impact on environment. Related to this, current study is tribological behavior of friction pad of commercial nature having material in dry sliding condition against disc of cast iron . The test of pin on disc was performed at room temperature under normal wear condition at concerned load and of rotating speed. In pad material some component like copper is presented when dynamic formation of tribology wear and layer is occurring there. The final result is related to a simple system which shows important indications in view of new break pad material development and also for system of braking nature.

Zmitrowicz [5] it presented that wear is a gradual removal type process of a material from solid surface which in contact and sliding. Contact surfaces damage are result of nothing but wear .Wear can have many patterns like ,abrasions ,corrugation ,cavitations ,erosion ,ploughing and fatigue. The abrasive wear results in irreversible changes in contours of body and also new gaps between contacting bodies. The removed material's useful measures are wear depth of profile surface. Deformation of bodies and wear profiles evolution are taken into account for the definition of the gap between contacting surfaces with the help of wear laws, depth of wear can be calculated. In this study, for anisotropic wear, constitutive equations which are extension of Archard's law was derived. Abrasion of materials with the help of microstructure has been described by this equation. The given example demonstrates mathematical calculation of the abraded amount of mass and temperature in the test rig of pin on disc machine.

Guicciardi et al. [6] Performed sliding tests of wear ten numbers on pin-on-disc. Each experiments were performed with tungsten carbide (WC) which is the material of pin and disc of silicon carbide. The purpose was to determine the wear and friction's data dispersion. The tests were performed using two sliding speed (v), 0.1m/s and 1m/s and two loads of 5N and 50N. The wear observed in dispersion range of 28-47 and 32-56% on disc and pin respectively. In the disc case, the dispersion reduced with increase of sliding speed and load applied. But in case of pin, no clear relationship was observed .The friction value ranged 5-15% with lower value of high load applied and it is independent of sliding velocity. From numeric point of view, it was observed that in all conditions of experiment, wear values was about 20% and friction value obtained was outliers.

Priit Podra, Soren Andersson [7] Conducted, with finite element method, the simulating of sliding wear. Wear is often a critical factor for the component which influences the product service life .Wear prediction is an important portion of engineering. In this paper, ANSYS, which is finite element software, has been used for wear simulation. A modeling with simulation is presented and adopted with linear law of wear and integration scheme of Euler .Extra care must be adopted to ensure validity of model and convergence of numerical solution wear mechanism was identified ,both experimentally and finite element method and Lim and Ashby wear map on spherical pin-on-disc steel contact in no lubricated condition. It was found that wear simulation result with FEA of a given shape and loading pattern can be considered on the base of wear coefficient 2'sliding distance change equivalence. The FEA software ANSYS is very much suitable for solving the contact problem and simulation of wear .The actual dispersion of the wear coefficient lies within ±40/60 % leading to effective deviation of result of wear simulation. All these result should therefore be analyzed on a relative scale for comparing different designs alternatives.

P. Prabhu [8] FEA normally neglects the wear contribution and surfaces changes due to the wear. However wear may be considered important part in any component subjected to repetitive nature load and may be very crucial for tribological applications ,considering sealing potential of surface prediction. In this study, a procedure has been proposed in which wear effect can be mathematically found and included in total analysis of structure. The Archard's equation has been taken as a base for calculating strain of wear , which can be used to change the strain of elasticity in structure in an explicit way. Theory extensions are also proposed and an example of wear adjustments is included using explicit creep.

Imam Syafa et al [9] Conducted experiment in which deformation takes place at the level of asperity for the two surfaces which are brought in contact. The important factors for wear are local pressure distribution and contacting surface deformation .This paper suggest a wear modal which can predict the rough sliding contact's wear .This wear model proposed is based on Archard's wear equation with the help of finite element analysis (FEA). In this paper, roughness is considered as spherical asperities of uniformly distributed nature. In this way, this proposed model which is combination of Archard's wear law in combination with FEA has proven to be powerful tool in finding wear of rough nature surface.

S. C. Lim [10] Formulated the wear map model. In this paper, author has presented his personal view about different type of wear mechanism, maps and also some recent maps which have been proposed. These maps present data of wear in a graphical way are effectively able to provide a more universal picture of material behavior in relative motion under different condition of sliding conditions. They also show the relation between various type of wear mechanism that occur under different condition of sliding and also the anticipated wear rates. Some future thought of research in the field are also presented.

Yang [11] Performed wear test on pin-on-disc type of tungsten carbide material inserts infront of hot worked tool steel disc .The experimental datas selected in the study are:

a) loads of 40 and 50 kgf.
b) Sliding speed of 100 and 130 m/min.
c) Temperatures selected of 25°c, 200°c,400°c and 600°c
d) Sliding distance of 1000 to 16000 m

it is mentioned that the load and sliding speed chosen are near to those values which were previously used in the experiment of turning. This is to perform the comparative study of the values of wear coefficient obtained in this experiment with previous study. In this experiment, two different types of pin settings were adopted. In first, the insert was kept in fully contact with disc during whole of the testing period. In the further second setting insert was kept at angle with the disc in the initial stage of wear test but at the end of the test, fully contact will be established. Two types of volume of wear loss with respect to distance values were obtained in this experiment, VA, VF for insert with initial angle setting with disc and full contact of inserts with disc respectively. In this work, the new insight has been given regarding the error of the coefficient of wear values which were obtained by earlier turning method and the same obtained by the standard testing method of pin on disc type.

Yucong Wang et al. [12] In their work, they explained a reproducible and quick bench scuffing and test of wear which has taken actual bore of engine cylinder sections and real piston skirts which ensures contacts to find out the tribological characteristic of contact of sliding nature between skirt of piston and the counter face of cylinder bore . For wear and scuffing simulation test, SATURN 1.91 L4 engine parts were utilized. Two rotating bench test machines were utilized in finding the behavior of tribological nature of coated piston and bore materials. To perform the piston motion against bores of cylinders in engine. Specimen were cut out the piston and cylinder liners such that skirt of piston match with the cylinder wall. Cylinder bore specimen was kept stationary in the wear and scuff type bench type tests. The test parameters which were used to simulate normal wear are 20 hr , Temperature 125 ° c, Stroke 6·77 mm, Frequency 10 Hz,and Load 120 N, Lubricant SAE 5W30 and quantity of lubricant -8ml (for wear test only). A software system kept track of value of the friction coefficient, the contact potential and variation in parameters like temperature.

R. Novak and T. Polcar [13] In detail, analyze the coefficient of friction uncertainty measured by the pin-on-disc apparatus and the respect coating rate of wear .Then they apply method to do large substrates ,one coated with titanium nitride (TiN),and the second type with hydrogenated diamond like carbon coating also called (DLC).They determined the most important contributors to the overall uncertainty measurement ,which can help to redesign the procedure of experiment to decrease the uncertainty measurement or to make it easy by ignoring some parameters . They concluded that calculations of uncertainties can help to differentiate between random variation of value and actual trends (means relation of measured value on those selected variables or man set of variables)

Sharma et al. [14] Work was concentrated on pin-on-disc tribometer which is a type of advanced tribometer with accurate measurement of friction and wear parameters of many combination of metals and lubricants under different selected load conditions, temprature and sliding speed. This model runs at very reduced rpm at a fixed sliding speed and constant radius of wear circular track. Data acquired system shows, maximum penetration depth and specific wear rate for both the mild steel (ms) and aluminium.The specific rate of wear is determined ,the specific rate of wear helps in finding the wear resistance exhibited by the metal under operating conditions. This study related to various parameters (friction coefficient, pattern of wear, testing of lubrication and graphs of different results) obtained by tribometer of pin on disc type.

