A Major Project II Report

TRIBOLOGICAL ANALYSIS OF TRIBOPAIRS OF CYLINDER LINER AND PISTON RING MATERIALS WITH AND WITHOUT CHROME PLATING.

A THESIS SUBMITTED IN PARTIAL FULFILLMENT OF THE REQUIREMENT FOR THE AWARD OF THE DEGREE OF

MASTER OF TECHNOLOGY

IN

COMPUTATIONAL DESIGN

BY

ANSHUL KUMAR

ROLL NO. - 2K15/CDN/04

UNDER THE GUIDANCE OF

Dr. RAMESH CHANDRA SINGH AND Dr. RAJIV CHAUDHARY

DELHI TECHNOLOGICAL UNIVERSITY

DEPARTMENT OF MECHANICAL, PRODUCTION & INDUSTRIAL AND AUTOMOBILE ENGINEERING
DELHI TECHNOLOGICAL UNIVERSITY
BAWANA ROAD, DELHI-110042

JUNE 2017

DECLARATION

I, **Anshul Kumar (2K15/CDN/04),** hereby certify that the work presented in this thesis entitled **"Tribological analysis of tribopairs of cylinder liner and piston ring material with and without chrome plating"** is submitted for the partial fulfillment of the requirement to the degree of **Master of Technology (Computational Design)** in Department of Mechanical Engineering at **Delhi Technological University** is an authentic record of my own work carried out under the supervision of **Dr. Ramesh Chandra Singh and Dr. Rajiv Chaudhary.** The matter embodied in this thesis has not been submitted in any other University / Institute for the award of any degree / certificate. Also, it has not been directly copied from any source without giving its proper reference.

Signature of Student

CERTIFICATE

This is to certify that the major project report II entitled, **"Tribological analysis of tribopairs of cylinder liner and piston ring material with and without chrome plating"** submitted by **Anshul Kumar (2K15/CDN/04)** in the partial fulfillment of the requirement for the award of **Master of Technology degree in the Mechanical engineering with specialization in Computational Design at Delhi Technological University, Delhi**, is an authentic work carried out by him under our supervision and guidance.

Further it is to be certify that the matter embodied in this report has not been submitted to any other university/ institute for award of any degree or certificate to the best of our knowledge and belief.

Dr. Rajiv Chaudhary

Associate Professor

Mechanical Engineering Department

Delhi Technological University

Delhi-110042

Dr. Ramesh Chandra Singh

Associate Professor

Mechanical Engineering Department

Delhi Technological University

Delhi-110042

ACKNOWLEDGEMENTS

First and foremost, praises and thanks to the God, the Almighty, for his showers of blessings throughout my research work to complete the research successfully.

I found my pen incompetent to express my thanks to my supervisors **Dr. Ramesh Chandra Singh and Dr. Rajiv Chaudhary, Associate Professor, DTU** under their kind and worthy guidance and supervision; I had the opportunity to carry out this work. It was only due to their advice, thoughtful comments, constructive criticism, and continuous vigil over the progress of my work with a personal interest that it has taken this shape. They have been a great source of encouragement. To get an opportunity to carry out the project work in the well-equipped, ever developing laboratories in our institution, I would like to pay my deep sense of thankfulness to **Prof. R. S. Mishra, HOD, Department of Mechanical Engineering, DTU.**

I would like to express my sincere gratitude and indebtedness to **Prof. Vikas Rastogi, DTU** for prevailing me to do my laboratory work at Design Centre. I am very thankful to him for allowing me to work in his lab and also for providing me guidance. I am highly thankful to **Dr. A.K. Agarwal, Associate Professor, DTU** as he helped me and motivated me towards my work. His suggestion was very thoughtful and impressive and that has inspired me to complete my project.

I am especially thankful to **Mr. Sumit Chaudhary, Ph.D. Scholar, DTU** as I have completed my project under their worthy guidance and supervision. Their advice and thoughtful comments inspired me and were very helpful to complete my project. I am also very thankful to **Mr. Rajesh Bohra and Mr. Rahul Verma**, Tribology Lab assistants, for assisting me throughout my experimentation work.

I am very much thankful to my parents for their moral support and encouragement, which was giving me the strength to chase my goal. Without their support and inspiration, I would not able to complete my degree.

I would especially like to acknowledge my gratitude to all my dear friends for their consistent support, valuable suggestions from time to time to make this project worthy.

With a silent prayer to the Almighty, I take this opportunity to express my gratitude to all those who have supported me in completing my fourth-semester project work as a part of my degree program.

ANSHUL KUMAR
M. Tech. (COMPUTATIONAL DESIGN)
2K15/CDN/04

ABSTRACT

The motor car is one of the most common machines in use today, and it is no exaggeration to state that it is crucial to the economic success of all the developing and developed nations of the world and to the quality of life of their citizens. The motor car itself consists of thousands of component parts, many of which rely on the interaction of their surfaces to function. There are many hundreds of tribological components, from bearings, pistons, transmissions, clutches, gears, to wiper blades, tires, and electrical contacts. The application of tribological principles is essential for the reliability of the motor vehicle, and mass production of the motor car has led to enormous advances in the field of Tribology. For example, many of the developments in lubrication and bearing surface technology have been driven by requirements for increased capacity and durability in the motor industry. It is observed that around 13-17% of total frictional losses occured in I.C. engine and 35-50% of total friction losses is due to piston ring assembly (PRA) system.

This research work focuses on the determining the coefficient of friction and the amount of the wear developed between the cylinder liner and piston ring. A comparison is also done between the effects of chorme plated piston ring and the uncoated piston rings. It is believed that the hard chromium coating proved to be effective in reducing the friction and wear in considerable amount. A Tribotester of pin on disc type is used for determining the friction force and wear amount.

In the End of this research work, it is seen from the results that the application of the chorme coating is having positive results on the service life of the piston ring. The specific wear rate and the coefficient of friction values for chrome coated pin were reduced by considerable amount when compared with the uncoated pin.

List of Figures

List of Tables

Table of Contents

CHAPTER 1: INTRODUCTION

1.1. Tribology

Tribology focuses on the three parameters; friction, wear and lubrication of interacting surfaces in relative motion. It is the new field of science defined in 1967 by the committee of the organization for the economic cooperation and development.

Tribology comprises of two Greek words: 'Tribos' meaning rubbing or sliding and 'logos' meaning science. Tribology is the field of science which applies an operational analysis to problem of great economic significance such as reliability, maintenance and wear of technical equipment ranging from household appliances to spacecraft.

Wear is the major cause of material wastage and loss of mechanical performance. Any reduction in the wear amount can save considerable amount of money and material. Friction is the main cause of wear and energy dissipation. It is estimated that the one third of the world's energy resources in present use is needed to overcome the friction in form or another. Lubrication proved to be the effective means of controlling the friction.

A surface interaction controls the functioning of every device available with surface contact. Every device made by human is subjected to friction and wear which the result of the relative motion between the surfaces. Human body also have many interacting surfaces e.g. knee joints, finger knuckles, hence these parts also subjected to the lubrication and wear.

Tribology effects life of living beings also and that to a greater degree. It is commonly observed that human skin becomes sweaty when it sense fear or stress. It has been discovered that the sweat on the palms and sole of the feet has the ability to increase the friction between palm or feet and the solid surface. This increase in friction is beneficial for humans because in the case of danger this sweat helps to firmly hold the weapon or climb the nearest tree.

The practical objective of tribology is to minimize the disadvantages of solid to solid contact; friction and wear. But in some situations one require friction and wear to occur. There are cases where minimizing friction and maximizing wear or minimizing wear and maximizing friction or maximizing both friction and wear is desirable. For example, reduction of wear but not friction is desirable in brakes and lubricated clutches, reduction of friction but not wear is desirable in pencils, increase in both friction and wear is desirable in erasers.

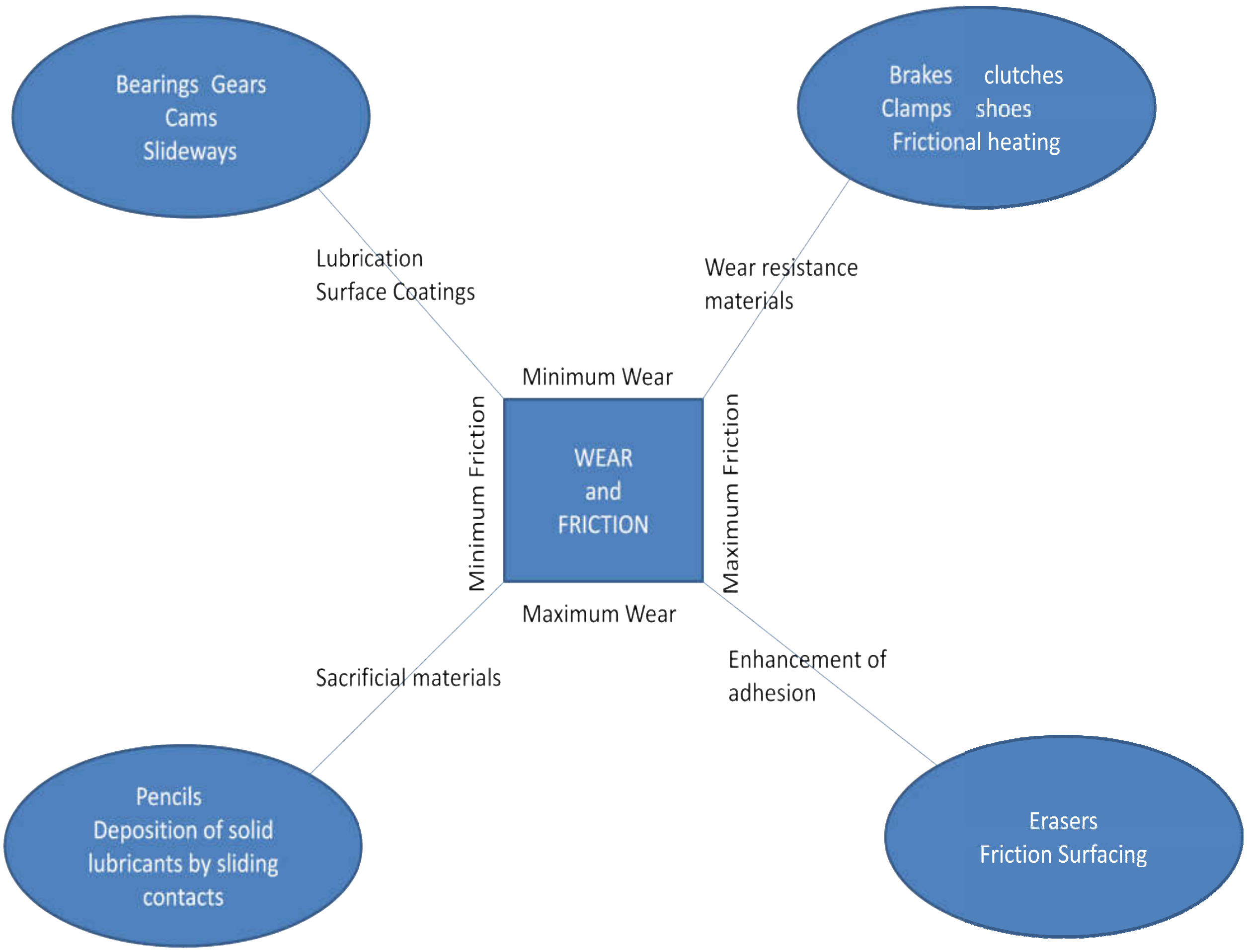


Fig 1: Objectives of Tribology [28]

1.1.1. Lubrication

To improve the smoothness of movement of one surface over another and to prevent damage to the solid surfaces, low shear strength layers of gases, liquids and solids were interposed between the two surfaces. The thickness of these liquid, gases and solid layers varies from 1 to 100 microns. Lubrication means the knowledge related to the enhancing the effectiveness of these films in preventing damage in solid contacts. Gaseous films are more suitable for low contact stress while the solid films are applied to the slow sliding speed contacts.

A lubrication which involved analysis of gaseous or liquid films is known as the hydrodynamic lubrication. Solid lubrication is the lubrication in which solid material is used as a lubricant. There are different types of lubrication used under different conditions. These types are given below:

- Elasto-hydrodynamic lubrication: A special type of lubrication which involves the physical interaction between the contacting surfaces and the liquid lubrication is known as elasto-hydrodynamic lubrication. This type of lubrication is used in gears.
- Boundary and extreme pressure lubricants: This type of lubrication the chemical interactions are present between the contacting bodies and liquid lubricant.

Whenever it is not possible to maintain the layer of film between the surfaces, an external force is used to maintain the thickness of the lubricant layer. Hydrostatic lubrication uses the same principle. In this type of lubrication, a forcible separation of the contacting bodies involves an external energy source which forces the liquid or gaseous lubricant into the space between the contacting surfaces. Liquid lubrication has its own disadvantages. Filters, pumps and cooling systems are to be maintained in good health for their efficient working. There are also environmental issues associated with the disposal of the liquid lubricants.

1.1.2. Wear

Wear occurs between the two contacting surfaces whenever the lubricating film failure happens. This failure lessen the relative movement between the solid bodies and ultimately damages the contacting surfaces. There are different mechanisms through which wear occurs. These mechanisms are as follows:

- Adhesive wear: This wear occurs due to the adhesion of the surfaces. Adhesion is more pronouncing when high contact pressure gets generated between the surfaces.

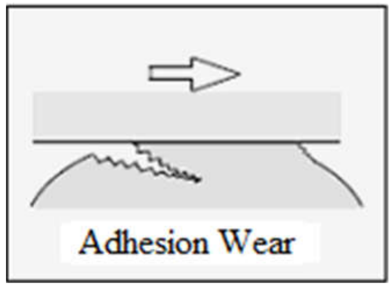


Fig 2: Adhesion Wear

- Fatigue wear: This wear is initiated by the fatigue processes due to the repetitive stresses under either sliding or rolling. Intervening films are effective then milder forms of wear. These milder form of wear are termed as fatigue wear.

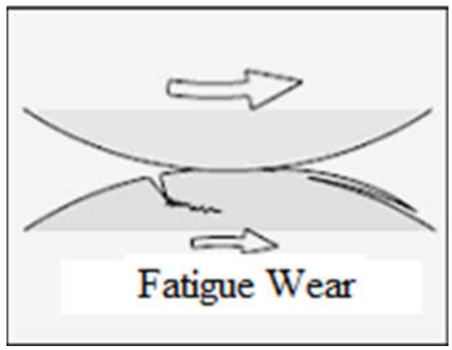


Fig 3: Fatigue Wear

- Abrasive wear: Sometimes the film material contains hard particles which flows over one body and causes wear to occur through ploughing action of the hard particles on the solid surface.

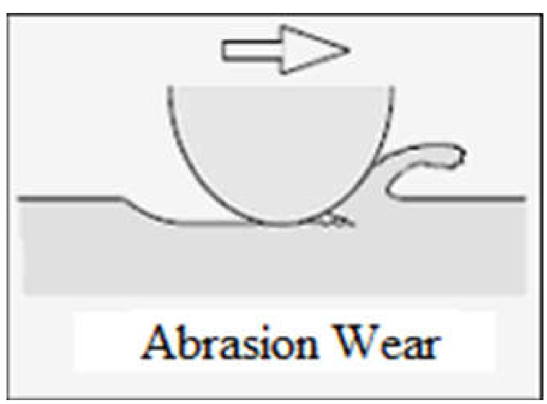


Fig 4: Abrasive Wear

- Erosive wear: In this type of wear, erosion of the solid surface occurs due to the fast movement of the lubricant over the body. This fast flowing lubricant causes impact on the solid surfaces and hence causes wear.
- Corrosive wear: There are some practical situations where solid surfaces are subjected to the chemical attack whenever the lubricant becomes reactive with the surface under favorable environment conditions.

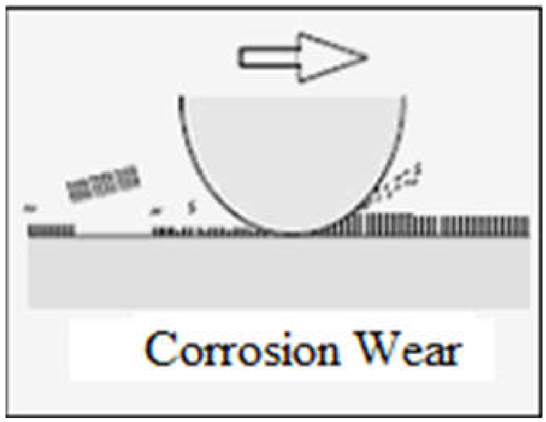


Fig 5: Corrosive Wear

1.2. Tribology: A short history

The historical backdrop of any cutting edge science is regularly very rich. Authentic proof is found through archeological reality and incidental composed record. The historical backdrop of tribology is not distinctive, with tribological accomplishments spreading over the time from the earliest starting point of humankind, to current. The field of tribology is generally more critical. It was created in 1966 by British physicist David Tabor. The training and basic parts of tribology are very old. The root word "tribos" is made an interpretation of from Greek to signify "rubbing", and numerous early imperative tribological perceptions were made by Greek logicians, researchers, and mechanicians. For instance, in 400 B.C., Aristotle mentioned the objective fact in his work Questions Mechanicae that grinding was known as an extremely discernible constrain and least for round items. Table 1 underneath highlights a couple of the chronicled accomplishments in tribology related to this exploration.

Table 1: Historical achievement of Tribology

Tribologist	Achievement	Time Period
Early Mankind	Earliest use of friction in conquest of fire through rubbing, drilling, or percussion.	c. 1,000,000-11,000 years ago
Egyptian Civilization	Transport of 500 ton obelisks on sledges	c. 2400 B.C.
Greek and Romans	Focus on philosophy and science by Greek and roman cultures, use of metal bearings and lubricant.	c. 900 B.C. – A.D. 400
Leonardo da Vinci	Observation of proportionality of friction. Wear studies for development of bearing materials. Invention of rolling element bearings.	c. 1400-1600
Guillaume Amontons Charles Augustin Coulomb Leonhard Euler	Development of Laws of Friction. Coefficient of friction defined.	c. 1600-1750
Heinrich Hertz	Development of elastic body contact theory over Christmas vacation at the age of 23.	1881

1.3. Consequences of friction on IC Engine

In the present era, Automobiles have become an inseparable part of human life. Every human being uses automobiles for comfort travelling, for transportation of goods, for providing emergency services etc. Therefore it becomes the duty of the human beings to maintain the serviceability of the automobiles to a longer period of time.

There are many reasons for the low efficiency of the automobile. Energy loss from different locations causes the low thermal efficiency of the engine. It is observed by Taylor, in fig 2., that around 60% of the energy from the engine is lost in the form of heat to the ambient from the surfaces of the engine. Along with this around 15% of the energy produced is lost in the mechanical actions. After all these losses around 25% of energy is available for the brake power and for transmission and hence ultimately after further losses in the transmission power is delivered to the wheels.

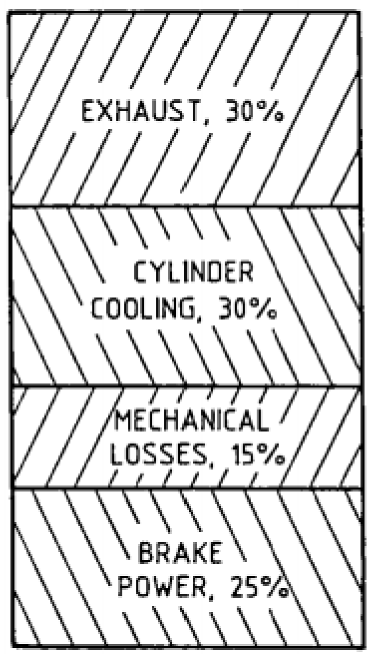


Fig 6: Energy Distribution in the IC engine [28].

Around 80% of the mechanical losses were associated with the frictional components of the engine i.e. valve train, piston assemblies and bearings. There were also some auxiliary losses which account around 20% of the mechanical losses. These auxiliary losses include pumping of the fluids. The oil pump in the limiting engine speeds is to be less than 10% efficient and consume 2 to 3 kW. To reduce these losses in the engine one should have to do film lubrication analysis along with the various tribological aspects. But one should also keep in mind that the reliability and durability of the components must not be degraded in order to reduce these losses.

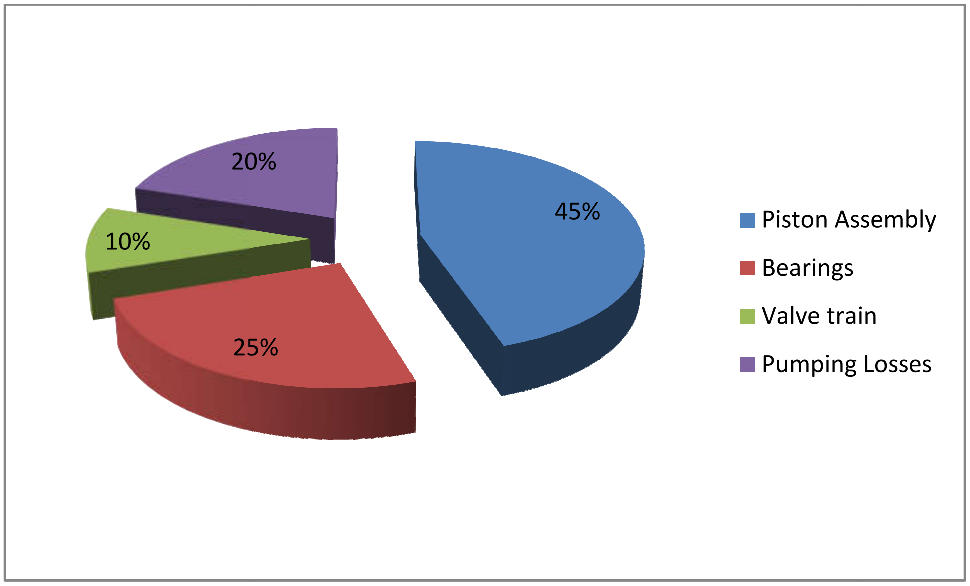


Fig 7: Distribution of Mechanical Losses in IC engine [28]

1.4. Piston Ring and Cylinder Liner

Cylinder frames a versatile limit into a motor barrel. On the off chance that the cylinder is too tight, it doesn't move uninhibitedly inside the chamber when the weight constrain connected amid control stroke and on the off chance that it is too free it would releases the weight compel. For legitimate working of an IC motor, cylinder should shape a middle of the road fit which give a decent fixing fit and less grinding resistance amongst cylinder and chamber divider. These prerequisites satisfied by cylinder rings.

In internal combustion engines, there are total of three rings are used, two compression rings and one oil scrapper rings. The compression rings are used to maintain the complete leak proof sealing between the cylinder liner and the piston. Compression rings prevent the movement of high pressure combustion products from the piston crown side to the crankshaft side. Oil scrapper rings maintain the uniform thickness of the lubricating oil between the cylinder liner and the piston rings. Whenever there is excess or deficiency of the lubricating oil, the oil scrapper ring satisfy the corresponding need of the oil.

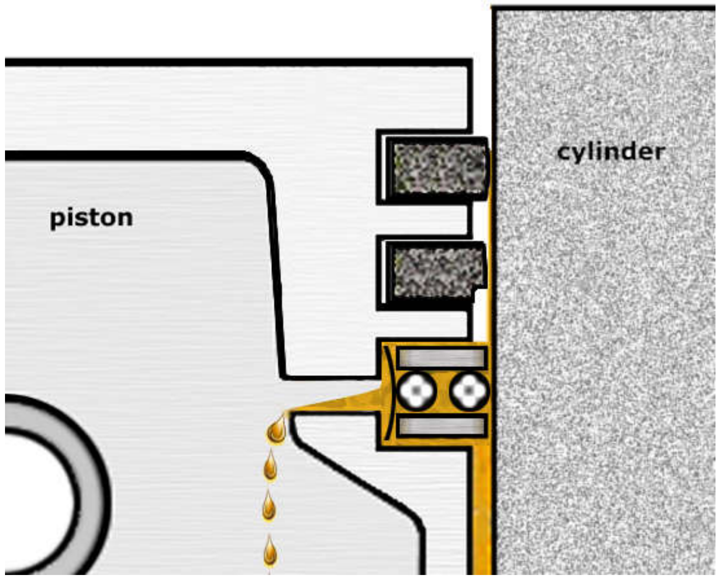


Fig 8: Compression Rings and Oil Scrapper Rings

Cylinder liner is the interior part of the cylinder block. It is just a cylinder with small thickness. It made interference fit with the cylinder block. It is one of the important surfaces of the engine. It is the surface on which the sliding of the piston ring occurs. It also retains the lubricating oil between the cylinder liner and the piston ring. A cylinder liner should have the large anti-galling characteristics, high wear resistance, also should cause less wear to the counter contacting surface i.e. piston ring surface. The liner also receives the heat from the combustion chamber through piston and piston ring and pass on this heat to the coolant. Therefore the material used for the cylinder liner should have the high the thermal conductivity.

