A Major Project Report
On

FINITE ELEMENT MODELING AND SIMULATION OF WEAR ON PIN ON DISC

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

MASTER OF TECHNOLOGY

IN

COMPUTATIONAL DESIGN

SUBMITTED BY

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STUDENT'S DECLARATION

I, **Pranit Kumar Singh (2k15/CDN/10),** hereby certify that the work which is being presented in this thesis entitled **"Finite Element Modeling and Simulation of Wear on Pin on Disc"** is submitted in the partial fulfilment of the requirement for 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. R. C. Singh.** The matter embodied in this thesis has not been submitted in any other University/Institute for the award of Master of Technology/certificate. Also, it has not been directly copied from any source without giving its proper reference.

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This is to certify that the report entitled, entitled **"Finite Element Modeling and Simulation of Wear on Pin on Disc"** submitted by **Pranit Kumar Singh (2k15/CDN/10)** in the partial fulfillment of the requirement for the award of **Master Of Technology Degree in the Computational Design at Delhi Technological University, Delhi** is an authentic work carried out by him under our supervision and guidance.

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ABSTRACT

The main objective of this work was to evaluate the state of stress and strain in Pin-on-Disc (POD) tribology test setup with structural steel as the tribo elements under self-mated conditions using finite element method (FEM). Structural steel is a major core and structural material in the Prototype Fast Breeder Reactor (PFBR. In PFBR there are many in-core and out-of-core component involving contact pairs undergoing sliding wear during operation as well as maintenance. Estimation of wear during operation of the machine would lead to developing appropriate wear mitigation approaches. However, measurement of in-situ wear in machine components is very difficult, if not impossible. Finite element method (FEM) based numerical modeling of machine operation with appropriate wear models would enable estimation of wear a-priority. As accuracy of calculated wear values strongly depends on the state of stress and strain in the components, accurate modelling of the state of stress and strain is essential. Apart from stress, strain, contact pressure, penetration and what type of friction is present can also be calculated. This simulation will give the exact total deformation and all other details regarding the assembly.

. The Finite Element Analysis was performed by using ANSYS 16.0 and various stresses such as maximum principal stresses, Von- Mises stresses, total deflection, total deformation, and penetration, contact pressure, occurred during working condition were evaluated. The results of the simulation depicts the stresses occurred is within the permissible limit of the piston material and the deformation is well within the tolerance limit.

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CHAPTER 1

INTRODUCTION

1. TRIBOLOGY

Tribology, the scientific name given to the study of mechanism of interacting surfaces in relative motion, is indeed one of the most basic concepts of engineering, especially of engineering design. The term tribology is derived from the Greek tribos, which means rubbing and includes all of the sciences and technologies of interacting surfaces in relative motion. The main area of analysis and application are friction, wear, and lubrication. The term itself was not coined till 1964. There are pictures of tribology in action from ancient Egypt, when early workers using the knowledge of tribology used oil to help facilitate sliding of large statues. The field is necessarily very interdisciplinary and utilizes skills sets from mechanical engineering, materials science and engineering, chemistry and chemical engineering and more. Tribology is especially vital in today's world because ample amount of energy is lost due to friction in mechanical parts. To use little energy, we need to reduce the amount of vital energy that is wasted. Significant energy is very much lost due to friction in sliding interfaces. Therefore, finding different ways to minimize friction and wear with the help of new technologies in tribology is very important to a greener and more sustainable world.

1.1. Importance of Tribology

Over the years, the governments of the foremost industrialized countries have developed in depth method to scale back energy waste. In 1966, the Jost Report demonstrated that the enormous waste of material and energy that occurred because of the misrecognition of tribology in the Britain. Similarly, in 1977, an American government-financial report suggested that US$16.25 billion might have been saved by the proper use of tribological science. Similar studies and research have also been done in other foreign countries; for example, W. Bartz carried out a research which is related to Germany between 1979 and 1983, while in 1990 P. Jost outlined the potential savings in GDP, to determine calculated values of 1.3–1.6 %.). Notably, whilst the study of W. Bartz takes under consideration only lubricated contacts, it categorizes the possibilities of saving into: (i) primary savings, because of reductions in mechanical friction; (ii) secondary savings, because a lesser exchange of machine elements because wear reduction; and (iii) tertiary savings, where a replacement material is employed for the production of new parts, thus increasing the energy content of the materials. Currently, the integration of tribology reflects a revolutionized aspect in life cycle assessment, with the main motive being environmental and economic performance. In this case, the main role of tribology is not only to reduce friction in a machine, but also to extend the machine's service life.

1.2 Basic principle of Tribology

Years of research in Tribology justifies the statement that friction and wear properties of a given material are not its intrinsic properties, but depend on many factors related to a specific application. Quantitative values for friction and wear in the forms of friction coefficient and wear rate, quoted in many engineering textbooks, depend on the following basic groups of parameters

(i) The structure of the system, i.e. its components and their relevant properties;

(ii) The operating variables, i.e. load (stress), kinematics, temperature and time;

(iii) Mutual interaction of the system's components.

1.3.TRIBOLOGICAL PROBLEM DESIGN

1.3.1. PLAIN SLIDING BEARING

When a journal bearing operates within the hydrodynamic regime of lubrication, a hydrodynamic film develops. Beneath these conditions conformal Surfaces are totally separated and a copious flow of lubricant is provided to stop heating. In these circumstances of complete separation, mechanical wear does not occur. Sometimes due to misalignment, inherently in the way the machine is assembled or of a changing nature generating from thermal as well as elastic distortion, could lead to metal-metal contact. Moreover, contact could occur at the moment of beginning (before the hydrodynamic film has had the opportunity to form totally), the bearing may be overloaded from time to time and external particles could enter the film space. In some applications, IC engines for example, acids and other corrosive substances could be generated during combustion and transmitted by the lubricant thus inducing a chemical kind of wear. The continuous application and removal of hydrodynamic pressure on the shaft could dislodge loosely held particles. In several cases, however, it’s the particles of foreign matter which are responsible for most of the wear in practical situations. Mainly, the hard particles are trapped between the journal and the bearing. Sometimes the particles are embedded in the surface of the softer material, relieving the situation. However, it is normal for the hard particles to be lapped in the bearing surface thus forming a lapping system, giving rise to wear on the hard shaft surface.

1.3.2. ROLLING CONTACT BEARING

Rolling contact bearings make up the widest class of machine elements which embody Hertzian contact problems. From a practical point of view, they are usually divided into two broad classes; ball bearings and roller-bearings, although the nature of contact and the laws governing friction and wear behavior are common to both classes. Although contact is basically a rolling one, in most cases an element of sliding is involved and this is particularly the case with certain types of roller bearings, notably the taper rolling bearings. The practicing designer will find the overwhelming number of specialized research papers devoted to rolling contact problems somewhat bewildering. He typically wishes to decide his stand regarding the relative importance of elastohydrodynamic (i.e. physical) and boundary (i.e. physicochemical) phenomena. He requires a frame of reference for the evaluation of the broad array of available contact materials and lubricants, and he will certainly appreciate information indicating what type of application is feasible for rolling contact mechanisms, at what cost, and what is beyond the current state of the art. As in most engineering applications, lubrication of a rolling Hertz contact is undertaken for two reasons: to control the friction forces and to minimize the probability of the contact's failure. With sliding elements, these two purposes are at least co- equal and friction control is often the predominant interest, but failure control is by far the most important purpose of rolling contact lubrication. It is almost universally true that lubrication, capable of providing failure- free operation of a rolling contact, will also confine the friction forces within tolerable limits. Considering failure control as the primary goal of rolling contact lubrication, a review of contact lubrication technology can be based on the interrelationship between the lubrication and the failure which renders the contact inoperative. Fortunately for the interpretive value of this treatment, considerable advances have recently been made in the analysis and understanding of several of the most important rolling contact failure mechanisms. The time is approaching when, at least for failures detected in their early stages, it will be possible to analyze a failed rolling contact and describe, in retrospect, the lubrication and contact material behavior which led to or aggravated the failure. These methods of failure analysis permit the engineer to introduce remedial design modifications to this machinery and, specifically, to improve lubrication so as to control premature or avoidable rolling contact failures. From this point of view, close correlation between lubrication theory and the failure mechanism is also an attractive goal because it can serve to verify lubrication concepts at the level where they matter in practical terms.

1.3.3. Piston, Piston rings and Cylinder liners:

One of the most common machine elements is the piston within a cylinder which normally forms part of an engine. The prime function of a piston assembly is to act as a seal and to counterbalance the action of fluid forces acting on the head of the piston. In the majority of cases the sealing action is achieved by the use of piston rings, although these are sometimes omitted in fast running hydraulic machinery finished to a high degree of precision. . On the other hand, most of the wear takes place in the vicinity of the top-dead-centre where the combination of pressure, velocity and temperature are least favorable to the operation of a hydrodynamic film. Conditions in the cylinder of an internal combustion engine can be introduction to the concept of tribodesign very corrosive due to the presence of sulphur and other harmful elements present in the fuel and oil. Corrosion can be particularly harmful before an engine has warmed up and the cylinder walls are below the 'dew-point' of the acid solution

1.3.4. Cam and cam followers:

Although elastohydrodynamic lubrication theory can now help us to understand how cam-follower contact behaves, from the point of view of its lubrication, it has not yet provided an effective design criterion. In a cam and tappet contact, friction is a relatively unimportant factor influencing the performance and its main effect is to generate unwanted heat. Therefore, the minimum attainable value is desired. The important design requirement as far as the contact is concerned is, however, that the working surfaces should support the imposed loads without serious wear or other form of surface failure. Thus it can be said that the development of cams and tappets is dominated by the need to avoid surface failure. The main design problem is to secure a film of appropriate thickness. It is known that a reduction in nose radius of a cam, which in turn increases Hertzian stress, also increases the relative velocity and thus the oil film thickness. The cam with the thicker film operates satisfactorily in service whereas the cam with the thinner film fails prematurely. Temperature limitations are likely to be important in the case of cams required to operate under intense conditions and scuffing is the most probable mode of failure. The loading conditions of cams are never steady and this fact should also be considered at the design stage.

1.3.5. Friction drive:

Friction drives, which are being used increasingly in infinitely variable gears, are the converse of hypoid gears in so far as it is the intention that two smooth machine elements should roll together without sliding, whilst being able to transmit a peripheral force from one to the other. Friction drives normally work in the elastohydrodynamic lubrication regime. If frictional traction is plotted against sliding speed, three principal modes may be identified. First, there is the linear mode in which traction is proportional to the relative velocity of sliding. Then, there is the transition mode during which a maximum is reached and, finally, a third zone with a falling characteristic. The initial region can be shown to relate to the rheological properties of the oil and viscosity is the predominant parameter. However, the fact that a maximum value is observed in the second zone is somewhat surprising. It is now believed that under appropriate circumstances a lubricant within a film, under the high pressure of the Hertzian contact, becomes a glass-like solid which, in common with other solids, has a limiting strength corresponding to the maximum value of traction. Regarding the third zone, the falling-off in traction is usually attributed to the fall in its viscosity associated with an increase in temperature of the lubricant.

1.3.6. Involute gears:

At the instant where the line of contact crosses the common tangent to the pitch circle, involute gear teeth roll one over the other without sliding. During the remaining period of interaction, i.e. when the contact zone lies in the addendum and dedendum, a certain amount of relative sliding occurs. Therefore the surface failure called pitting is most likely to be found on the pitch line, whereas scuffing is found in the addendum and dedendum regions. There is evidence that with good quality hardened gears, scuffing occurs at the point where deceleration and overload combine to produce the greatest disturbance. However, before reaching the scuffing stage, another type of damage is obtained which is located in the vicinity of the tip of both Introduction to the concept of tribodesign pinion and gear teeth. This type of damage is believed to be due to abrasion by hard debris detached from the tip wedge. There are indications of subsurface fatigue due to cyclic Hertzian stress. The growth of fatigue cracks can be related to the effect of lubricant trapped in an incipient crack during successive cycles. Because of conservative design factors, the great majority of gear systems now in use is not seriously affected by lubrication deficiency. However, in really compact designs, which require a high degree of reliability at high operating stresses, speeds or temperatures, the lubricant truly becomes an engineering material.

Over the years, a number of methods have been suggested to predict the adequate lubrication of gears. In general, they have served a design purpose but with strong limits to the gear size and operating conditions. The search has continued and, gradually, as the range of speeds and loads continues to expand, designers are moving away from the strictly empirical approach. Two concepts of defining adequate lubrication have received some popularity in recent years. One is the minimum film thickness concept; the other is the critical temperature criteria. They both have a theoretical background but their application to a mode of failure remains hypothetical.

1.3.7. Hypoid gears:

Hypoid gears are normally used in right-angle drives associated with the axles of automobiles. Tooth actions combine the rolling action characteristic of spiral-bevel gears with a degree ofsliding which makes this type of gear critical from the point of view of surface loading. Successful operation of a hypoid gear is dependent on the provision of the so-called extreme pressure oils, that is, oils containing additives which form surface protective layers at elevated temperatures. There are several types of additives for compounding hypoid lubricants. Lead-soap, active sulphur additives may prevent scuffing in drives which have not yet been run-in, particularly when the gears have not been phosphate. They are usually not satisfactory under high torque but are effective at high speed. Lead-sulphur chlorine.Tribology in machine design additives are generally satisfactory under high-torque low-speed conditions but are sometimes less so at high speeds. The prevailing modes of failure are pitting and scuffing.

1.3.8. Worm gears:

Worm gears are somewhat special because of the degree of conformity which is greater than in any other type of gear. It can be classified as a screw pair within the family of lower pairs. However, it represents a fairly critical situation in view of the very high degree of relative sliding. From the wear point of view, the only suitable combination of materials is phosphor-bronze with hardened steel. Also essential is a good surface finish and accurate, rigid positioning. Lubricants used to lubricate a worm gear usually contain surface active additives and the prevailing mode of lubrication is mixed or boundary lubrication. Therefore, the wear is mild and probably corrosive as a result of the action of boundary lubricants.

It clearly follows from the discussion presented above that the engineer responsible for the tribological aspect of design, be it bearings or other systems involving moving parts, must be expected to be able to analyze the situation with which he is confronted and bring to bear the appropriate knowledge for its solution. He must reasonably expect the information to be presented to him in such a form that he is able to see it in relation to other aspects of the subject and to assess it’s relevant to his own system. Furthermore, it is obvious that a correct appreciation of a tribological situation requires a high degree of scientific sophistication, but the same can also be said of many other aspects of modern engineering. The inclusion of the basic principles of tribology, as well as tribodesign, within an engineering design course generally does not place too great an additional burden on students, because it should call for the basic principles of the material which is required in any engineering course. For example, a study of the dynamics of fluids will allow an easy transition to the theory of hydrodynamic lubrication. Knowledge of thermodynamics and heat transfer can also be put to good use, and indeed a basic knowledge of engineering materials must be drawn upon.

1.4.1 WEAR

There are several precise definitions for wear. However, for engineering purposes the following definitions contains the essential elements.

- Wear is damage to a surface as a result of relative motion with respect to another substance. One key point is that wear is damage and it is not limited to loss of material from the surface. However, loss of material is definitely one way in which apart can experience wear.
- Another way included in this definition is by movement of material without loss of mass. An example of this would be the change of geometry or dimension of a part as a result of plastic deformation (e.g., from repeated hammering).
- There is also a third mode implied, which is damage to a surface that does not result in mass loss or dimensional changes. An example of this third mode might be development of network of cracks in a surface. This might be of significance in applications where maintaining optical transparency is a prime engineering concern. Lens and aircraft windows are examples where this is an appropriate definition of wear.

1.4.2 WEAR MECHANISM

It is their mechanisms customary to divide wear occurring in engineering practice into four broad general classes, namely: adhesive wear, surface fatigue wear, abrasive wear and chemical wear. Wear is usually associated with the loss of material from contracting bodies in relative motion. It is controlled by the properties of the material, the environmental and operating conditions and the geometry of the contacting bodies. As an additional factor influencing the wear of some materials, especially certain organic polymers, the kinematic of relative motion within the contact zone should also be mentioned. Two groups of wear mechanism can be identified; the first comprising those dominated by the mechanical behavior of materials, and the second comprising those defined by the chemical nature of the materials. In almost every situation it is possible to identify the leading wear mechanism, which is usually determined by the mechanical properties and chemical stability of the material, temperature within the contact zone, and operating conditions.

1.4.2.1. Adhesive wear

Adhesive wear is a phenomenon which occurs when two metals rub together with sufficient force to cause the removal of material from the less wear-resistant surface. This wear is dependent on physical and chemical factors such as material properties, presence of corrosive atmosphere or chemicals, as well as the dynamics such as the velocity and applied load.This phenomenon is considered corrosion by means of mechanical action rather than chemical reaction.

A number of well-defined steps leading to the formation of adhesive-wear particles can be identified: (i) deformation of the contacting asperities; (ii) removal of the surface films; (iii) formation of the adhesive junction (Fig. 2.7); (iv) failure of the junctions and transfer of material; (v) modification of transferred fragments; (vi) removal of transferred fragments and creation of loose wear particles. The volume of material removed by the adhesive-wear process can be estimated from the expression proposed by Archard

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

Where k= wear coefficient,

L= sliding distance and

H= hardness of the softer material in contact.

The adhesive wear is influenced by the following parameters characterizing the bodies in contact:

(i) electronic structure;

(ii) crystal structure;

(iii) crystal orientation;

(iv) cohesive strength;

For example, hexagonal metals, in general, are more resistant to adhesive wear than either body-centred cubic or face-centred cubic metals.

1.4.2.2. Abrasive Wear

Abrasive wear is a very common and, at the same time, very serious type of wear. It arises when two interacting surfaces are in direct physical contact, and one of them is significantly harder than the other. Under the action of a normal load, the asperities on the harder surface penetrate the softer surface thus producing plastic deformations.

The amount of material removed in this process can be estimated from the expression:

Simplified,

$$V_{\mathrm{abr}} = \frac{2}{\pi} \frac{\tan \Theta}{H} WL,$$

Refined,

$$V_{\mathrm{abr}} = n^2 \frac{P_y E W^{3/2}}{K_{\mathrm{Ic}}^2 H^{3/2}} L,$$

Where E= elastic modulus,

H= hardness of the softer material,

K= fracture toughness,

N= work-hardening factor and

P= yield strength.