E.M. Bortoleto et al. [15] Analyzed the wear regime of T1 taking the pin on disc system consideration with two different steels in contact of dry type. Analysis was done both by means of numerical simulation (FEM) and experimental test also. Pin-on-disc test performed in dry condition and followed ASTM G-99-05 standard, which clarifies the conditions for sample and its preparation. The experiment was done for 3600 second corresponds to a distance of 360m .The load varies from 5N to 140N with five repetitions for every load conditions . The coefficient of friction values were found by the ratio of friction load to the applied normal load. The variation of mass was calculated from scale measurement with precision of Le-5g.
To develop a general FEM model ,which able to guess wear in some more complex geometries by taking input parameters obtained from wear test which performed furthermore ,this is the first step to investigate in depth the friction and wear phenomenon in actual system for example , piston ring cylinder contact in IC engines.

Johanssona et al. [16] experimental work was related to develop and verify a methodology which is capable of copy and show the real type engine behavior in mixed type and boundary lubrication regimes to reduce the frictional losses and wear. This work is focused on the test rig development method and to analyzed present and future materials.

You Wang et al. [17] Analyzed and evaluated resistance of abrasive wear of coatings of ceramics by using diamond abrasives. They took the help of plasma spray technique to deposition of coatings with nanostructured and reconstituted AL_2O_3/TiO_2 powdery forms. Modified grinding and polishing machines were used for abrasive wear test. In this study abrasion rates or wear rates are calculated by using the average (mean) measurement value of three specimen in terms of the material volume removed per unit load and sliding distance against the abrasive pad of diamond. Both before and after wear,AL_2O_3/TiO_2 coatings were checked by x-ray diffraction , SEM and test of indentation . The wear mechanism of abrasive was also discussed .Titanium carbide, and like other transitions carbide of metal has important properties including less coefficient of friction ,high value of surface hardness ,superior melting point (3067°C) including high thermal and chemical stability

Alexander Sivkov et al. [18] Discussed the possibility of coating deposition TiC/Ti on substrate of copper by using high speed jet of plasma, produced by the co-axial magneto type plasma accelerator (CMPA) .The average value of hardness of the coating deposited was found and it was 1900 HV . The critical adhesion strength value was found to be 5200 MPa . This type of strong adhesion may be explained by mixing of the main elements hydro dynamically in the liquid state while in crystallization process and in depth penetration of coating particle into substrate.

Tom Peat et al. [19] explained in-depth the erosion and corrosion performance result of HVOF deposited type WC-COCr, Cr_3C_2 –NiCr with AL_2O_3 based coating with in slurry of liquid impingement .By the use of liquid impingement arrangement , the result will exhibit a symbol of three coating performance situations reflecting a flowing type environment .The loss of mass ,due to erosion ,corrosion and the factor of synergy have been calculated for every coating so that comparative data can be generated for the relative type performance of every coating material with multiple attack angle. The analysis of metallographic was done to establish a relation between properties of coating and the coating degradation mode by wear scar damage assessment. This part of work seeks to give a unique insight on the erosion and corrosion result of WC-COCr, Cr_3C_2-NiCr and Al_2O_3 based coatings liquid impingement of slurry by relative analysis of three coatings by consistent corrosion-erosion flowing conditions.

Guo Yan Fu et al. [20] Developed a new type of glass coating to prevent the oxidation of the 20 MnSiNb structure steel. The main glass coating used in this study consisted of metal silicates, chromium oxides and binder of sodium silicate. They also calculated the weight changes of 20 MnSiNb samples by anti-oxidation ability and samples were heated in a muffle furnace. The specimens were heated from room temperature to many different temperatures for instance, 800°C, 900°C, 1000°C, 1050°C and 1150°C and it maintained for 60 Min. at last possible mechanism of protection of coating was also explored and investigated .

S. Chaudhary et.al. [21] Studied the high temperature tribological behavior of Ni coated tribopair and compared with the uncoated. The coated tribopair showed less coefficient of friction and wear i.e better tribological performance as compared to the uncoated tribopair.

2.1 Conclusions from Literature Review

Thus, many studies focus on determining parameters which influence the friction coefficient between two surfaces and have tried to replicate the rotating pairs of automobiles by subjecting them to almost similar conditions. Various textured surfaces have been studied and the mechanisms by which they improve the tribological performances have been stated. Similarly, comparisons on the basis of coefficient of friction have been made between hard chrome coated surfaces and uncoated surfaces. However not much research study has been done to compare the results of different coatings on same type of material when subjected to different test conditions. Hence with this work, I have attempted to evaluate the tribological results of mild steel pin of different sizes, with cast iron plate under different test conditions.

High friction takes a toll on the life of the component and hence it is important to devise means to control friction. Research work done over the past years has evidently revealed that surface texturing is an adequate means of improving the tribological properties of material. Various parameters of the textured pattern also determine the extent upto friction can be reduced. Though surface texturing considerably reduces wear, it can also have negative influence on tribological properties under starved lubrication conditions. Hence optimal conditions must be maintained so that desirable effects of texturing can be achieved.

Chapter 3

Machine and Equipments

3.1 High Temperature Rotary Tribometer Ball/Pin on Disc

This high temperature rotary type tribometer ball/pin on disc is a standalone floor type model with integrated controller unit. It provides study of friction and characteristics of wear in case of sliding contact under different type test conditions of dry, lubricated, ambient, only pin heated, only disc heated and both pin and disc heated.

Principle:-

The principle is to press a fixed type cylindrical pin over a rotating disc of hardened type. The one line friction and wear thus found are used to find the material characteristic. Sliding happens between the stationary type pin specimen and a rotating hardened disc. The load, rotational velocity and track diameter of sliding are the input values for the test, while the tangential type frictional force and wear on specimen are continuously captured with electronic sensors and recorded on PC as function of time.

Sensors for measurement of frictional force, wear, temperature and speed are part of testing units

The operation of tester is by software, the parameters for operation are input into the software for operation of testing unit and output from the sensors are processed within the instrumentation controller and displayed on software screen

The System — The system includes testing unit with controller, the instrumentation controller is housed within the frame of the tester and connected to PC through NI 64 pin cable.

The PC is loaded with software WINDUCOM 2010 compatible with windows-7 operating system. The WINDUCOM 2010 software has three distinct screens. The ACQUIRE screen is for inputting the parameters for test, displaying the acquired parameter values in graphical form and store these values for post evaluation of test result. The COMPARE screen is for displaying the result of each parameter of test in graphical form and also to facilitating to compare the result for four test of each parameter at a glance

Component and Assembly— All components and assemblies of tester are housed within a floor standing frame with the loading and rotating assemblies exposed for easy accessibility. The disc specimen Φ 100 x 8 mm thick is fastened to disc holder mounted on spindle, driven by an induction motor, the motor and the belt drive are within the frame and covered with panels on all sides.

The upper specimen is stationary during test, usually cylindrical pin or spherical ball is used as specimens. It is held between Vee-Jaws on specimen holder. The specimen holder is an attachment to the loading lever. The loading lever is a long bar pinned at middle of its length on sliding plate. And the bearing at pivot provides friction free motion about the pivoted axis along vertical and horizontal planes. A steel cable is attached at the other end of loading lever and passed over pulleys to suspend dead weights to apply normal load.

