ADDIS ABABA UNIVERSITY

INSTITUTE OF TECHNOLOGY

SCHOOL OF GRADUATE STUDIES

Finite Element Based Surface Wear Analysis of Cam and Follower System

A thesis submitted to the school of Graduate studies of Addis Ababa University in partial fulfillment of the Degree of Masters of Science

In

Mechanical Engineering (Mechanical Design Stream)

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ABSTRACT

A particular type of contact condition, known as cam and follower contact, exists in the direct valve train system of an engine and is partly responsible for wear. In this thesis the wear analysis of cam and follower contact system are analyzed and finite element modeling of cam and follower assembly is done using the ABAQUS analysis software. The contact pressure , von misses stress are calculated theoretically and the ABAQUS results are presented in contour plot and numerically. The results showed that the cam rotational angle and the pressure angle had an effect on the contact pressure, on stress distribution and also plays a great role on the surface wear of the contact. The results also showed that the contact pressure, von misses stress and the wear increases with of cam rotational angle and the wear increases linearly with the contact pressure.Based upon the theoretical analyses and ABAQUS analysis, a theoretical model for evaluating the tribological performance of the valve train was developed. A multi-aspect comparison between theoretical and ABAQUS results was made. A good agreement between theoretical and experimental results showed that the model provided a reliable prediction o f the tribological characteristics of the cam/roller follower. Three critical portions of the cam could be identified these are the cam basic surface region, cam flank and cam lobe region.

Acknowledgments

I express my humility and utmost gratitude to God for his blessings in my life. Verily nosuccess would have been achieved without his grace and mercy.

I would like to express my sincere appreciation and gratitude to Dr. Daniel Tilahun for being my advisor and for his valuable guidance, help and encouragement during the process of this thesis. Without his guidance and support this work would have been impossible. He was the one who inspire and encourage me to work on this project. His supervision, advice, guidance and encouragement have helped me tremendously.

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My deepest appreciation is extended to my parents whose love and support has among other things helped me to achieve my goals as a student so far.

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Nomenclature

μ	coefficient of friction
g	the gravitational acceleration
W	the weight of follower
ω	the angular velocity of cam
v	the velocity of follower
θ	cam angle or cam angular position
a	the acceleration of follower
Y1	pring compression at the start of lift
Y2	spring compression
K	spring constant
ϕ	pressure angle
m	mass of follower
Fin	inertial force
Fs	spring force
b	half contact width
N1 & N2	normal forces of the guide
Fn	normal force of the contact
L	contact length
$\sigma 1$	the maximum principal stress at the point of maximum shear stress
$\sigma 2$	the minimum principal stress at the point of max. shear

σ_y	plane stress along y-axis
σ_z	plane stress along z-axis
P_o	maximum pressure
P(y)	pressure distribution
E_i	Modulus of elasticity
v_1	poisons ratio
σh	hertiz contact stress
K	wear coefficient
H	hardness
V	wear rate/wear volume
Q	wear volume per sliding
A	contact area
δ	plastic deformation
X	the distance from the center of cam to the point of Contact
v_s	sliding velocity
$\dot{\theta}$	angular velocity of cam
$\theta_b, \theta_a, \theta_c$	transition point angles
h	maximum lift

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CHAPTER ONE: INTRODUCTION

1.1 Background of the project

Camshaft is one component of the internal combustion engine that engineers are always concerned about how to predict and extend the service life. Variables like lift profile and material of the cam, valve trains configuration and manufacturing process are responsible for the fatigue performance of the camshaft. High values of stress in the peak of the cam are the main responsible of cam damage. Cams are commonly used in opening and closing of valves in internal combustion engines. Both the inlet and outlet valves are regulated using cam and follower. The study of cam and follower mechanism becomes important for desired and required performance of the engines [1].

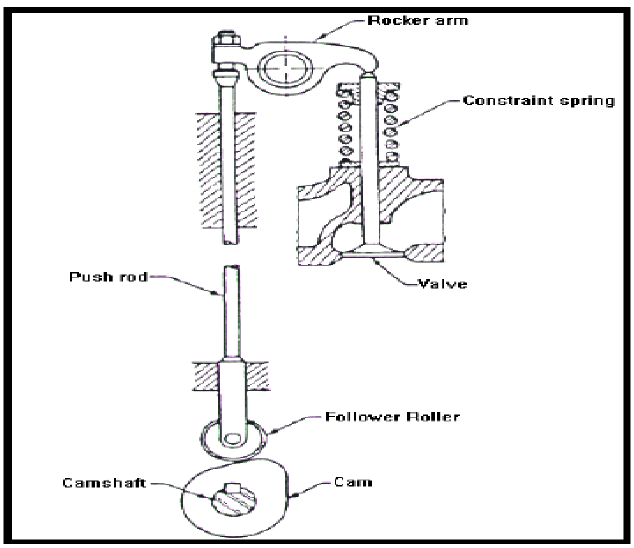


Fig 1.1 Over head valve and roller follower configuration

the fig shows the the assembly of over head valve and roller follower configuration with respect toi the constrain spring, push rod and rocker arm.

A cam–follower kinematic pair works under complicated conditions of mechanical load, and wears during operation. The contact surfaces of the cam and the follower are usually surface hardened. The hardening may be due to phase transformation or precipitation processes occurring in the material during heat treatment or thermo-chemical treatment[2].