1.4.2.3. Wear due to surface fatigue

Load carrying nonconforming contacts, known as Hertzian contacts, are sites of relative motion in numerous machine elements such as rolling bearings, gears, friction drives, cams and tappets. The relative motion of the surfaces in contact is composed of varying degrees of pure rolling and sliding. When the loads are not negligible, continued load cycling eventually leads to failure of the material at the contacting surfaces. The failure is attributed to multiple reversals of the contact stress field, and is therefore classified as a fatigue failure.

A number of steps leading to the generation of wear particles can be identified. They are:

(i) Transmission of stresses at contact points;

(ii) Growth of plastic deformation per cycle;

(iii) Subsurface void and crack nucleation;

(iv) Crack formation and propagation;

(v) Creation of wear particles.

For sliding contacts, the amount of material removed due to fatigue can be estimated from the expression:

$$V_f = C \frac{\eta \gamma}{\bar{\varepsilon}_1^2 H} WL.$$

Where q= distribution of asperity heights,

y= particle size constant,

El= strain to failure in one loading cycle and

H= hardness.

1.4.2.4. Wear due to chemical reactions

Wear resulting from chemical reactions induced by friction is influenced mainly by the environment and its active interaction with the materials in contact. There is a well-defined sequence of events leading to the creation of wear particles.

At the beginning, the surfaces in contact react with the environment, creating reaction products which are deposited on the surfaces. The second step involves the removal of the reaction products due to crack formation and abrasion. In this way, a parent material is again exposed to environmental attack. The friction process itself can lead to thermal and mechanical activation of the surface layers inducing the following changes :

(i) Increased reactivity due to increased temperature. As a result of that the formation of the reaction product is substantially accelerated;

(ii) Increased brittleness resulting from heavy work-hardening.

A simple model of chemical wear can be used to estimate the amount of material loss:

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

where k = velocity factor of oxidation,

d= diameter of asperity contact,

p= thickness of the reaction layer, (critical thickness)

H= hardness

1.4.3. Wear model (Archard's law)

It is postulated by Archard that the total wear volume is proportional to the real contact area times the sliding distance. A coefficient K which is proportionality constant between real contact area, sliding distance and the we are volume has been introduce,

$$V = KA_r l = Kl\frac{W}{H}$$

Where V = represents the wear volume [m^3];

K = represents the proportionality constant;

A_r= represents the real area of the contact [m^2];

W = represents the load [N];

H = represents the Vickers hardness of the softer surface [Pa];

l = represents the sliding distance [m];

The K coefficient is called as Archard coefficient, wear coefficient or wear constant. The low value of K indicates that wear is caused by only a very small proportion of small asperity contact.

The same term wear coefficient is defined by the following equation

$$k = \frac{V}{W * L} \qquad \textbf{(3-18)}$$

Where

k is the specific wear rate (wear rate) [mm^3/N-mm];

V is the wear volume [mm^3];

W is the normal load [N];

L is the sliding distance [mm];

1.4.4. Contact Mechanics

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

- There are following type of contact models:
    - Frictionless point contact
    - Point contact with Coulomb friction
    - Soft-finger contact'

1.4.4.1. Point Contact

1. Consider rigid objects A and B that make contact over region R
2. Contact pressures $\rho(x) \geq 0$ for all $x \in R$
3. If R is a planar region, with uniform friction and uniform normal, then all pressure distributions over R are equivalent to
    a. A combination of forces on convex hull of R
    b. If R is polygonal, a combination of forces on the *vertices* of the convex hull of R

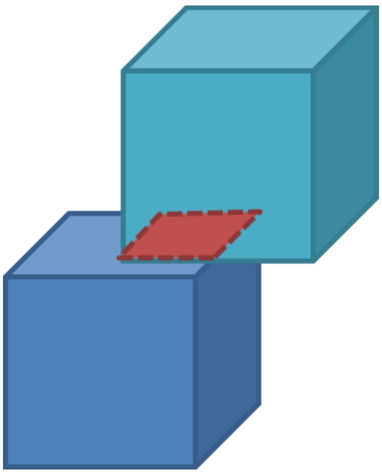


Fig1: Point Contact

1.4.4.2. Frictionless contact points :

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

- Unilateral constraint

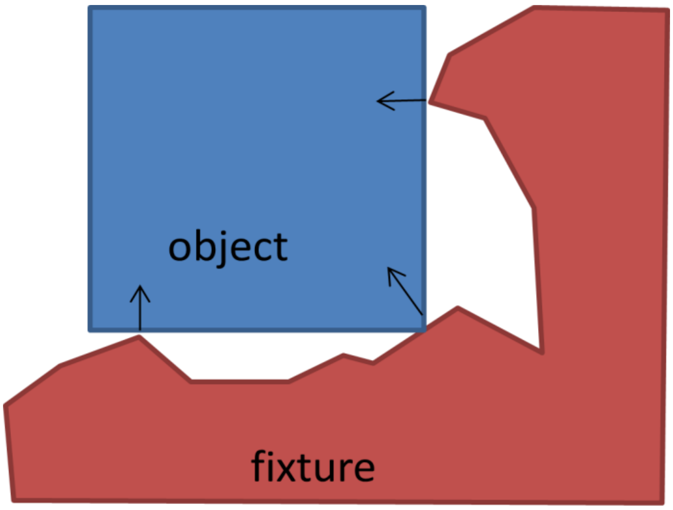


Fig2: Frictionless Point Contact

1.4.4.3 Frictionless Dynamics

- Assume body at rest
- Consider pre-contact acceleration a, angular accel $\dot{\omega}$
- Nonpenetration $n_i^T \ddot{c}_i \geq 0$ must be satisfied post-contact
- **Solve** for nonnegative contact forces f_i that alter acceleration to satisfy constraints

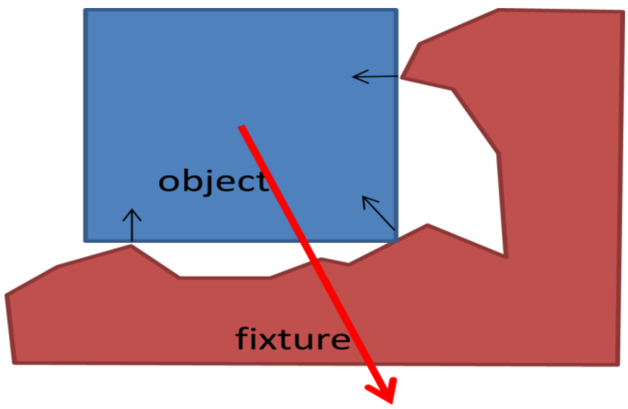


Fig3: Frictionless Dynamics

1.5. Motivation

Currently, the integration of tribology represents a modern aspect in life cycle assessment, with the main targets being environmental and economic performance. In this case, the main roles of tribology are not only to reduce friction in a machine, but also to extend the machines service life.

Tribology is especially vital in today's world because ample amount of energy is lost due to friction in mechanical parts. To use little energy, we need to reduce the amount of vital energy that is wasted. Significant energy is very much lost due to friction in sliding interfaces. Therefore, finding different ways to minimize friction and wear with the help of new technologies in tribology is very important to a greener and more sustainable world.

CHAPTER 2

LITERATURE REVIEW

Bortoleto et al. [4] presented Experimental and numerical analysis of dry contact in the pin on disc test this work presents a computational study based on the linear Archard's wear law and finite element modeling (FEM), in order to analyze un lubricated sliding wear observed in typical pin on disc tests. Such modeling was developed using finite element software ABAQUSS with 3-Ddeformable geometries and elastic–plastic material behaviour for the contact surfaces. Archard's wear model was implemented into a FORTRAN user subroutine (UMESHMOTION) in order to describe sliding wear. Modelling of debris and oxide formation mechanisms was taken into account by the use of a global wear coefficient obtained from experimental measurements.

Kennedy et al [5] carried out a research on Contact temperatures and their influence on wear during pin-on-disk tribotesting presents some of the most useful analytical and numerical methods that can be used to predict surface temperature rises in dry or boundary lubricated pin-on-disk tribotests. The development of relatively simple, accurate, and easy-to-use expressions that can be used to predict contact temperatures in pin-on-disk sliding contacts. Wear of a ceramic (zirconia), metal (stainless steel) and polymer (polyethylene) in pin- on-disc tests are carried out.

Xinmin et al. [6] uses of pin-on-disc, they studied of the tribology characteristics of sintered versus standard steel gear materials and they simulate the sliding part of gear tooth contact in boundary and mixed lubricated regions, comparing the tribological characteristics of two sintered gear materials with those of a standard gear material. The comparison considered damage mechanisms, wear, and friction between these materials in different configurations (i.e., standard versus standard, sintered versus sintered, and sintered versus standard). The results indicate that, for pairings of the same gear materials, i.e., RS–RS (16MnCr5), AQ–AQ (Distaloy AQþ0.2% C), and Mo–Mo (Astaloy 85Moþ0.2% C), RS has a lower friction coefficient. For PM and RS combinations, both PM pins have lower friction coefficients with RS disc material than do RS pins with PM disc materials. For the wear coefficient, at low and high speeds, RS pins always display better wear resistance than do AQ or Mo pins because of their high hardness and compacted microstructure. For RS–PM combinations, Mo pins display higher wear resistance than do AQ pins because their larger and more numerous pores enable good lubrication. Pins in the Mo–RS combination displayed the highest wear resistance, mainly because the pores in Mo discs hold lubricant, lubricating the contact surface and preventing adhesive wear. For the RS pin in the Mo–RS combination and the AQ pin in RS–AQ, the damage mechanism is slight adhesive wear and scuffing. For pins in the PM–PM, RS–PM, AQ–RS, and RS–RS combinations, the damage mechanism is a heavier scuffing-type adhesive wear.

Verma et al [7], Braking pad-disc system: Wear mechanisms and formation of wear fragments. The tribological phenomena that occur in brake systems are interesting even in other respects than just the braking action. One important issue, that is gaining increasing importance over the last years, concerns the environmental impact of wear debris produced by the braking action. In this context, the present study is focused on the tribological behavior of a commercial friction pad material dry sliding against a cast iron disc. Pin-on-disc tests were conducted at room temperature under mild wear conditions, as concerns load and rotating speed. The effect of some components of pad material, in particular of copper, on the dynamic of formation of tribological layer and wear debris is presented. The results obtained so, although referring to quite a simpler system that real brake systems, still may provide interesting indications, for instance in view of the development of novel brake pad materials and braking control systems.

Zmitrowicz [8] published a paper on ‘Wear patterns and laws of wear: a review’ and presented that wear is a process of gradual removal of a material from surfaces of solids subject to contact and sliding. Damages of contact surfaces are results of wear. They can have various patterns (abrasion, fatigue, ploughing, corrugation, erosion and cavitation). The results of abrasive wear are identified as irreversible changes in body contours and as evolutions of gaps between contacting solids. The wear depth profile of a surface is a useful measure of the removed material. The definition of the gap between contacting bodies takes into account deformations of bodies and evolutions of wear profiles. The wear depth can be estimated with the aid of wear laws. Derived in this study, constitutive equations of anisotropic wear are extensions of the Archard's law of wear. The equations describe abrasion of materials with microstructures. The illustrative example demonstrates calculations of the abraded mass and temperatures in pinon-disc test rig.

Guicciardi et al. [9] performed ten pin-on-disk sliding wear tests for each experimental condition were carried out with a commercial tungsten carbide (WC) pin on silicon carbide (SiC) disks in order to determine the wear and friction data dispersion. The tests were repeated using two sliding speeds (v), 0.1 and 1.0 m/s, and two applied loads (P), 5 and 50 N. The wear data showed a dispersion in the range of 28–47 and 32–56%, for disk and pin, respectively. For the disk, the dispersion decreased when increasing both sliding speed and applied load; for the pin, no clear relationship was found. The friction values spread in the range of 5–15%, with a lower dispersion at high applied load, independent of the sliding speed. From a statistical point of view, it was found that, in all the experimental conditions adopted, about 20% of the wear and friction values can be considered outliers.

PriitPodra, Soren Andersson [10] performed simulating sliding wear with finite element method; wear of components is often a critical factor influencing the product service life. Wear prediction is therefore an important part of engineering. The wear simulation approach with commercial finite element (FE) software ANSYS is presented in this paper. A modeling and simulation procedure is proposed and used with the linear wear law and the Euler integration scheme. Good care, however, must be taken to assure model validity and numerical solution convergence. A spherical pin-on-disc un lubricated steel contact was analyzed both experimentally and with FEM, and the Lim and Ashby wear map was used to identify the wear mechanism.
It was shown that the FEA wear simulation results of a given geometry and loading can be treated on the basis of wear coefficient2sliding distance change equivalence. The finite element software ANSYS is well suited for the solving of contact problems as well as the wear simulation. The actual scatter of the wear

coefficient being within the limits of ±40–60% led to considerable deviation of wear simulation results. These results must therefore be evaluated on a relative scale to compare different design options.

P. Prabhu [11] Finite element analysis usually neglects the contributions of wear and the changes in the surface due to wear. However, wear may be important in any structure subjected to repeated loadings and may be critical for certain tribological applications including the prediction of the sealing potential of surfaces. In this paper, a procedure is proposed whereby the effects of wear may be calculated and included in the overall analysis of the structure. The Archard's equation is used as the basis for calculating wear strain which is used to modify the elastic strain in an element in an explicit manner. Extensions of the theory are also proposed and an example using explicit creep for the wear adjustments is included.

Imam Syafa at al [12] performed experiment two surfaces are brought in contact, deformation takes place at asperity level. The local pressure distribution and deformation of the contacting surfaces are importance with respect to wear. This paper describes a wear model to predict the wear of rough sliding contacts. The wear model is based on the general Archard's wear equation in combination with finite element analysis (FEA). In this paper the roughness is represented by uniformly distributed spherical asperities. The proposed model, FE in combination with Archard's wear law, has proven to be a powerful tool in predicting wear of rough surfaces.

S. C. Lim [13] developed the wear map model, this paper presents a summary of the author's personal view of the development of wear-mechanism maps, culminating in the presentation of some recently proposed maps. These maps, which present wear data in a graphical manner, are able to provide a more global picture of how materials in relative motions behave when different sliding conditions are encountered; they also provide the relationships between various dominant mechanisms of wear that are observed to occur under different sliding conditions as well as the anticipated rates of wear. Some thoughts on future directions for research in this area are also presented.

Yang [14] 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.

Yucong Wang et al. [15] in their research describes a quick and reproducible bench scuffing and wear test which has utilized real piston skirts and engine cylinder bore sections with conformed contacts to determine the tribological characteristics of the sliding contact between the piston skirt and the cylinder bore counter face. Saturn 1.9 l L-4 engine parts were used for the wear and scuffing simulation tests. Two reciprocating bench test machines were used in evaluating the tribological behaviour of coated piston/bore materials. To simulate the motion of pistons against cylinder bores in engine operation, test specimens were cut from the pistons and cylinder liners so that the piston skirt conforms to the cylinder wall. In the wear and scuff bench tests, the cylinder bore specimen was stationary.

The following test conditions were used to simulate normal wear: duration- 20h; temperature- 125◦C; stroke- 6.77 mm; frequency- 10 Hz; load- 120 N; test lubricant- SAE 5W-30 and quantity- 8 ml (for wear tests only). A data acquisition system kept track of the coefficient of friction, the contact potential (or electric contact resistance), and changes in test parameters, such as the temperature.

R. Novak and T. Polcar [16] 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).

Sharma et al. [17] 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.

E.M. Bortoleto et al. [18] analyzed the T1 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

Johanssona et al. [19] work was related to the development and verification of a methodology capable of mimicking the real engine behavior at boundary and mixed lubrication regimes in order to minimize frictional losses and wear. This work focused on the development of test rig methodology and to investigate present and future candidate materials.

By using real production components (as opposed to simplified geometries) that contain the correct surface morphology, coating thickness and material composition the best component comparison to the real engine tests is made.

It was however not possible in a simplified tribometer experiment to utilize the same speeds, loads and temperatures as can be found in the engine. The objective was thus to find a parameter that can compare the contact of the top piston ring against the cylinder liner in the engine and in the tribometer.

You Wang et al. [20] evaluated the abrasive wear resistance of the ceramic coatings using diamond abrasives. They used the plasma spray technique to deposit coatings with reconstituted nanostructured Al_2O_3/TiO_2 powders. The abrasive wear tests were performed on a modified grinding/polishing machine. Abrasion rates (wear rates) are calculated using the mean measurement value of three samples in terms of the volume of material removed per unit load and distance of sliding against the diamond abrasive pad.

Both as fabricated and after wear Al_2O_3/TiO_2 coatings were investigated by X-ray diffraction, SEM and indentation tests. The abrasive wear mechanism is also discussed.

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.

Alexander Sivkov et al. [21] 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.

Tom Peat et al. [22] provides insight on the erosion-corrosion performance of HVOF deposited WC-CoCr, Cr_3C_2-NiCr and Al_2O_3 based coatings under slurry liquid impingement. Through the use of liquid impingement apparatus, results will provide an indication of the three coatings' performance under conditions reflecting a flowing environment. The mass loss as a result of erosion, corrosion and the synergy factor has been calculated for each coating in order to generate comparative data for the relative performance of each coating material under multiple angles of attack. Metallographic analysis was carried out to establish a link between coating properties and the mode of coating degradation through assessment of wear scar damage. This body of work seeks to provide a novel insight on the erosion corrosion performance of WC-CoCr, Cr_3C_2-NiCr and Al_2O_3 based coatings under slurry liquid impingement through comparative analysis of the three coatings under consistent flowing erosion-corrosion conditions.

Guo Yan Fu et al. [23] developed a new glass coating to prevent the oxidation of the 20MnSiNb structural steel. The glass coating used in this study mainly consisted of metasilicate, chromium oxides and sodium silicate binder.

They also evaluated the anti-oxidation ability by the weight changes of 20MnSiNb samples which were heated in a muffle furnace. The samples were heated from room temperature to different temperatures, such as 800 °C, 900 °C, 1000 °C, 1050 °C, 1100 °C and 1150 °C, and maintained for 60 min. Finally, the possible protection mechanism of the coating was also investigated.

C. Verdon et al. [24] 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.

Xiaohong Shi et al. [25] 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.