The lower specimen is horizontally mounted cylindrical disc. It is rotated by a motor and its speed controlled by a variable frequency drive. The specimen pin is clamped vertically on specimen holder

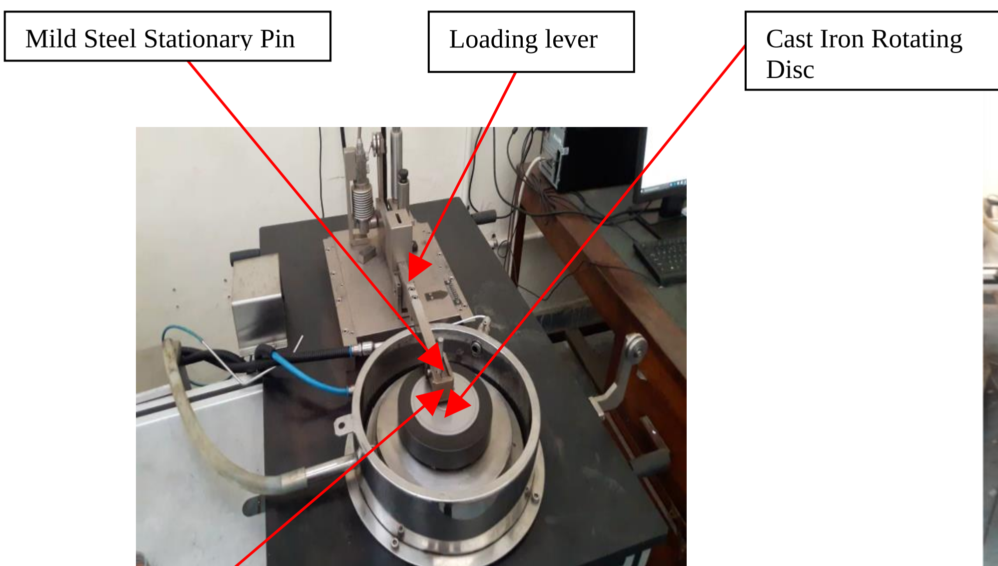

Fig. 3.1: Test Specimen Setup

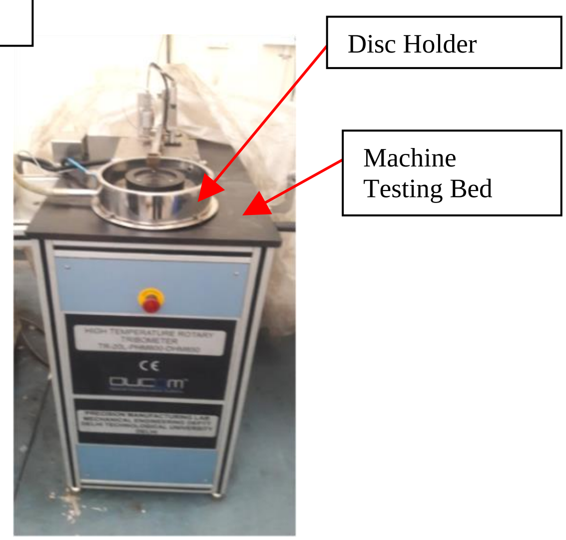

Fig. 3.2: High Temperature Rotary Tribometer Ball/Pin to Disc type

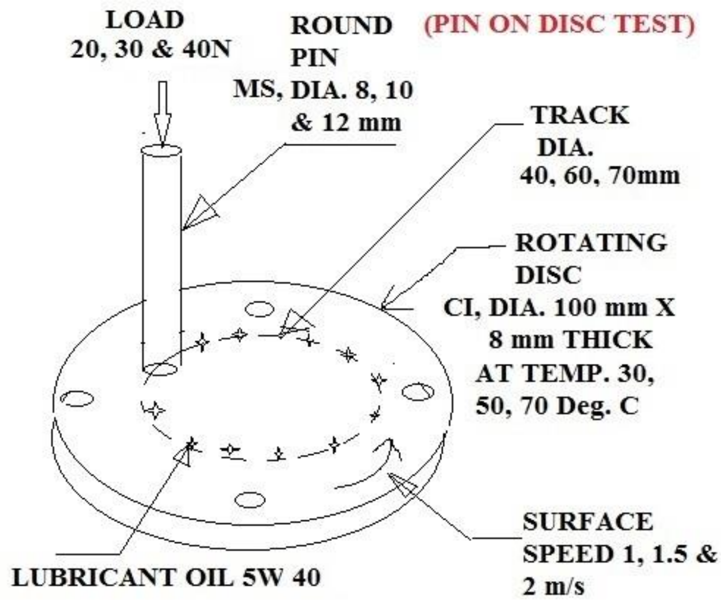

Fig. 3.3: Testing Mechanism

3.1.1 Test Specimens and Specimen Preparation

Materials — The simplicity of this experiment enables an array of combination of materials can be assessed. Tribological pairs must be machined into required dimensions to accommodate into the experimental setup, and tribological materials under test must have higher stress resistance than the stress experienced during the testing, without failure or excessive flexure. Thus, the materials must possess certain physical characteristics like dimensional accuracy, material type, form and processing treatment; and certain mechanical characteristics like surface finish, composition, indentation hardness and microstructure.

Test Specimens—Two specimens are taken under test, i.e. a pin, which is cylindrical; and a plate, in round form with Dia. 100 mm X 8mm Thickness as shown in Fig. 3.4 (a) and 3.4 (b). Pins of the selected round shape have a pin diameter in the range of 8 mm, 10 mm and 12 mm. as shown in fig. 3.4 (a).

Specimen Preparation—The pins are prepared by turning and grinding operation. A mild steel rod machined and turned to a diameter range of 8 mm to 12 mm and the total length of 65 mm on a conventional lathe machine. Turning is followed by parting operation, at the required length of 65 mm, which is performed on the conventional lathe machine. Pins are then sent for grinding operation, to ensure a smooth flat circular surface at both ends of the pins.

A plate of cast iron, having a thickness of 12 mm, are casted and sent for machining using conventional lathe machine followed by grinding operation for smooth flat surface, reducing the thickness to 8 mm. Holes and counter bores are performed as per drawing shown in Fig 3.4 (a)