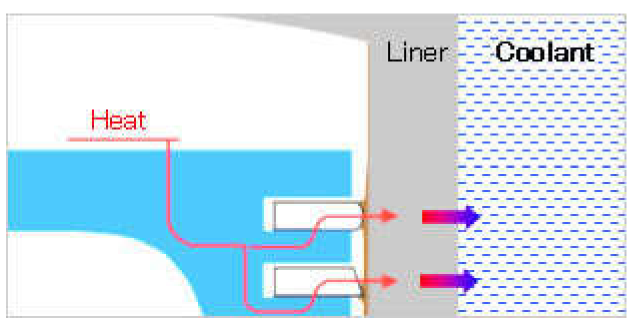


Fig 9: Heat transfer in the piston ring and liner assembly.

The cylinder liner should have the excellent surface finish. It also prevents the high pressure and temperature gas from leaking out to outside. The cylinder liner material should be hard so that it cannot be transformed easily under the influence of high pressure and high temperature.

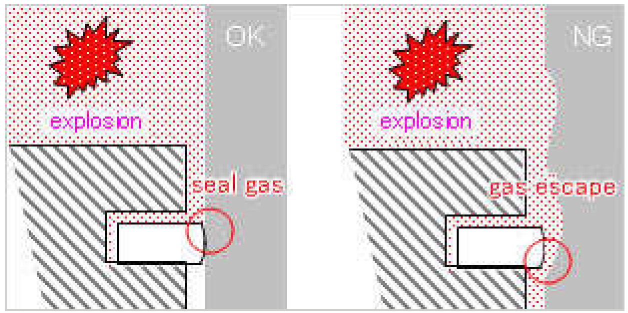


Fig 10: Leak proof Cylinder liner.

Cast iron is the commonly used material for the cylinder liner as it provides good wear resistance properties. Steel is also used for the liner material as it has high stiffness and strength. These two properties are essential for the liner. Wear rate is reduced after using the steel as the liner material. For this research work, mild steel is used as testing material corresponding to the liner.

1.5. Blow By in Internal Combustion engines.

Blow by is the condition in which the high pressure and high temperature gases from the combustion chamber side leaks out on the crank case side. This condition occurs when the gap between the piston ring and the cylinder liner increases. An increase in the gap between these two surfaces is the result of wearing out of the material. This leak out of the gases causes loss of the energy which otherwise can be utilized in the turbocharger. So this makes the blow by problem a serious issue which needs to be resolved while designing the piston ring and cylinder liner assembly. By changing the method of lubrication, materials used for these parts, use of coating or nano particles are proved to be effective ways to solve this problem.

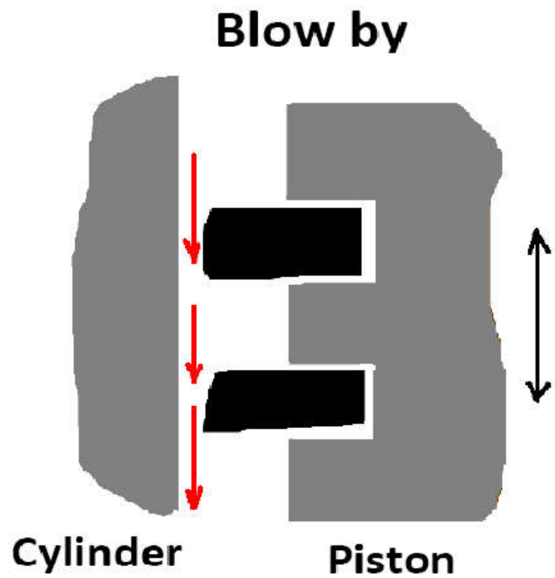


Fig 11: Blow By problem in IC engine.

1.6. Types of Wear Test

There is different variety of wear test available to determine the amount of wear and the friction force between the two surfaces. The test was performed at different wear conditions and test parameters. Wear tests are mainly divided into the following categories:

A. Scratch Test

Solid surfaces of the materials are checked for scratch resistance through a scratch test. The apparatus contains a stylus which slides over the surface of the specimen. The normal load, sliding speed, direction, stylus geometry and stylus material can be varied. Windows based data acquisition software is used to evaluate the tribology data to determine the contact forces at the interface mainly in the tangential direction. A built in camera is used to capture the image of the generated scratch scar. It was investigated scratching an alumina coating, using conical diamond indenters with different tip radii under either progressively increasing or constant loading. The interaction between the disc and the indenter was studied by performing single scratching on a polished virgin surface, repeated scratching over the same track and closely-spaced, multiple parallel interacting scratching.

B. Slurry Abrasion Test

A slurry is a semi liquid mixture of solid particles and a liquid mainly water. This test is used to check the abrasive resistance of the solid material surfaces. The setup of the test contains a rectangular shaped test specimen and a cup filled with abrasive slurry. The amount of material lost from the specimen is measured by weighing the specimen before and after the test. The temperature of the testing environment is maintained by immersing the cup containing slurry in a water bath. Different materials are suitable for this test like polymers, metals, composites, heat treated materials. The necessity of using this test comes when it was seen that slurry erosion problems frequently occurs in the hydroelectric power plants mainly during the rainy season. This occurs due to increased number of particles impinging on the surfaces.

C. Friction Test

The frictional behavior of the materials under dry, starved lubrication or fully lubrication conditions is determined by the friction tester. In this test, the test specimen is rubbed against a ball, pin or a cylinder. The ball or pin is given a reciprocating linear motion. A force transducer is mounted on the specimen to measure the frictional force developed between the interfaces. An unusual velocity profile is generated due to the reciprocating motion of the ball. This profile is used to monitor the static and dynamic friction force over a wide range of sliding speeds. The amount of wear may also be determined by evaluating the scars on the surface by using profilometer. Yucong Wang et al [23]used a plint reciprocating machine to determine the wear and scuffing behavior of the nickel tungsten coated piston against aluminium engine cylinder bore.

D. Air Jet Erosion test

Erosive wear is caused by the impact of particles air jet impinging on a solid surface. Erosion leads to loss of life of components in aerospace, gas turbines, boilers and power plants. In order to maximize life, proper selection of materials used in such applications is required. In this test the erosion resistance of the surface of solid materials are tested by the use of jet of gases. The jet of gases contains hard abrasive particles. The test is performed by passing the jet through the nozzle of specified diameter towards the test metal surface. Metals, polymers, coatings are the range of materials which can be tested. Different parameters of the test like temperature, angle of incidence of the jet, speed of particles, can be varied to replicate the actual conditions.

E. Pin on Disc test

The pin on disc test is the one of the methods used to determine the coefficient of friction and the amount of wear occurred between two rubbing surfaces. This test is versatile in nature when the range of materials used is considered. This test considers mainly two types of motion between the surfaces. First is the rotational motion and second is reciprocating motion. In case of rotating type of pin on disc, the position of the pin is fixed and definite load is applied on it. The disc is made to rotate with the definite angular velocity. A uniform surface contact is maintained between the pin and the disc. In case of reciprocating type of pin on disc, the reciprocating motion is given to the pin on the stationary disc. The sliding distance is one of the parameters which is to be changed according to the different conditions. A wide range of materials can be easily tested on this setup. Materials include metals, ceramics, heat treated samples, composites and polymers. The advantage of this method over other methods is that it measure the lifetime of a particular system. This test can be used for different coatings also. These coatings can be thick or thin and can be of any material.

CHAPTER 2: LITERATURE REVIEW

B. Podgornik, D. Mandrino et al (2016) [1] showed in their research work the influence of the additive type and concentration on the tribological behaviour of CrN coatings when operating under boundary lubrication. Also focused on the effect of the contact conditions, including load, sliding speed and oil temperature. The anti wear additive has the smallest influence and results in complete protection of the CrN coating with respect to the wear.

Friedrich et. al (1997) [2] Studied tribological behaviour of chromium nitride (CrN) deposited by RF magnetron sputtering. Mechanical properties like hardness, residual stress, adhesion and thickness are used for characterization. Sliding kinematics is used to understand the contact mechanics of tribological system at high loading condition at the range of 7 mm thickness. Result shows the moderate residual stress and surface adhesion from the substrate followed by reducing wear rate comparison to uncoated material in tribological application.

Broszeit et. al. (1999) [3] investigated the deposition and application of CrN coating deposited by physical vapour deposition (PVD) and discussed the problems associated with chromium (Cr) electroplating and behaviour of titanium nitride (TiN) which are not up to the mark with various process parameters. CrN is one of the answers of these problems suggested by author. Coating thickness of CrN slightly decreases by increasing the temperature while thickness of TiN increase at elevated temperature. Electroplated chromium (Cr) shows highest deposition rate but deposition rate of TiN and CrN, chromium nitride shows higher deposition rate. It replaces the TiN and electroplating of Cr with their promising corrosion resistance and several applications. Titanium (Ti) alloy against alumina-bronze 630 is reported by M.Y.P.

Costa et. al. (2010) [4] investigated that the high velocity oxygen fuel (HVOF) WC–10%Co–4%Cr thermal sprayed and TiN, CrN and DLC physical vapour deposition (PVD) coatings were applied to increase titanium substrate wear and friction resistance. Quantitative comparison was done at force of 5N and speed of 0.5m/s at various co0ndition.Graphite carbon structures for Ti–6Al–4V alloy DLC coated and aluminium–bronze 630 tribological pair works as a solid lubricant to prevent the wear.

Navinsek et. al. (1997) [5] studied single layer, low and high temperature CrN coatings and double TiN + CrN coatings. CrN PVD coatings deposited at 450°C high temperature and 250°C lower temperature mainly in new field of application. Thickness of 5-10 mm of CrN coating shows good adhesion to substrate more corrosion resistant than traditional coating of TiN which are thermally sTable up to 700°C.Double TiN + CrN coatings were used as a highly abrasive resistant coating in the production of rotors (in the electromotor industry), and in cold forming and forging in mass manufacturing of screws.

Unal et al. (2004) [6] studied and explored the influence of test speed and load values on the friction and wear behavior of pure polytetrafluoroethylene (PTFE), glass fiber reinforced (GFR) and bronze and carbon (C) filled PTFE polymers. Friction and wear experiments were run under ambient conditions in a pin on- disc arrangement. Tests were carried out at sliding speed of 0.32, 0.64, 0.96 and 1.28 m/s and under a nominal load of 5, 10, 20 and 30 N. The results showed that, for pure PTFE and its composites used in this investigated, the friction coefficient decrease with the increase in load. The maximum reductions in wear rate and friction coefficient were obtained by reinforced PTFE+17% glass fiber. The wear rate for pure PTFE was in the order of 10-7 mm2/N, while the wear rate values for PTFE composites were in the order of 10-8 and 10-9 mm2/N. Adding glass fiber, bronze and carbon fillers to PTFE were found effective in reducing the wear rate of the PTFE composite. In addition, for the range of load and speeds used in this investigation, the wear rate showed very little sensitivity to test speed and large sensitivity to the applied load, particularly at high load values.

Ashkan Moosavian et al. (2016) [7] in this study the effects of piston scuffing fault on engine performance and vibrations are investigated in an internal combustion (IC) engine ran under a specific test procedure. Three body abrasive wear mechanism was employed to produce piston scuffing fault it caused the engine performance to reduce significantly. According to Continuous wavelet transform (CWT) analysis wavelet, piston scuffing fault appeared in the scales of 7–14 (frequency band of 2.4–4.7 kHz) and ore at the scale of 9 (frequency of 3.7 kHz)

P.C. Mishra et al. (2014) [8] in this study the piston compression ring tribology and the theoretical and experimental works developed to analyze ring liner contact friction. Because of micro conjunction effect. The friction is comparatively less in case of a rough liner 80 % Power

Loss is in compression and power stroke together of total power loss in an engine cycle. Broad literature survey is carried out in the research area of piston compression ring to know about the simulation and experimental methods developed to study its performance.

A.S Shah et al. (2014) [9] Measure the Piston Ring Assembly friction by the measuring the ‘‘Power consumption’’ under the different operating parameter like speed and lubricant on a motorized multi cylinder engine test rig. Initially the power consumption is reduces till 900 rpm but then it increases with increase in speed of the engine and lubricants properties varies with different manufacture.

B.M Sutaria et al (2013) [10] have used Sewing machine oil as blend oil in the Castrol GTX oil for oil analysis on the four ball tribotester. The coefficient of friction, wear scar diameter and frictional torque these parameters are measured with five different blending ratios two different loads and of oils. Coefficient of friction decreases and Wear scar diameter increases with the increases of the blending ratio of oil.

P.C. Mishra (2013) [11] have used a four stroke four cylinder engine is modeled for lubrication performance. The detailed parameters related to engine friction and lubrication is computed numerically for the 1-3-4-2 engine firing order. To avoid friction and subsequent wear, the liner surface is textured with cross h pattern and the ring is coated with thermal and wear resistant coatings.

S. Ingole et al. (2013) [12] added Titanium dioxide and P25 additives to re-refined base oil and the friction and wear characteristics were examined at a constant applied load and rate of reciprocation using reciprocating pin-on-disk apparatus. From this investigation it was found that the nanoTiO2 particles addition to the base oil slightly reduced the coefficient of friction.

N. Morris et al. (2012) [13] compared an analytic solution to the average flow model is presented for this contact with a new analytical thermal model. Analysis carried out here corresponds to a typical cylinder of a V12 engine with an output power of 510 BHP. The combustion pressure variation piston sliding speed for engine speeds of 2000 and 6000 rpm respectively .These parameters are analyzed and compare between isothermal and thermal condition.

Murat Kapsiz et al. (2011) [14] In this paper's study The Taguchi design method with three process parameters sliding velocity, applied load and oil type was applied to optimize the reciprocating wear test for different commercial oil conditions of cylinder liner (CL)/piston. It was observed that the interactions between the control factors do not have significant influence on the weight loss and friction of the CL and PR pair.

H.R Mistry et al. (2011) [15] in this paper's study paper reports a set of experiments were carried out on developed experimental setup at laboratory scale to measure PRA friction of multi cylinder 800 cc engine system indirectly by measurement of power consumption by Strip Method with variable frequency drive (VFD) is used to vary the engine speed. Frictional power loss contribution by individual piston ring varies under different speed.

P.C. Mishra et al. (2008) [16] studied the compression ring tribology at the vicinity of top dead center in Compression and power stroke transition. The aim is to attain full fluid film lubrication, thus reducing friction because of boundary interactions.

George et al. [2007] [17] predicted and compared their results with results from other semi-empirical models. In early studies, the squeeze film effect was neglected and a simplified hydrodynamic lubrication theory was applied to predict the oil film thickness. The model proposed by the authors considers that the complete ring pack can be reduced to a set of several compression rings and one twin-rail oil control ring. Each rail of the oil control ring is manipulated as a separate single ring. For the simulation of the oil film action between the piston ring and the cylinder liner, the one-dimensional Reynolds equation is used, considering sliding and squeeze ring motion.

C.T Simon (2004) [18] In this paper 's study the current status and future trends in automotive lubricants including discussion of current and anticipated future requirements of automotive engine oils review the current standard ASTM (American Society for Testing and Materials) test methods for engine lubricants and other compilations of automotive standards. Overview of various lubrication aspects of a typical power train system including the engine, transmission, driveline, and other components to improve the efficiency and productivity.

Hugh Spikes et al. (2001) [19] this paper attempts to predict and discuss some of the many challenges facing fundamental research in tribology over the first twelve years based on extrapolation of existing trends, and then, more speculatively considering possible driving forces over subsequent decades. Applications of tribology become more disparate, extending from conventional engineering machines and manufacture to micro electro mechanical systems (MEMS) and hair conditioners, groups working in various areas may lose communication or interest in one another.

L.J Yang [20] performed pin-on-disc wear tests of tungsten carbide inserts against hot-work tool steel disc. The experimental parameters selected in this research work are: (i) applied loads of 40 and 50 kgf; (ii) speeds of 100 and 130 m/min; (iii) temperatures of 25, 200, 400 and 600 C; (iv) distances from 1000 to 16,000 m. It should be noted that the load and speed selected are close to those values used previously in the turning experiment. This is to facilitate the comparison of the wear coefficient values obtained from the previous study with the current one.

Two types of insert (pin) settings were used. With the first type, the insert was set in full contact with the disc throughout the whole testing cycle. In the second type, the insert was set with an initial angle with the disc at the beginning of the wear test, but would have a full contact at the end of the testing cycle. Two types of wear volume loss against distance values were obtained in this investigation, VA and VF, for inserts with an initial angular setting with the counter disc and those with a full contact with the counter disc, respectively.

This work has provided a significant new insight into the discrepancy of the wear coefficient values obtained previously by the turning method and those by the standard pin-on-disc testing method.

R. Novak and T. Polcar [21] analyze in detail the uncertainty of friction coefficient measured by the standard pin-on-disc apparatus and the corresponding coating wear rate. Then we report application of the method to two large set of substrates, one coated by titanium nitride (TiN), the second with hydrogenated diamond like carbon coating (DLC). We determine the most significant contributors to the overall measurement uncertainty, which could help to either re-design the experiment procedure to reduce the measurement uncertainty or to simplify it by neglecting some parameters. We show that estimation of uncertainties could help to distinguish between random value variation and true trends (i.e. dependence of measured values on selected variable or set of variables).

S M. Tului, F. Ruffini et al. [22] work was focused on pin on disc tribometer, which is an advanced tribometer with precise measuring of friction and wear properties of combination of metals and lubricants under selected conditions of load, speed and temperature. The model used runs at very low rpm, at constant sliding velocity and fixed radius of wear circle. Data acquisition includes maximum depth of penetration and specific wear rate for both aluminium and mild steel. The specific wear rate is calculated; the specific wear rate helps in determining wear resistance provided by the metal under running conditions. This research relates to the various aspects (coefficient of friction, wear patter, lubrication testing, result graphs) obtained by pin on disc tribometer.

H. Wang et al. [23] analyzed the Ti wear regime transition considering the pin on disc system and two different steels in dry contact. Analysis was conducted by means of both numerical simulation (FEM) and experimental tests. Pin on disc tests were conducted without lubrication, following ASTM G99-05 standard, which describes the conditions for sample geometry and preparation. The test was carried out for 3600 s, corresponding a distance of 360 m. The normal load applied varied from 5 to140 N, with 5 repetitions for each load condition. The values of friction coefficient were determined by the ratio between friction load and applied normal load. The variation of mass was determined from measurements conducted on a scale with precision of 1e-5 g. To produce a general FEM model, able to predict wear in more complex geometries systems by using input parameters obtained from experimental wear tests. Furthermore, this is a first step to create the background needed to investigate in depth the friction and wear phenomena in real systems, e.g. piston ring-cylinder contact in internal combustion engines.

Alexander Sivkov et al. [24] explained the possibility of TiC/Ti coating deposition on a copper substrate using a high speed plasma jet, generated by the coaxial magneto plasma accelerator (CMPA). The average hardness of the deposited coating was determined and found to be about 1900 HV. The value of the critical adhesion strength was estimated to be 5200 MPa. Such a strong adhesion can be explained by hydrodynamic mixing of the main elements in the liquid state, during crystallization process, and deep penetration of coating particles into the substrate. Titanium carbide, like other transition metal carbides, has useful properties including a low friction coefficient, high surface hardness, high melting point (3067 °C) as well as thermal and chemical stability.

C. Verdon et al. [25] studied the measurement of the wear resistance of WC-Co coatings deposited by the HVOF process, as a function of the deposition conditions.

The HVOF thermal spraying of the powders on the austenitic stainless steel substrate was performed by Sulzer-Metco AG. Two different coatings were produced from the same powder of nominal composition WC88-Co12%wt using two different sets of spraying parameters. The main difference between those was the fuel gas used, i.e. H_2 or C_3H_8 for coating 1 and 2, respectively.

The erosion damages were observed in SEM along coating surface and transverse section. Special attention is paid to correlating the wear resistance to the coating microstructure, the latter being closely related to the spraying conditions.

X. Shi et al. [26] prepared a novel kind of SiC and La-Mo-Si-O-C coating by pack cementation and SAPS, respectively to enhance the oxidation resistance of C/C composites. To alleviate the mismatch of thermal expansion coefficient between outer coating and C/C composites, the SiC internal coating was first prepared by pack cementation for C/C composites. The oxidation test of coating was conducted under 1773 K with static air. The phase composition, microstructures and oxidation resistance property of the La-Mo-Si-O-C coating have been studied.

Y. Zhang at al. [27] in their work selected the ZrB2-SiC-ZrC ceramic as the ablation coating for C/C composites and supersonic atmosphere plasma spraying (SAPS) was employed to prepare this coating due to the high temperature of a plasma arc and the high velocity of particles. Between the C/C substrate and the UHTC coating, a SiC buffer layer was sprayed to alleviate the mismatch. The purpose of this work is to describe the microstructure evolution of the coating as a function of different ablation time. The ablation mechanism of C/C composites with the multilayer coating is also discussed.

CHAPTER 3: EXPERIMENTATION

3.1. MATERIALS

Steel is an alloy of iron having a carbon percentage lower than 2%. By varying the carbon percentage one can obtain the different forms of steel. Steel is mainly divided into four types depending upon the chemical composition: carbon steel, alloy steel, stainless steel and tool steel. Every form of steel has iron and carbon in its composition.

Carbon steel is further divided into

1. Low carbon steel: It is commonly known as the mild steel. It has the carbon percentage varying from 0.08% to 0.30%. The products of low carbon steel are relatively softer when compared to the other carbon steel.
2. Medium carbon steel: This form of carbon steel has the carbon percentage varying from the 0.31% to 0.60% and also has the manganese percentage varying from 0.06 % to 1.65%. The products of this form of carbon steel are having relatively more strength than the low carbon steel. Because of the more carbon content, medium carbon steel becomes difficult to weld, form and even cut. These steel are hardened by using heat treatment techniques.
3. High carbon steel: This form of carbon steel is having maximum percentage of carbon content. The carbon percentage varies from 0.61% to 1.50 %. Because of the high carbon percentage, these steel are very strong and it becomes very difficult to weld, bend or cut this type of steel.