Cecilia Bartulia et al. [26] studied fabrication of plasma sprayed ZrB_2-SiC ceramic coatings and tested them for high temperature oxidation resistance. The particular attention was focused on ZrB_2-SiC ceramic composites. An alternative fabrication technology, based on plasma spraying, was investigated to produce near net shape components. Thick Coatings were produced by controlled atmosphere plasma spray (CAPS), over a wide range of processing pressures.
Thermogravimetric and high temperature X-ray diffraction analyses were carried out to investigate the mechanism of formation of a potentially protective layer on the surface of exposed samples.

Elizabeth Withey et al. [27] attempted to reduce the compatibility issues between Haynes 230 and the Cu-Mg-Si PCM using three plasma sprayed metal oxide coatings: ZrO_2-20 wt.%Y_2O_3, Y_2O_3, and Al_2O_3. To understand the capability of these coatings as effective barriers against PCM attack of Haynes 230, sub-sized containment vessels were used to simulate the conditions of a single cycle of the thermal energy storage system. The chosen coating materials are known to be stable oxides. Reaction of the oxide coatings with the PCM, along with penetration of the coatings by one or more elements of the PCM, was found to determine failure in the investigated coatings.

Yulei Zhang at al. [28] 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.

M. Tului et al. [29] work aims at developing a thermal spraying process, alternative to the CVD. A powder suitable for thermal spraying was prepared by means of spray dryer agglomeration. ZrB_2 powder was sprayed using a commercial plasma spray equipment called controlled atmosphere plasma spray (CAPS). The usefulness of the simplified model to optimize the spray parameters was verified. Then, a simplified model of the plasma spray process was applied to optimize the spray parameters. Finally, the tribological and electrical properties of sprayed coatings were tested.

Pengf-ei He et al. [30] presented an internal plasma spraying fabrication process for a TiO_2-based ceramic coating reinforced by 12 wt%Al_2O_3 (referred to as "TAC"). The microstructure, mechanical properties and areal surface topographies of the coating were characterized. And the reciprocating tribometer experiments were carried out to simulate engine acceleration from cruise to full power. The corresponding friction parameters of load, frequency and temperature were varied from 53 to 157N, from 35 to 45Hz, and from 85 to 197.5 °C respectively. Then, the tribological results and the wear mechanisms were analyzed and compared with uncoated cylinder barrels as a reference material (referred to as "REF"). As the objects for all tests, Lycoming 6-cylinder air-cooled horizontally-opposed aircraft piston engine IO-540-K was used.

CHAPTER 3

MATHEMATICAL MODEL

3.1 Numerical Analysis

Numerical analysis is the study of algorithms that use numerical approximation for the problems of mathematical analysis. Such problems originate generally from real-world applications of algebra, geometry and calculus, and they involve variables which vary continuously; these problems occur throughout the natural sciences, social sciences, engineering, medicine, and business.

In our problem of the frictional contact analysis, we used ANSYS to solve and solver of the software uses the solution technique as NEWTON RAPHSON technique which is Newton's method as a numerical technique for solving nonlinear equilibrium equations.

3.2. Non Linear Analysis

Nonlinear Structural Analysis is the prediction of the response of nonlinear structures by model based simulation. Simulation involves a combination of mathematical modeling, discretization methods and numerical techniques. A response diagram characterizes only the gross behavior of a structure, as it might be observed simply by conducting an experiment on a mechanical testing machine. Further insight into the source of nonlinearity is required to capture such physical behavior with mathematical and computational models for computer simulation. For structural analysis there are four sources of nonlinear behavior. The corresponding nonlinear effects are identified by the terms material, geometric, force B.C. and displacement B.C., in which B.C. means “boundary conditions.” In this course we shall be primarily concerned with the last three types of nonlinearity, with emphasis on the geometric one.

3.1.1. Causes of Nonlinear Behavior

Nonlinear structural behavior arises from a number of causes, which can be grouped into three principal categories:

i. Changing Status (Including Contact)

ii. Geometric Nonlinearities

iii. Material Nonlinearities

3.1.1.1. Changing Status (Including Contact)

Many common structural features exhibit nonlinear behavior that is *status-dependent*. For example, a tension-only cable is either slack or taut; a roller support is either in contact or not in contact. Status changes might be directly related to load (as in the case of the cable), or they might be determined by some external cause. Situations in which *contact* occurs are common to many different nonlinear applications. Contact forms a distinctive and important subset to the category of changing-status nonlinearities.

3.1.1.2. Geometric Nonlinearities

If a structure experiences large deformations, its changing geometric configuration can cause the structure to respond nonlinearly. Geometric nonlinearity is characterized by "large" displacements and/or rotations.

3.1.1.3 Material Nonlinearities

Nonlinear stress-strain relationships are a common cause of nonlinear structural behavior. Many factors can influence a material's stress-strain properties, including load history (as in elasto-plastic response), environmental conditions (such as temperature), and the amount of time that a load is applied (as in creep response).

3.1.2. Basic Information about Nonlinear Analyses

ANSYS employs the "Newton-Raphson" approach to solve nonlinear problems. In this approach, the load is subdivided into a series of load increments. The load increments can be applied over several load steps.

Before each solution, the Newton-Raphson method evaluates the out-of-balance load vector, which is the difference between the restoring forces (the loads corresponding to the element stresses) and the applied loads. The program then performs a linear solution, using the out-of-balance loads, and checks for convergence. If convergence criteria are not satisfied, the out-of-balance load vector is re-evaluated, the stiffness matrix is updated, and a new solution is obtained. This iterative procedure continues until the problem converges. A number of convergence-enhancement and recovery features, such as line search, automatic load stepping, and bisection, can be activated to help the problem to converge. If convergence cannot be achieved, then the program attempts to solve with a smaller load increment. In some nonlinear static analyses, if you use the Newton-Raphson method alone, the tangent stiffness matrix may become singular (or non-unique), causing severe convergence difficulties. Such occurrences include nonlinear buckling analyses in which the structure either collapses completely or "snaps through" to another stable configuration. For such situations, you can activate an alternative iteration scheme, the *arc-length method*, to help avoid bifurcation points and track unloading. The arc-length method causes the Newton-Raphson equilibrium iterations to converge along an *arc*, thereby often preventing divergence, even when the slope of the load vs. deflection curve becomes zero or negative. The ANSYS program gives you a number of choices when you designate convergence criteria: you can base convergence checking on forces, moments, displacements, or rotations, or on any combination of these items. Additionally, each item can have a different convergence tolerance value. For multiple-degree-of-freedom problems, you also have a choice of convergence norms.

To summarize, a nonlinear analysis is organized into three levels of operation:

- The "top" level consists of the *load steps* that you define explicitly over a "time" span (see the discussion of "time" Loads are assumed to vary linearly within load steps (for static analyses).

- Within each load step, you can direct the program to perform several solutions (*sub steps* or *time steps*) to apply the load gradually.
- At each sub step, the program will perform a number of *equilibrium iterations* to obtain a converged solution.

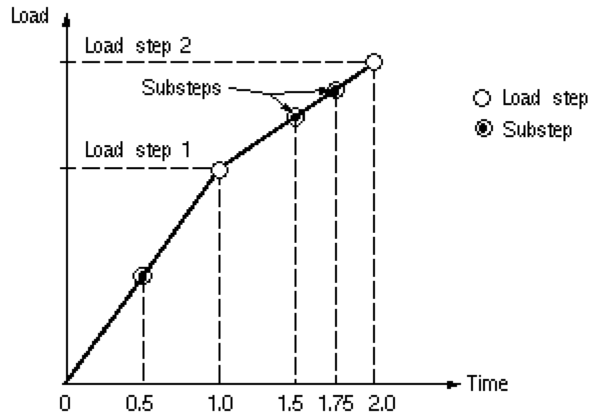


Figure: Load steps, substeps, and "time"

Sub steps:

When using multiple sub steps, you need to achieve a balance between accuracy and economy: more sub steps (that is, small time step sizes) usually result in better accuracy, but at a cost of increased run times. ANSYS provides automatic time stepping that is designed for this purpose. *Automatic time stepping* adjusts the time step size as needed, gaining a better balance between accuracy and economy. Automatic time stepping activates the ANSYS program's *bisection* feature. Bisection provides a means of automatically recovering from a convergence failure. This feature will cut a time step size in half whenever equilibrium iterations fail to converge and automatically restart from the last converged sub step. If the halved time step again fails to converge, bisection will again cut the time step size and restart, continuing the process until convergence is achieved or until the minimum time step size (specified by you) is reached.

Load and Displacement Directions:

Consider what happens to loads when your structure experiences large deflections. In many instances, the loads applied to your system maintain constant direction no matter how the structure deflects. In other cases, forces will change direction, "following" the elements as they undergo large rotations. The ANSYS program can model both situations, depending on the type of load applied. Accelerations and concentrated forces maintain their original orientation, regardless of the element orientation. Surface loads always act normal to the deflected element surface, and can be used to model "following" forces. Figure 8-7 illustrates constant-direction and following forces. Note-Nodal coordinate system orientations are not updated in a large deflection analysis. Calculated displacements are therefore output in the original directions.

Nonlinear Transient Analyses:

The procedure for analyzing nonlinear transient behavior is similar to that used for nonlinear static behavior: you apply the load in incremental steps, and the program performs equilibrium iterations at each step. The main difference between the static and transient procedures is that time-integration effects can be activated in the transient analysis. Thus, "time" always represents actual chronology in a transient analysis. The automatic time stepping and bisection feature is also applicable for transient analyses.

3.3. Contact Mechanics in ANSYS

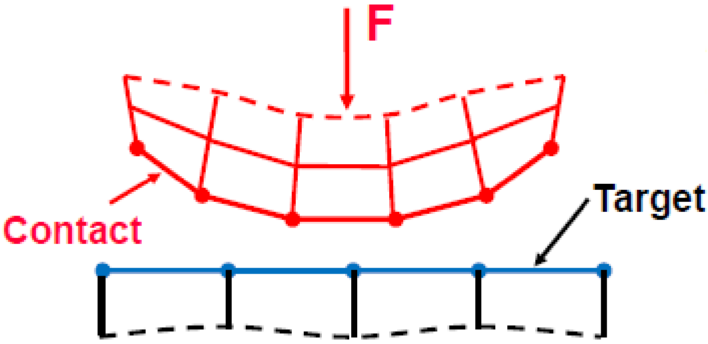
When two separate surfaces touch each other such that they become mutuall yg y tan gent, the y are said to be in contact. In the common physical sense, surfaces that are in contact have these characteristics:

- They do not interpenetrate.
- They can transmit compressive normal forces and tangential friction forces.
- They often do not transmit tensile normal forces.

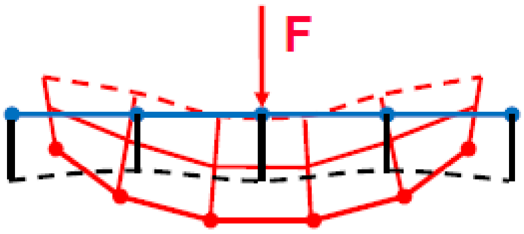
3.3.1. How compatibility is enforced in a contact region:

Physical contacting bodies do not interpenetrate. Therefore, the pg p ro gram must establish a relationship between the two surfaces to prevent them from passing through each other in the analysis.

- When the program prevents interpenetration, we say that it enforces contact compatibility
- Workbench Mechanical offers several different contact formulations to enforce compatibility at the contact interface.



Penetration occurs when contact compatibility is not enforced.



For nonlinear solid body contact of faces, *Pure Penalty* or *Augmented Lagrange* formulations can be used:

– Both of these are penalty-based contact formulations

$$F_{normal} = k_{normal} x_{penetration}$$

The main difference between *Pure Penalty* and *Augmented Lagrange* methods is that the latter augments the contact force (pressure) calculations:

Pure Penalty: $$F_{normal} = k_{normal} x_{penetration}$$

Augmented Lagrange: $$F_{normal} = k_{normal} x_{penetration} + \lambda$$

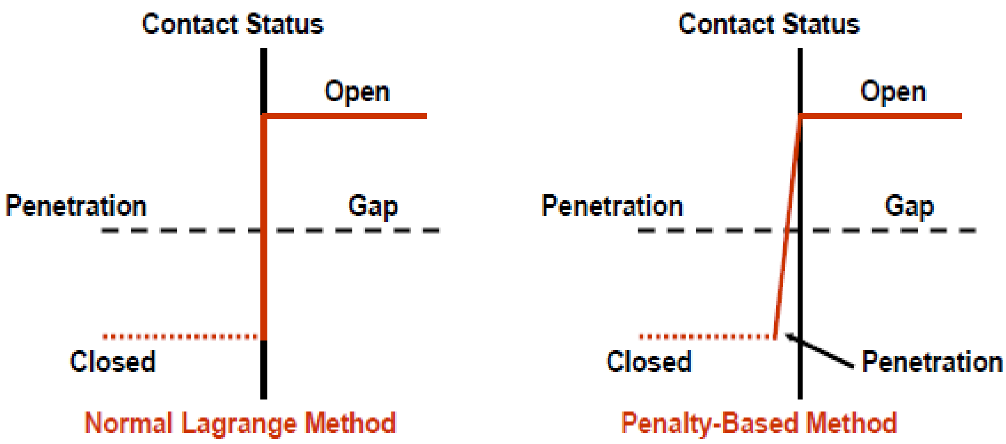
Because of the extra term λ, the augmented Lagrange method is less sensitive to the magnitude of the contact stiffness *knormal*.

3.4. Lagrange multiplier Formulation:

The Normal Lagrange Formulation adds an extra degree of freedom (contact pressure) to satisfy contact compatibility. Consequently, instead of resolving contact force as contact stiffness and penetration, contact force (contact pressure) is solved for explicitly as an extra DOF.

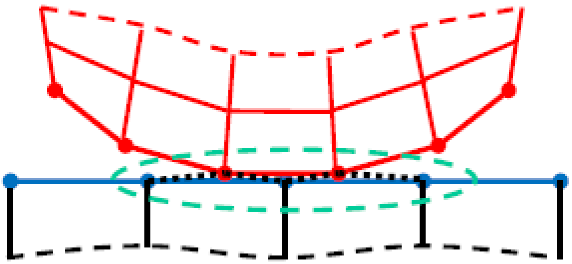
Chattering is an issue which often occurs with Normal Lagrange method:

- If no penetration is allowed (left), then the contact status is either open or closed (a step function). This can sometimes make convergence more difficult because contact points may oscillate between open/closed status. This is called *chattering*
- If some slight penetration is allowed (right), it can make it easier to converge since contact is no longer a step change.



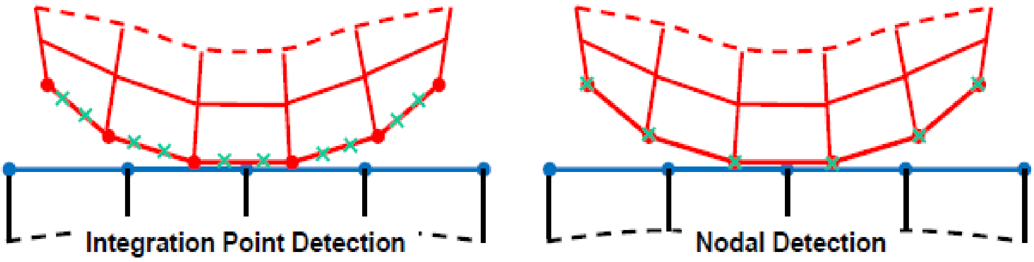
For the specific case of “Bonded” and “No Separation” type of contact between two faces, a multi-point constraint (*MPC)* formulation is available.

- MPC internally adds constraint equations to “tie” the displacements between contacting surfaces.
- This approach is not penalty-based or Lagrange multiplier-based. It is a direct, efficient way of relating surfaces of contact regions which are bonded.
- Large-deformation effects also are supported with MPC-based bonded contact.



Contact is detected differently, depending on the formulation used:

- Pure Penalty and Augmented Lagrange Formulations use integration point detection. This results in more detection points (10 in this example on left)
- Normal Lagrange and MPC Formulation use nodal detection (normal direction from Target). This results in fewer detection points (6 in the example on right)
- Nodal detection may handle contact at edges slightly better, but a localized, finer mesh will alleviate this situation with integration point detection.



A summary of the contact formulations available in Workbench Mechanical is listed below**:**

Formulation	Normal	Tangential	Normal Stiffness	Tangential Stiffness	Type
Augmented Lagrange	Augmented Lagrange	Penalty	Yes	Yes[1]	Any
Pure Penalty	Penalty	Penalty	Yes	Yes[1]	Any
MPC	MPC	MPC	-	-	Bonded,No Separation
Normal Lagrange	Lagrange Multiplier	Penalty	-	Yes[1]	Any

* 1 **Tangential stiffness is not directly input by user**

*The "Normal Lagrange" method is so named because Lagrange multiplier formulation is used in the Normal direction while penalty-based method is used in the tangential direction.

	Pure Penalty		Augmented Lagrange		Normal Lagrange		MPC
+	Good convergence behavior (few equilibrium iterations)	-	May require additional equilibrium iterations if penetration is too large	-	May require additional equilibrium iterations if chattering is present	+	Good convergence behavior (few equilibrium iterations)
-	Sensitive to selection of normal contact stiffness		Less sensitive to selection of normal contact stiffness	+	No normal contact stiffness is required	+	No normal contact stiffness is required
-	Contact penetration is present and uncontrolled		Contact penetration is present but controlled to some degree	+	Usually, penetration is near-zero	+	No penetration
+	Useful for any type of contact behavior	+	Useful for any type of contact behavior	+	Useful for any type of contact behavior	-	Only Bonded & No Separation behaviors
+	Either Iterative or Direct Solvers can be used	+	Either Iterative or Direct Solvers can be used	-	Only Direct Solver can be used	+	Either Iterative or Direc Solvers can be used
+	Symmetric or asymmetric contact available	+	Symmetric or asymmetric contact available		Asymmetric contact only		Asymmetric contact only
+	Contact detection at integration points	+	Contact detection at integration points		Contact detection at nodes		Contact detection at nodes

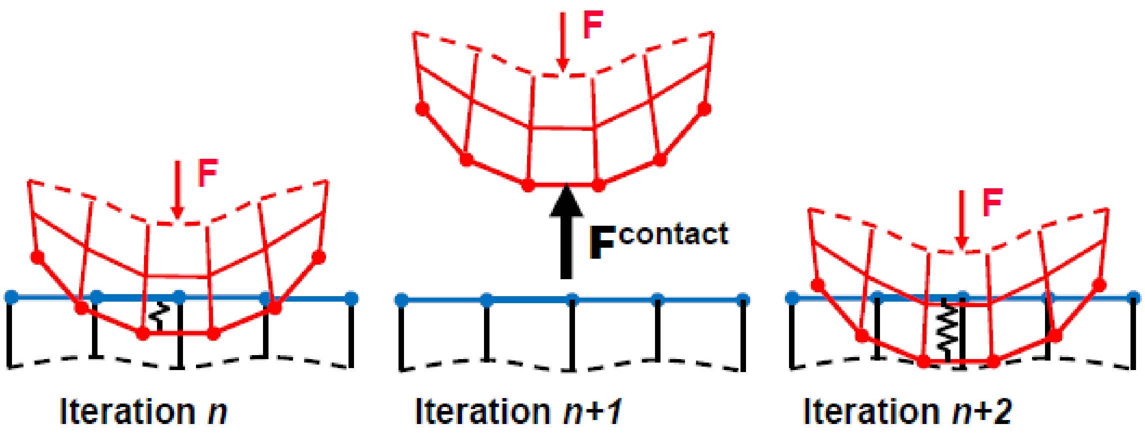
"Augmented Lagrange" is recommended for general frictionless or frictional contact in large-deformation problems.