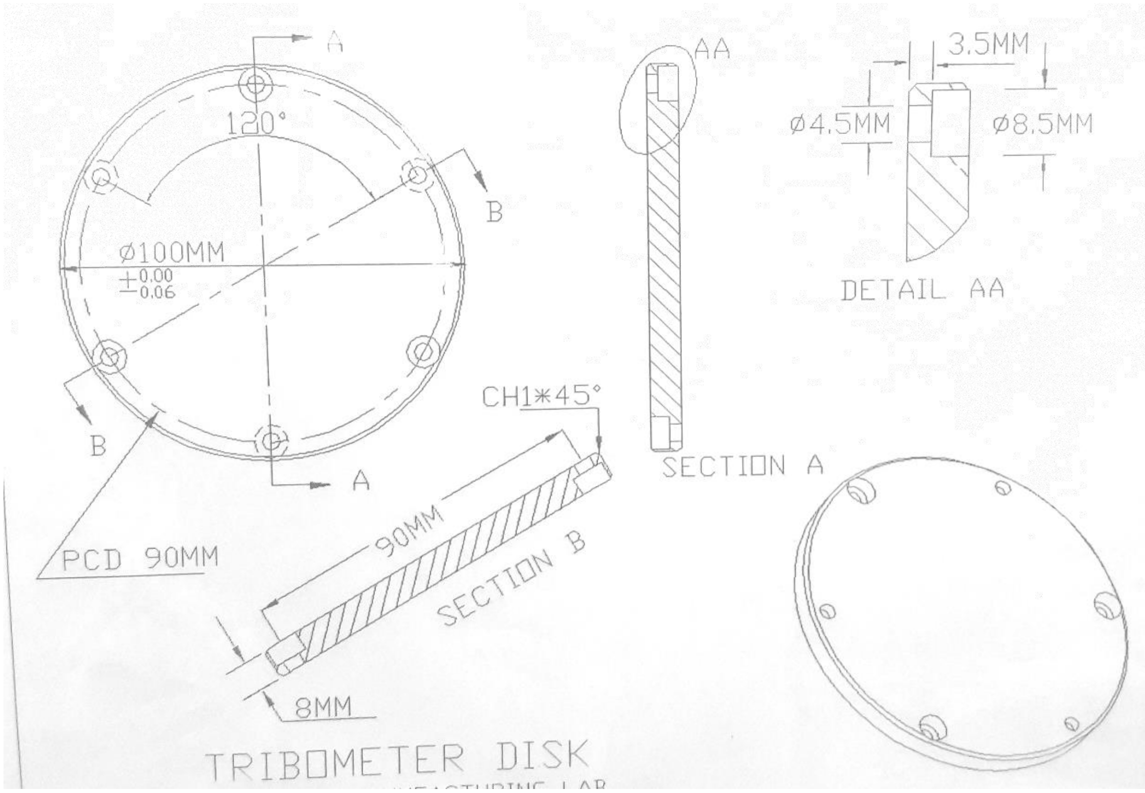

Fig. 3.4 (a)

Plate Dia. 100mm x 8mm Thick

Fig.-3.4 (b)

MS PIN USED FOR TESTING

Fig.-3.4 (c)

Test Specimens (Pins and Disc)

Surface Finish— Smoothness of both tribological pair material surface should be of 0.8 µm (32 µin.) surface finish as an arithmetic average or less, for desire of realistic conditions. A higher value of surface roughness will lead to drastic deviation in wear measurement and difficult to determine. Surface machining and preparation needs to be taken care of, to elude surface layer and sub-surface damage, which affects the physical properties of the surface layers of the material significantly. Some experiments require specific and enhanced surface preparation, for certain parameter evaluation.

Initial and the final weights of all the pins and plates are calculated using an electronic weighing machine, which has a least count of 0.001g. Weight is calculated to determine the total wear of plates and pins, during the experiment, at certain specified values of different parameters. Wear is expressed in the form of specific wear rate, to obtain another decision variable.

As per Table 3.1 there are some common parameter ranges, which can be achieved using a rotating tribometer system:

Table 3.1 High Temperature Rotary Tribometer Specifications

PARAMETERS	ROTARY TRIBOMETER
Speed	300-3000 rpm
Normal Load	20-200 N by dead weights
Frictional Force	0.1 – 200N, Least count 1 N
Wear	0-2000µm, Least Count 1µm
Wear Track Diameter	20-80 mm. Manually set in steps of 2mm
Specification size (Pin and Ball)	Pin Dia. 8, 10 and 12mm, Ball Dia. 4, 6 and 10mm
Wear Disc Size	Dia. 100 mm x8mm Thick, EN-31 Hardened to 62 HRc. Ground to surface and Roughness 1.6 Ra
Pin Heating (Temperature)	Ambient-800 Deg. C (Heating by coil Heater)
Disc Heating (Temperature)	Ambient-850 Deg. C (Heating by induction heater)

3.2 Electronic Weighing Balance

Fig 3.5: Digital Weighing Scale

Electronic weighing scales are widely used in industries, owing to their high accuracy and precision. Digital weighing scales make use of strain gauge load cell to measure weights. Digital type scales change the force of the weights into an electrical signal. The digital scales have a flat tray upon which the specimen is kept. Beneath the tray are four raised pegs which are used to distribute the weight force equally. When the specimen is put on the tray, the design of the digital scale applies force of the weight on one end of the load cell. As a result, the end bends downwards. The strain gauge measures this deformation and since load cell carries electric charge as it moves in the downward direction, there is a change in resistance and thus the deformation gets converted into electrical signal. This signal is sent to an analog into digital converter and again by a microchip which translates the obtained data. As a result of these calculations, the number indicating specimen weight is displayed on the LCD screen of the digital scale. A glass pane is used to cover the weighing scale to avoid error in the output due to air.

Chapter 4

Results and Discussion

4.1 Test Input Parameters

Trobological test of 3 Nos. were performed with mild steel pin of Dia 8mm, 10mm and 12mm. The Tribological Disc was of cast iron. The combination of materials so chosen was due to fact that in many applications of machine tool and automobile assemblies, cast iron and mild steel components relative motion are quite prevalent like Work head, spindle pair, Pulley, Shaft combination flywheel crankshaft combinations etc. All the input parameters are tabulated in Table 4.1 as shown below.

Table 4.1: All Three Test Input Parameters

Pin Dia. (mm)	Wear Track Dia. (mm)	Surface speed (m/s)	Load (N)	Temperature (deg.C)	Distance (km)	Time (Min.)	RPM	Lub. oil	Pin Mtrl	Disc Mtrl
8	40	1	20	Ambient (30)	1	17	478	5W40	Mild steel	Cast iron
10	60	1.5	30	40	1	11	478	5W40		
12	70	2	40	50	1	8	546	5W40		

4.2 Test Output Parameters

4.2.1 For Dia. 8mm Pin

Table 4.2: Test Input Parameters for Dia. 8mm Pin

Pin Dia. (mm)	Wear Track Dia. (mm)	Surface speed (m/s)	Load (N)	Temperature (deg.C)	Distance (km)	Time (Min.)	RPM	Lub. oil	Pin Mtrl	Disc Mtrl
8	40	1	20	Ambient (30)	1	17	478	5W40	Mild steel	Cast iron

4.2.1.1 Wear rate result-

The results of wear rate with respect to sliding distance has been shown in the below Fig.-4.1 and the test input detail data have been depicted in Table 4.2.

From the below graph it was observed that initially the wear rate reach up to maximum value 7 x10^{-6} mm^3/N-m which is due to initial peaks and valleys available on tribo pair surfaces.

The value of wear rate then decreases to and maintains at average stable value of 2x10^{-6} mm^3/N-m after having attaining constant relative velocity.