For this research work, Mild steel is selected as it has certain advantages over the other type of steel. Mild steel is readily weldable as the electric current passes through the material without distorting the shape of the material. It is not the case with other carbon steel. Mild steel is ductile in nature. This ductility of the mild steel is due to the low carbon content. The carbon particles act as the hindrance to the movement of the dislocations in the material. As these hindrances are less in mild steel, hence the movement of dislocation is more, making the mild steel ductile. Mild steel is hardened by the carburizing process. In this process the mild steel is heated in the environment rich of carbon. The carbon particles then decompose on the surface of the mild steel. Due to this decomposition, the surface gets hardened while the core remains soft. This process is mainly used to increase the strength and wear resistance property that is hardness of the mild steel. Fatigue strength of the mild steel is also increased after carburizing. The economic advantage of the mild steel over other is that it is readily available in bulk and also it is cheaper than other carbon steel. One can easily procure mild steel from number of sources. Like many other metals, mild steel is also recyclable without having a change in properties and quality. Magnetic property of mild steel helps in separating it from the waste. Recycling is always cheaper and faster process than the mining

Mild steel is selected as the material for the disc having following properties:

- Density = 7890 kg/m^3
- Young's Modulus = 2e11 Pa
- Specific heat = 465 J/Kg-K
- Thermal Conductivity = 53.6 W/m-K
- Poisson's Ratio = 0.3

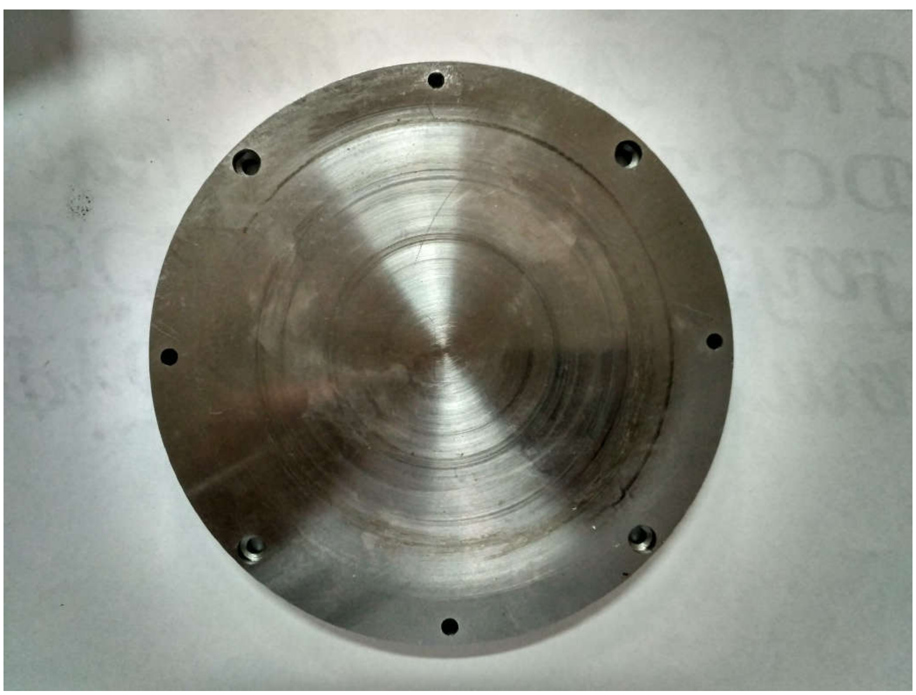


Fig 12: Mild Steel Disc

Cast iron is an alloy of iron and carbon, containing more than 2% carbon. In addition to carbon, cast iron contains other materials like silicon, manganese, phosphorus and sulphur. There is a basic difference between steels and cast iron. Steels usually contain less than 1% carbon while cast iron normally contains 2% to 4% carbon. Composition of cast iron:

- Percentage of carbon = 3.41%
- Percentage of Manganese = 0.63%
- Percentage of Silicon = 2.80%

Fig 13: Chrome Coated Pin

The mechanical properties of the cast iron components are inferior to the parts, which are machined from the rolled steels. However even with this drawback, cast iron offers many advantages:

1. It is available in large quantities and is produced on a mass scale. The tooling required for the casting process is relatively simple and inexpensive. This reduces the cost of cast iron products.
2. Cast iron components can be given any complex shape without involving costly machining operations.
3. Cast iron has a higher compressive strength. The compressive strength of cast iron is three to five times that of steel. This is an advantage in certain applications.

4. Cast iron has an excellent ability to damp vibrations, which makes it an ideal choice for machine tool guides and frames.
5. Cast iron has more resistance to wear even under the conditions of boundary lubrication.
6. The mechanical properties of cast iron parts do not change between room temperature and 350 deg Celsius.

Cast iron has certain drawbacks. It has a poor tensile strength compared to steel. Cast iron parts are section sensitive. Cast iron doesn't show plastic deformation before failure and exhibits no yield point. The failure of the cast iron is sudden and total. Cast iron is brittle and has poor impact resistance. The machinability of cast iron parts is poor compared to the parts made of steel.

In our study, cast iron is used as a pin material. The following are the properties of the cast iron:

- Density = 7078.5 kg/m^3
- Young's Modulus = 1.4e11 Pa
- Specific heat = 460 J/Kg-K
- Thermal Conductivity = 46 W/m-K
- Poisson's Ratio = 0.26

3.2. Chromium Plating

Chromium is bluish-white and lustrous metal that is resistant to corrosion in most atmospheres. Chromium plating, therefore, is extensively used as a final finishing operation. There are two principal classes of chromium plating: "decorative, in which thin coatings serve as a non-tarnishing, durable surface finish; and industrial or "hard" chromium, where heavy coatings are used to take advantage of the special properties of chromium, which include resistance to heat, wear, corrosion, erosion, low coefficient of friction and anti galling.

Chromium is the hardest of the most commonly deposited metals. Hard chrome is used as a wear resistant coating not only on steel but also on a wide variety of other metals. Hard chromium differs from decorative chromium not only because of its use but also because of the difference in deposit thickness. A typical hard chrome deposit is in the range of 12 microns to 250 microns thick.

The electro-deposition of hard chrome is a recognized means of prolonging the life of all types of metal parts subjected to wear, friction, abrasion and corrosion. These parts can be protected when newly manufactured, or they can be salvaged when they are worn and would otherwise be scrapped. As an example, it is much less expensive to reclaim worn hydraulic components by rebuilding tolerances with a hard chrome deposit than it is to buy or fabricate the part from scratch. Hard chromium, because it has a low surface energy, is more often deposited on sliding or revolving parts and is, therefore, used on things like engines, pumps, compressors, hydraulic and pneumatic rods, etc. The hard chromium deposit is also highly resistant to corrosion, and a large number of applications where it is used to protect parts from corrosion are in popular demand. Another attribute of the process is that it is a relatively cold one and can, therefore, be used to impart a very hard surface onto delicate parts without fear of distortion or changing the substrates properties. Thus, it can be regarded as a means of surface hardening

Theory of chrome plating

It has long been known that the electrolysis of pure chromic acid solutions produces only hydrogen evolution under most conditions. In fact, hydrogen evolution at the cathode and oxygen evolution at the anode are two reactions that can be achieved when any conductive liquid has an electric current passed through it. The diagram below shows these reactions for an aqueous solution of sodium chloride (NaCl).

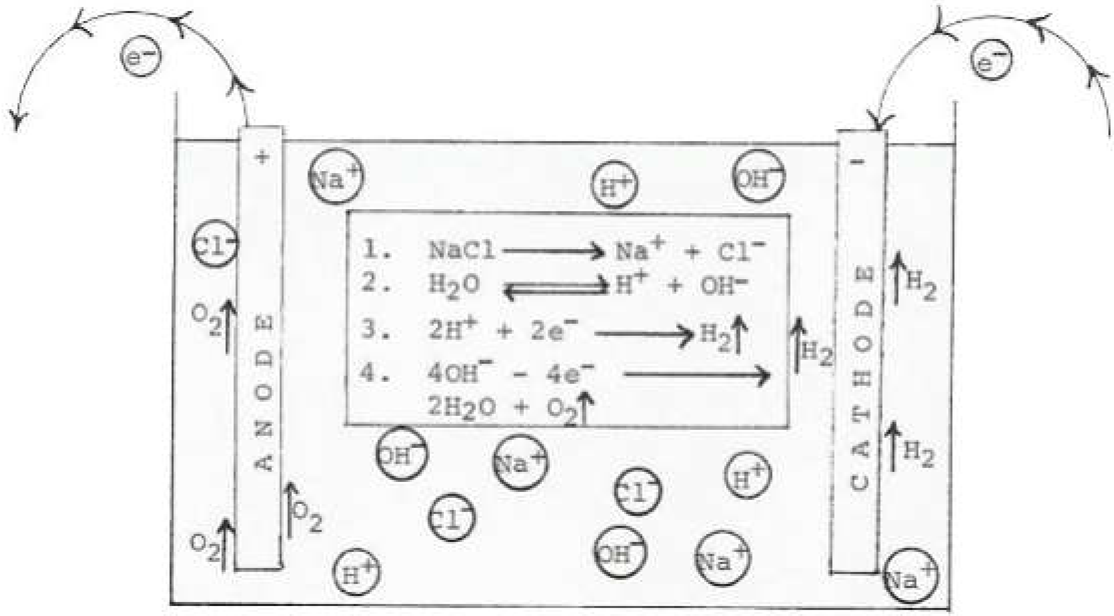


Fig 14: Chemistry involved in Electro-Plating. [30]

The first two reactions show the disassociation of NaCl and H2O. These reactions occur in any aqueous solution. The double arrow indicates the reaction can go in either direction. Reactions 3 and 4, however, can only occur under the influence of an electric current where electrons flow from the cathode to the anode. The third reaction shows how two positive hydrogen ions and two electrons produce the neutral hydrogen molecule that leaves the solution as a gas. The fourth reaction shows how four negatively charged hydroxide ions lose their four electrons to produce two water molecules and the neutral oxygen molecule, which also leaves the solution as a gas.

The chemical reactions in a chrome plating system are not quite that simple. Besides oxygen and hydrogen evolution, a positively charged chromium complex is reduced to chromium metal on the cathode surface; some Cr (VI) is also partially reduced to Cr (III) at the cathode surface. First, it acts as a blocking agent so that the desired chromic-dichromate complex can be formed. Second, it catalyzes the reduction of Cr (II) to Cr metal, which cannot be obtained without the formation of chrome-sulfate complexes. The distribution of these chrome-catalyst complexes (some of which can be reduced to Cr metal and some of which cannot) depends on the concentration of the catalyst. Hoare's mechanism shows through chemical equations that the best distribution of these complexes, (if one's object is to arrive at a metallic chrome plate) is achieved by employing an l00: l chromic acid to sulfate ratio.

A second reaction involves the formation of Cr (III) which cannot be reduced to Cr metal and is, therefore, considered a contaminant in the solution. While it forms continuously at the cathode surface in any chrome plating bath, fortunately, with sufficient anode area, it can be oxidized back to Cr (VI) just as swiftly at the anode surface. Cr (III) also contributes to the resistance of the solution to current flow. In other words, more voltage will have to be supplied by the rectifier to achieve the desired amperage. This contributes to the additional side reaction of excess heat evolution.

$$\underset{\text{(Chromium trioxide)}}{CrO_3} + \underset{\text{(water)}}{H_2O} \longrightarrow \underset{\text{(Chromic acid)}}{H_2CrO_4}$$

$$2CrO_4^{=} + 2H^+ \rightleftharpoons Cr_2O_7^{=} + H_2O$$

$$Cr_2O_7^{=} + 14H^+ + 12e^- \longrightarrow 2Cr + 7H_2O$$

$$2Cr^{+3} + 7H_2O \longrightarrow Cr_2O_7^{=} + 6e^-$$

The chromium lost from the bath by plating on the work piece, or drag-out, is replenished by adding chromic acid. Chromium must be present as the complex salt in order to plate satisfactorily. Metallic chromium does not dissolve to form this salt when used as the anode and, consequently, this is one of the reasons why insoluble lead anodes are used instead. Also, if pure chrome anodes were used the bath would continuously build-up in Cr as the anode would dissolve at 100% efficiency while the deposit only takes place at 15-35% efficiency.

Hard chrome plating is used for this project work. It provides number of advantages which is beneficial in the conditions present inside the internal combustion engine. The hard chrome coating have following advantages:

- Resistance to wear.
- Resistance to corrosion
- Resistance to heat.
- Resistance to erosion.
- Low value of coefficient of friction
- Anti Galling in nature.

In this work, cast iron is used as base metal and hard chrome coating is done over it. Coating thickness used in this work is of 25 microns which well within the prescribed range of 12 microns to 225 microns.

The testing of the hard chrome is done in the laboratory of Anton Paar, gurugram. A micro hardness test is performed over the specimen to measure the hardness of the coated surface. The micro hardness test is chosen for the measurement of hardness because of the drawbacks of the classic hardness measuring techniques. The drawbacks of classical methods are:

- All classical hardness measurement are based on the manual optical measurements.
- There is no automatic calculation of hardness from several indentations.
- At low loads resolution limits of the optical microscope are reached (imprint is too small for precise diagonal measurement)

The instrument has the ability to measure the indenter depth penetration under the applied force throughout the testing cycle. It is capable of measuring both the plastic and elastic deformation of the material under test. The instrument measured not only the hardness but also the elastic modulus of the test specimen.

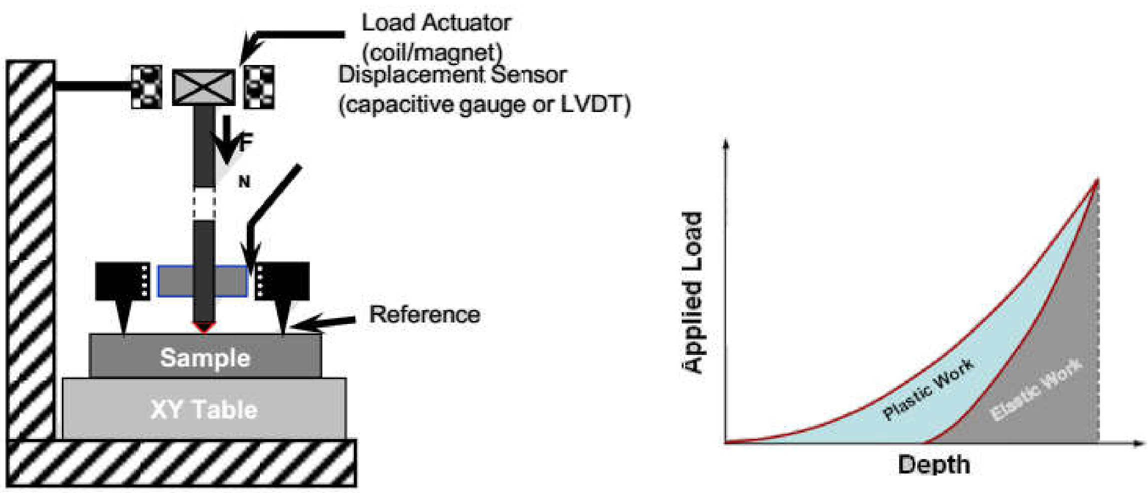


Fig 15: Experimental setup of Micro indenter and typical Graph obtained from micro indenter

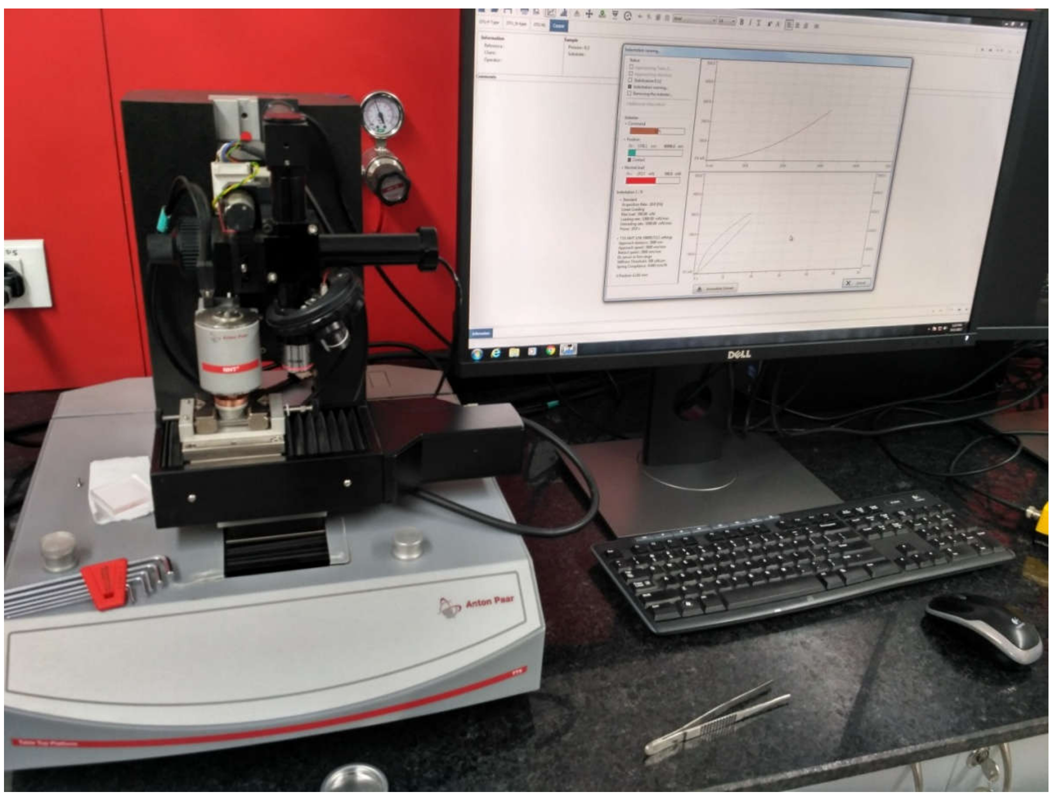


Fig 16: Micro Indenter setup in Anton Paar.

The indentation parameters of test are given below:

- Acquisition Rate: 10.0 Hz
- Linear loading is provided.
- Maximum load applied is 200 mN.
- Loading rate: 400 mN/min
- Unloading rate: 400 mN/min
- Pause: 10 seconds
- Material of indenter : Diamond

The results and curves obtained from the micro hardness test are given below:

- Normal Force vs time vs Penetration depth curve.

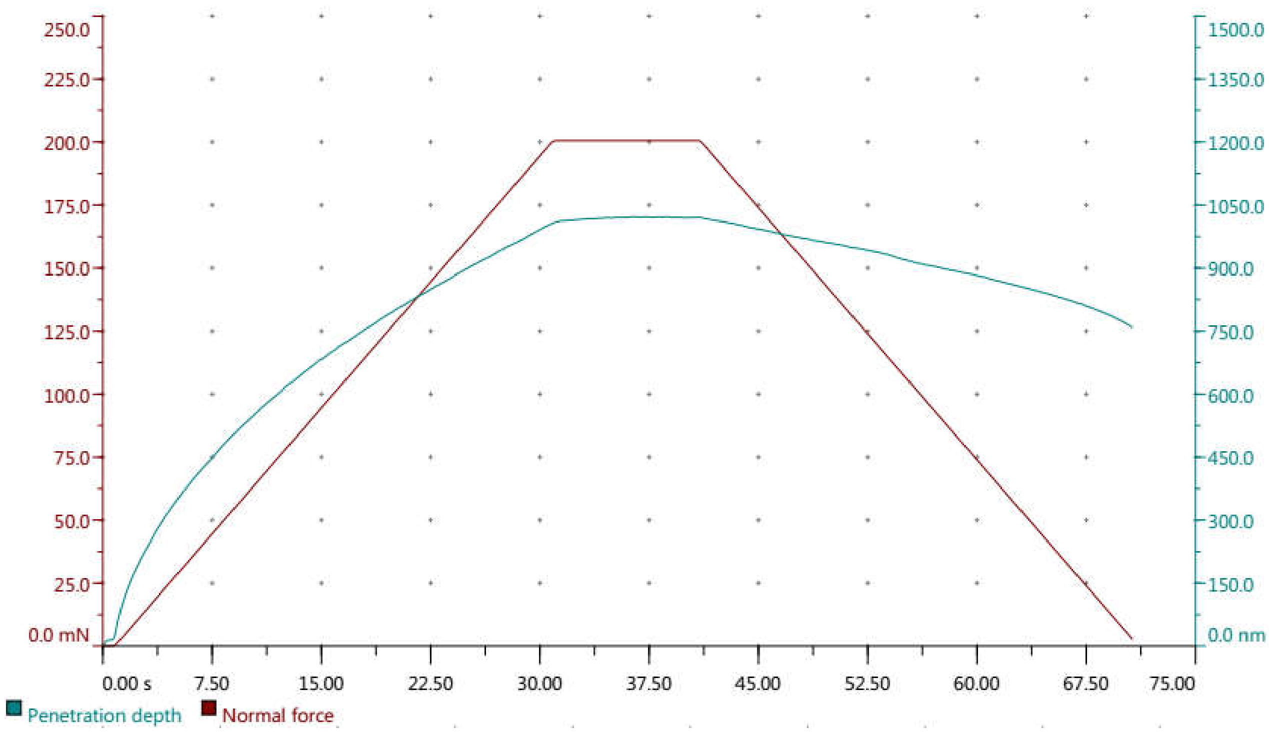


Fig 17: Normal Force vs time vs Penetration depth curve obtained from Indenter.

- Load vs Indentation

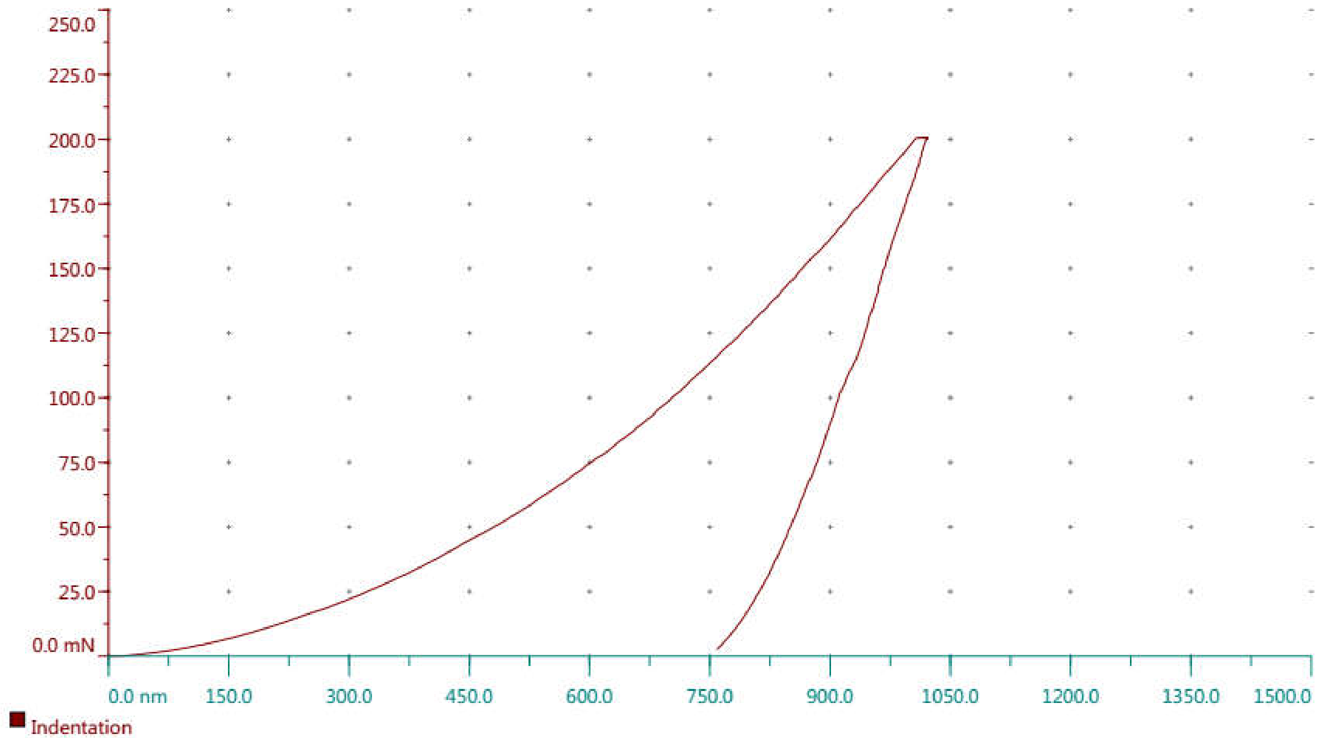


Fig 18: Load vs. Indentation depth curve

Results of the test are as follows:

- Hardness : 10632 MPa or 984.62 Vickers
- Elastic Modulus : 226.62 GPa
- Maximum Force: 200.55 mN
- Maximum Penetration : 1019.45 nm

3.3. Experimentation Considerations

A pin on disk tribotester is used to perform the experiment. At the end of the experiment one can easily determine the amount of wear and the coefficient of friction present between the two rubbing surfaces. Before conducting the test, it is important to deter mine the conditions of the test.

There are three possible lubricating conditions present:

- Fully flooded condition: In this condition, there is a continuous supply of the lubricating oil between the contacting surfaces. This maintains a uniform layer of lubricant between the disk and the pin surfaces.
- Dry condition: In this condition, one does not use any kind of lubricant between the surfaces. There is dry friction present between the contacting surfaces. This condition represents the condition when there is no lubricant is present in the real life application.
- Starved lubrication condition: In this condition, before carrying out the experiment, small amount of lubricant is placed over the disc surface. This creates a thin film of lubricant on the contact region of pin and disk.

For this project work, full flooded lubrication and starved lubrication is chosen.