- Augmented Lagrange formulation adds additional controls to automatically reduce penetration
- The "Normal Stiffness" is the contact stiffness knormal explained earlier, used only for "Pure Penalty" or "Augmented Lagrange"
- This is a relative factor. The use of 1.0 is recommended for general bulk deformation dominated problems. For bending-dominated situations, a smaller value of 0.1 may be useful if convergence difficulties are encountered.

The Normal Contact Stiffness knormal is the most important parameter affecting both accuracy and convergence behaviour.

- A large value of stiffness gives better accuracy, but the problem may become more difficult to convergence.

- If the contact stiffness is too large, the model may oscillate, with contacting surfaces bouncing off of each other



The default Normal Stiffness is automatically determined by WB Mechanical.

- The user may input a “Normal Stiffness Factor” (FKN) which is a multiplier on the code calculated stiffness. The lower the factor, the lower the contact stiffness.
- Default FKN =10 (for Bonded and No Separation behaviors)
- Default FKN=1.0 (for all other behaviours)

Some general guidelines on selection of Normal Stiffness for contact problems:

- For bulk-dominated problems: Use “Program Controlled” or manually enter a “Normal Stiffness Factor” of “1”.
- For bending-dominated problems: Manually enter a “Normal Stiffness Factor” of “0.01” to “0.1”.

The normal stiffness can also be automatically adjusted during the solution. If difficulties arise, the stiffness will be reduced automatically.

- “Each Iteration” sets the program to update stiffness at the end of each equilibrium iteration. This choice is recommended if you are unsure of a Normal Stiffness Factor to use in order to obtain good results.
- “Each Iteration, Aggressive” also sets the program to update stiffness at the end of each equilibrium iteration, but, this option allows for a broader range in the adjusted value.

Internally, the designation of Contact and Target surfaces can be very important:

- In WB-Mechanical, under each "Contact Region," the Contact and Target surfaces are shown. The normal's of the Contact surfaces are displayed in red while those of the Target surfaces are shown in blue.
- The Contact and Target surfaces designate which two pairs of surfaces can come into contact with one another.

By default, WB-Mechanical uses Symmetric Behavior:

This means that the Contact surfaces are constrained from penetrating the Target surfaces and the Target surfaces are constrained from penetrating the Contact surfaces.

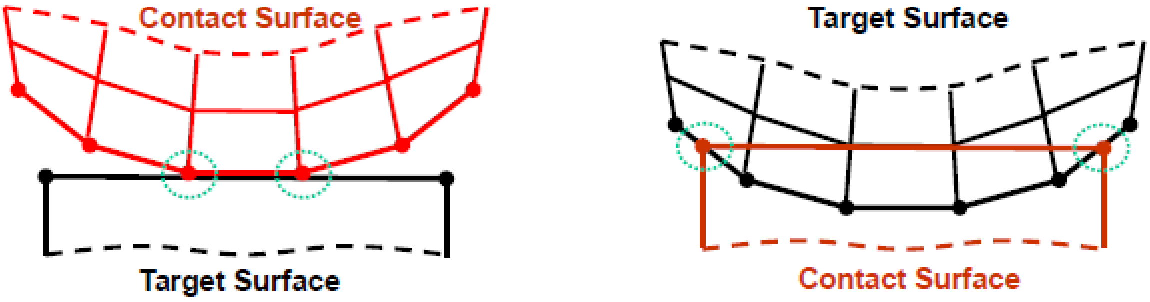
If the user wishes, Asymmetric Behavior can be used:

For Asymmetric Behavior, only the Contact surfaces are constrained from penetrating the Target surfaces.

- In Auto-Asymmetric Behavior, the Contact and Target surface designation may be reversed internally
- Although it is noted that surfaces are constrained from penetrating each other, recall that with Penalty-based methods, some small penetration may occur.

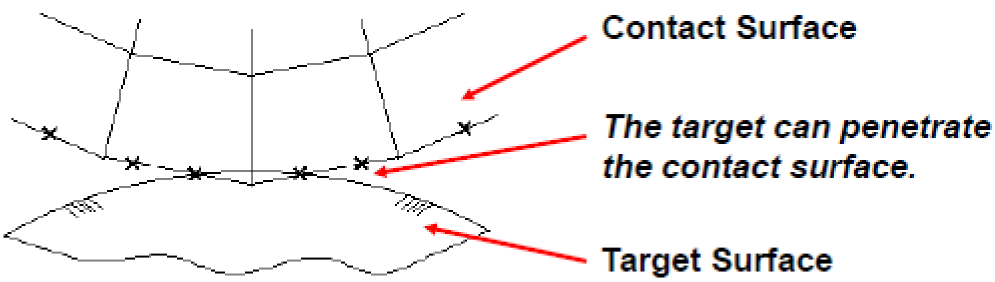
For Asymmetric Behavior, the nodes of the Contact surface cannot penetrate the Target surface. This is a very important rule to remember. Consider the following:

- On the left, the top red mesh is the mesh on the Contact side. The nodes cannot penetrate the Target surface, so contact is established correctly
- On the right, the bottom red mesh is the Contact surface whereas the top is the Target. Because the nodes of the Contact cannot penetrate the Target, too mucactual penetration occurs.



For Asymmetric Behavior, the integration point detection may allow some penetration at edges because of the location of contact detection points.

– The figure on the bottom illustrates this case:



On the other hand, there are more contact detection points if integration points are used, so each contact detection method has its pros and cons.

The following guidelines can be beneficial for proper selection of contact surfaces for Asymmetric behavior:

- If a convex surface comes into contact with a flat or concave surface, the flat or concave surface should be the Target surface.
- If one surface has a coarse mesh and the other a fine mesh, the surface with the coarse mesh should be the Target surface.
- If one surface is stiffer than the other, the stiffer surface should be the Target surface.
- If one surface is higher order and the other is lower order, the lower order surface should be the Target surface.
- If one surface is larger than the other, the larger surface should be the Target surface.

Normal Lagrange and MPC require Asymmetric Behavior.

- Because of the nature of the equations, Symmetric Behavior would be over constraining the model mathematically, so Auto-Asymmetric behavior is used when Symmetric Behavior selected.
- It is always good for the user to follow the general rules of thumb in selecting Contact and Target surfaces noted on the previous slide for any situation below where Asymmetric Behavior is used.

	Specified Option	Pure Penalty	Augmented Lagrange	Normal Lagrange	MPC
Behavior	Symmetric Behavior	Symmetric	Symmetric	Auto-Asymmetric	Auto-Asymmetric
Internally	Asymmetric Behavior	Asymmetric	Asymmetric	Asymmetric	Asymmetric
Used	Auto-Asymmetric Behavior	Auto-Asymmetric	Auto-Asymmetric	Auto-Asymmetric	Auto-Asymmetric
Reviewing	Symmetric Behavior	Results on Both	Results on Both	Results on Either	Results on Either
Results	Asymmetric Behavior	Results on Contact	Results on Contact	Results on Contact	Results on Contact
	Auto-Asymmetric Behavior	Results on Either	Results on Either	Results on Either	Results on Either
Notes	Symmetric Behavior	Easier to set up	Easier to set up	Let program designate	Let program designate
	Asymmetric Behavior	Efficiency and control	Efficiency and control	User has control	User has control
	Auto-Asymmetric Behavior	Let program designate	Let program designate	Let program designate	Let program designate

3.5. Frictional Contact

Frictional contact is available with *ANSYS.* In general, the tangential or sliding behaviour of two contacting bodies may be frictionless or involve friction.

- Frictionless behaviour allows the bodies to slide relative to one another without any resistance.
- When friction is included, shear forces can develop between the two bodies.

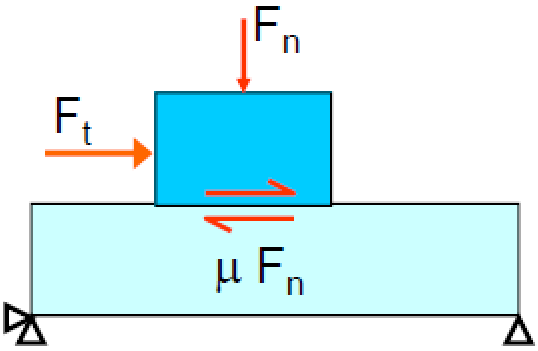
Frictional contact may be used with small-deflection or large deflection analyses:

Friction is accounted for with Coulomb's Law:

$$F_{tangential} \leq \mu \cdot F_{normal}$$

μ= Coefficient of static friction

Once the tangential force *F tangential* exceeds the above value, sliding will occur.



For frictional contact, a “friction coefficient” must be input

- A Friction Coefficient μ of 0.0 results in the same behaviour as “frictionless” contact.
- The contact formulation, as noted earlier, is recommended to be set to “Augmented Lagrange”.

If frictional contact is present, additional contact output is available

- Contact Frictional Stress and Contact Sliding Distance can be reviewed to get a better understanding of frictional effects
- For Contact Status, “Sticking” vs. “Sliding” results differentiate which contacting areas are moving.

CHAPTER 4

METHODOLOGY

ANSYS (version 16.0) is a finite element program, which is used for the simulations of the engineering real life problem. The software uses three different stages that are pre-processing, solution & post processing to solve problems. There are ten module define in this software. Pre-processing stage involves the creation of geometry and preparation of FEM model, defining Element Type, Material Property & Interaction. In Solution stage, ANSYS software generates mesh that describe the behaviour of each node and element, computes the unknown values of output field variables such as total deformation, stress-strain induced, frictional behaviour, pressure distribution, wear rate, contact pressure, etc. In the post processing stage that is Visualization. In this stage results can be analyzed and plotted.

ANSYS is a software application used for both the modelling and analysis of mechanical components and assemblies (pre-processing) and visualizing the finite element analysis result. It is a general-purpose Finite-Element analyzer that employs implicit integration scheme. This runs as a background process and does the real numerical calculations. After completion of simulation, the results can be monitored in the post processing phase. ANSYS is also used to view, plot and process the data.
There are 9 modules to perform analysis in ANSYS.

4. Pin on Disc Wear Model and Analysis

4.1. Assumptions

The general assumption that helps to simplify and reduce the simulation process of contact mechanic's on ANSYS WORKBENCH 16.0. The assumptions are start from modeling of wear analysis using software, using pin on disc model instead of contact wire and pantograph contact point.

The following are the assumptions used on this thesis paper to simulate wear analysis on ANSYS WORKBENCH 16.0.

- The assembly contact between pin and disc is frictional contact option of ANSYS without any clearance.
- The analysis conducted is a non linear.
- The analysis of modelled elements is considered as rigid to flexible contact, where one element of model taken as fixed and other as rigid and vice versa.
- The attachment for model helps to support and transfer load to analyzed model.
- The contact behaviour of modelled elements are taken as asymmetric
- The contact formulation used in Augmented Langrange.
- The stiffness value used is 0.1.
- The analysis is performed by assuming the deflection of modeled elements is small and neglected.

4.1.1. Method of Wear Simulation

To conduct wear simulation on ANSYS software the contact analysis is performed following five basic steps:

- Select whether the contact analysis is on ANSYS Parametric Design Language(APDL) or Workbench in this paper the wear simulation is conducted on ANSYS Workbench
- Create the model geometry and mesh
- Identify the contact surfaces and material used for each part of model
- Define solution options
- Apply boundary conditions and normal force and rotational velocity of disc in case of pin on disc simulation
- Solve the contact problem.

4.2. Geometric Modeling

Two solid models are to be created in this module, which are pin and disc. The assembled model of pin and disc setup will as follows.

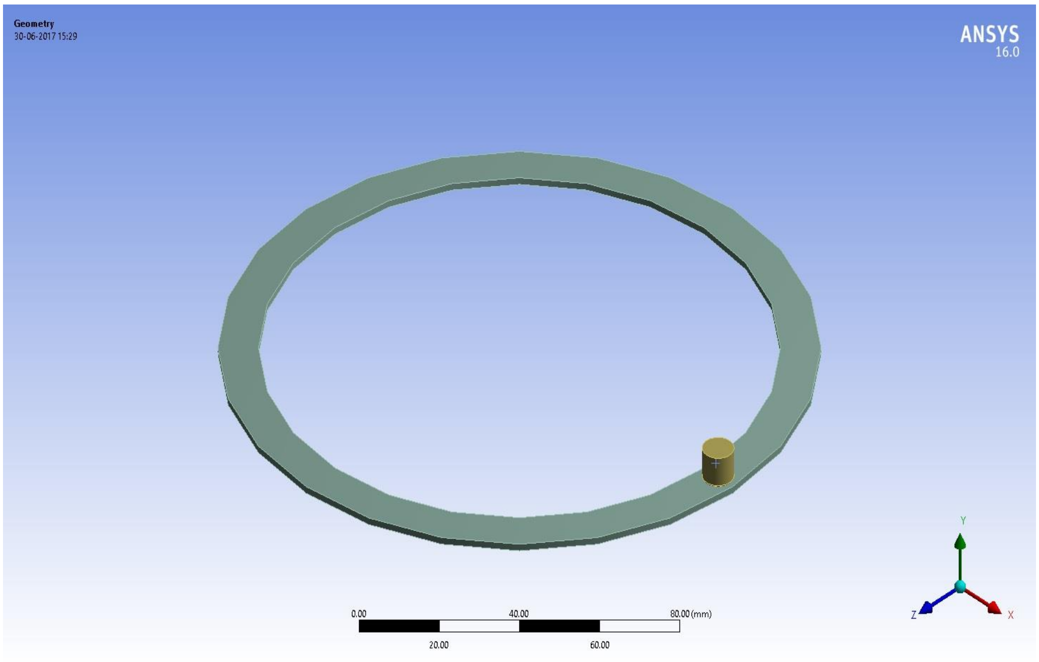


Fig: 1(i)

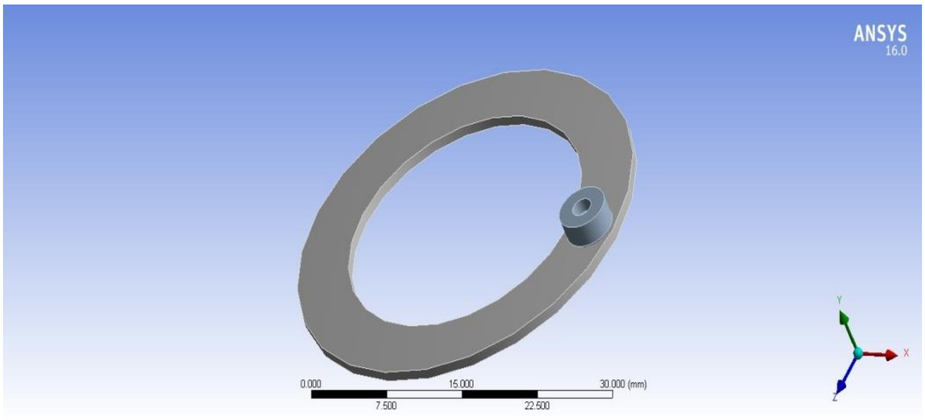


Fig: 1(ii) Modeling of pin and Disc

- **Dimension of Pin**
- Diameter: 10 mm
- Height: 32 mm
- **Dimension of Disc**
- Diameter: 165 mm
- Thickness: 10 mm

Properties of Pin and Disc materials:

Properties of material	Material properties of pin	Material properties of disc
Young's Modulus	200 GPa	200 GPa
Poisson's Ratio	0.3	0.3
Shear Modulus	76GPa	76 Gpa
Bulk Modulus	166GPa	166 Gpa
Density	7850 Kg/m^3	7850 Kg/m^3
Expansion coefficient	1.2 10^{-6} m/mK	1.2 10^{-6} m/mK

4.3. Connections

For the assembly of Pin and Disc we have two basic type of connections. The Connections are defined as different type of mating surfaces in the assembly. There are various types of connections that could be defined to stimulate the desired motion. We are having primarily two connections that required defining different contact and joint features for them.