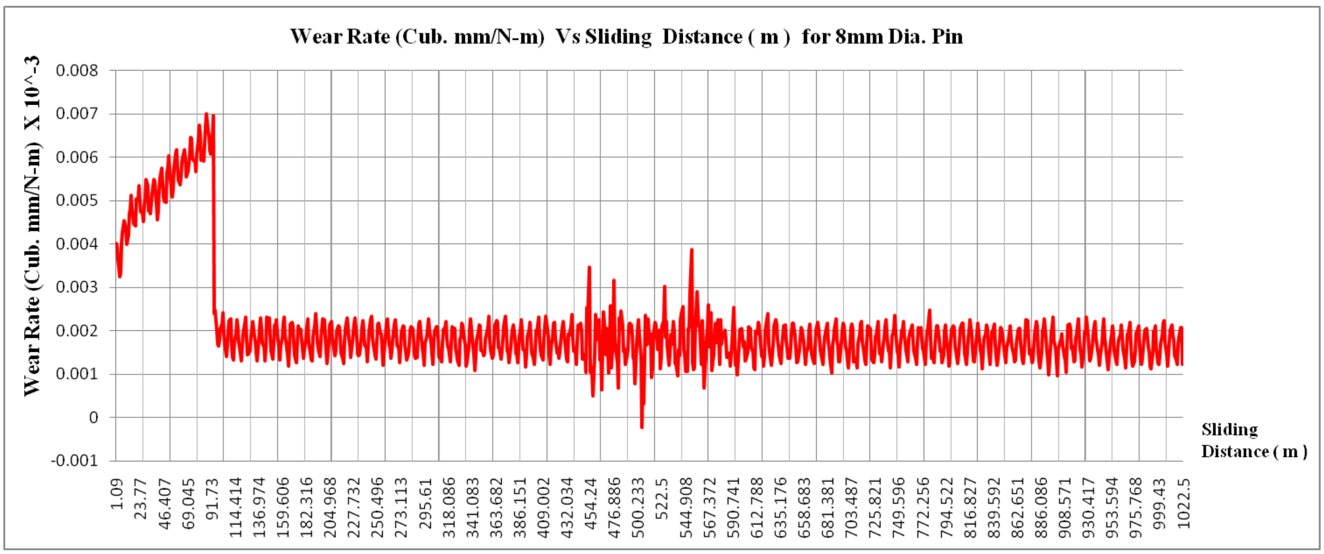

Fig.- 4.1 Wear Rate Vs Sliding Distance Graph (8mm Dia Pin)

4.2.1.2 Frictional force result-

The results of frictional force with respect to sliding distance has been shown in the below Fig.-4.2 and the test input detail data have been depicted in Table 4.2.

From the below graph it was observed that initially the frictional force reach its minimum value 7 N and the value of friction force then increases to its maximum value of 20N and maintains its average stable value of 20N after having attaining constant relative velocity.

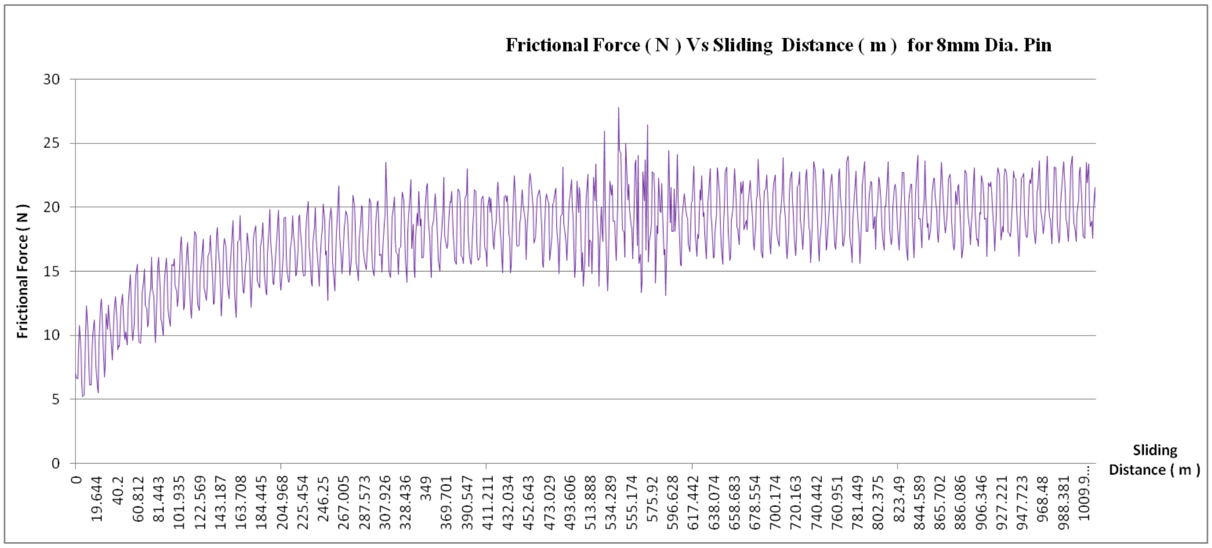

Fig.- 4.2 Frictional Force Vs Sliding Distance Graph (8mm Dia Pin)

4.2.1.3 Coefficient of friction result-

The results of friction coefficient with respect to distance of sliding has been shown in the above Fig.-4.3 and the test input detail data have been depicted in Table 4.2.

From the below graph, it was observed that initially the friction coefficient reaches upto maximum value 0.14 which is due to initial peaks and valleys available on tribo pair surfaces. The value of coefficient of friction then decreases to 0.04 and maintains at average stable value of 0.04 after having attaining constant relative velocity.

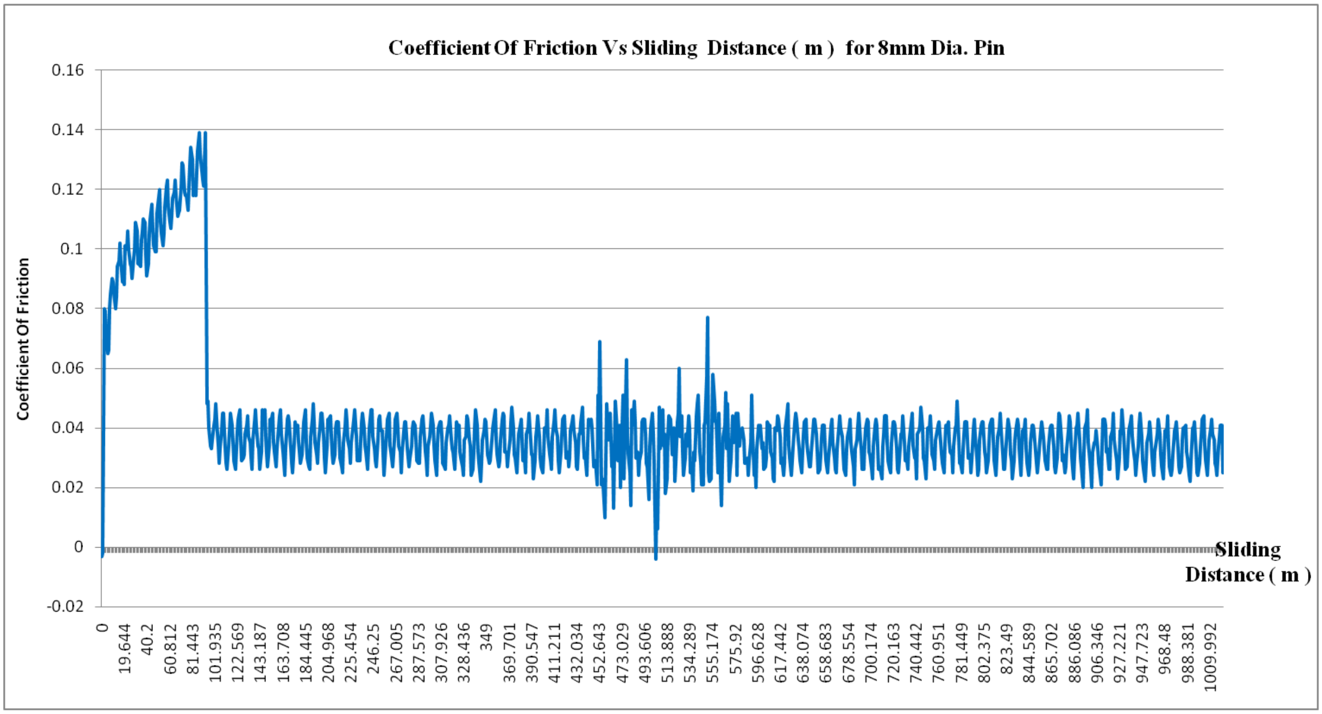

Fig.- 4.3 Coefficient of friction Vs Sliding Distance Graph (8mm Dia Pin)