For the experiment, along with determining the lubricating system, one need to determine the normal load applied on the pin and the sliding speed. In this experiment, three normal loads of 10 N, 30 N and 50 N are used. Also the experiment is conducted on three sliding speed i.e. 2.6178 m/s, 3.663 m/s and 5.2631 m/s. A total running distance considered for the test is 2000 meters.

Table 2: Conditions for the pin on disk test.

S. No.	Sliding speed (in m/s)	Track diameter (in mm)	Revolution per minute	Normal Force (in Newton)	Time of run (in sec)
1.	2.6178	50	1000	10, 30 & 50	764
2.	3.663	70	1000	10, 30 & 50	546
3.	5.2631	100	1000	10, 30 & 50	380

3.4. Pin on Disc tribotester

Pin on disc tribotester comes in two variant. First one is having a rotating motion of the disc and a stationary pin. This is known as the rotating type of pin on disk setup. Second one is having a reciprocating motion of the pin over the stationary disc. This is known as the reciprocating type of the pin on disk setup.

Pin on disk test is used to evaluate the amount of wear occurs between the disc and pin surface and also to determine the friction force generated between two contacted surfaces. Tribotester used for this project work is developed by the DUCOM, India. This tribotester comes with a data acquisition system. The complete setup consists of mainly three units: a Tribotester, wear and friction monitor system and a computer system.

Fig 19: Pin on Disc Tribotester

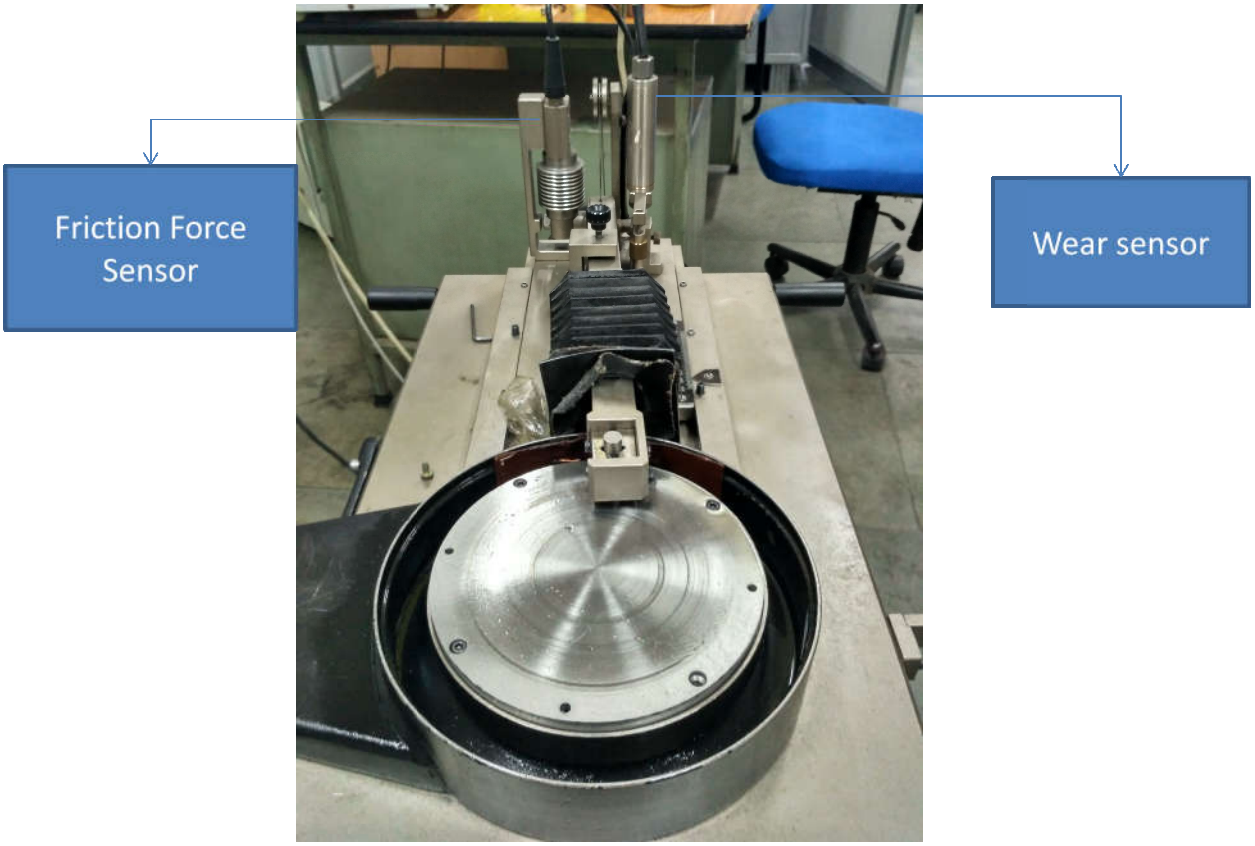


Fig 20: Friction Force sensor and Wear sensor

Fig 21: DUCOM Wear and Friction Monitor.

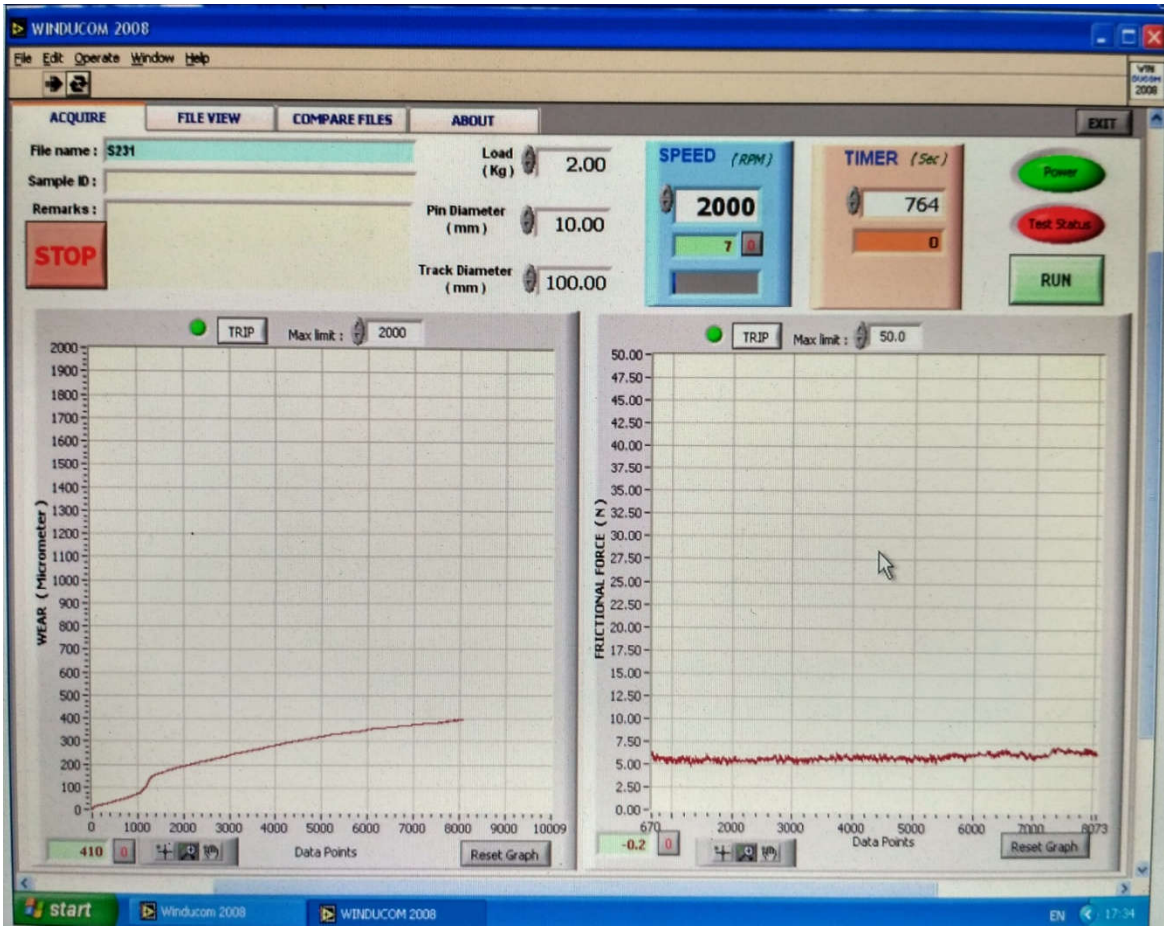


Fig 22: WINDUCOM 2008 Software

A tribotester is of a rotating type of pin on disc tribotester. It consist of lubricating oil sump, wear sensor, friction force sensor, a clamp to hold the pin in required position, a pulley and lever to transfer load normally to the pin. The maximum load can be applied is 200 N. The specification of the tribotester is given in the table below:

Table 2: Specification of pin on disc tribotester.

Parameter	Unit	Minimum value	Maximum Value
Pin Diameter	Mm	3	12
Track diameter	Mm	0	165
Disk rotating speed	Rpm	200	2000
Friction Force	N	0	200
Normal Force	N	1	200

Wear and friction monitor gets the signals from the wear sensor and friction sensor as an input signals. These signals are converted into the machine language of the computer system. Computer system read machine language and converts it into a form which is readable by the complier of the software. DUCOM software (WinDucom 2008) utilizes input signals to show the results of the experiments. A graph of friction force appears on the screen of the computer system. The software allows us to select and vary the various parameters of the experiment like load, time of run, track diameter etc according to the working condition. The tribotester directly gives the friction force developed between the pin and the disc at a given loading and sliding speed.

The amount of material loss from the two components is measured by regularly measuring the weight of the component before and after the experiment. The difference of the weight gives the amount of wear loss. This comes out to be in grams. An accurate weighing balance is used for measuring the weight. It has a least count of 0.0001 grams.

The specific wear rate is then calculated by given below formula:

$$Sw = \frac{Vw}{P \times D}$$

Where S_w = specific wear rate, mm^3 / N-m

V_w = Volume of wear loss, in mm^3

P = Applied load in newtons, N

D = Sliding Distance in meters, m

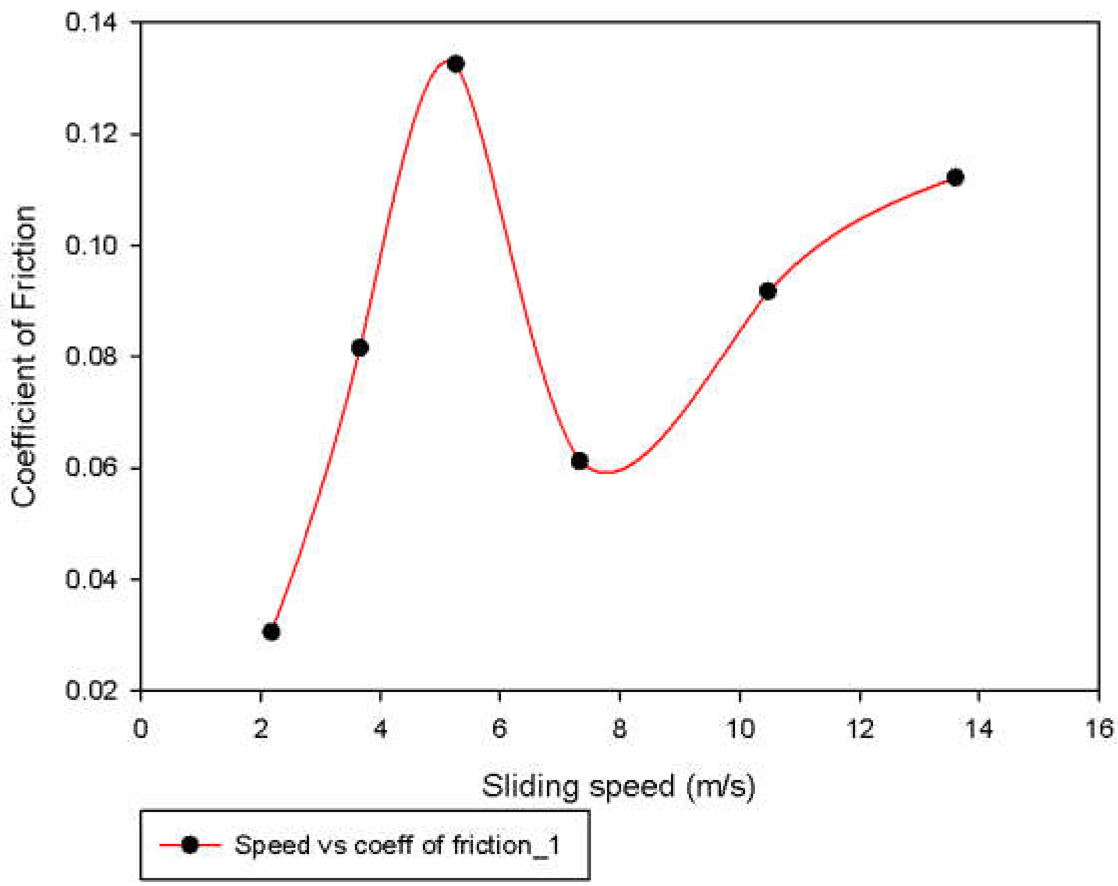
The specimen should be cleaned properly by acetone to remove the oil particles attached on the surface of the components. Otherwise it would hamper the weight of the specimen.

CHAPTER 4: RESULTS & DISCUSSIONS

The experiment was conducted on low speed, medium speed and high speed range. In fully flooded condition the lubricant used was SAE 40. The normal load varied from 10 N to 60 N.

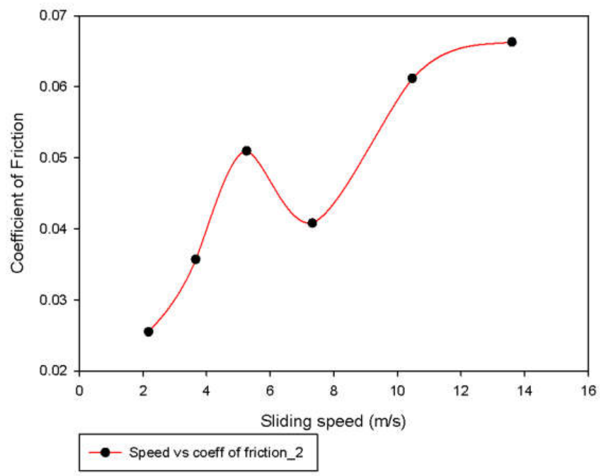
1. **Sliding speed vs coefficient of friction under various testing conditions.**

Test Condition: Full flooded lubrication with uncoated pin under 10N load



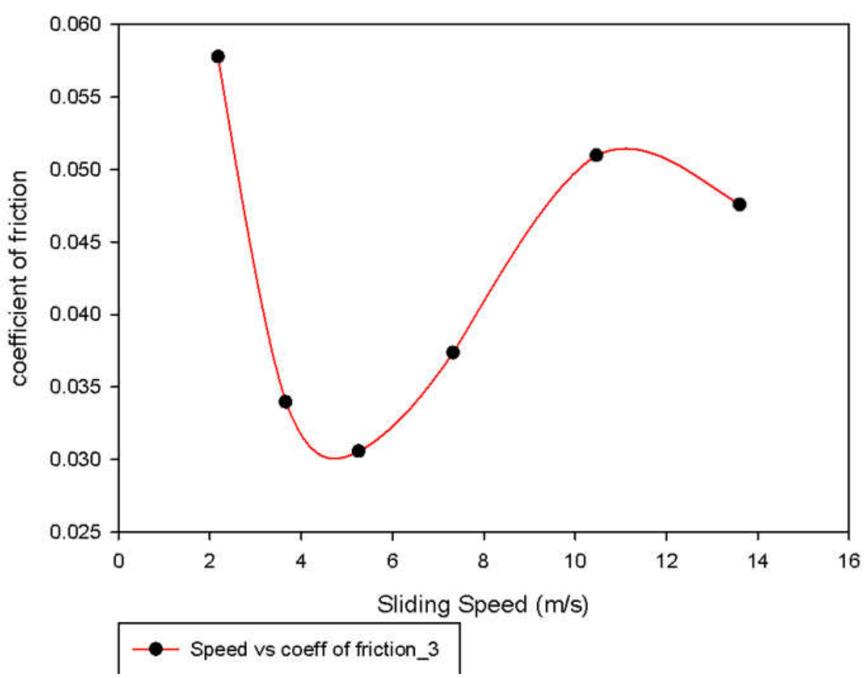
At 10 N load, the Coefficient of friction increased in the start when the speed was increased 2.18 m/s to 5.2631 m/s due to the boundary lubrication. As we reach in the range of medium speed the coefficient of friction followed Stribeck Curve in the end at high velocity range the slopee of the curve increased.

Test Condition: Full flooded lubrication with uncoated pin under 20N load

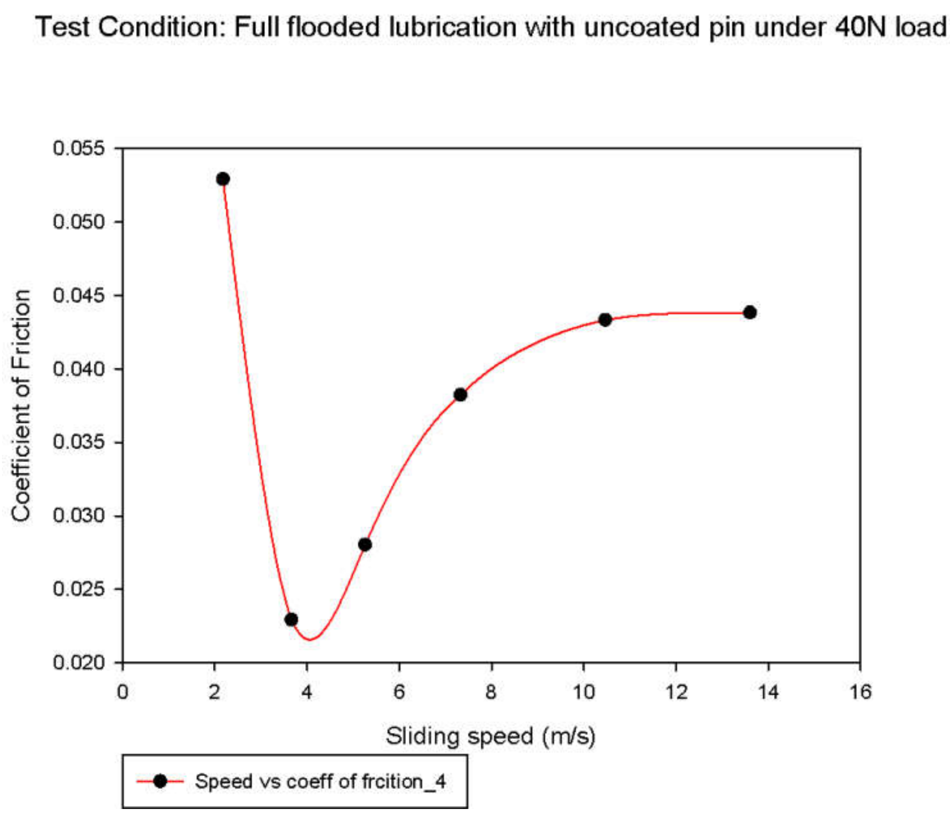


At 20 N load, the Coefficient of friction increased at the start when the speed was increased 2.18 m/s to 5.2631 m/s due to the boundary lubrication. As we reach in the range of medium speed the coefficient of friction followed Stribeck Curve in the end at high velocity range the slopee of the curve became stable indicating the more flow of lubricant in the interface.

Test Condition: Full flooded lubrication with uncoated pin under 30N load

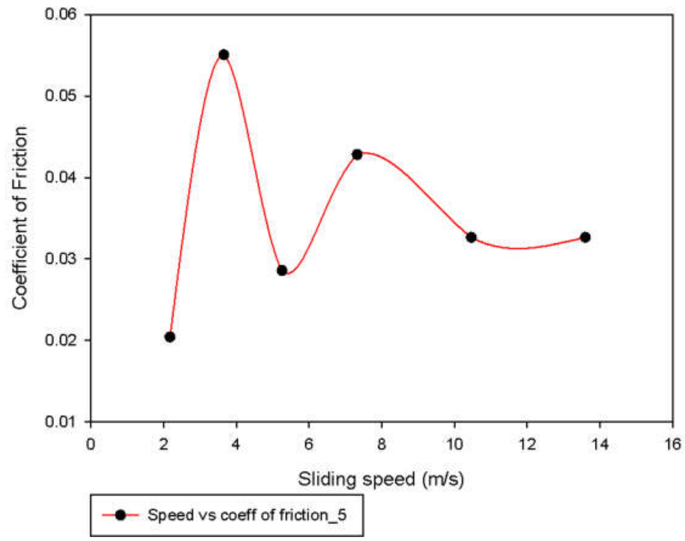


At 30 N load, the Coefficient of friction increased at the start the speed was high 2.18 m/s due to the boundary lubrication. As we reach in the range of medium speed the coefficient of friction followed Stribeck Curve in the end at high velocity range the slopee of the curve showed decline indicating the increased mass flow rate at the interface.



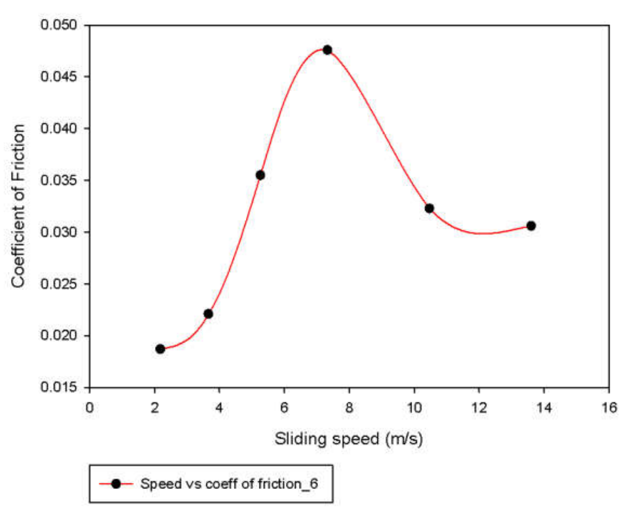
At 40N load, the Coefficient of friction increased at the start the speed was high 2.18 m/s due to the boundary lubrication. As we reach in the range of medium speed the coefficient of friction followed Stribeck Curve in the end at high velocity range the slopee of the curve showed stable slopee indicating the increased mass flow rate at the interface.

Test Condition: Full flooded lubrication with uncoated pin under 50N load

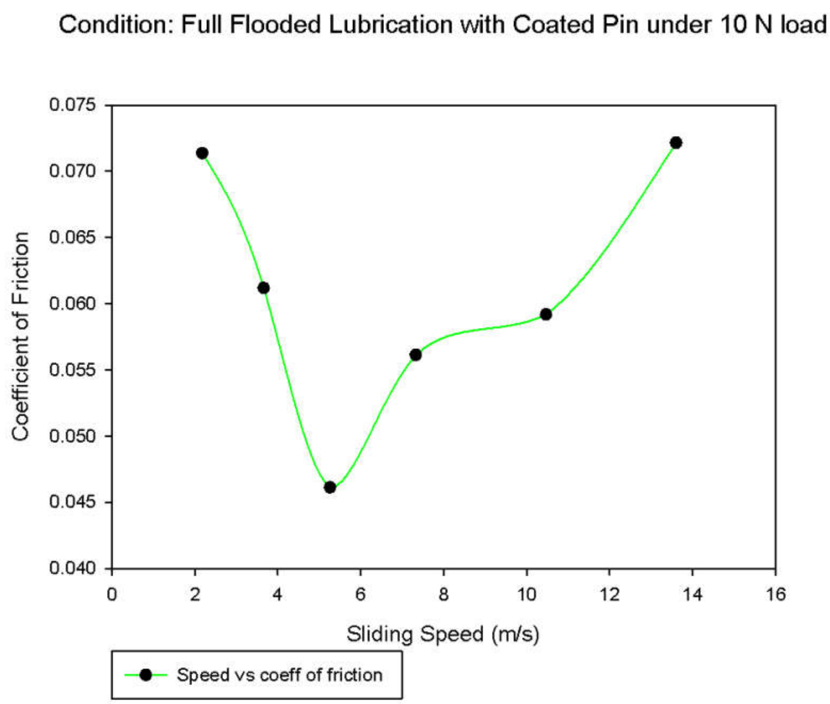


At 50N load, the Coefficient of friction increased at the start the speed increased from 2.18 m/s to 3.663 m/s due to the boundary lubrication. As we reach in the range of medium speed the coefficient of friction followed Stribeck Curve but at the speed of 7.33 m/s the coefficient of friction again increased due to the increased velocity and less mass flow rate. In the end at high velocity range the slope of the curve decreased indicating the increased mass flow rate at the interface.