4.3.1. Contacts:

Contacts are basically of three types

- Frictional
- Frictionless

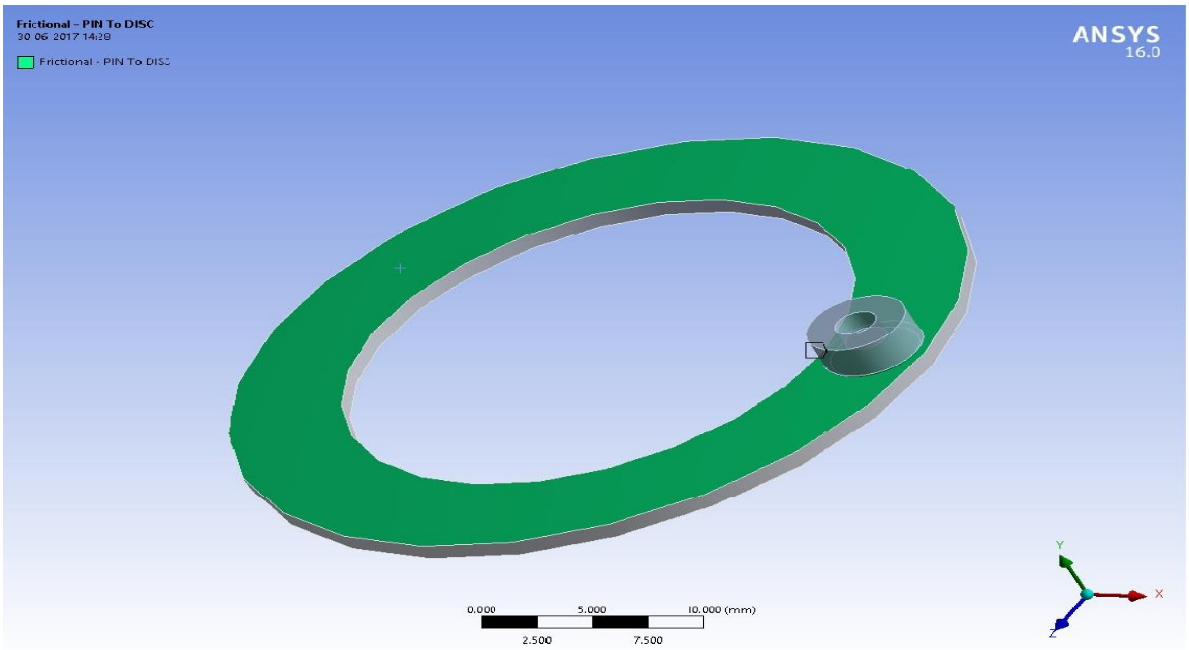
- Bonded
- Rough

We are using Frictional setting for our simulation. Generally it is guided to keep COF less than 0.3 for faster and easy convergence. In contacts we have used three different assumptions for our simulations:

- Disc is Rigid and Pin is flexible
- Pin is Rigid and Disc is flexible
- Pin and Disc both are flexible.

4.3.1.1. Disc is Rigid and Pin is flexible

Contact Type: We have considered Disc as rigid and Pin as a flexible because we want to calculate deformation on the pin itself. We have selected frictional contacts with a very low COF and asymmetric behaviour. Our detection method is nodal normal to target. We have kept trim contact to off. We are updating stiffness in aggressive manner. These all setting parameters are necessary for fast convergence.



- Fig: 2(i) Contact Define, Disc is Rigid and Pin is flexible

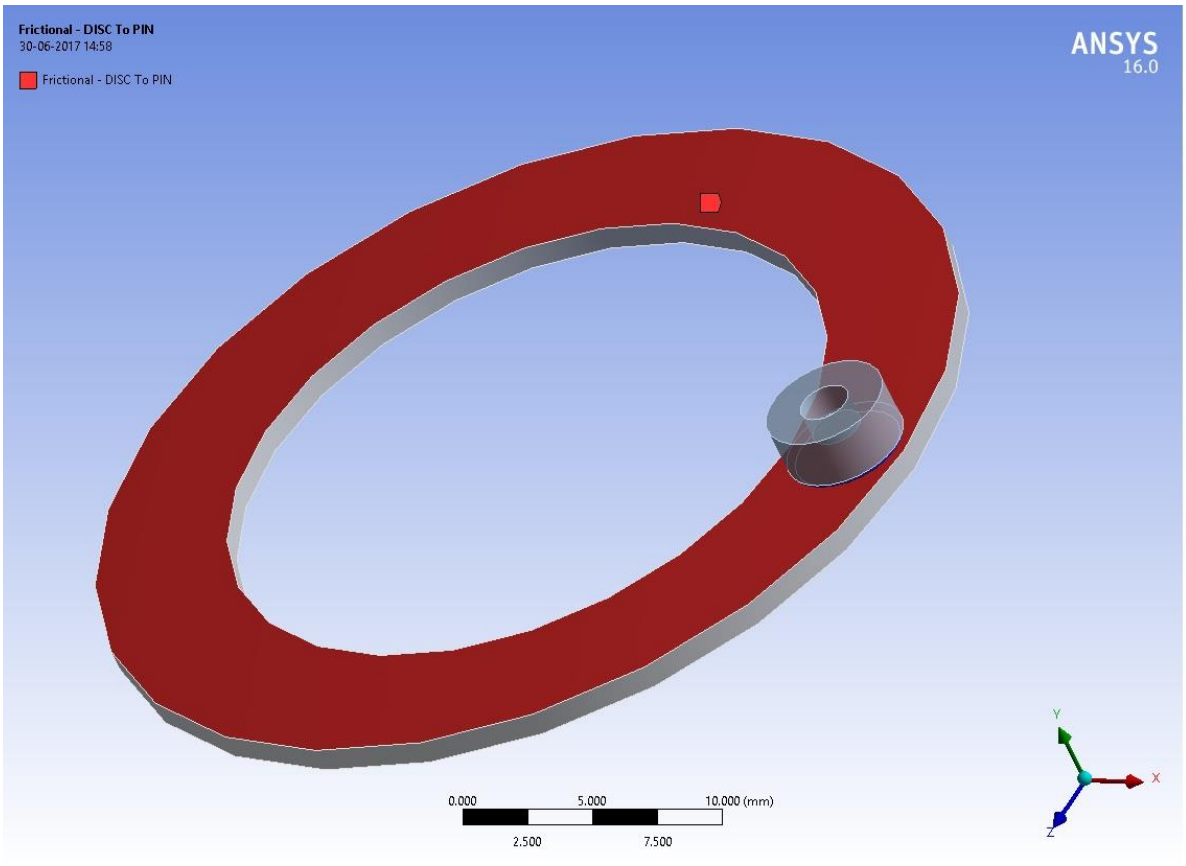


Fig: 2(ii) Contact Define, Pin is Rigid and Disc is flexible

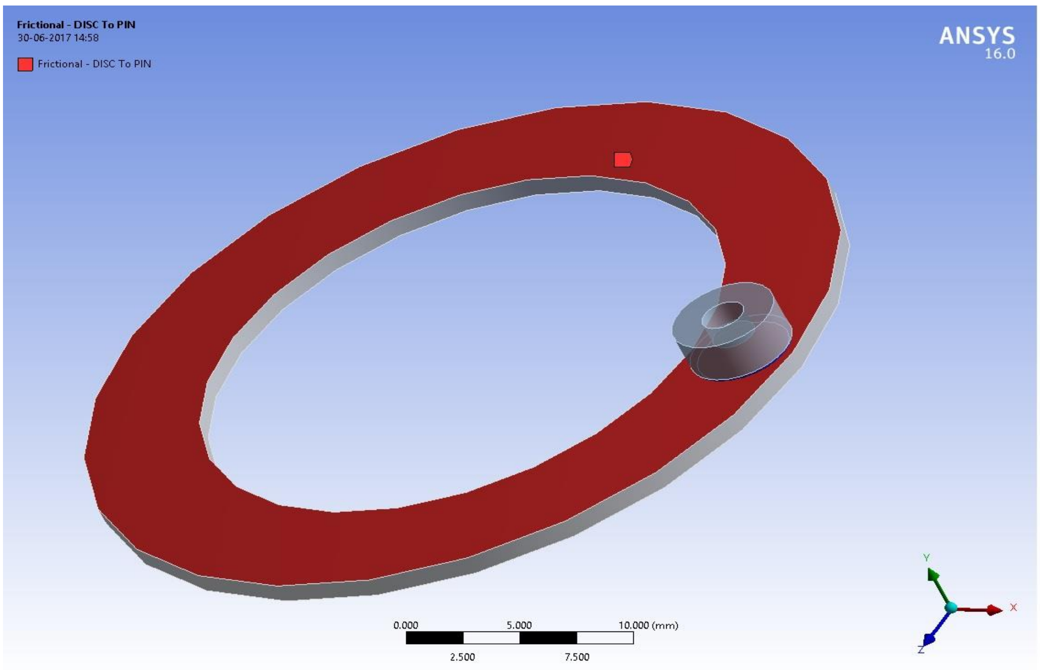


Fig: 2(iii) Contact Define, Pin and Disc both are flexible

As ANSYS 16.0 does not has any module for simulating wear mechanism, there is an imperative need to develop external program script in order to do such. The program is could be incorporated to implement the wear mechanism in ANSYS 16.0 and higher version only. This command is to be executed, for contact definition between pin and disc.

The programming langiage can be varied as oper the compatibility of ANSYS module under usage, although FORTRON, PYTHON, C++ and other programming languages are easily acceptable by ANSYS 16.0. The code script produced here for the simulation of this problem has taken PYTHON language script. Same program could be model using other programming languages too.

The script is as follows:

```
!   Commands inserted into this file will be executed just after the contact region definition.
!   The type number for the contact type is equal to the parameter "cid".
!   The type number for the target type is equal to the parameter "tid".
!   The real and mat number for the asymmetric contact pair is equal to the parameter "cid".
!   The real and mat number for the symmetric contact pair(if it exists) is equal to the parameter "tid".

!   Active UNIT system in Workbench when this object was created:  Metric (m, kg, N, s, V, A)
!   NOTE:  Any data that requires units (such as mass) is assumed to be in the consistent solver unit system.
!                See Solving Units in the help system for more information.


K=1
H=5e5 !hardness, lb/in^2
m=0.8 !exponent on contact pressure
n=1.3 !exponent on relative pressure velocity
!
TB,WEAR,cid,,,ARCD
TBDATA,1,K,H,m,n
!
```

4.4 Mesh

Mesh generation is the practice of generating a polygonal or polyhedral mesh that approximates a geometric domain. The term "grid generation" is often used interchangeably. Typical uses are for rendering to a computer screen or for physical simulation such as finite element analysis. Three-dimensional meshes created for finite element analysis need to consist of tetrahedra, pyramids, prisms or hexahedra. Those used for the finite volume method can consist of arbitrary polyhedra. Those used for finite difference methods usually need to consist of piecewise structured arrays of hexahedra known as multi-block structured meshes. A mesh is otherwise a discretization of a domain existing in one, two or three dimensions.

Structured Meshes

A structured mesh is characterized by regular connectivity that can be expressed as a two or three dimensional array. This restricts the element choices to quadrilaterals in 2D or hexahedra in 3D. The above example mesh is a structured mesh, as we could store the mesh connectivity in a 40 by 12 array. The regularity of the connectivity allows us to conserve space since neighbourhood relationships are defined by the storage arrangement. Additional classification can be made upon whether the mesh is conformal or not.

Mesh adaptation, often referred to as Adaptive Mesh Refinement (AMR), refers to the modification of an existing mesh so as to accurately capture flow features. Generally, the goal of these modifications is to improve resolution of flow features without excessive increase in computational effort. We shall discuss in brief on some of the concepts important in mesh adaptation.

Mesh adaptation strategies can usually be classified as one of three general types: r-refinement, h-refinement, or p-refinement. Combinations of these are also possible, for example hp-refinement and hr-refinement. We summarise these types of refinement below.

r- Refinement

Refinement is the modification of mesh resolution without changing the number of nodes or cells present in a mesh or the connectivity of a mesh. The increase in resolution is made by moving the grid points into regions of activity, which results in a greater clustering of points in those regions. The movement of the nodes can be controlled in various ways. On common technique is to treat the mesh as if it is an elastic solid and solve system equations (subject to some forcing) that deforms the original mesh. Care must be taken, however, that no problems due to excessive grid skewness arise.

h-Refinement

h-refinement is the modification of mesh resolution by changing the mesh connectivity. Depending upon the technique used, this may not result in a change in the overall number of grid cells or grid points. The simplest strategy for this type of refinement subdivides cells, while more complex procedures may insert or remove nodes (or cells) to change the overall mesh topology.

p-Refinement

A very popular tool in Finite Element Modelling (FEM) rather than in Finite Volume Modelling (FVM), it achieves increased resolution by increasing the order of accuracy of the polynomial in each element (or cell).

Pin and plate is taken as dependent part assembly so the meshing is to done separately for both parts. And the element type for the loading condition we choose element type for both pin and plate whose shape is hexagonal.

- **Pin-** **Element shape-** Hexagonal
  Geometric order- Linear
  Number of elements-786
- **Plate-** **Element Shape-** Hexagonal
  Geometric order- Linear
  Number of elements-1454

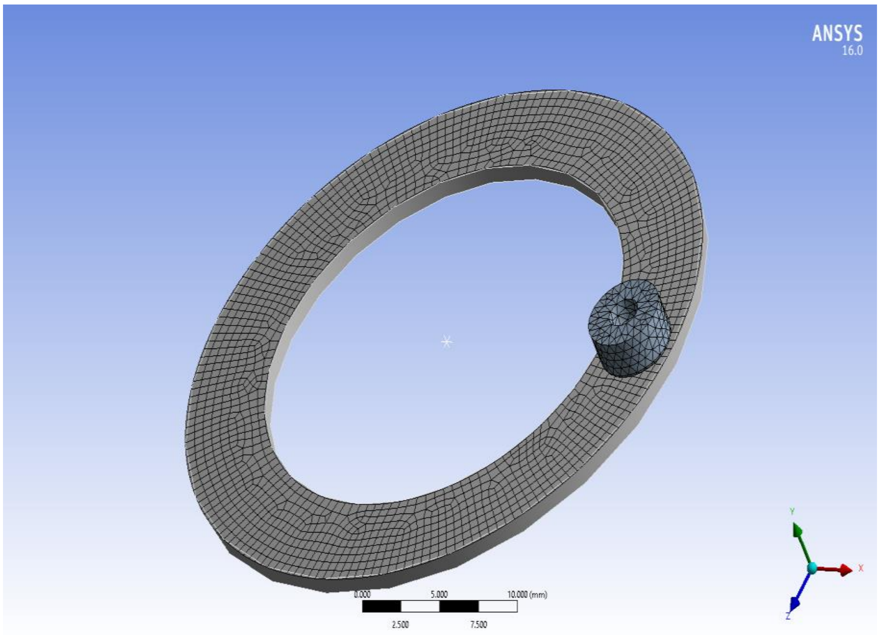


Fig: 3(i)

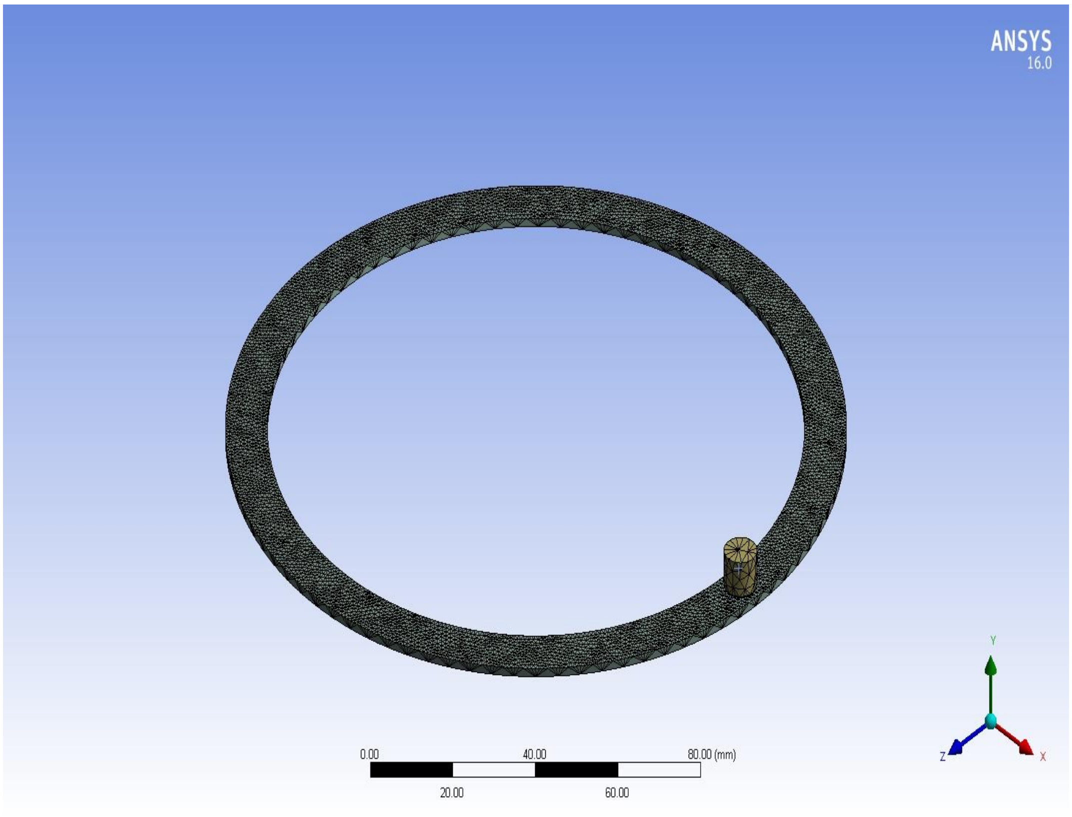


Fig: 3(ii) Mesh

4.5. Analysis Setting

Sub stepping is done by giving number of steps. For each time steps load, rotational velocity can be varied. Initial time step is set to 0.1 sec and minimum time step depends upon the user itself. Smaller the minimum time step, larger the number of iteration. Similarly RPM can be varied.

4.6 Load and Boundary Condition

4.6.1. Boundary condition

The boundary condition is attained for getting accurate and compatible to real conditions. Here the displacement in the X and Z direction of the outer round faces of the pin arrested, so given the value as zero. At the same time, Y direction displacement given as free, for allowing the movement of pin only in the Y direction. Then displacement in the X and Z direction of the outer round faces of the hollow disc made free at the same time, Y direction displacement is arrested, so given the value as zero, for allowing the movement of hollow disc in the X and

Z direction. Now the wear simulation of pin on disc displacement effect is analyzed.