4.2.2 For Dia. 10mm Pin

4.2.2.1

Table 4.3: Test Input Parameters for Dia. 10mm Pin

Pin Dia. (mm)	Wear Track Dia. (mm)	Surface speed (m/s)	Load (N)	Temperature (deg.C)	Distance (km)	Time (Min.)	RPM	Lub. oil	Pin Mtrl	Disc Mtrl
10	60	1.5	30	40	1	11	478	5W40	Mild steel	Cast iron

4.2.2.2 Wear rate result-

The results of wear rate with respect to sliding distance has been shown in the below Fig.-4.4 and the test input detail data have been depicted in Table 4.3.

From the below graph it was observed that initially the wear rate reach upto maximum value 6.5 x10^{-6} mm^3/N-m which is due to initial peaks and valleys available on tribo pair surfaces.

The value of wear rate then decreases to and maintains at average stable value of 5x10^{-6} mm^3/N-m after having attaining constant relative velocity.

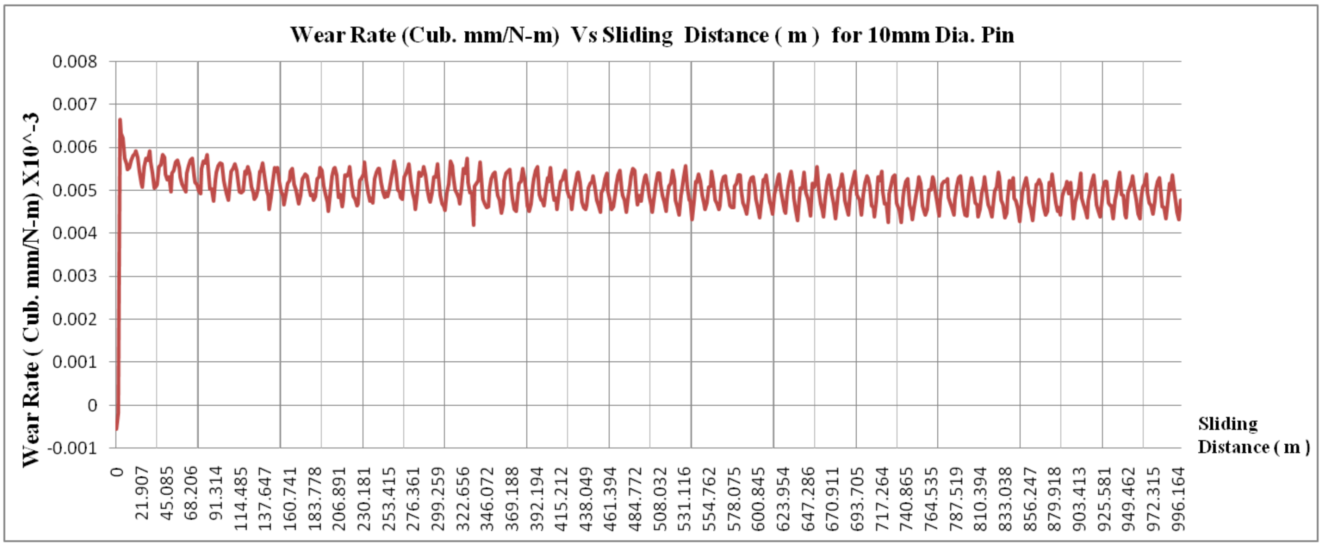

Fig.- 4.4 Wear Rate Vs Sliding Distance Graph (10mm Dia Pin)

4.2.2.3 Frictional force result-

The results of frictional force with respect to sliding distance has been shown in the below Fig.-4.5 and the test input detail data have been depicted in Table 4.3.

From the below graph it was observed that initially the frictional force reach its minimum value 10 N and the value of friction force then increases to its maximum value of 40 N and maintains its average stable value of 50N after having attaining constant relative velocity.

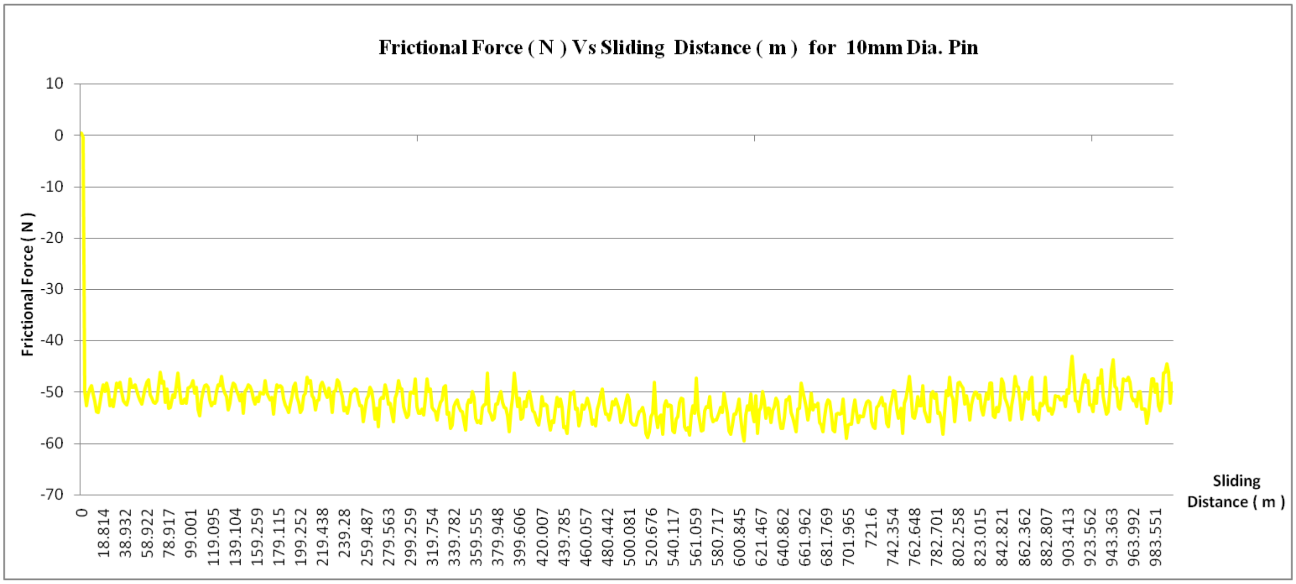

Fig.- 4.5 Frictional Force Vs Sliding Distance Graph (10mm Dia Pin)

4.2.2.4 Coefficient of friction result-

The results of coefficient of friction with respect to sliding distance has been shown in the below Fig.-4.6 and the test input detail data have been depicted in Table 4.3.

From the above graph it was observed that initially the coefficient of friction reach upto maximum value 0.09 which is due to initial peaks and valleys available on tribo pair surfaces. The value of coefficient of friction then decreases to 0.05 and maintains at average stable value of 0.06 after having attaining constant relative velocity.