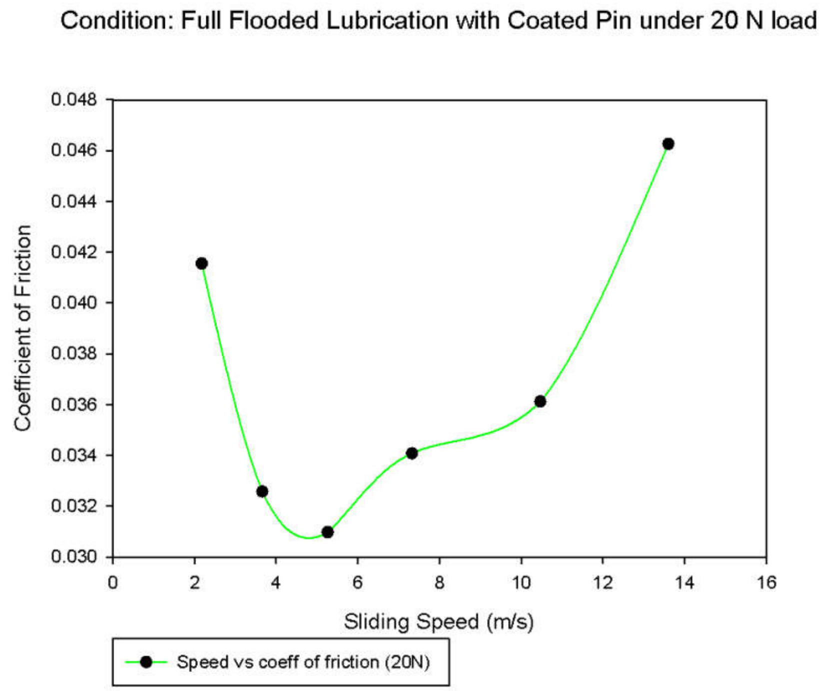
Test Condition: Full flooded lubrication with uncoated pin under 60N load



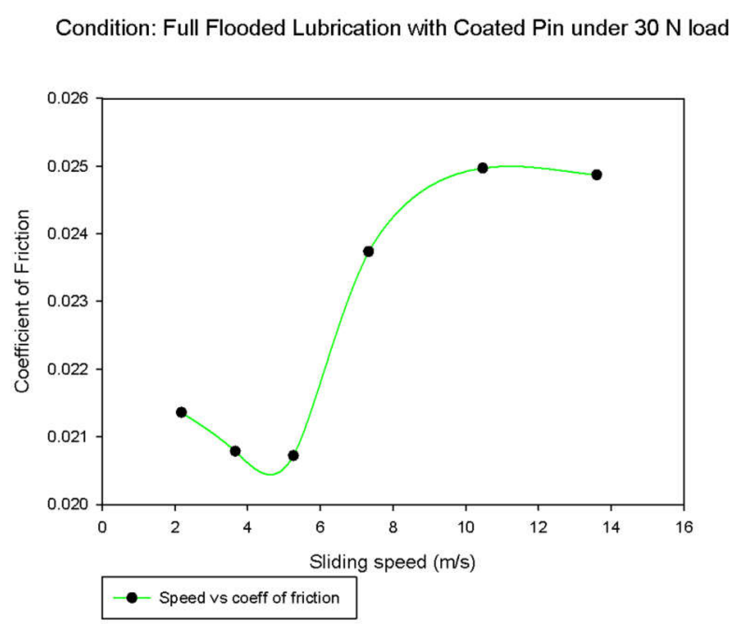
At 60N load, the Coefficient of friction increased at the start the speed increased from 2.18 m/s to 7.33 m/s due to the boundary lubrication and then as we reach in the range of medium speed upto 7.33 m/s the coefficient of friction keeps on increasing due to the increased velocity and less mass flow rate. In the end at high velocity range the slopee of the curve decreased indicating the increased mass flow rate at the interface.



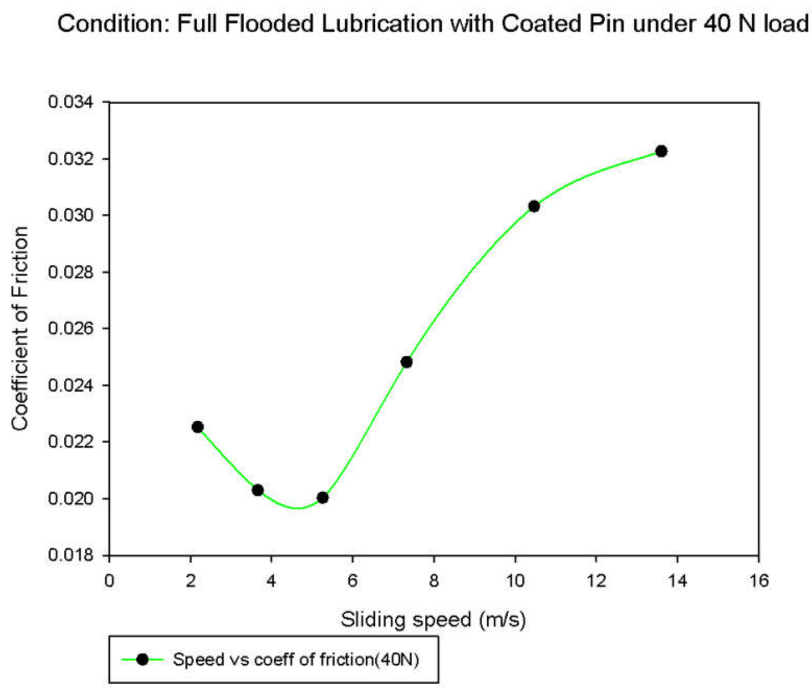
At 10N load, the Coefficient of friction decreased at the start the speed increased from 2.18 m/s to 5.2631 m/s due to the conversion of boundary lubrication to hydrostatic lubrication. As we reach in the range of medium speed the coefficient of friction followed Stribeck Curve and increased upto the speed of 10.47 m/s with larger slopee at the beginning then the curve became flat due to the entry of more lubricant at the interface. The slope again showed an increase due to the dominance of the viscous forces.



At 20N load, the Coefficient of friction decreased at the start the speed increased from 2.18 m/s to 5.2631 m/s. As we reach in the range of medium speed the coefficient of friction followed Stribeck Curve and increased upto the speed of 10.47 m/s with lesser slopee and then the slopee increased sharply due to the viscous forces of the lubricating oil.

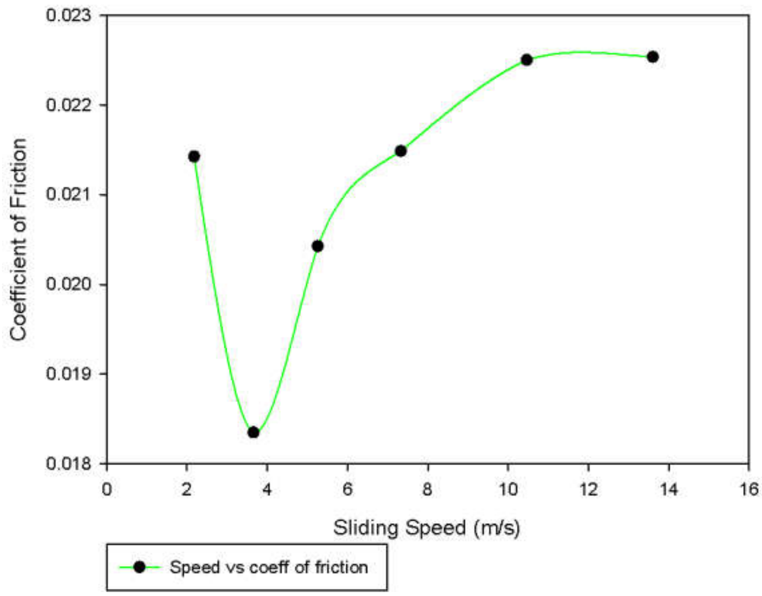


At 30N load, the Coefficient of friction decreased at the start the speed increased from 2.18 m/s to 3.663 m/s. As we reach in the range of medium speed the coefficient of friction followed Stribeck Curve and increased upto the speed of 10.47 m/s with larger slopee due to the increased load and less entry of lubricating oil in the interface and then the slopee decreased slowly due to the entry of more lubricant at the interface.



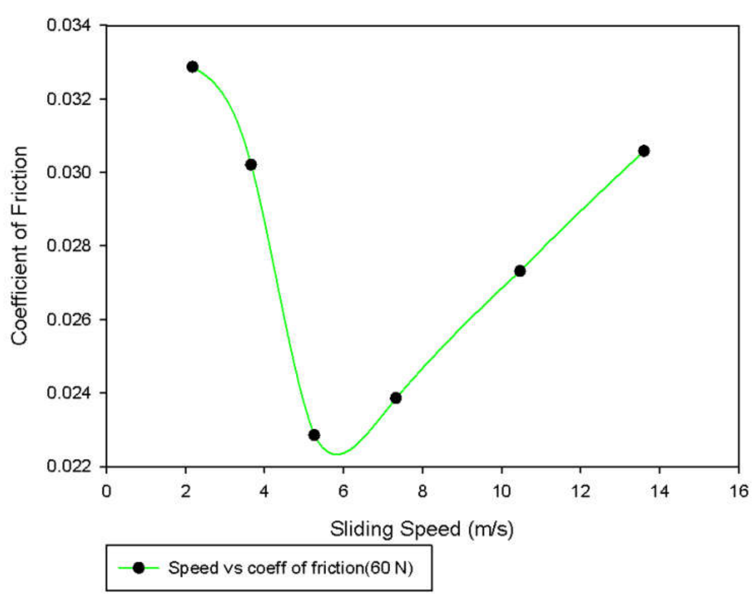
At 40N load, the Coefficient of friction decreased at the start the speed increased from 2.18 m/s to 3.66 m/s. As we reach in the range of medium speed the coefficient of friction followed Stribeck Curve and increased upto the speed of 10.47 m/s with larger slopee due to the increased load and less entry of lubricating oil in the interface and then the slopee increased slowly due to the entry of more lubricant at the interface.

Condition: Full Flooded Lubrication with Coated Pin under 50 N load

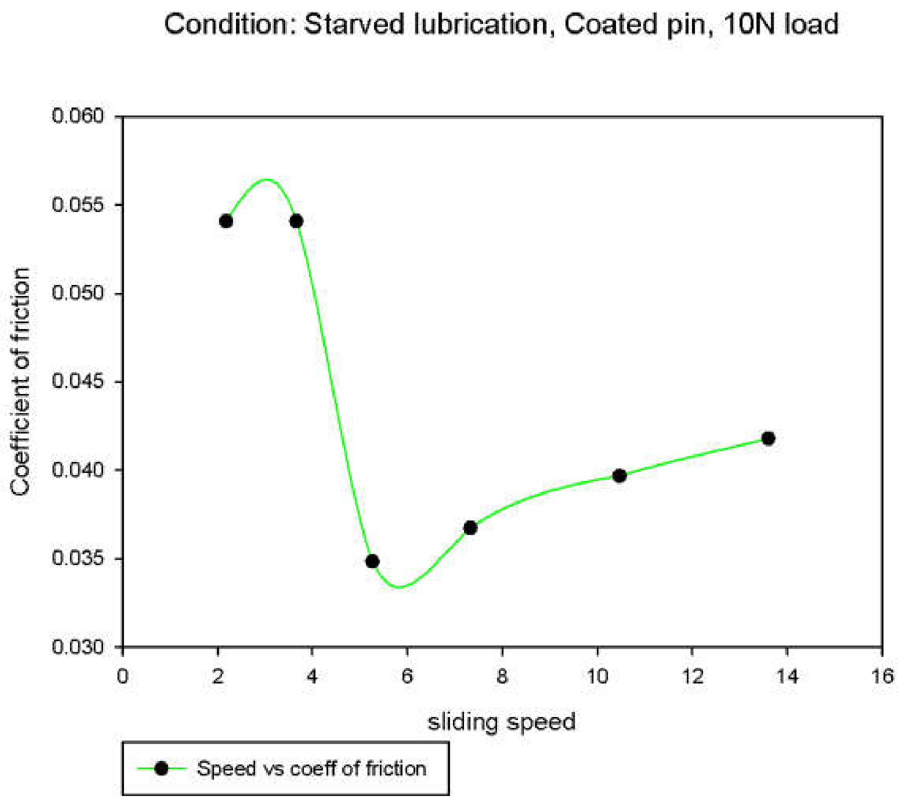


At 50N load, the Coefficient of friction decreased at the start the speed increased from 2.18 m/s to 3.663 m/s. As we reach in the range of medium speed the coefficient of friction followed Stribeck Curve and increased sharply upto the speed of 7.33 m/s and then the slopee decreased as the spend increased indicating the increased mass flow rate at the interface.

Condition: Full Flooded Lubrication with Coated Pin under 60 N load

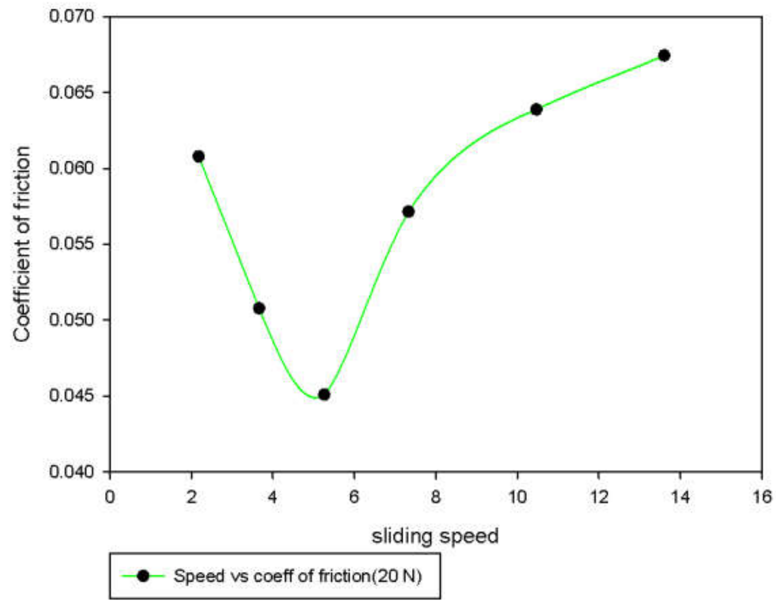


At 60N load, the Coefficient of friction decreased at the start the speed increased from 2.18 m/s to 5.2631 m/s. As we reach in the range of medium speed the coefficient of friction followed Stribeck Curve and increased sharply upto the speed of 13.6054 m/s.



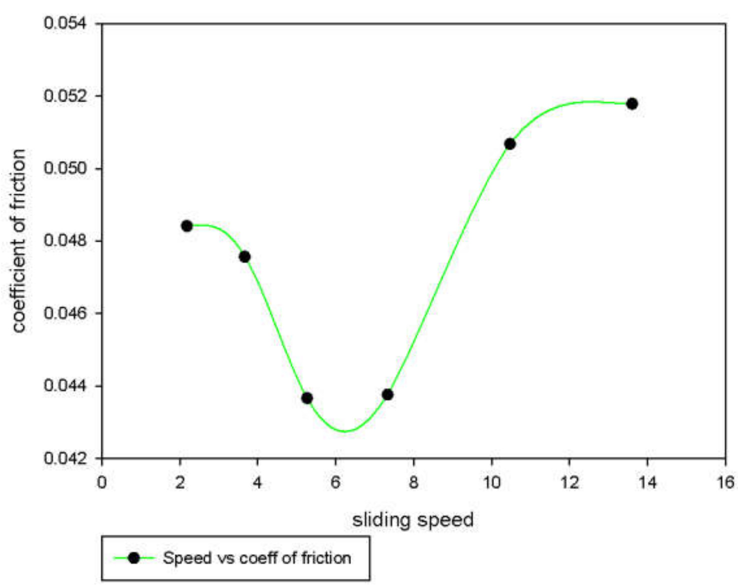
At 10N load, the Coefficient of friction remains almost constant as we increase the speed from 2.18 m/s to 3.66 m/s. As we reach in the range of medium speed the coefficient of friction followed Stribeck Curve and increased with constant slopee upto the speed of 13.6054 m/s.

Condition: Starved lubrication, Coated pin, 20N load

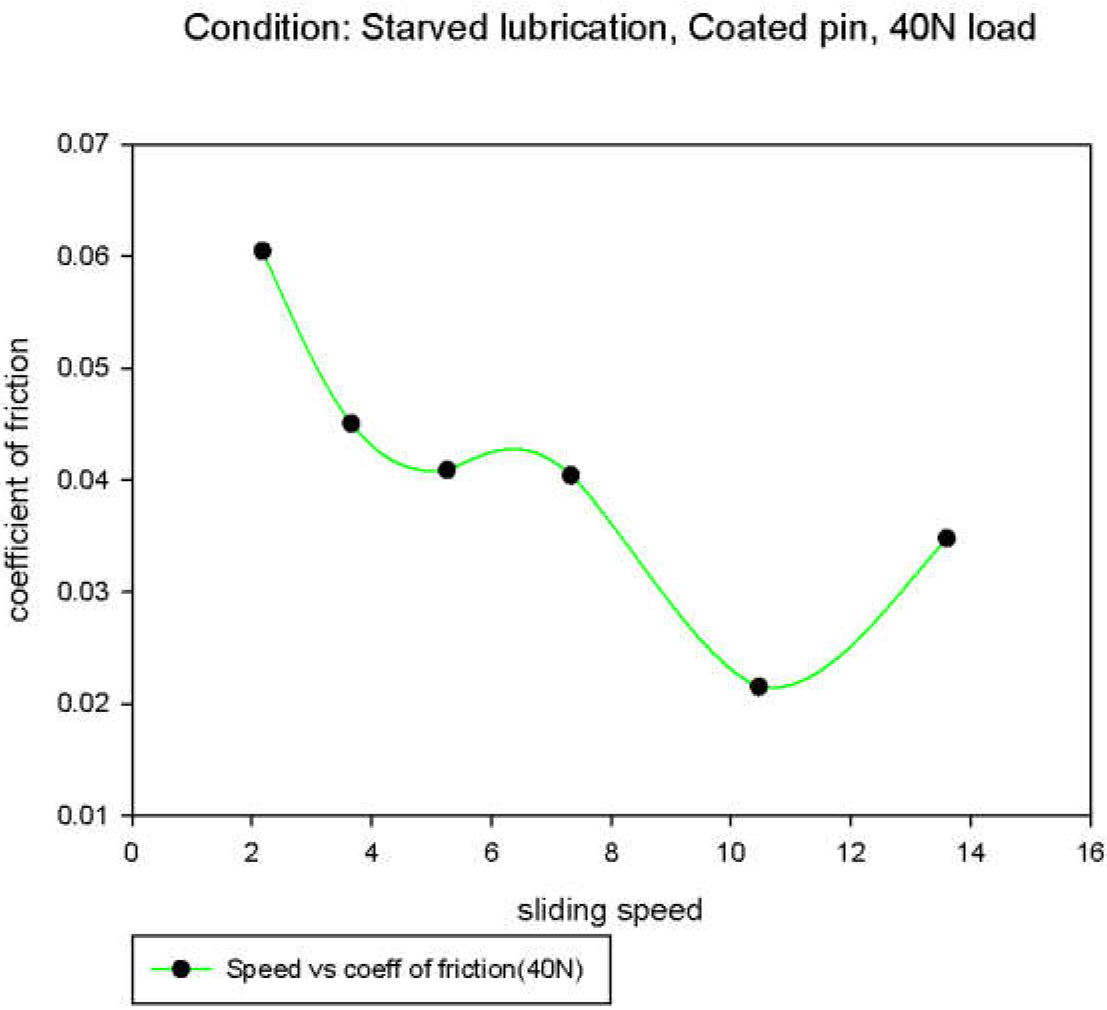


At 20N load, the Coefficient of friction decreased at the start the speed increased from 2.18 m/s to 5.2631 m/s. due to the transformation of boundary lubrication to hydrodynamic lubrication. As we reach in the range of medium speed the coefficient of friction increases with a greater slope and then keeps on increasing but the slope is not that much this is due to the increased flow rate of lubricant between the pin and disc.

Condition: Starved lubrication, Coated pin, 30N load

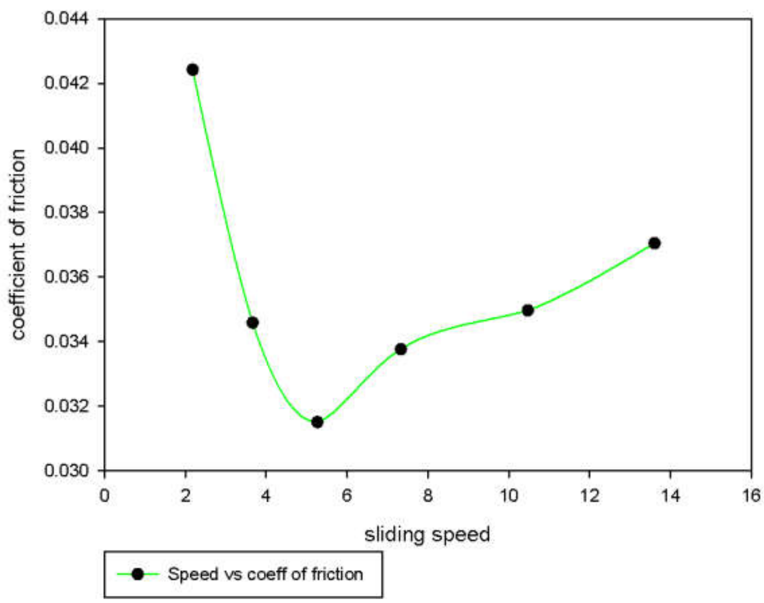


At 30N load, the Coefficient of friction decreased at the start the speed increased from 2.18 m/s to 5.2631 m/s, due to the transformation of boundary lubrication to hydrodynamic lubrication. As we reach in the range of medium speed the coefficient of friction increases with a greater slope and then keeps on increasing but the slope is not that much this is due to the increased flow rate of lubricant between the pin and disc.



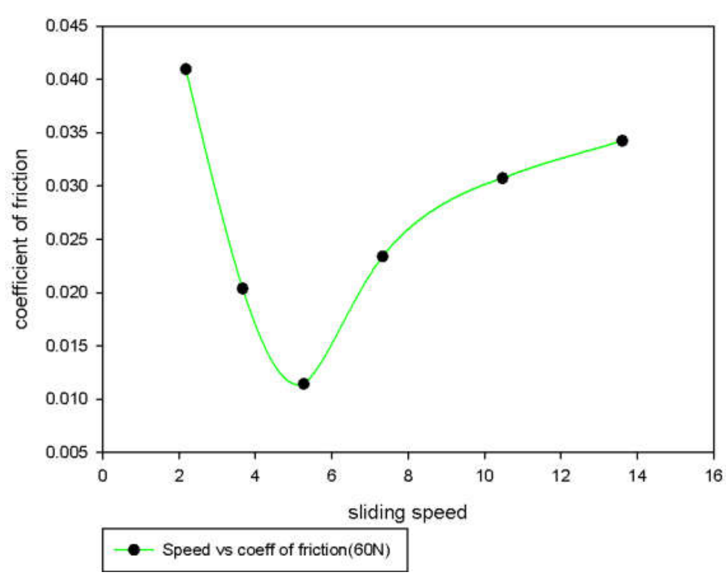
At 40N load, the Coefficient of friction decreased at the start the speed increased from 2.18 m/s to 10.47 m/s. due to the transformation of boundary lubrication to hydrodynamic lubrication, the slope becomes constant in the speed range of 5.2631 m/s to 7.326 m/s due to the constant layer of lubrication. As we reach in the range of high speed the coefficient of friction increases due to the higher drag force present which pulls the lubricant in the space.