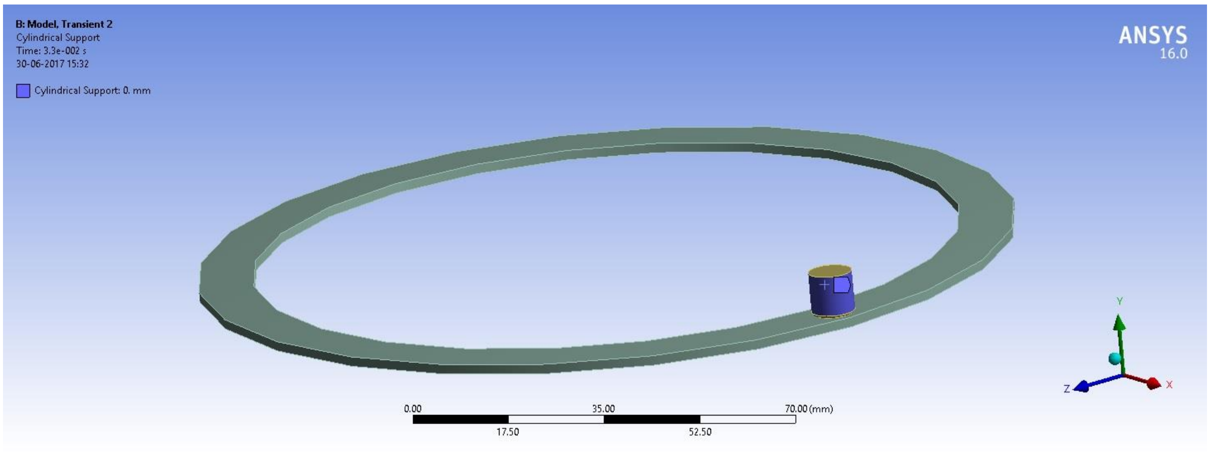


Fig 4: Cylindrical Support

For simulation purpose a load 10N, 20N and 30N, and on force is applied on top surface of pin holder. Force 10N is taken as the lowest possible tension load condition. The above three load conditions are applied on pin and the analysis continues.

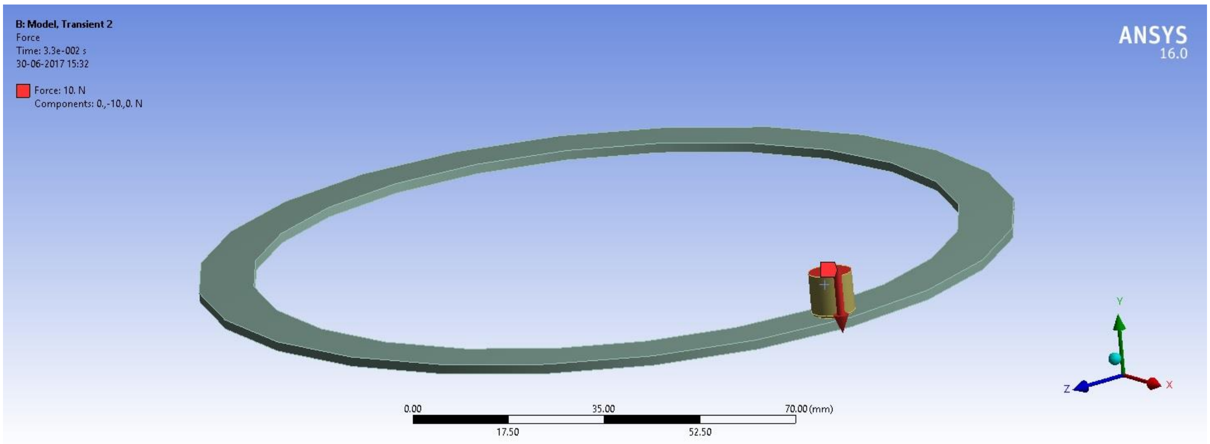


Fig 5: Applying Load

4.6.2. Rotational Velocity

The rotational velocity is applied on disc model, the Tribometer device disc rotate with a specified rotational speed which given by the electric motor installed on it. The rotational velocity applied on the model is value that used by Tribometer.

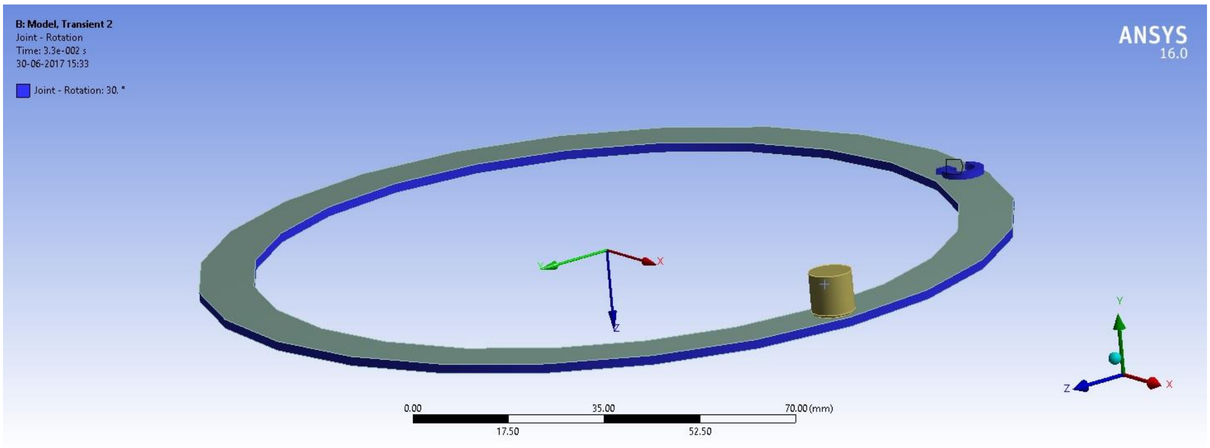


Fig 6: Applying Rotational Velocity

4.7. Calculation of Eroded Volume

For calculating actual lost volume of material a script is required to be fed in solution modular. The script is as follows:

```
!   Commands inserted into this file will be executed immediately after the ANSYS /POST1
command.

!   Active UNIT system in Workbench when this object was created:  Metric (mm, kg, N, s,
mV, mA)
!   NOTE:  Any data that requires units (such as mass) is assumed to be in the consistent
solver unit system.
!                See Solving Units in the help system for more information.


/post1
set,last,last
esel,s,mat,,2
etable,a,volu
ssum
*get,my_volume,ssum,,item,a
alls
```

CHAPTER 5

RESULTS AND DISCUSSION

The wear model and analysis is performed using finite element method with software ANSYS Workbench that consist of a transient structural (ANSYS); for purpose of getting the maximum equivalent stress, total deformation, pressure, frictional stress, wear, penetration etc. On this model the location and depth highly stressed zone also determined with implication of red color area. On same boundary conditions and different load applications used to perform the analysis where the model geometry is same as stated above on pin on disc simulations. Model is for performing the analysis and gets the solution of modeled geometry. The boundary conditions, contact behavior of system, and applying different load cases also performed on this part of FEM analysis. Final stage of analysis on FEM is gating the maximum equivalent stress and maximum equivalent stress locations on pin on disc model.

5.1. Total Deformation

Using the ANSYS Workbench software, along with the boundary conditions, applied loads is given then we got a value of total deformation on Pin for this load. Then the maximum and the minimum value is noted and the portion where this value is obtained, i.e. some portion of the pin showing the maximum deformation. In real condition that portion is worn out and it taken as wear depth.

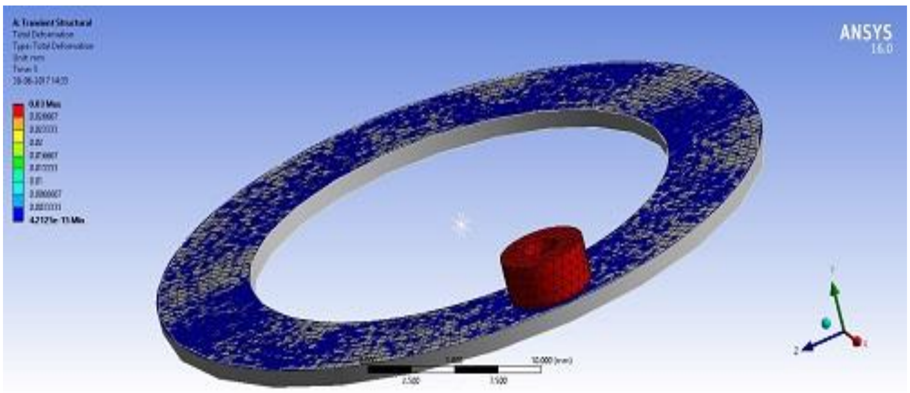


Fig: 7(i) Deformation when Disc is Rigid, Pin is Flexible

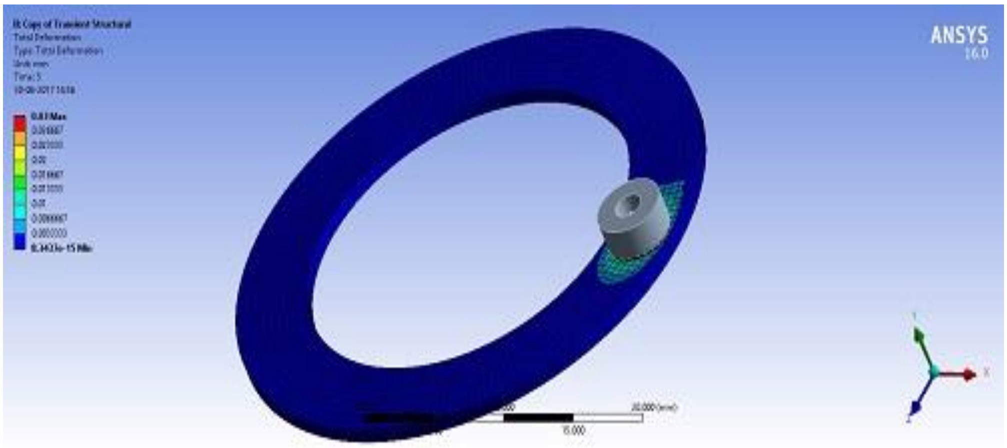


Fig: 7(ii) Deformation when, Disc is Flexible, Pin is Rigid

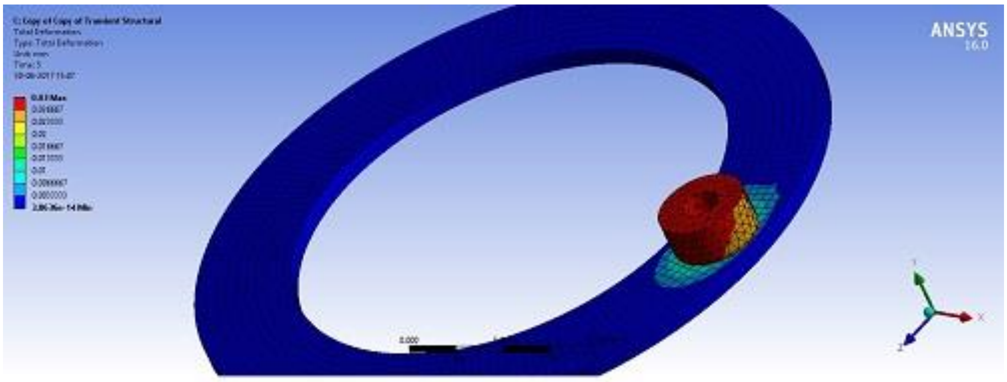


Fig: 7(iii) Deformation when, Both Flexible

As shown in fig 5(a) deformation is found only on pin as the disk is rigid. In the other case as shown in fig5 (b) deformation is found only disc as pin is rigid. In the case of when both pin and disc are flexible, maximum deformation is obtained in pin itself.

5.2. Equivalent (Von-Mises) stress

Using the ANSYS Workbench software, along with the boundary conditions, applied loads is given then we got a value for this load, the values equivalent (von-Mises) stress. Von Mises stress is widely used by designers, to check whether their design will withstand given load condition and it arises from distortion energy failure theory. Then the maximum and the minimum value is noted and the portion where it's value is obtained, i.e. some portion of the pin showing the maximum equivalent stress. In real condition that portion is worn out and it taken as wear depth.

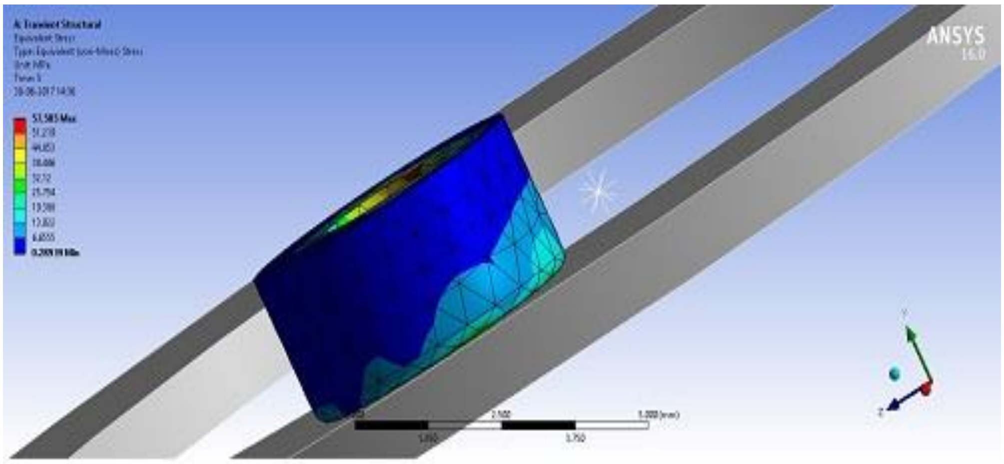


Fig: 8(i) Equivalent (Von Mises) Stress, when Disc is Rigid, Pin is Flexible

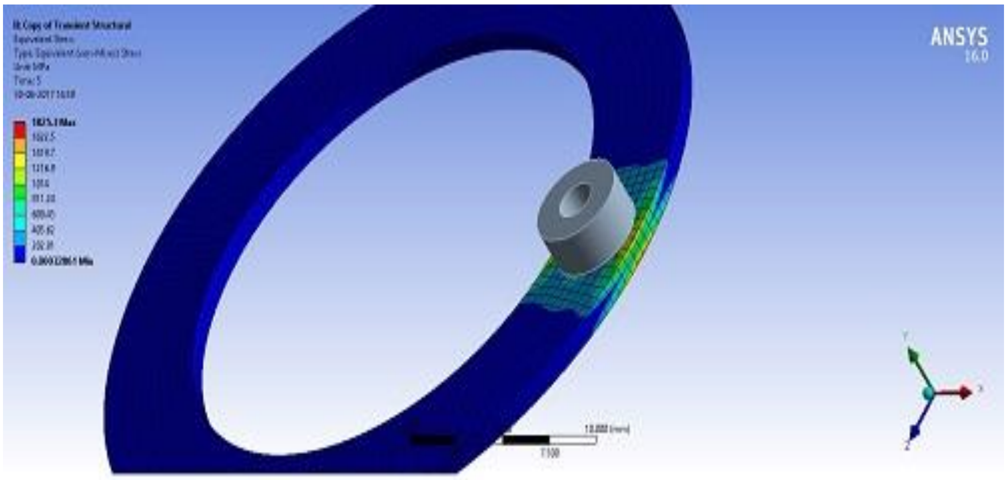


Fig: 8(ii) Equivalent (Von Mises) Stress, when Disc is Flexible, Pin is Rigid

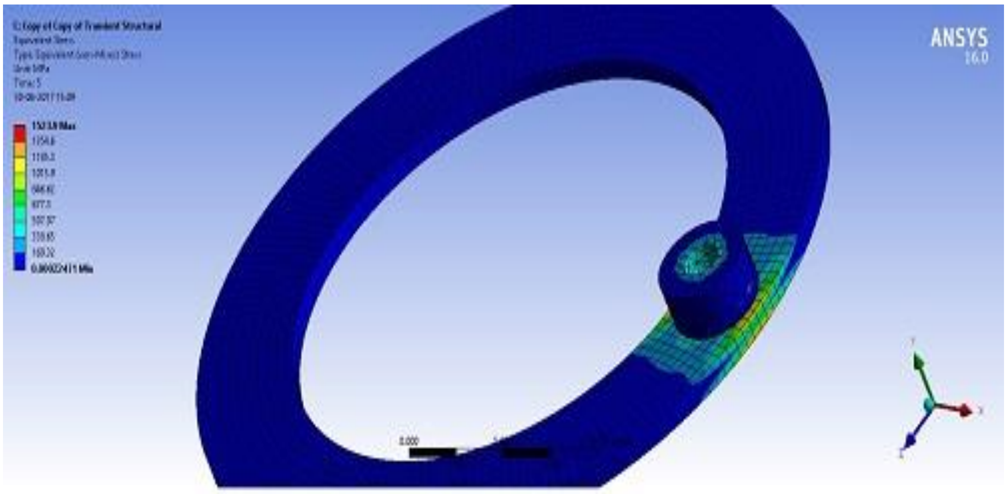


Fig: 8(iii) Equivalent (Von Mises) Stress, Both Flexible

In the fig 5(d) it is seen maximum equivalent stress on pin that to on part of pin which is attached to the disc. Stress varies from maxima at the interface of pin and disc to minima at the upper portion of the pin. In the case where disc is flexible we get maximum stress at the disc plane. But when both are flexible as shown in fig 5(f) It can inferred that stress is generated maximum at the interface of the pin and disc. Stress is generated in the pin up to some length and on the surface of disc itself. This shows that area under contact which varies throughout rotation stress is maximum at that area only. As disc surface area is more it will not be under any danger. So bottom portion of pin is the most vulnerable section of our assembly.

5.3. Contact Pressure

The values of contact pressure are obtained The value of contact pressure is recorded for every step to get the wear depth. It can be seen that the magnitude of contact pressure is increasing by increasing the load at the constant speed.