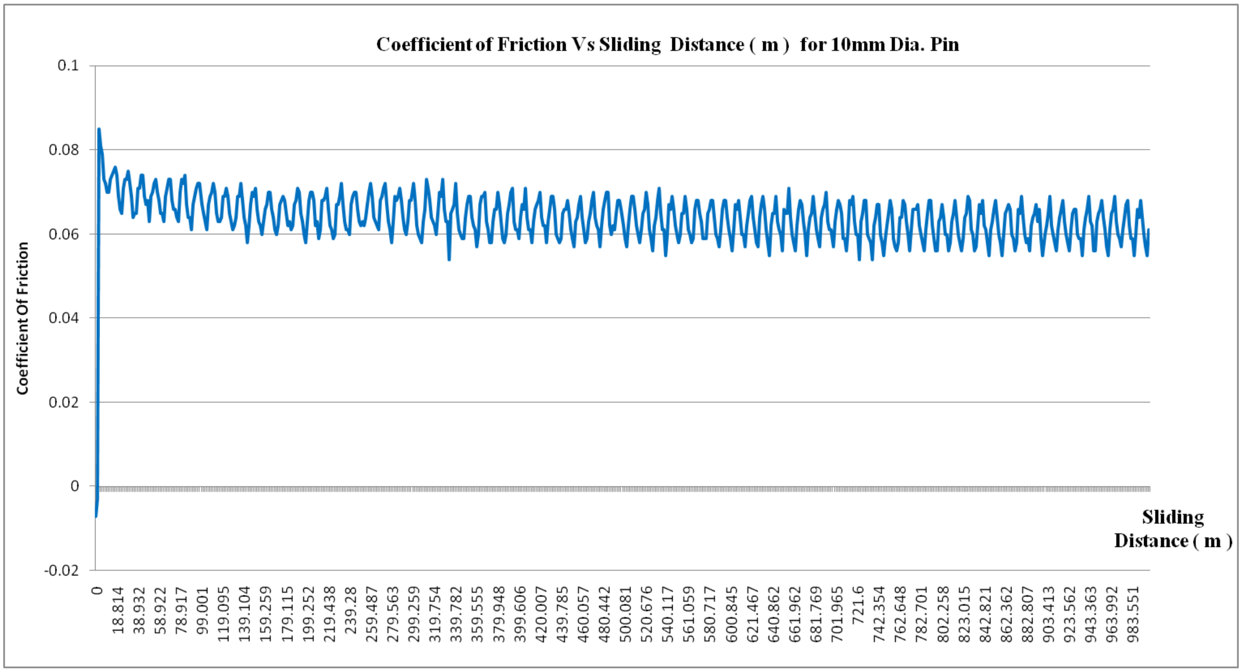

Fig.- 4.6 Coefficient of friction Vs Sliding speed Graph (10mm Dia Pin)

4.2.3 For Dia. 12mm Pin

4.2.3.1 **Table 4.4**: Test Input Parameters for Dia. 12mm Pin

Pin Dia. (mm)	Wear Track Dia. (mm)	Surface speed (m/s)	Load (N)	Temperature (deg.C)	Distance (km)	Time (Min.)	RPM	Lub. oil	Pin Mtrl	Disc Mtrl
12	70	2	40	50	1	8	546	5W40	Mild steel	Cast iron

4.2.3.2 Wear rate result-

The results of wear rate with respect to sliding distance has been shown in the below Fig.- 4.7 and the test input detail data have been depicted in Table 4.4.

From the below graph it was observed that initially the wear rate reach upto maximum value 1.45 x10^{-6} mm^3/N-m which is due to initial peaks and valleys available on tribo pair surfaces.

The value of wear rate then decreases to and maintains at average stable value of 1.2x10^{-6} mm^3/N-m after having attaining constant relative velocity.

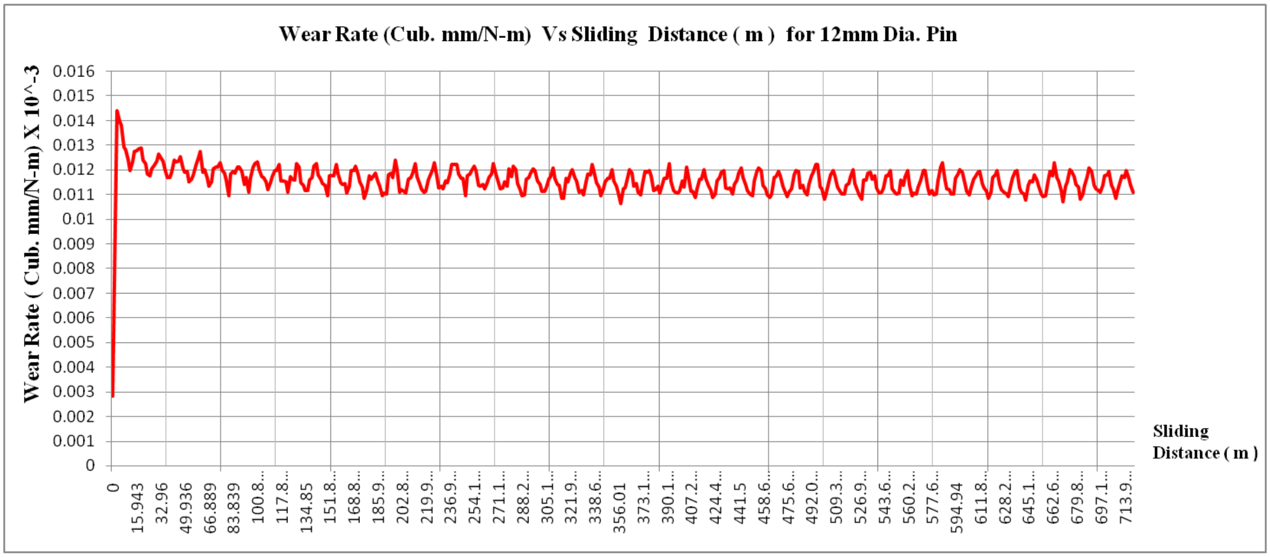

Fig.- 4.7 Wear Rate Vs Sliding Distance Graph (12mm Dia. Pin)

4.2.3.3 Frictional force result-

The results of frictional force with respect to sliding distance has been shown in the below Fig.-4.8 and the test input detail data have been depicted in Table 4.4

From the below graph it was observed that initially the frictional force reach its minimum value 4 N and the value of friction force then increases to its maximum value of 6 N and maintains its average stable value of 7N after having attaining constant relative velocity.