Condition: Starved lubrication, Coated pin, 50N load

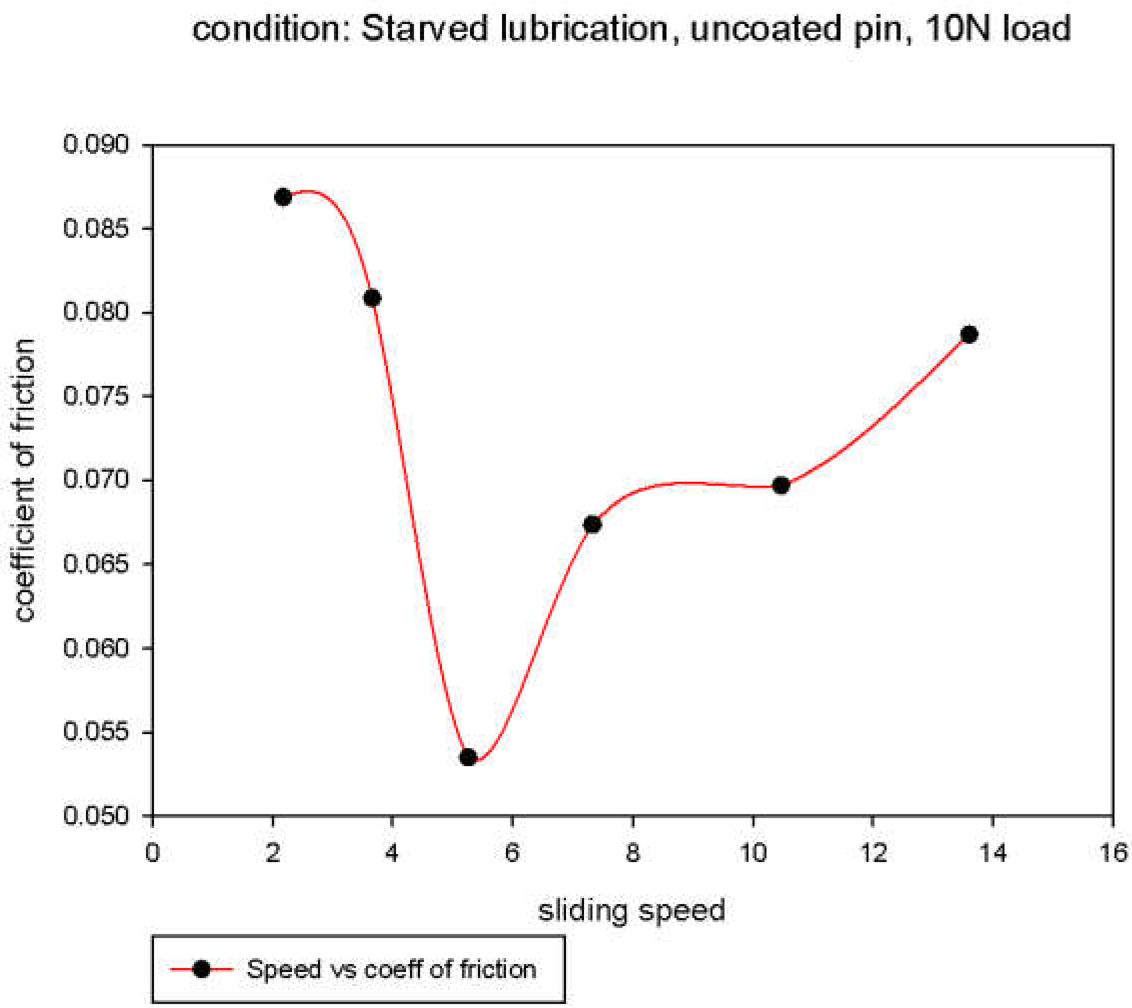


At 50N load, the Coefficient of friction decreased at a faster rate as the speed increased from 2.18 m/s to 5.2631 m/s. due to the transformation of boundary lubrication to hydrodynamic lubrication. As we reach in the range of medium speed the coefficient of friction increases with a lesser slope and keeps on increasing but the slope is not that much this is due to the increased flow rate of lubricant between the pin and disc.

Condition: Starved lubrication, Coated pin, 60N load

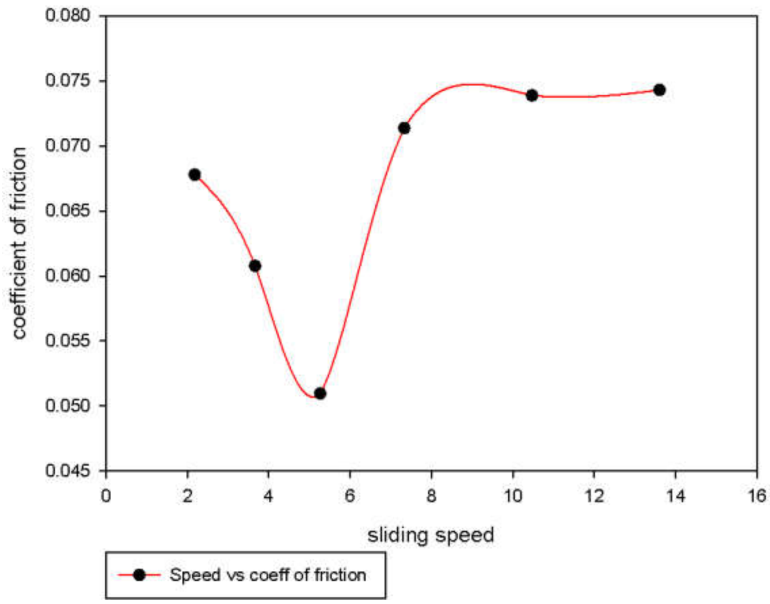


At 60N load, the Coefficient of friction decreased at a faster rate as the speed increased from 2.18 m/s to 5.2631 m/s. due to the transformation of boundary lubrication to hydrodynamic lubrication. As we reach in the range of medium speed the coefficient of friction increases and keeps on increasing in the high speed range but the slope is not that much this is due to the increased flow rate of lubricant between the pin and disc.



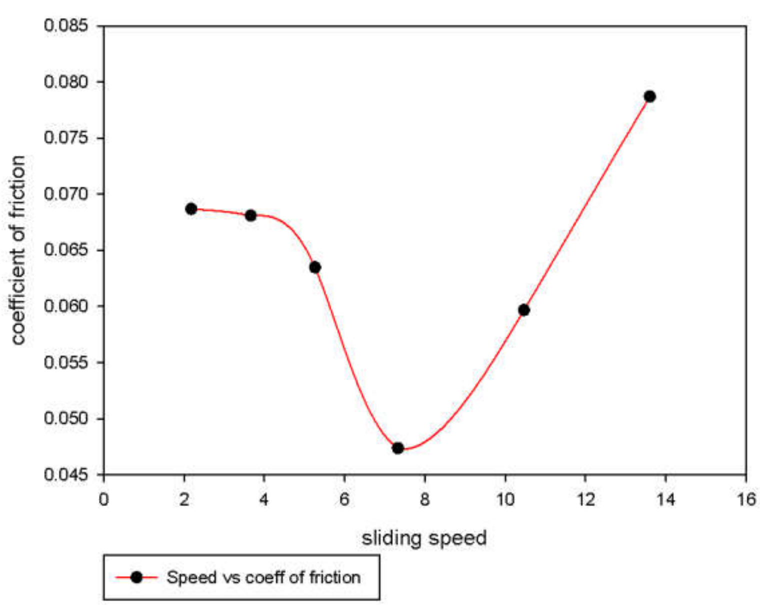
At 10N load, the Coefficient of friction decreased at a faster rate as the speed increased from 2.18 m/s to 5.2631 m/s. due to the transformation of boundary lubrication to hydrodynamic lubrication. As we reach in the range of medium speed the coefficient of friction increases due to increase in speed and then becomes constant due to increased flow of lubricant. In the high speed range the coefficient of friction again increase due to the drag on the lubricant.

condition: Starved lubrication, uncoated pin, 10N load

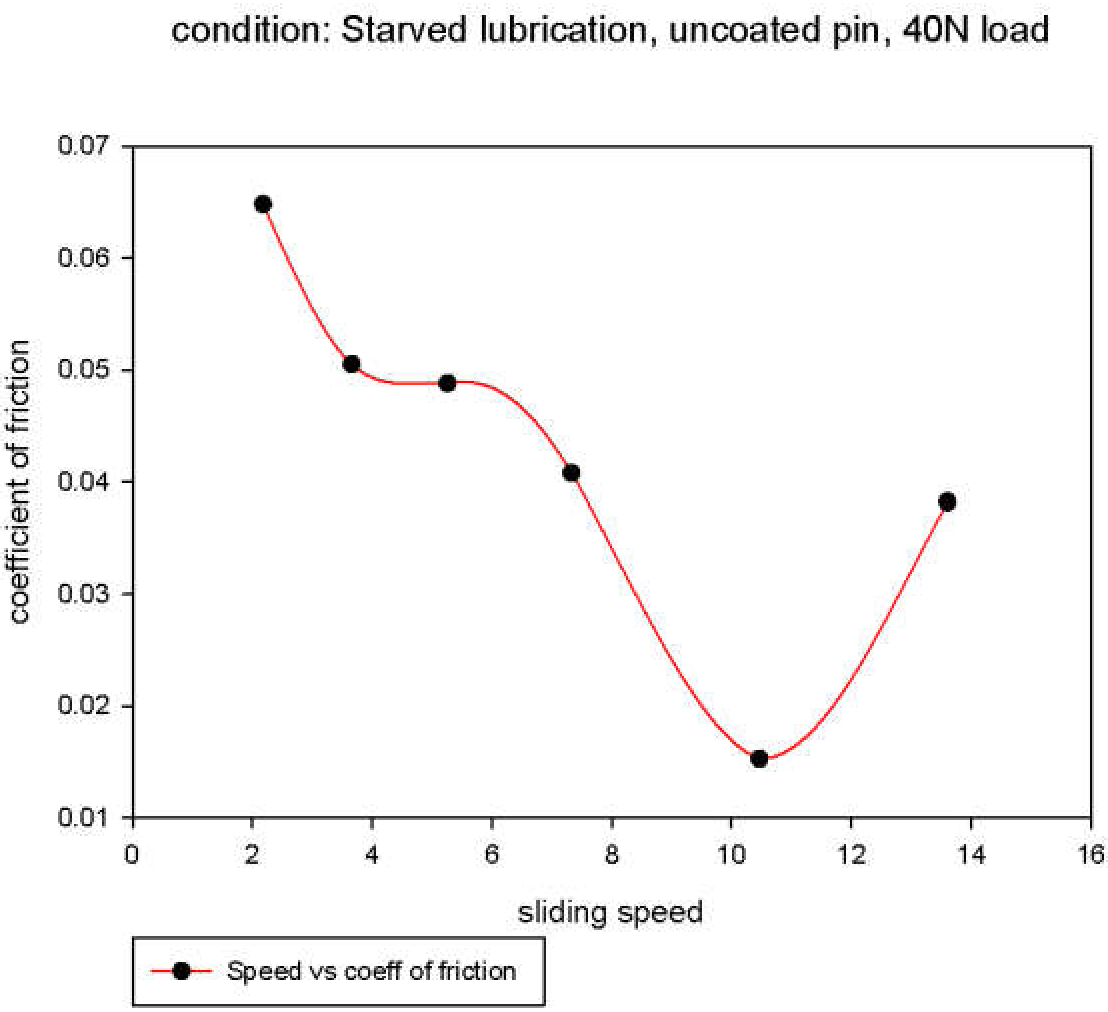


At 20N load, the Coefficient of friction decreased at a faster rate as the speed increased from 2.18 m/s to 5.2631 m/s. due to the transformation of boundary lubrication to hydrodynamic lubrication. As we reach in the range of medium speed the coefficient of friction increases due to increase in speed and then becomes stable due to the presence of a stable lubricating film between the pin and disc.

condition: Starved lubrication, uncoated pin, 30N load

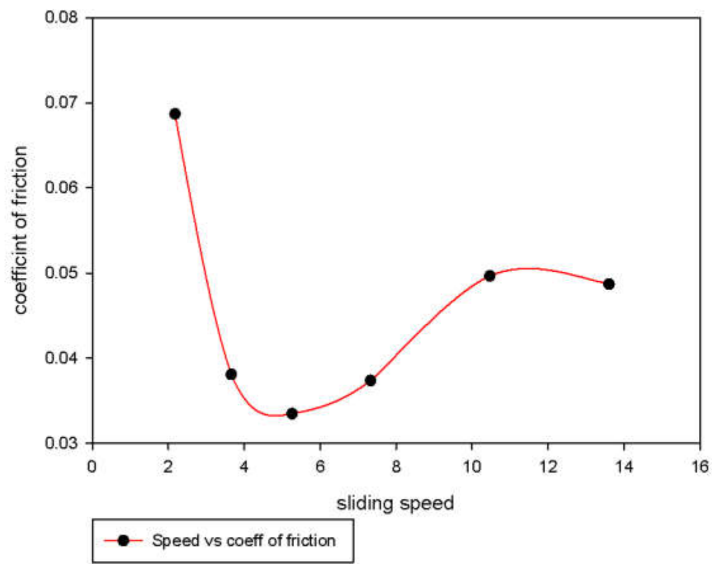


At 30N load, the Coefficient of friction is almost constant as the speed increased from 2.18 m/s to 5.2631 m/s. due to the presence of boundary lubrication. Then the coefficient of friction decreases as the hydrodynamic lubrication comes into play. As we reach in the range of high speed from medium the coefficient of friction increases due to increase in speed.



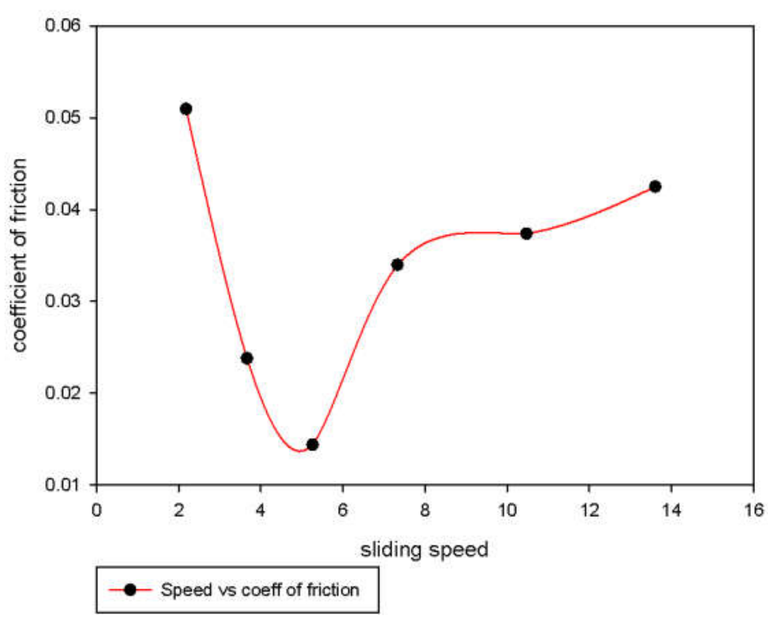
At 40N load, the Coefficient of friction decreased at a faster rate as the speed increased from 2.18 m/s to 10.47 m/s. due to the transformation of boundary lubrication to mixed lubrication and then hydrodynamic lubrication. In the high speed range the coefficient of friction again increase due to the drag on the lubricant.

condition: Starved lubrication, uncoated pin, 50N load



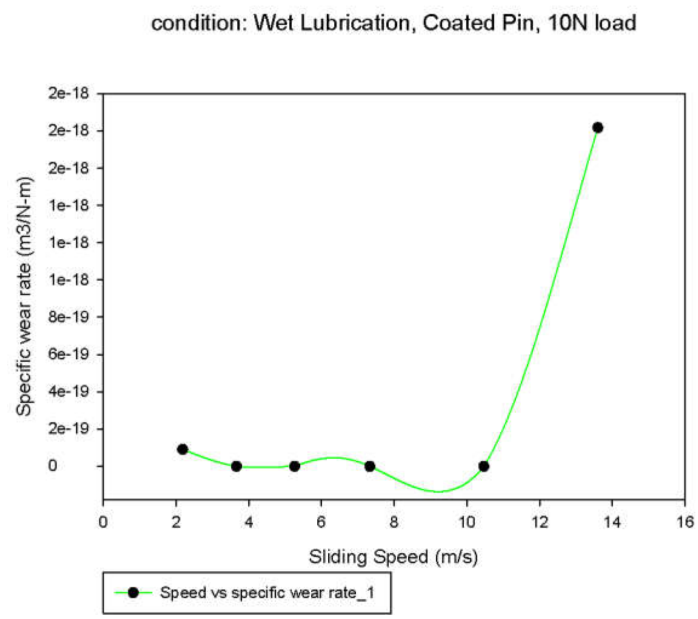
At 50N load, the Coefficient of friction decreased at a faster rate as the speed increased from 2.18 m/s to 5.2631 m/s. due to the transformation of boundary lubrication to hydrodynamic lubrication. As we reach in the range of medium speed the coefficient of friction increases due to increase in speed with very less slope. In the high speed range the coefficient of friction again decrease due to the increased mass flow rate of the lubricant.

condition: Starved lubrication, uncoated pin, 60N load

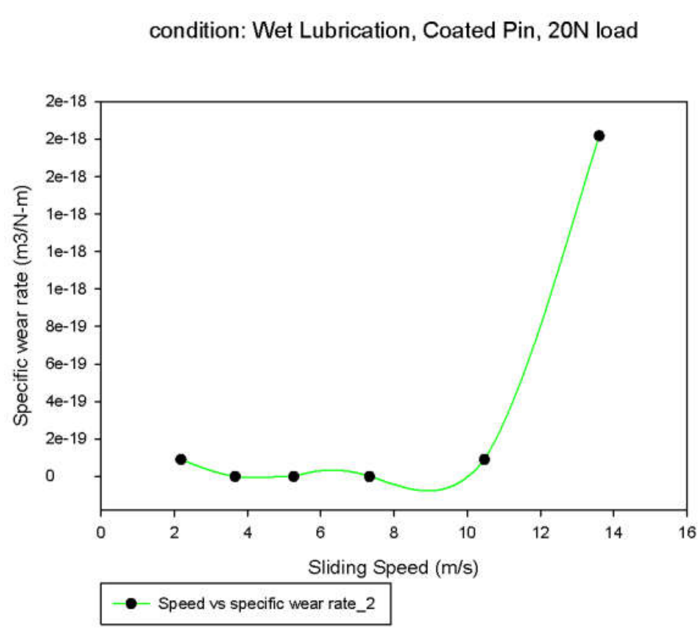


At 60N load, the Coefficient of friction decreased at a faster rate as the speed increased from 2.18 m/s to 5.2631 m/s. due to the transformation of boundary lubrication to hydrodynamic lubrication. As we reach in the range of medium speed the coefficient of friction increases due to increase in speed and then becomes constant due to increased flow of lubricant. In the high speed range the coefficient of friction again shows slit increase due to the drag on the lubricant.

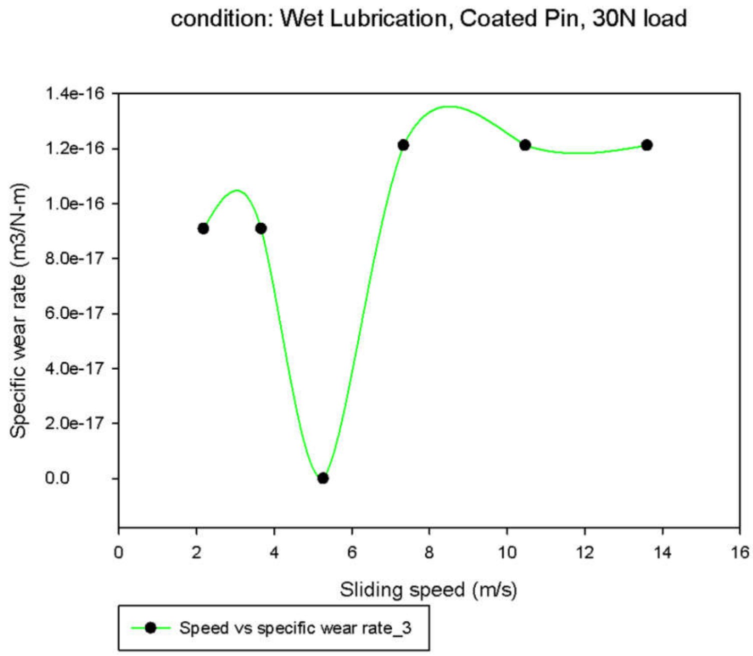
2. Specific Wear rate vs. Sliding speed at various test conditions



At 10 N load the specific wear rate drops as we reach from low speed range to medium speed range. The higher values of specific wear rate at the start may be due to the virgin surfaces with higher surface roughness. In the high speed range the specific wear rate increased due to the increased friction force between the pin and the disc.

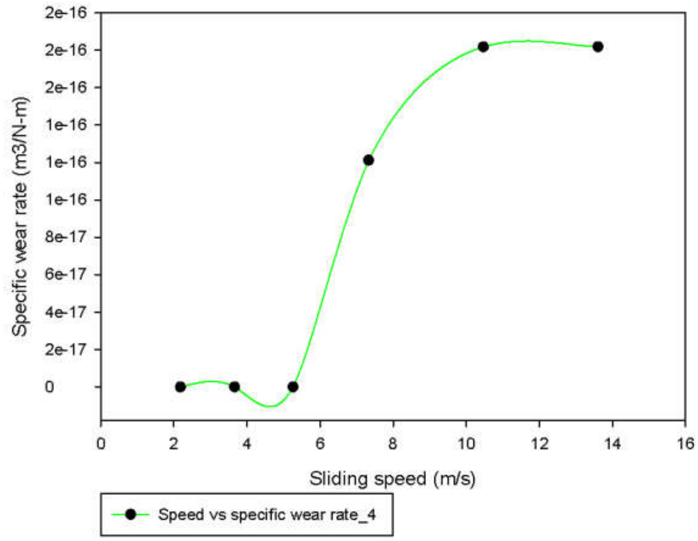


At 20 N load the specific wear rate drops as we reach from low speed range to medium speed range. The higher values of specific wear rate at the start may be due to the virgin surfaces with higher surface roughness. As the speed increased from 7.33 m/s to 13.6054 m/s the specific wear rate again increased due to the increased metal to metal contact.



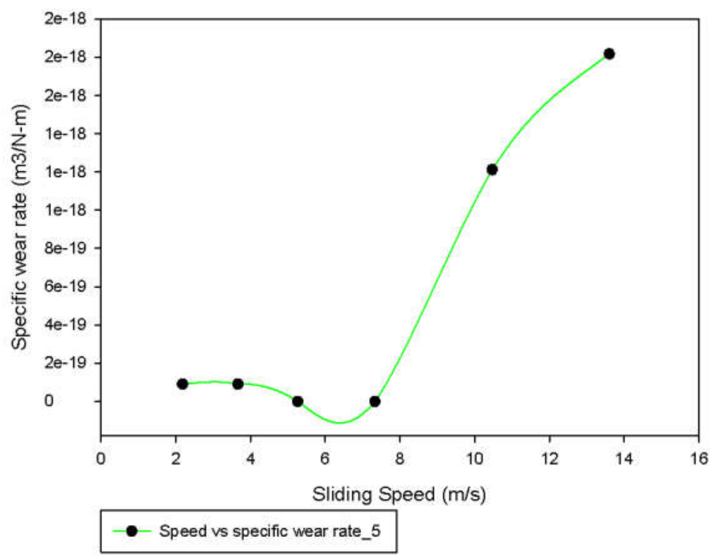
At 30 N load the specific wear rate drops as we reach from low speed range to medium speed range this is due to the hydrodynamic lubrication. The specific wear rate again increased due to the increased metal to metal contact. Then the specific wear rate shows constant trend because as the speed increased the lubricating film thickness between the pin and disc also increased.

condition: Wet Lubrication, Coated Pin, 40N load

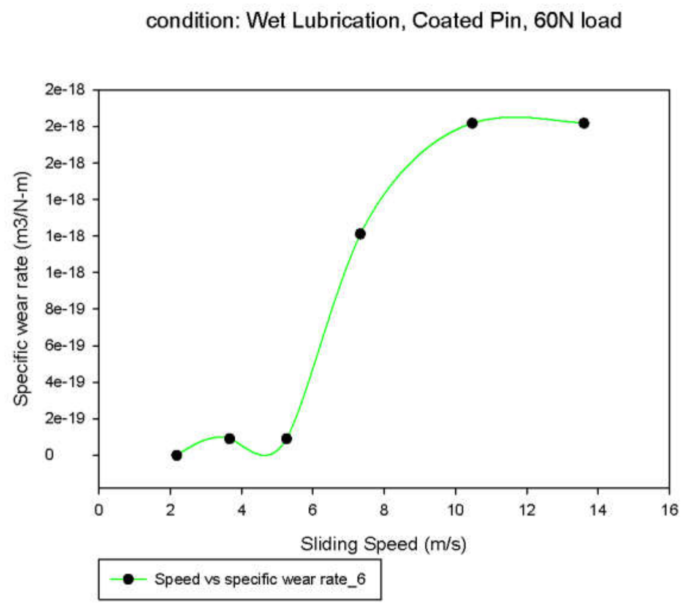


At 40 N load the specific wear rate have no significant value in the low speed range this may be due to the layer of lubricant between the pin and disc. Then as the friction force increased the specific wear rate also increased.