The contact pressure distribution at the disc surface, showing a non-uniform stress field under the flat pin surface contact. Higher pressure values are observed along the edge of the contact of the pin with the disc, which can lead to local wear and plastic deformation effects. n

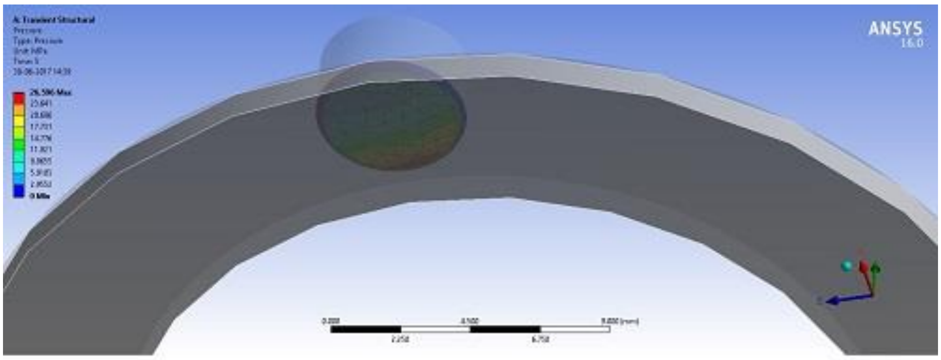


Fig: 9(i) Contact Pressure, when Disc is Rigid, Pin is Flexible

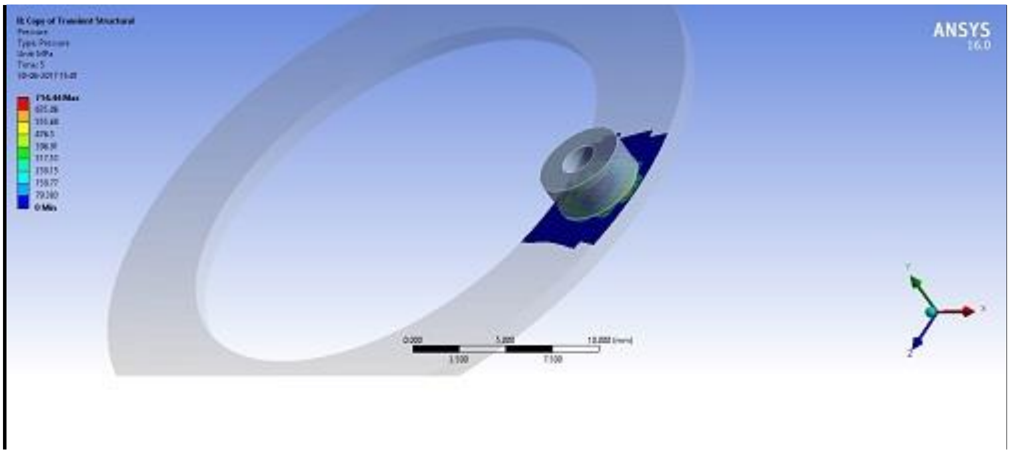


Fig: 9(ii) Contact Pressure, when Disc is Flexible, Pin is Rigid

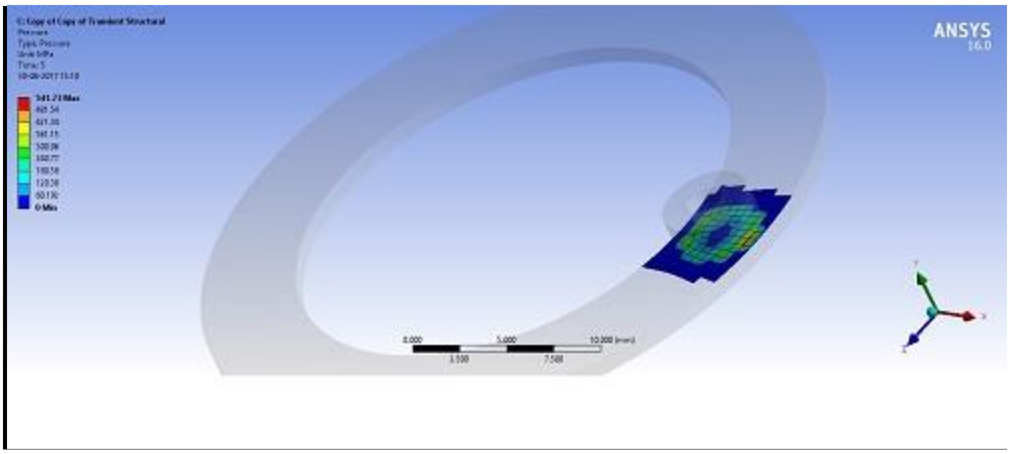


Fig: 9(iii) Contact Pressure, Both Flexible

It concluded from the above figures that contact pressure is maximum at the interface. In the case when the disk is rigid bottom face of pin has maximum contact pressure. But in the case where the pin is rigid the disc portion comes under max contact pressure. In the case of both being flexible it is evident that contact pressure is maximum at the interface of pin and disc.

Because of that pin comes under heavy duress and maximum deformation occurs at pin only.

5.4. Frictional Stress

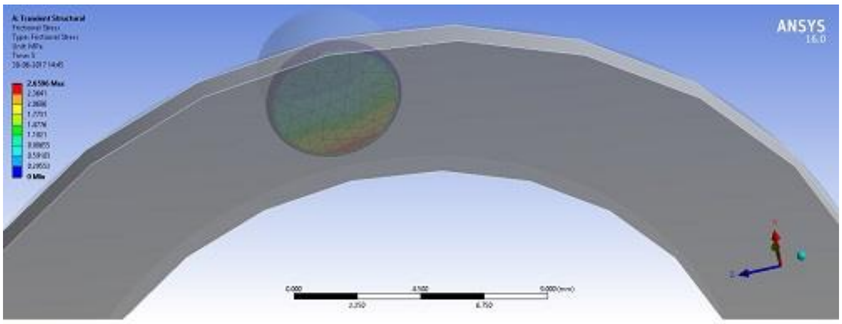


Fig: 10(i) Frictional Stress, when Disc is Rigid, Pin is Flexible

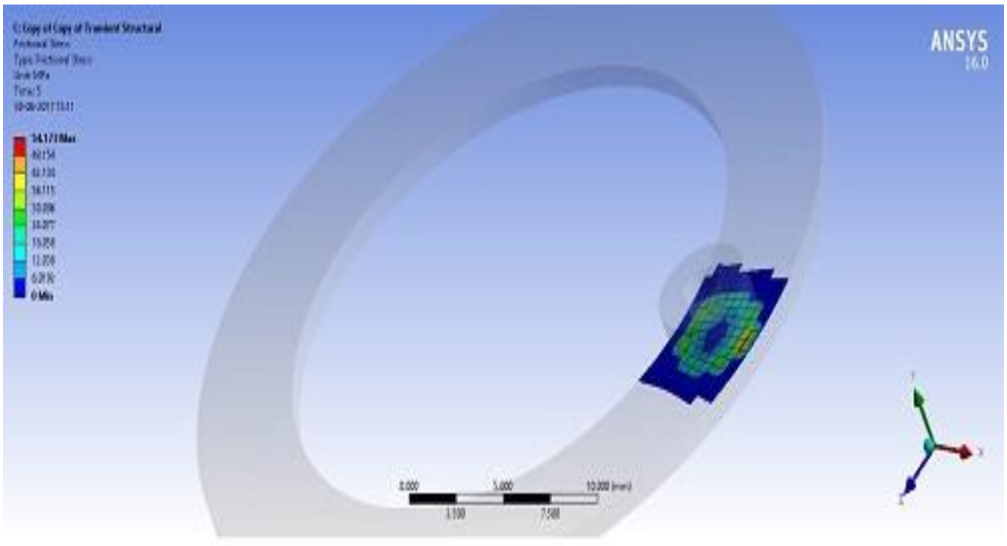


Fig: 10(ii) Frictional Stress, Both Flexible

It concluded from the above figures that frictional stress is maximum at the interface. In the case when the disk is rigid bottom face of pin has maximum frictional stress. But in the case where the pin is rigid the disc portion comes under max frictional stress. In the case of both being flexible it is evident that frictional stress is maximum at the interface of pin and disc.

Because of that pin comes under heavy duress and maximum deformation occurs at pin only.

5.5. Sliding Distance

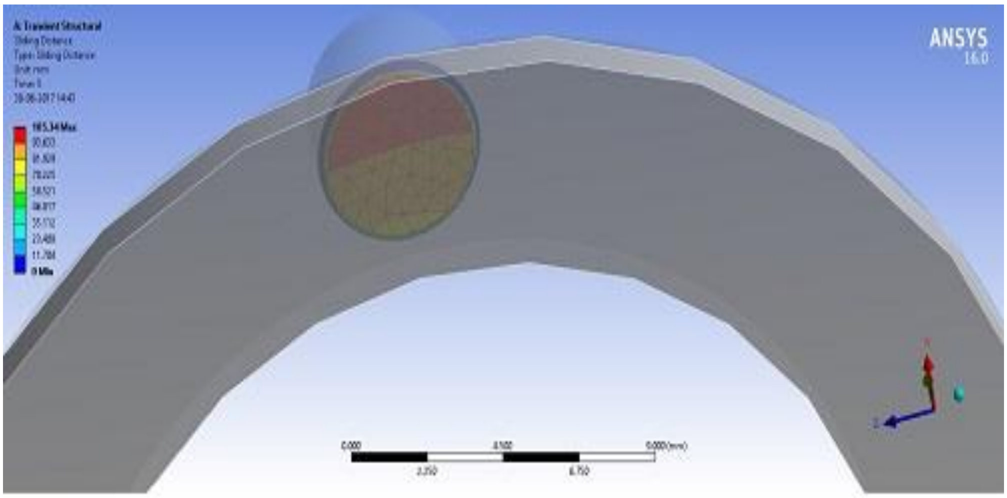


Fig: 11(i) Sliding Distance, when Disc is Rigid, Pin is Flexible

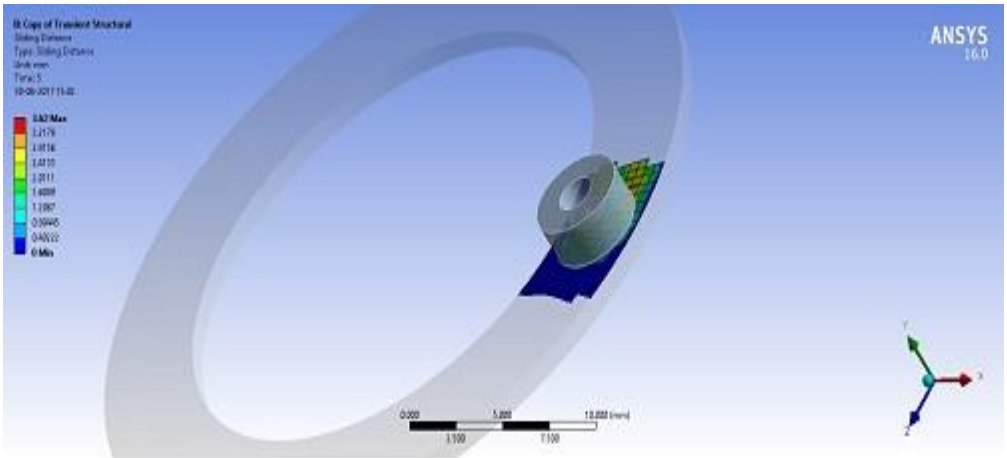


Fig: 11(ii) Sliding Distance, when Disc is Flexible, Pin is Rigid

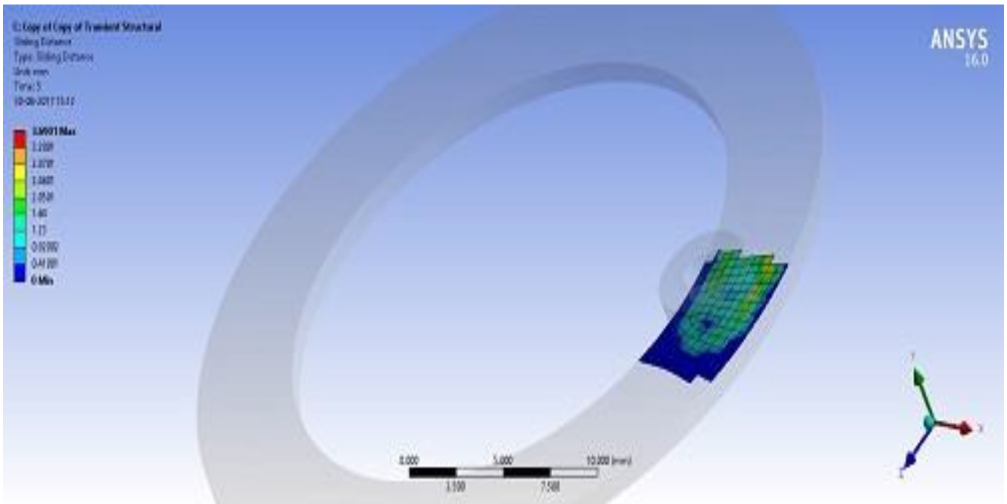


Fig: 11(iii) Sliding Distance, Both Flexible

5.6. Penetration

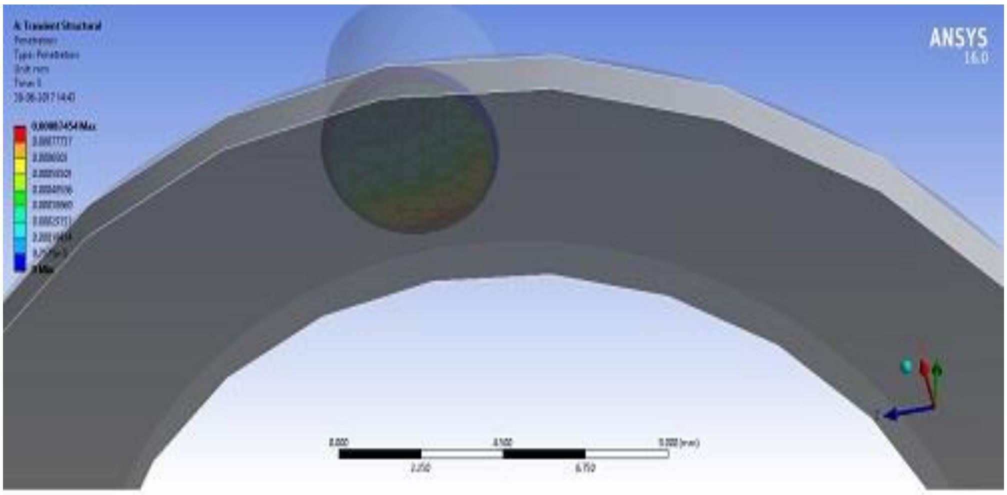


Fig: 12(i) Penetration, when Disc is Rigid, Pin is Flexible

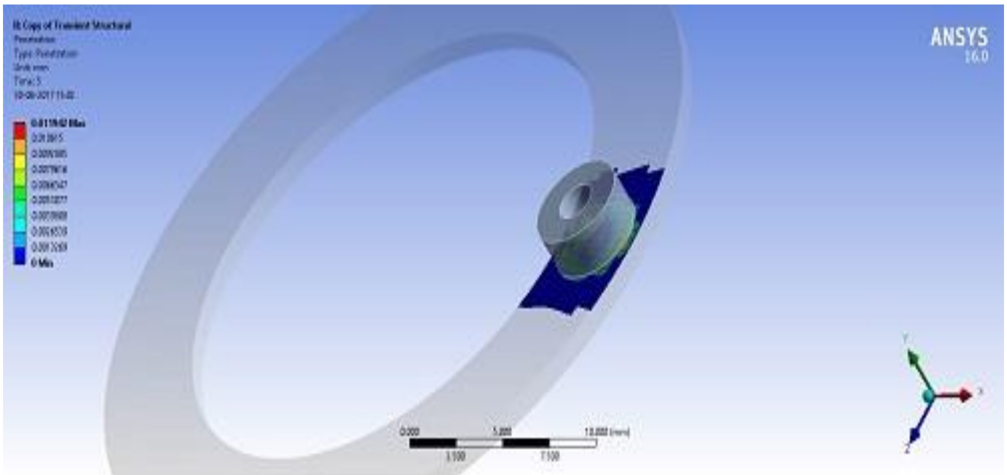


Fig: 12(ii) Penetration, when Disc is Flexible, Pin is Rigid

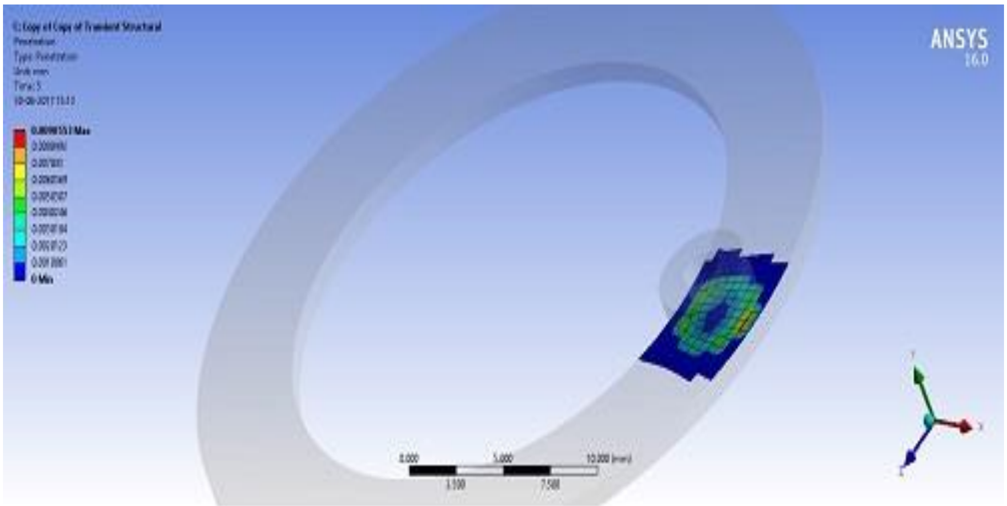


Fig: 12(iii) Penetration, Both Flexible

Because of high values of frictional stress and von mises stress at the bottom part of the pin it may be assumed that penetration would have happened at the bottom part of the pin. From the above figures it is evident that penetration has occurred at the bottom part of the pin itself.