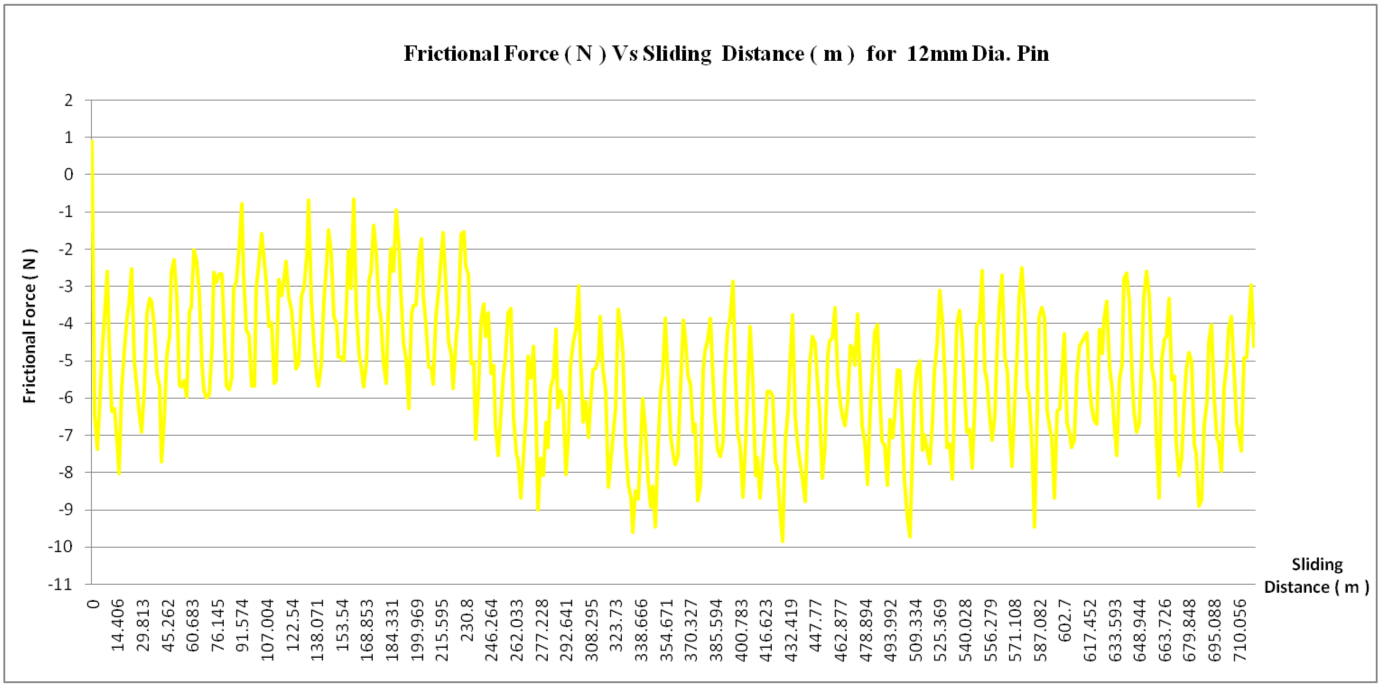

Fig.- 4.8 Frictional force Vs Sliding distance Graph (12mm Dia. Pin)

4.2.3.4 Coefficient of friction result-

The results of coefficient of friction with respect to sliding distance has been shown in the below Fig.-4.9 and the test input detail data have been depicted in Table 4.4.

From the above graph it was observed that initially the coefficient of friction reach upto maximum value 0.12 which is due to initial peaks and valleys available on tribo pair surfaces. The value of coefficient of friction then decreases to 0.1 and maintains at average stable value of 0.115 after having attaining constant relative velocity.

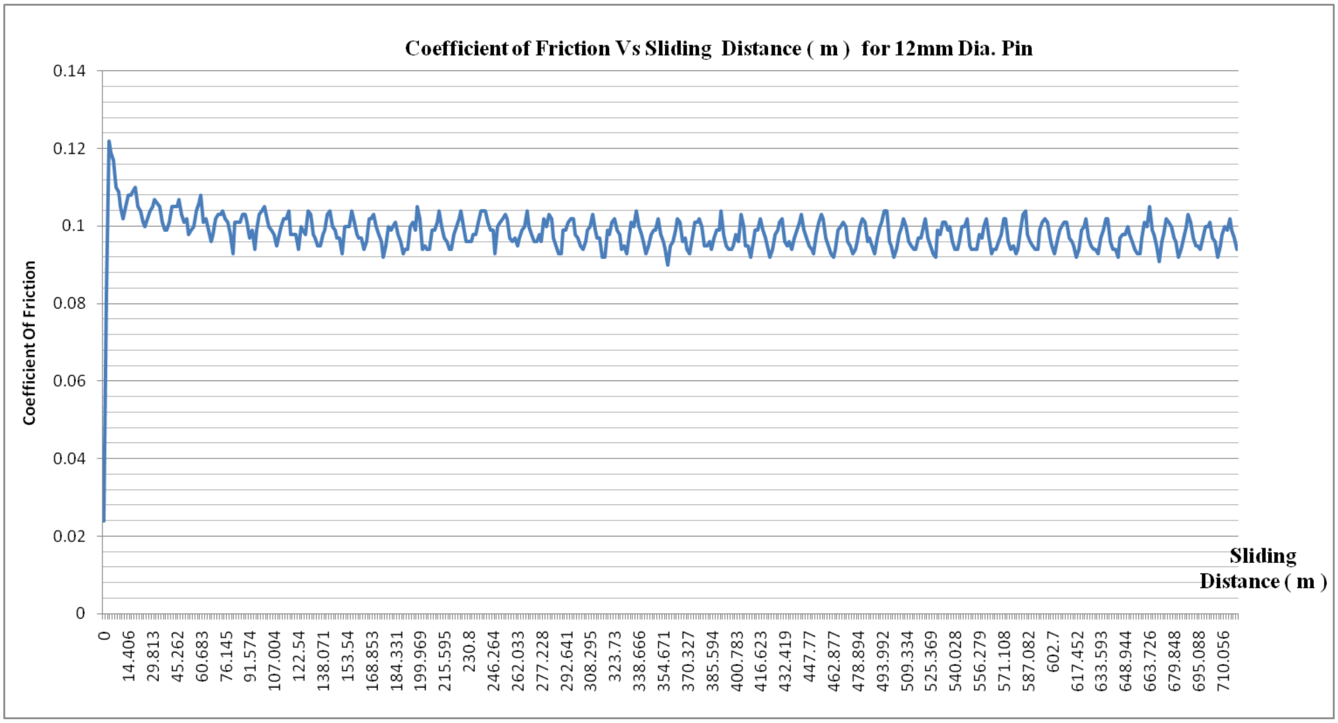

Fig.- 4.9 Coefficient of friction Vs Sliding Distance Graph (12mm Dia. Pin)

4.3 Specimen (Pins) weight result-

The pins of different diameters were weighed before test and also thorough chemical cleaned pins after test were weighed with the help of digital weighing machine of least count 0.0001 gm as well and data as observed has been tabulated below in Table 4.5.

It is clear from the table that average loss of Wight of Mild steel pin as wear is 0.93 milli gram.

Table 4.5: Pins weight before and after wear

Pin Dia (mm)	**Weight before wear (gm)**	**Weight after wear(gm)**	**Weight loss as wear (milli gram)**
8	24.9205	24.9194	1.1
10	39.0881	39.0871	1
12	56.0561	56.0554	0.7

Chapter 5

Conclusions

In this experimental study of tribological characteristics, as a tibo pair, MS pin of dia 8mm, 10mm and 12mm were taken along with cast iron disc with lubrication oil 5W40 at different temperatures of 30°C, 40°C and 70°C with surface speed of 1, 1.5 and 2m/s for sliding distance of 1000 m on track dia of 30, 50 and 70mm with load of 20, 30 and 40N respectively used for finding tribological parameters like wear rate in mm^3/N-m and coefficient of friction.

From results so obtained, it was found that the wear rate initially had high value of $7x10^{-6}$ mm^3/Nm and then decreased to a stable value of $2x10^{-6}$ mm^3/Nm. The value variation may be due to removal of initial peaks and valleys present in micro structure of tribopair and subsequent attaining constant relative speed.

The coefficient of friction initially had value of 0.14 and then decreased to a stable value of 0.04 after having attaining constant relative speed.

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