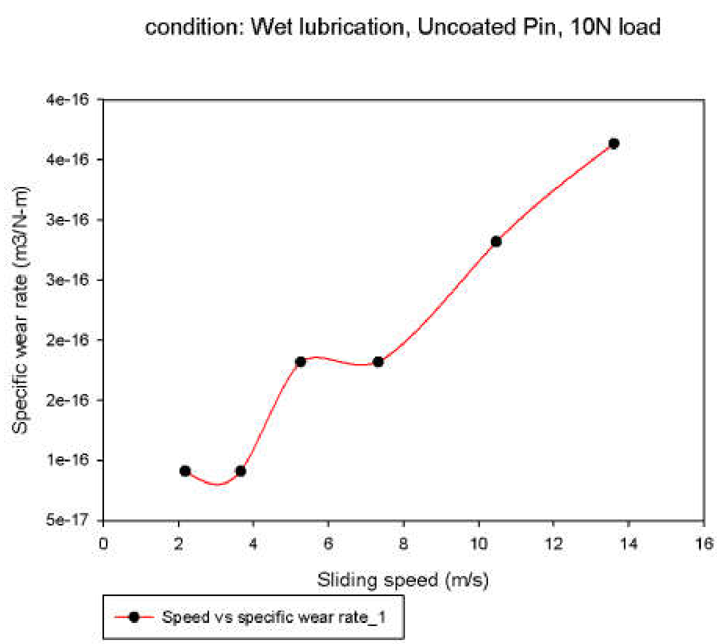
condition: Wet Lubrication, Coated Pin, 50N load



At 50 N load the specific wear rate drops as we reach from low speed range to medium speed range. The higher value of specific wear rate is due to the boundary lubrication. In the medium speed range as the hydrodynamic lubrication comes into play the wear is negligible. in high speed range the specific wear rate increased due to the increased friction force.

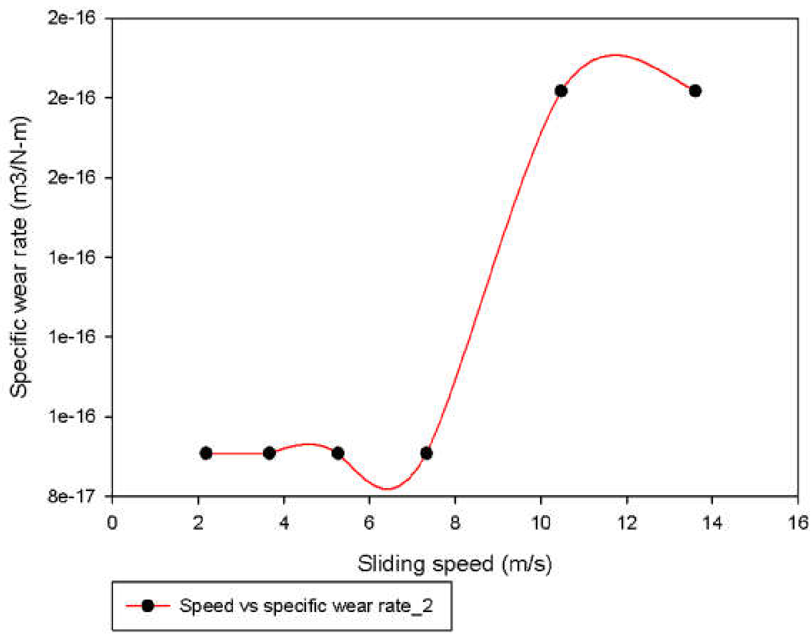


At 60 N load the specific wear increased as we reach from the low speed range to high speed range due to the increase in friction force.

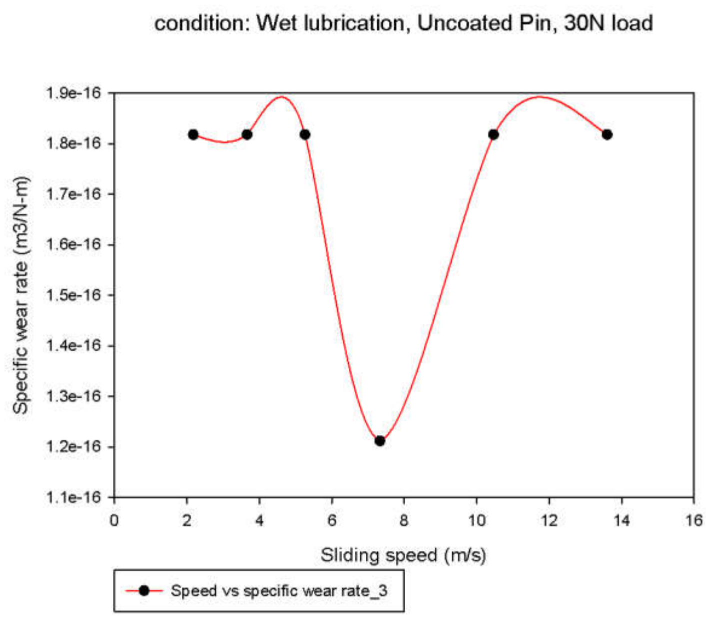


At 10 N load the specific wear has no significance in the low speed range due to the presence of boundary lubrication between the pin and disc. The specific wear rate increased as we reach from the low speed range to high speed range due to the increase in friction force.

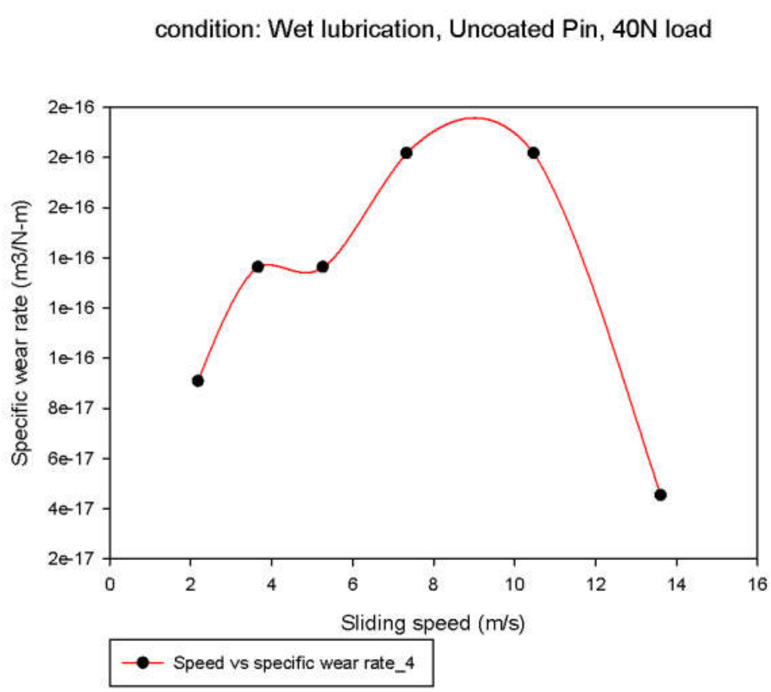
condition: Wet lubrication, Uncoated Pin, 20N load



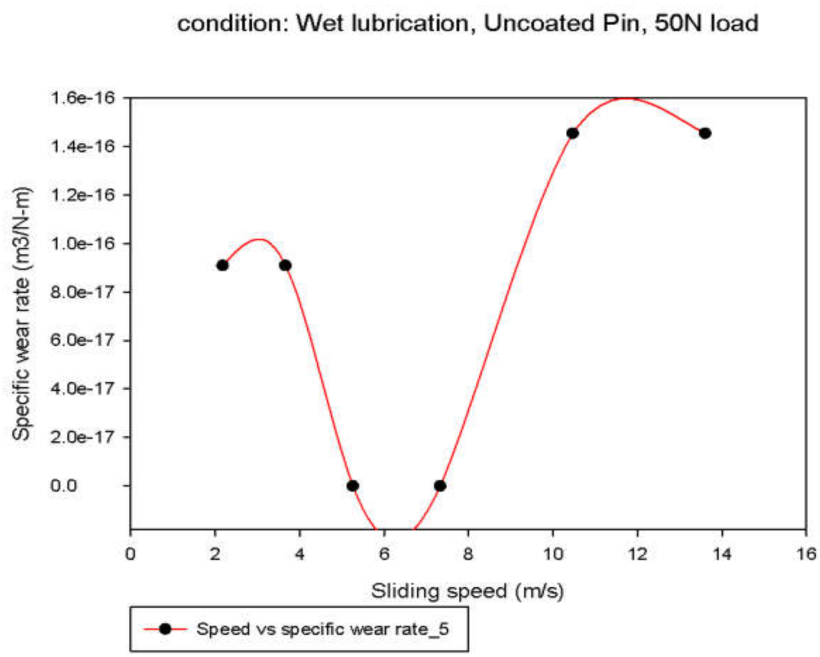
At 20 N load the specific wear rate drops as we reach from low speed range to medium speed range. The higher values of specific wear rate at the start may be due to the virgin surfaces with higher surface roughness. As the speed increased from 7.33 m/s to 13.2631 m/s the specific wear rate again increased due to the increased metal to metal contact.



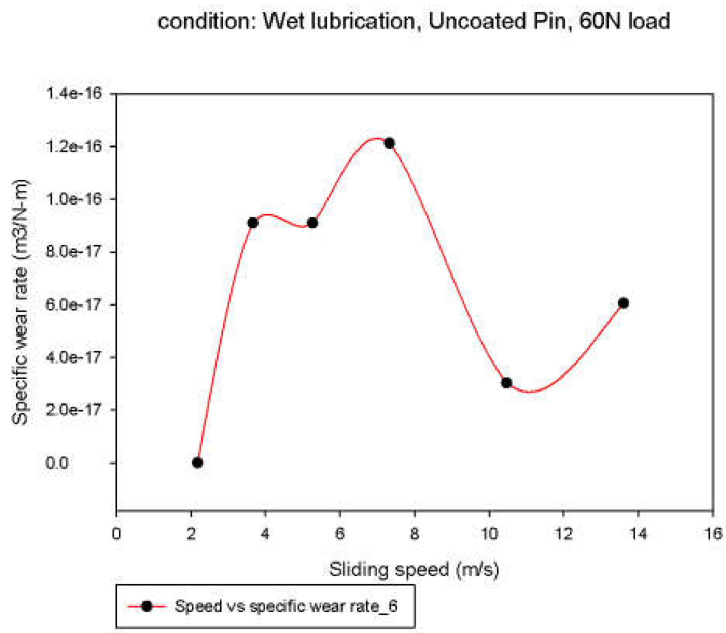
At 30 N load the specific wear rate drops as we reach from low speed range to medium speed range this is due to the hydrodynamic lubrication. The specific wear rate again increased due to the increased frictional force. Then the specific wear rate shows constant trend in high speed region because as the speed increased the lubricating film thickness between the pin and disc also increased.



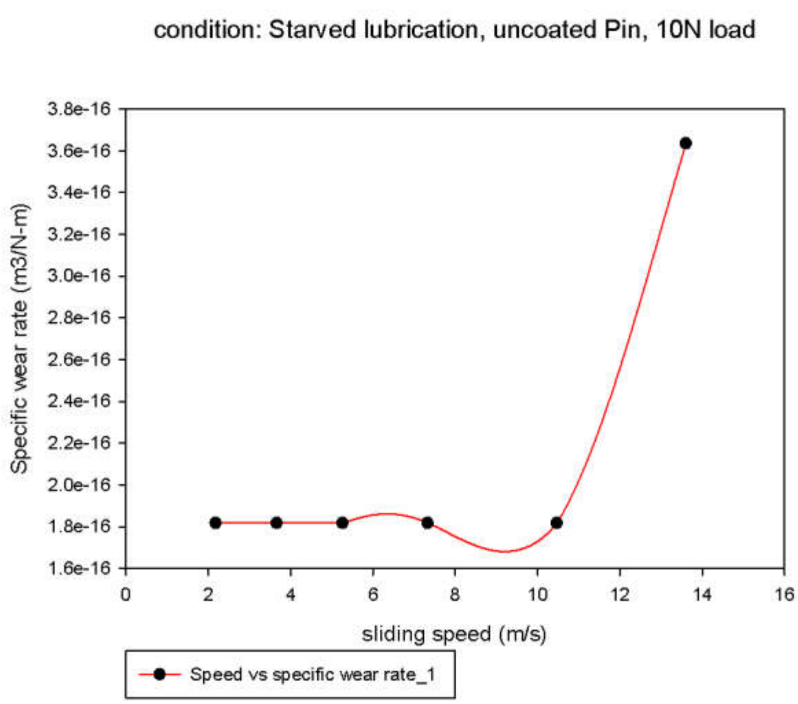
At 40 N load the specific wear rate have less value in the low speed range this may be due to the layer of lubricant between the pin and disc. In the medium speed region the hydrodynamic lubrication between the pin and the disc comes into play but the thickness of lubricating film is to small to prevent the metal to metal contact that results in higher specific wear rate. As we increase the speed to the high speed range the specific wear rate again drooped due to the presence of thick layer of lubricant.



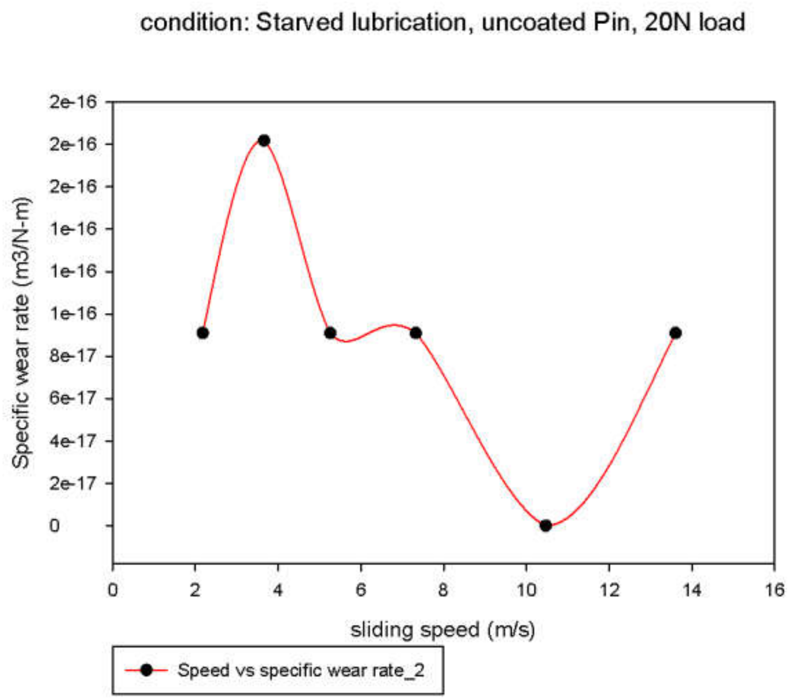
At 50 N load the specific wear rate drops as we reach from low speed range to medium speed range. The higher value of specific wear rate is due to the boundary lubrication. In the medium speed range as the hydrodynamic lubrication comes into play the wear is negligible. In high speed range the specific wear rate increased due to the increased friction force.



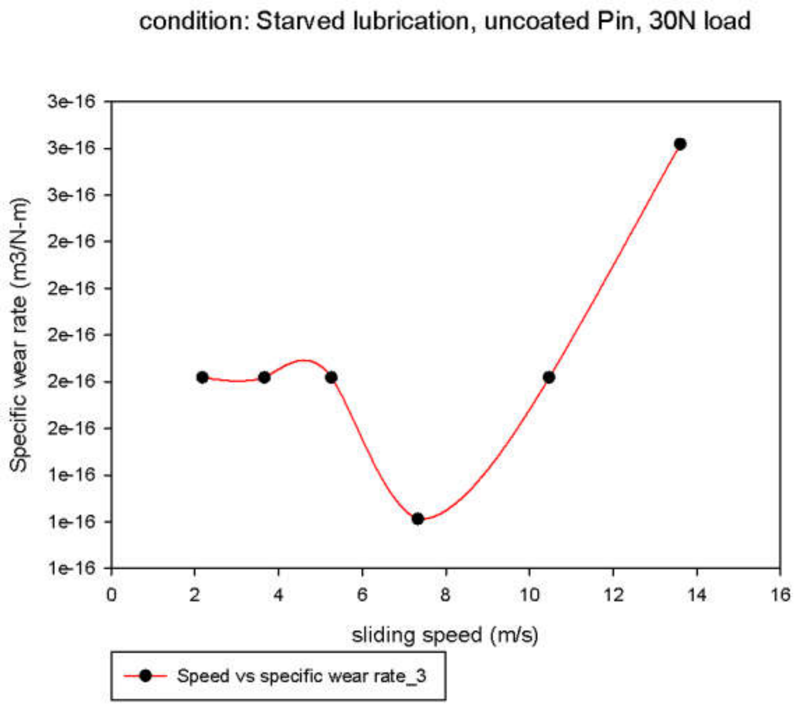
At 60 N load the specific wear increased as we reach from the low speed range to medium speed range due to the increase in metal to metal contact between the pin and the disc. As the speed increased from7 m/s to 13 m/s the specific wear rate again drooped due to the increased thickness of lubrication film.



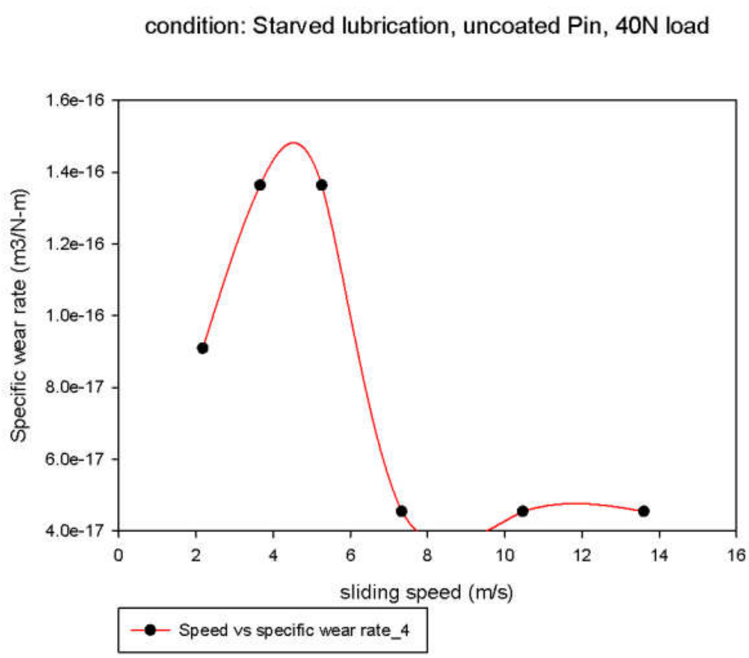
At 10 N load the specific wear has no significance in the low speed range due to the presence of boundary lubrication between the pin and disc. The specific wear rate increased as we reach from the low speed range to high speed range due to the increase in friction force.



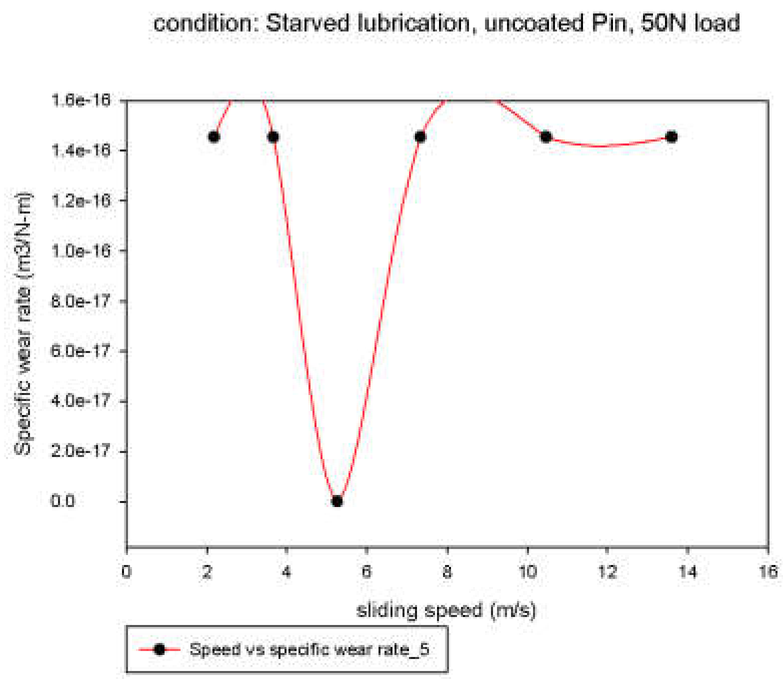
At 20 N load the specific wear rate drops as we reach from low speed range to medium speed range. This is due to the presence of thick boundary lubrication between the pin and the disc. As the speed increased from 10 m/s to 13 m/s the specific wear rate again increased due to the increased frictional force.



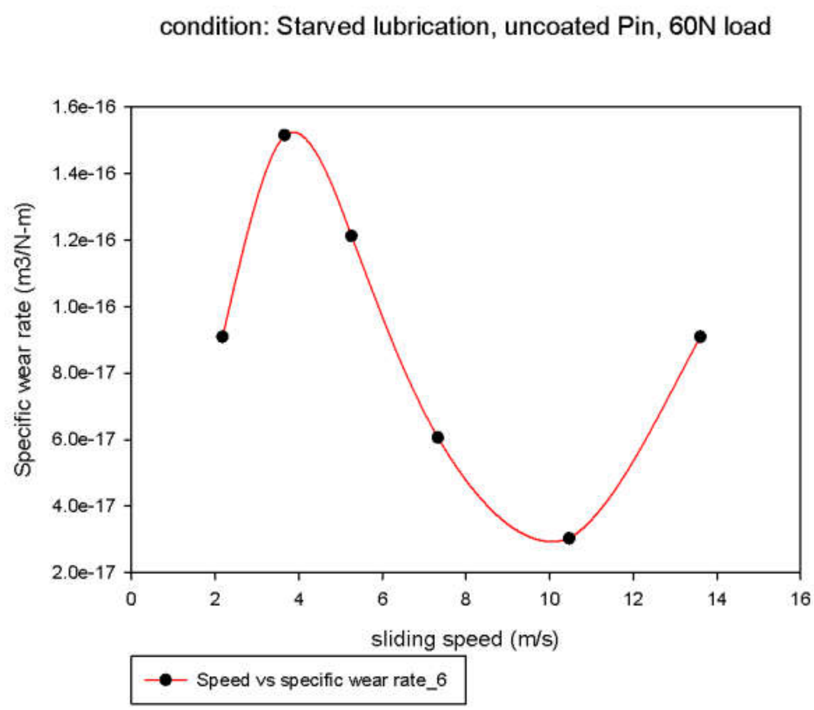
At 30 N load the specific wear rate drops as we reach from low speed range to medium speed range this is due to thick boundary lubrication between the pin and disc. The specific wear rate again increased due to the increased frictional force.