5.7. Status

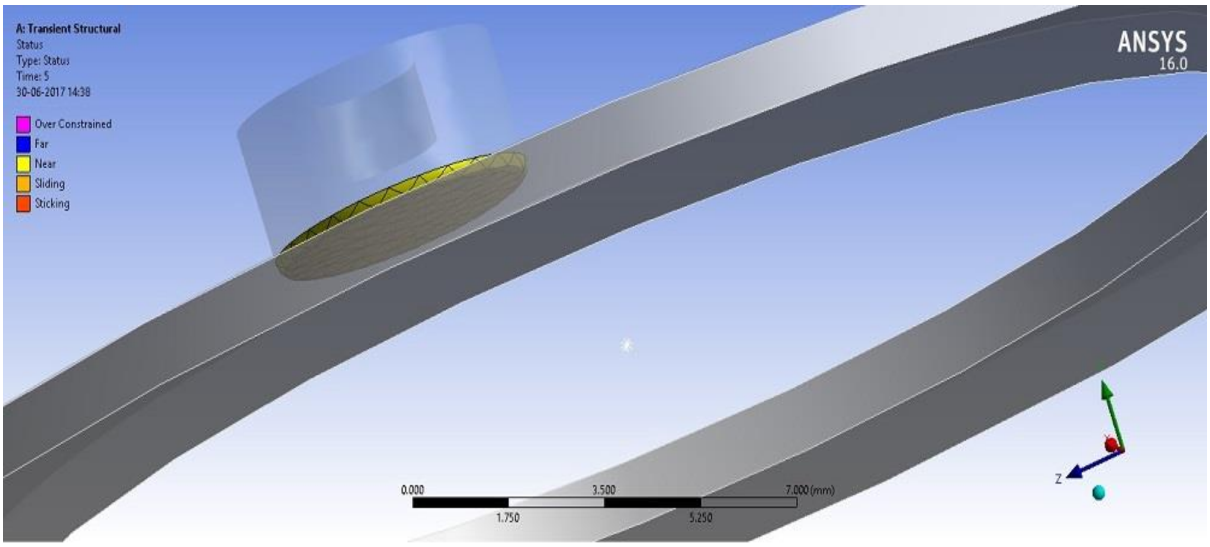


Fig: 13(i) Status, when Disc is Rigid, Pin is Flexible

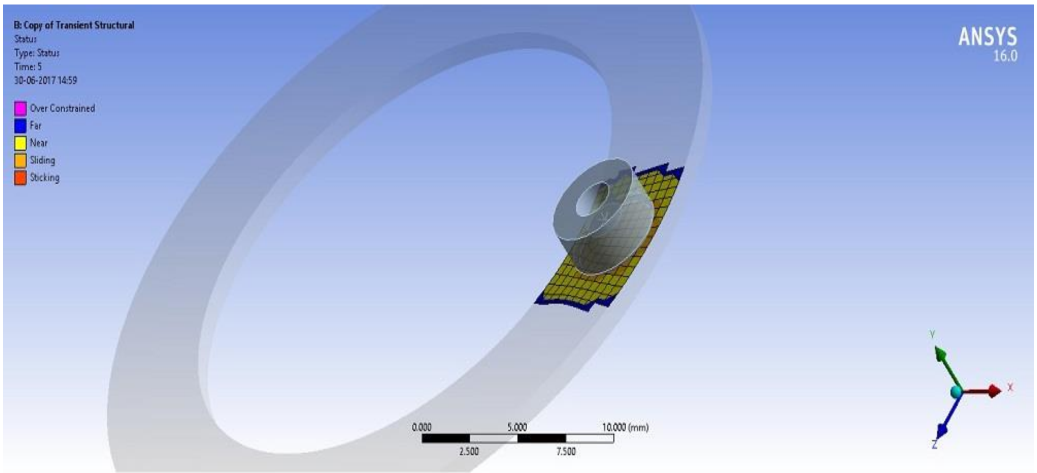


Fig: 13(ii) Status, when Disc is Flexible, Pin is Rigid

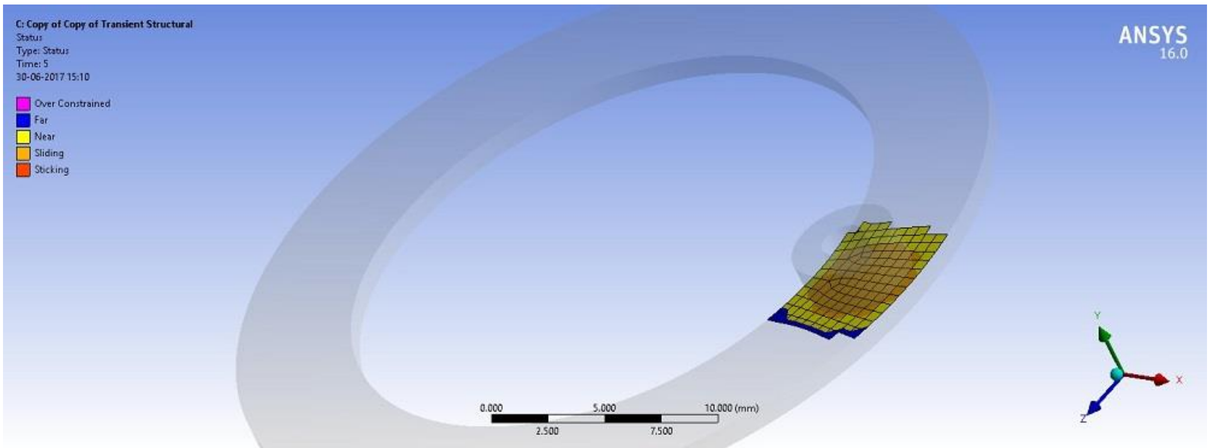
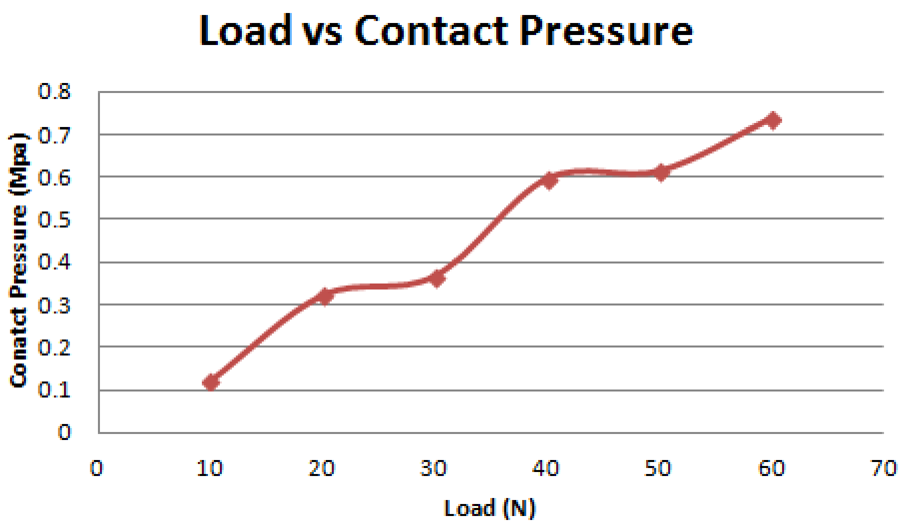


Fig: 13(iii) Status, Both Flexible

It's always heavy debate that about what type of friction is happening at the interface. Whether it is sliding friction or sticking friction. Using ANSYS for the first time we can see that at interface sliding friction is present at bottom part. Sticking friction is evidently absent at fewer loads.

(a)



This graph shows that on varying load on pin the contact pressure at the interface of pin and disc increases in nearly linear fashion. This supports the point that when there is increase in load there heavy contact pressure at interface of the assembly. This suggests that we can increase load to a higher value because it can directly lead to failure of the assembly.

(b)

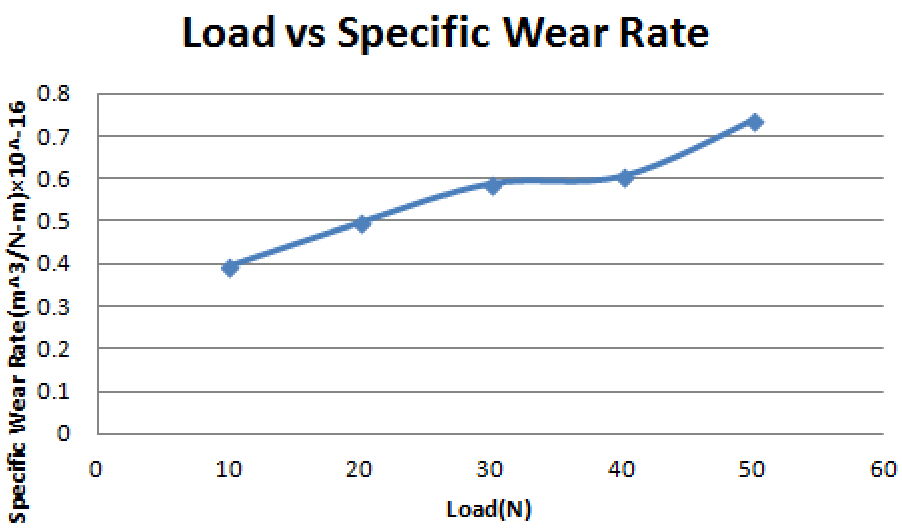


Fig: 15

The graph shows the near linear variation of Specific wear rate with respect to load. As load increases the contact pressure increases which brings the interface under critical loading condition. Due to this the specific wear rate increases accordingly.

(c)

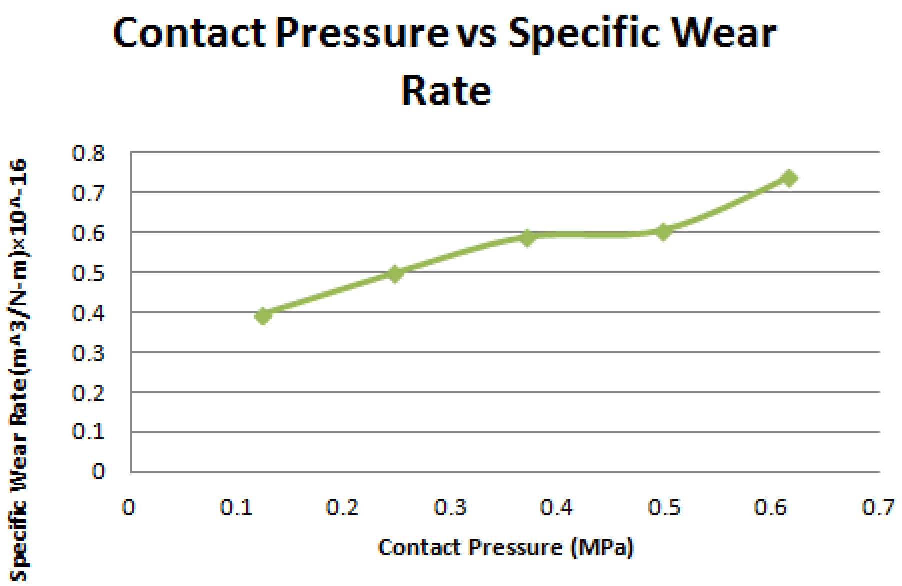


Fig: 16

This graph shows the similar variations as given earlier graph between load vs. specific wear rate. This simply justifies the earlier data. As contact pressure increases, specific wear rate increases in the same fashion.

(d)

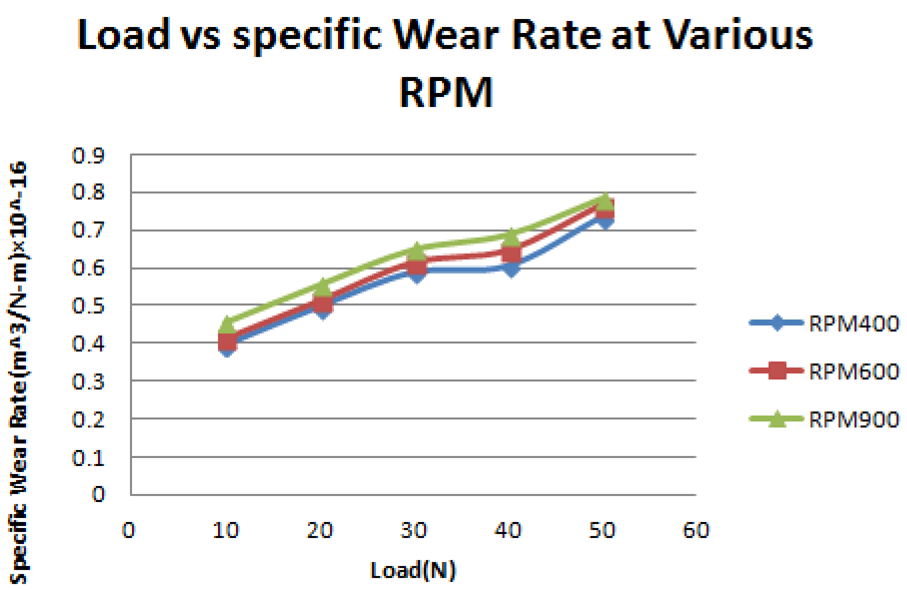


Fig: 17

This graph justifies that as there is increase RPM the wear rate increase in dry conditions. Here as there is increase in RPM the value for specific wear rate increases.

(e)

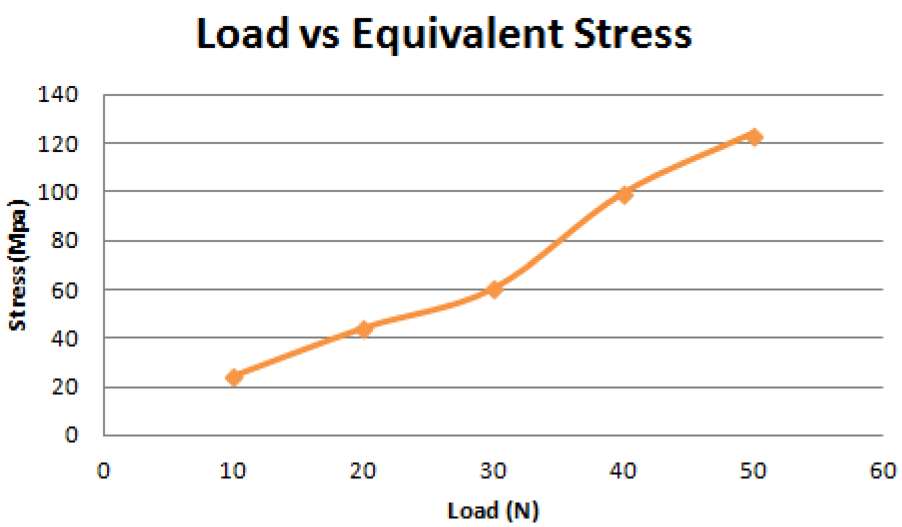


Fig: 18

As load increases stress increases in near linear fashion.

(f)

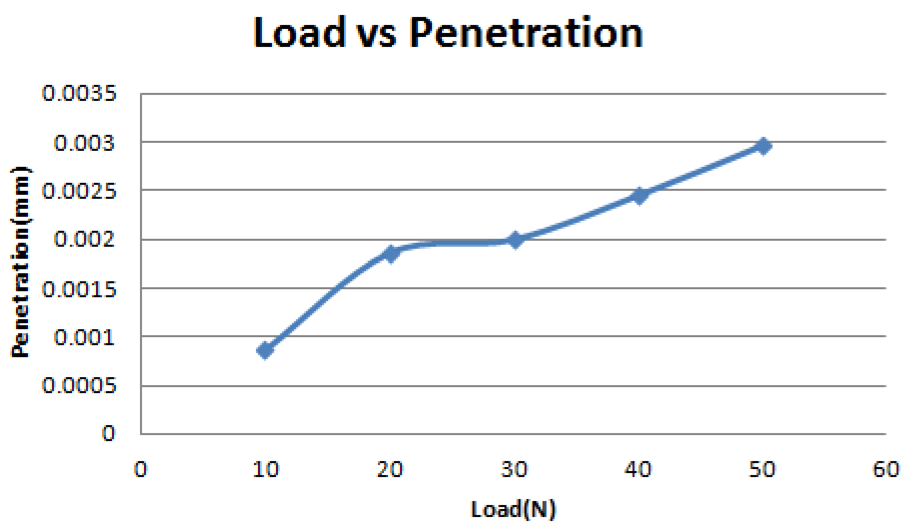


Fig: 19

The entire graph states the same variations in the conditions. As the load increases, penetration in the pin increases.

(g)

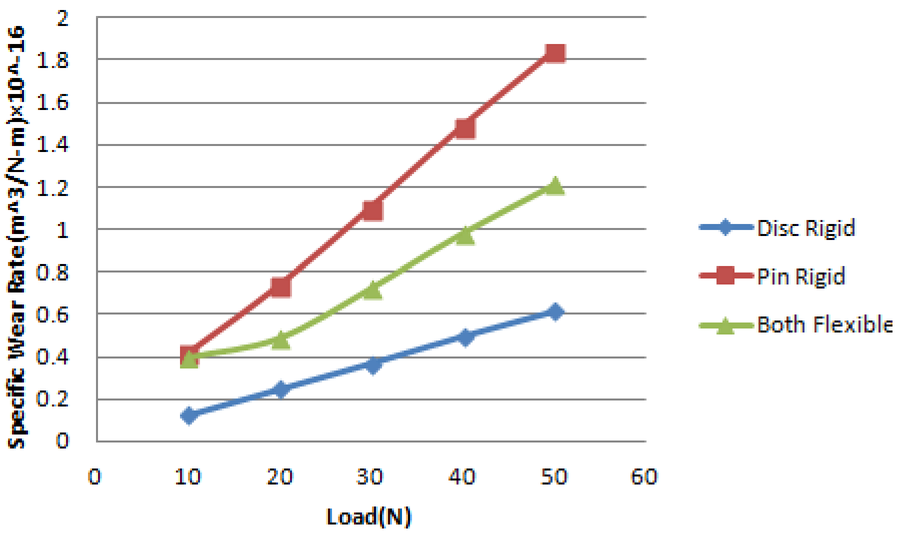


Fig: 20

This graph is comparison between various assumptions that we have taken in the beginning of our analysis. The three conditions are there, once the pin is rigid, other time disc is rigid and lastly both flexible. There use a huge difference between the data at different condition. The general pattern of the graph is same. This shows that specific wear rate increase with load regardless of any assumptions. But the value of wear rate differs. The first assumption in which we have disc rigid, provides the most convincing data.

CHAPTER 6

CONCLUSION AND FUTURE SCOPE

6.1 Conclusion

According to results and simulations conducted in this work allowed the following conclusions:

- The specific wear rate is increases as load increases.
- The contact pressure increases as load increases.
- The specific wear rate increases as RPM increases in dry conditions.
- Assumption that our Disc is rigid provides most convincing result.
- Equivalent stress increases with load
- Penetration increases as load increases.

7.2 Future scope

Tribology is the vast field of science, simulation of Tribological problem gives a numerical analysis of the Tribological problems and real life. The main future perspectives of this research work are as follows:

- Different combination of pin and disc material can be explored and simulated with giving different roughness to the mating surfaces.
- A more advance software technology can be used in future to analyze the simulation more accurately.
- As the processing power of our system will increase we can increase the number of nodes and element which in turn will provide more convincing results.
- Frictional contact model problems are to be solved by making a solver using programming in FORTRAN.

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