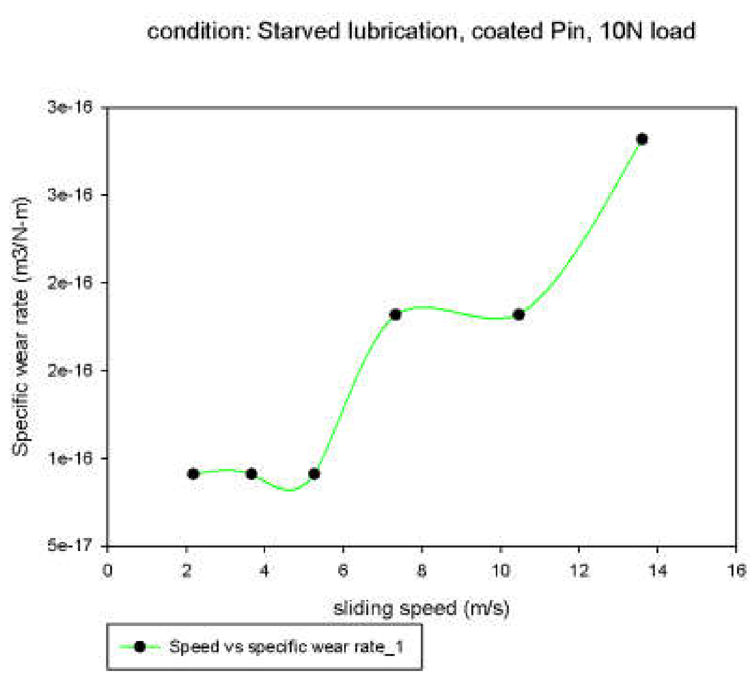
At 40 N load the specific wear rate have less value in the low speed range this may be due to the layer of lubricant between the pin and disc. As the speed is increased further to medium speed range and high speed range the thick boundary lubrication comes into play more over there may not be enough time for the surfaces to have adhesive wear at high speed.



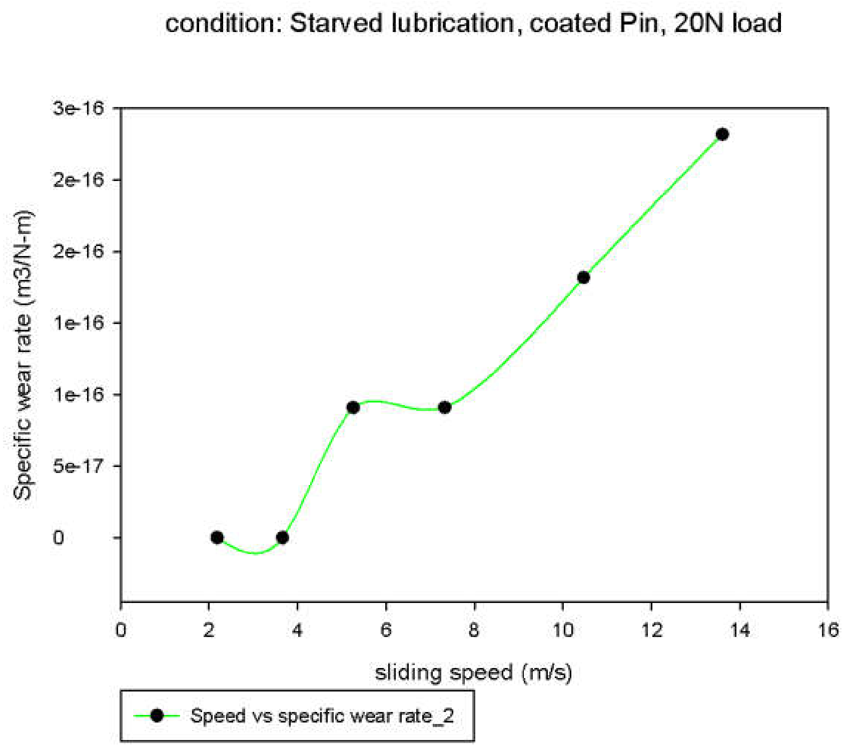
At 50 N load the specific wear rate drops as we reach from low speed range to medium speed range. The higher value of specific wear rate is due to the boundary lubrication. In the medium speed range as the thickness of boundary lubrication increased and the wear is negligible. In high speed range the specific wear rate increased due to the increased friction force and becomes almost constant because as the speed increases the amount of lubricant flowing between the two surfaces also increased.



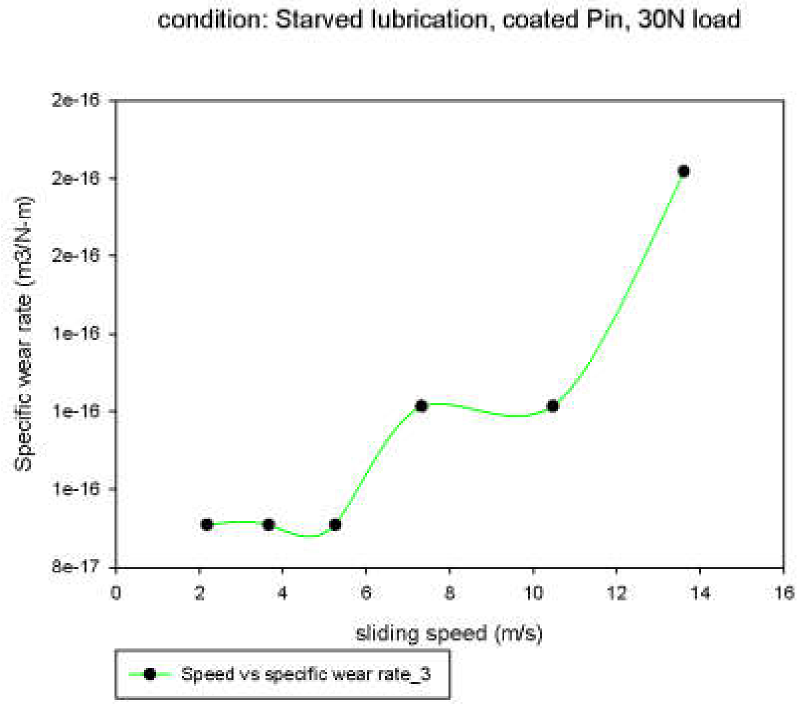
At 60 N load the specific wear decreased as we reach from the low speed range to medium speed range due to the lesser metal to metal contact at high speed between the pin and the disc. As the speed increased to high speed range the specific wear rate again increased due to the increased frictional force.



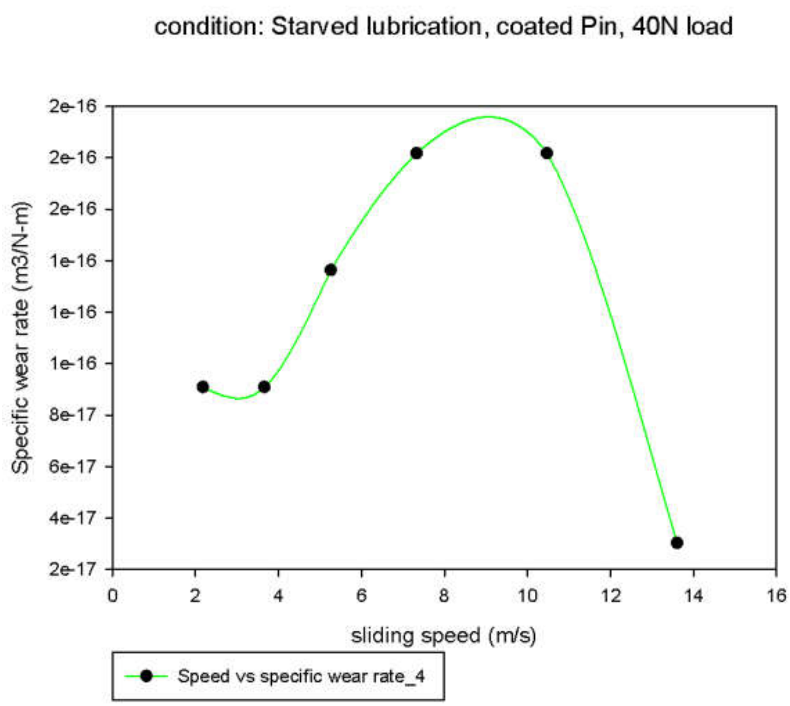
At 10 N load the specific wear has no significance in the low speed range due to the presence of boundary lubrication between the pin and disc. The specific wear rate increased as we reach from the low speed range to high speed range due to the increase in friction force.



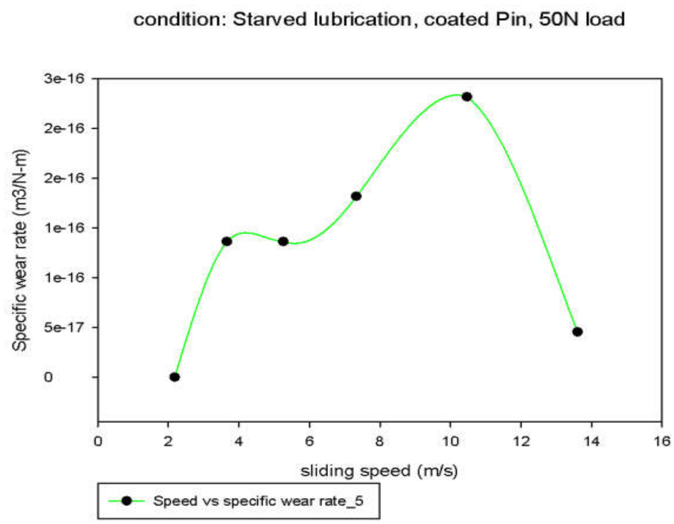
At 20 N load the specific wear has no significance in the low speed range due to the presence of boundary lubrication between the pin and disc. The specific wear rate increased as we reach from the low speed range to high speed range due to the increase in friction force.



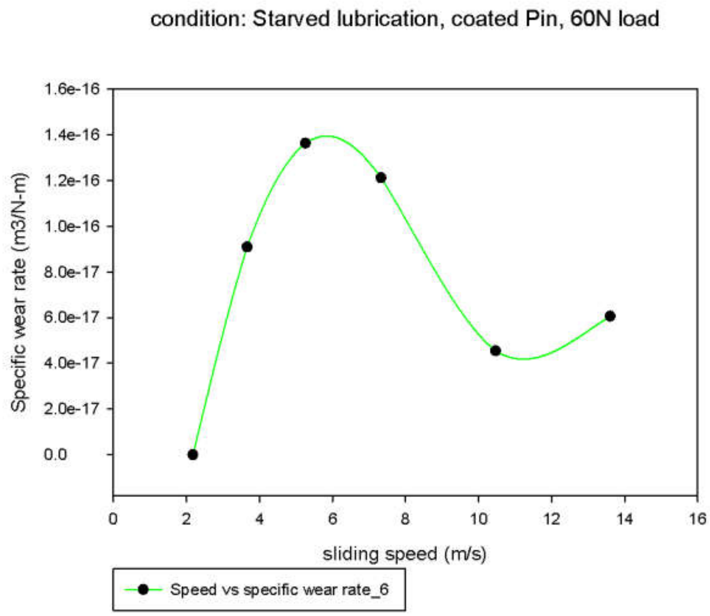
At 30 N load the specific wear rate has no significance in the low speed region as the boundary lubrication comes into play. The specific wear rate increased in the medium to high speed region due to the increased frictional force.



At 40 N load the specific wear rate have less value in the low speed range this may be due to the layer of lubricant between the pin and disc. In the medium speed range the specific wear rate increases because the lubricating layer can't prevent the metal to metal contact. As the speed is increased further to medium speed range and high speed range the thick boundary lubrication comes into play more over there may not be enough time for the surfaces to have adhesive wear at high speed.

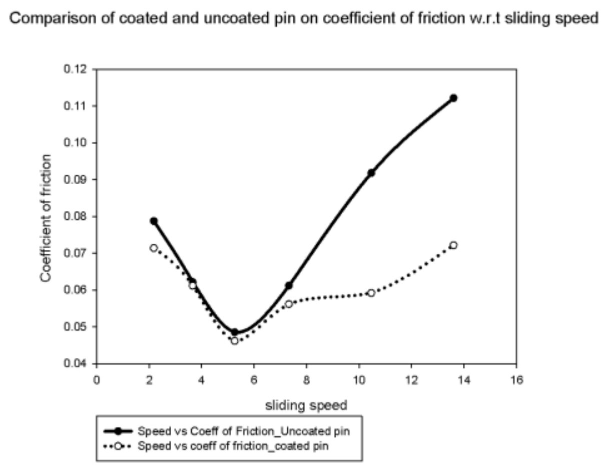


At 50 N load the specific wear rate have less value in the low speed range this may be due to the layer of lubricant between the pin and disc. In the medium speed range the specific wear rate increases because the lubricating layer can't prevent the metal to metal contact. As the speed is increased further to medium speed range and high speed range the thick boundary lubrication comes into play more over there may not be enough time for the surfaces to have adhesive wear at high speed.

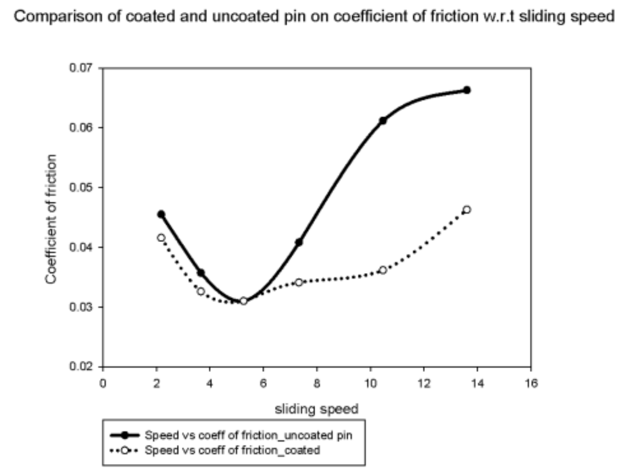


At 60 N load the specific wear rate have less value in the low speed range this may be due to the layer of lubricant between the pin and disc. In the medium speed range the specific wear rate increases at a very fast rate because the lubricating layer cant prevent the metal to metal contact. As the speed is increased further to medium speed range and high speed range the thick boundary lubrication comes into play more over there may not be enough time for the surfaces to have adhesive wear at high speed.

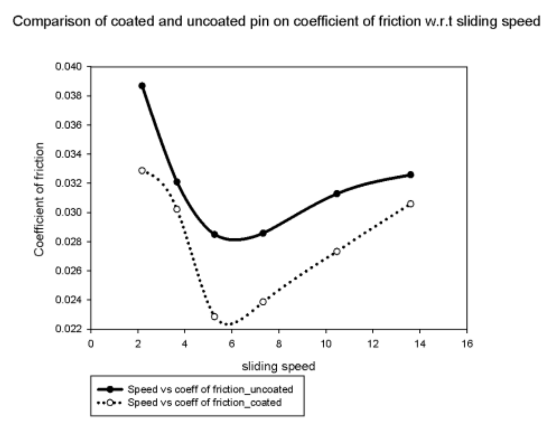
CHAPTER 5: CONCLUSION & FUTURE SCOPE



(i)



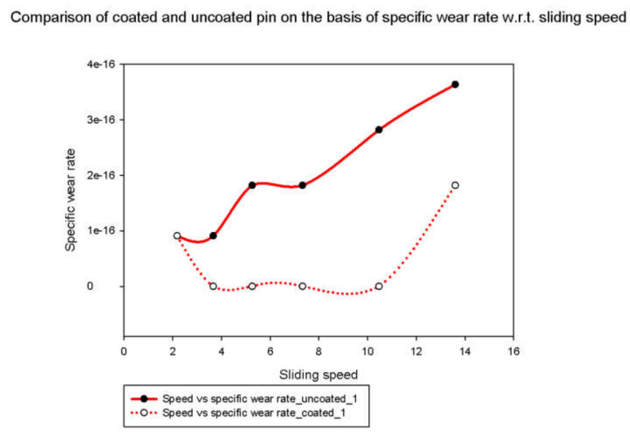
(ii)



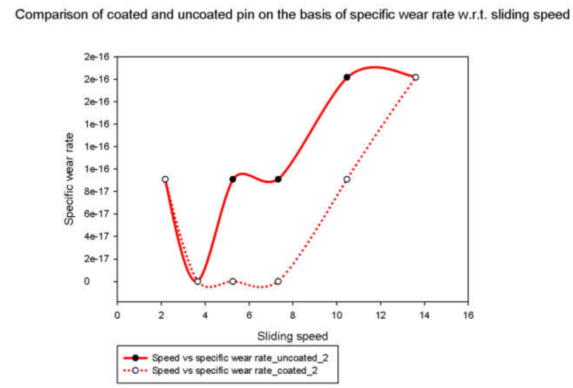
(iii)

Fig 23: Comparison of Coated and uncoated pin on coefficient of friction w.r.t sliding speed under wet lubrication and on load 10N, 20N and 60N respectively

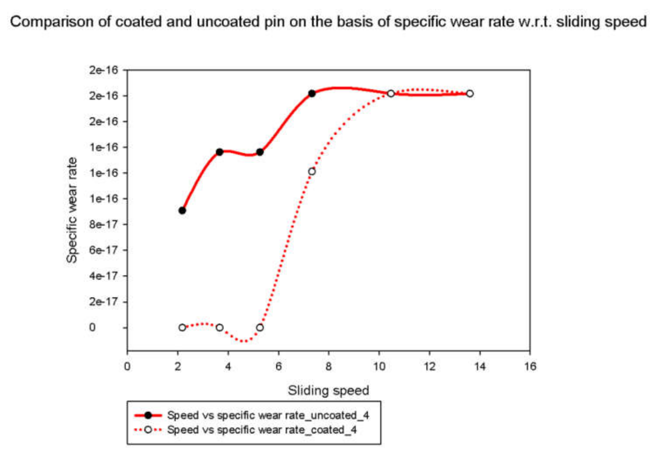
It is concluded from the above fig: , that the coefficient of friction is reduced by an considerable amount after using the chrome coated pin instead of the uncoated pin. Hence it is desirable to use the chrome plating in piston ring assembly. This will improve the service life of the piston ring as well as the cylinder liner. Maintenance cost is also reduced as no frequent replacement of the liner or piston ring is required.



(i)



(ii)



(iii)

Fig 24: Comparison of the coated and uncoated pin on the basis of the specific wear rate w.r.t. sliding speed under wet lubrication at 10N, 20N and 40N load respectively.

It is concluded from the above fig: that there is a reduction of specific wear rate for the chrome plated pin when compared with the uncoated pin. This reduction in wear rate results in the increased service life of the piston ring. Also the blow by in the internal combustion engine could also be reduced by using the chrome plating on the piston ring. The gap between the piston ring and the cylinder liner is maintained for the longer period of time. This also makes the reduction in maintenance cost.

Future Scope

It is for sure that the usage of chrome plating has resulted in reduction the friction coefficient and the specific wear rate. But still there is scope of further reduction in friction coefficient and wear rate. This can be achieved by using different coatings or the method of coating like the plasma arc spray method or the Physical vapor deposition method. The application of coating could also be beneficial for the roller bearings. As in roller bearings there is solid to solid contact, hence the reduction in friction coefficient and wear rate becomes necessary for the improved service life of the bearings.

REFERENCES

1. B. Podgornik, D. Mandrino et al., "Performance of CrN coatings under boundary lubrication", Tribology international 96 (2016), pp 247-257.

2. C Friedrich et al.,"PVD CrN coatings for tribological application on piston rings", Surface and Coatings Technology (1997), 97(1–3), pp 661-668.

3. E Broszeit et al, "Deposition, properties and applications of PVD CrN coatings", Surface and Coatings Technology (1999), 115(1), pp 9-16.

4. M.Y.P. Costa et al., "An investigation on sliding wear behavior of PVD coatings", Tribology International (2010), 43(11) pp 2196-2202.

5. B Navinšek, P Panjan, I Milošev, "Industrial applications of CrN (PVD) coatings, deposited at high and low temperatures" , Surface and Coatings Technology (1997),97(1–3), pp 182-191.

6. H.Unal, A.Mimaroglu, Kadroglu, and, H.Ekiz, "Sliding Friction and Wear Behaviour of Polytetrafluoroethylene and its Composites under Dry Conditions", Material and Design (2004), pp 239-245.

7. A. Moosavian et al., "Piston scuffing fault and its identification in an IC engine by vibration analysis", Applied Acoustics 102 (2016), pp 40-48.

8. P.C. Mishra, "A Review of Piston Compression Ring Tribology", Tribology in Industry Vol. 36, No. 3 (2014), pp 269-280.

9. A.S. Shah, D.V. Bhatt, "Experimental study to measure piston ring assembly friction of multi cylinder I.C. Engine on motorized engine test Rig: A case study", International conference on Advances in Tribology and engineering systems, springer Indian 2014, pp 60-63.

10. D.V. Bhatt, B. M. Sutaria, M. K. Bhatt, B. K. Sonigra, "ASTM Performance Tests to Study the Effect of Blend Oil with Based Oil Analysis for Automobile Engines",

International Conference on Mechanical, Production and Automobile Engineering (ICMPAE'2013).

11. P.C. Mishra, “Modeling for Friction of Four Stroke Four Cylinder In-Line Petrol Engine”, Faculty of Engineering, Vol. 35, No. 3 (2013), pp 237-245.

12. S. Ingole, A. Charanpahari et al., “Tribological behavior of nano TiO2 as an additive in base oil”, International conference on wear of materials 301 (2013), Elsevier, pp 776-785.

13. N. Morris, R.Rahmani, H.Rahnejat, P.D.King, B.Fitzsimons, “Tribology of piston compression ring conjunction under transient thermal mixed regime of lubrication”, Tribology International 59 (2013) pp 248–258.

14. M. Kapsiz, M. Durat, “Friction and wear studies between cylinder liner and piston ring pair using Taguchi design method”, Advances in Engineering Software 42 (2011) pp 595–603.

15. H.R. Mistry, D.V. Bhatt, “Study of tribological parameters on SI engine – a case study”, International Journal of Advances in Engineering & Technology, (2011), pp 6-10.

16. P.C. Mishra, S. Balakrishnan, H. Rahneja: “Tribology of compression ring-to-cylinder Contact at reversal”, Engineering Tribology (2008), pp 23-31.

17. G. A. Livanos, N.P. Kyrtatos,“Friction model of a marine diesel engine piston assembly”. Tribology International (2007), pp 42-44.

18. C.T Simon, M.L. McMillan, “Automotive tribology overview of current advances and challenges for the future”, Tribology International 37 (2004), pp 517–536.

19. H. Spikes, “Tribology research in the twenty first century”, Tribology International 34 (2001), pp 23-28.

20. L. J. Yang, “Wear coefficient of tungsten carbide against hot-work tool steel disc with two different pin settings”. International conference on wear of materials, 257(5), pp 481-495.

21. R. Novak., T. Polcar, "Tribological analysis of thin films by pin-on-disc: evaluation of friction and wear measurement uncertainty", Tribology International 74, pp 154-163.

22. M. Tului, F. Ruffini,"Some properties of atmospheric air and inert gas high-pressure plasma sprayed ZrB2 coatings". Surface and Coatings Technology, 2014 pp 483-489.

23. H. Wang, Q.S Yong,"Tribological behaviors of internal plasma sprayed TiO 2-based ceramic coating on engine cylinder under lubricated conditions", Tribology International (2016), pp 124-128.

24. A.Sivkov, I. Shanenkov, A. Pak, D. Gerasimov, "Deposition of a TiC/Ti coating with a strong substrate adhesion using a high-speed plasma jet". Surface and Coatings Technology (2016), 291, pp 1-6.

25. C. Verdon, A. Karimi."Microstructural and analytical study of thermally sprayed WC-Co coatings in connection with their wear resistance". Materials Science and Engineering (1997), 234, pp 731-734.

26. X. Shi, C. Wang et al., "Oxidation resistance of a La–Mo–Si–O–C coating prepared by Supersonic Atmosphere Plasma Spraying on the surface of SiC-coated C/C composites." Surface and Coatings Technology (2016), 300, 10-18.

27. Y. Zhang, P. Zhang.. "SiC/ZrB 2–SiC–ZrC multilayer coating for carbon/carbon composites against ablation". Surface and Coatings Technology (2016), 300, pp 1-9.

28. C.M. Taylor, Engine Tribology, Elsevier (1993)

29. T.A. Stolarski, Tribology in Machine design, Butterworth-Heinemann publication, pp 13-64.

30. E. Svenson, "DuraChrome Hard Chromium Plating", Plating Resources Inc., USA 1980,2006.