Wear can be defined as the removal of material from solid surfaces as a result of mechanical action. The amount of material removed is often quite small for contact surfaces in high performance machines or machines with long life. Wear can appear in many ways depending on the material of the interacting contact surfaces, the environment and the running conditions. From an engineering point of view, wear is often classified as mild or severe. Mild wear is what the engineers strive for. That can be obtained by proper form and topography of the contact surfaces. Convenient materials and treatments of the surfaces are also necessary in order to get a mild wear condition. However, often the lubrication of the surfaces is the most important factor in order to secure that a mild wear condition is obtained. Sometimes, severe wear may occur, giving rough or failed surfaces. A severe wear situation is not acceptable in any machine and must therefore be avoided[3].

Wear may also be classified from a fundamental micro scale wear mechanism view. The different wear mechanisms often described in the literature are: adhesive wear, abrasive wear, corrosive wear and surface fatigue wear. *Adhesive wear* is a type of wear which occur when wear particles are formed due to the adhesive interaction between the rubbing surfaces. This type of wear may also be named scuffing, scoring, seizure and galling due to the appearances and behavior of the worn surfaces. Adhesive wear is often associated with severe wear but its role in mild wear conditions is unclear. Abrasive wear occurs when a harder surface or particles plough a series of groves on a softer surface. Often generated wear particles by adhesive or corrosive mechanisms will be abrasive particles, which will wear the contact surfaces when they move through the contact. Corrosive wear occurs when the surfaces chemically react with the environment and form reaction layers on the surfaces, which will be worn off due to the mechanical action. Mild wear of metals is often thought to be of a corrosive wear type. Another corrosive wear is fretting, which occur in contacts with small oscillating motions. Corrosive wear generate small sometimes flake-like wear particles which may be hard and abrasive. Surface fatigue wear appears as pits on the surfaces. This type of wear may be found in rolling contacts. The surfaces are fatigued due to repeated high contact stresses[3]

It is difficult to distinguish between the different wear mechanisms, since they often occur together. The different types of wear may also be dependent on each other. An adhesive and corrosive wear process may generate particles that are abrasive in a contact.

The possibility to predict the type and amount of wear is limited, although, many wear models can be found in the literature. The models are either simple models describing a wear mechanism from a fundamental view or simple empirical relationships. They are normally not convenient to use or at least very difficult to use in many practical cases.

Wear of cams and cam followers operating in timing gear systems of vehicle combustion engines takes place as an effect of friction. Durability of these components should match the engine's service life. Their wear should be little intense, However, it turns out in practical operation that components of the cam/follower system may be subject to excessive wear. It may take place after the engine runs an insignificant mileage, i.e. from several to several hundred kilometers. The excessive cam wear is also accompanied by accelerated wear of cam followers. The factors influencing excessive wear of the cam/follower system components may be divided into the categories of material related, technological and operational. The material factors include the type of material selected and , surface layer hardness . The surface layer properties depend on the graphite size and arrangement as well as the structure of the matrix usually composed of martensite and retained austenite. Fractions of these components are decisive for the surface layer hardness as well as the internal stresses which may contribute to the surface layer fatigue cracking by decreasing the number of cycles necessary for it to be initiated. In such a case, the fatigue wear intensity of a cam's surface layer may be considerably increased compared to its abrasive wear intensity, which may subsequently contribute to excessive cam wear and damaging of the cam/follower system. The technological reasons for excessive wear of the cam/follower friction may result from an incorrect technological process (heat treatment), and particularly from inappropriate parameters of the cam induction quenching and tempering, poor mechanical working (grinding) or a failed choice of the engine's timing gear system components during assembly. Nonconformity with the required heat treatment parameters may cause a decrease of the surface layer hardness and an increase of abrasive wear. The operational reasons for excessive wear of the said friction components may include inappropriate pressure forces and poor lubrication(dry condtion). Increased pressure forces are triggered by technological factors and an additional operational factor. All these aspects can lead to excessive abrasive wear caused by an increase of pressure and friction forces as well as fatigue wear in the surface layer(3).

Scuffing , Pitting polish, and scoring are the kinds of modes cam and follower wear occurs from the surface due to the high contact stresses that are present between cam and follower contact. Pitting most often occurs in a follower type timing gear system with the camshaft inside the engine block. It is a failure of surface manifested by break out of small roughly triangular portions of a material surface due to compressive stresses causing fatigue at a point below a surface. Scuffing is a local welding of two heavinly loaded surfaces specialy when a high degree of relative sliding occurs under poor lubrication conditions, followed by tearing apart of the welded material. it starts at high stressed zone with poor surface finish and generally during the running-in period of new parts. Polishing wear is because of the general attrition of contacting surfaces. scoring is scratches in the surface. Most methods of scuffing investigation concern an engine with overhead camshafts. The thickness of the oil film is too small to ideally separate all the irregularities of the mating surfaces It may lead to welding and breaking of the connections between the peaks of the irregularities in the contact micro areas [4].

Most often, the face of the follower shows even wear over the whole area, and greater wear values sometimes occur in the central area. It particularly refers to abrasive wear due to pitting or scuffing. If scuffing is responsible for intensive wear, higher wear values occur on the side of the follower face or the wear process becomes more complicated.

As part of the continuing effort to understand the tribological performance of the cam and follower, the present study will analyze surface wear analysis of cam and follower system.it also consisted of two methods; these are theoretical analyses and finite element analysis using abaqus software. Finally the comparison between the theoretical and finite element analysis results will be attained in this thesis.It is believed that the present work including both theoretical analyses and finite element analysis using abaqus has improved the understanding of the tribological performance of the cam and follower.

1.2 Statement of problem

The property of wear is crucial in many areas of tribology.cam and follower isa component of Tribology so the wear analysis of cam and follower is crucial for the performance of engine, and other areas of application. Many researches are done on cam and follower their main concern is contact pressure, fatigue and wear analysis. They used experimental, theoretical and FEM. This thesis I will use finite element analysis using ABAQUS software for wear and stress analysis of cam and follower, and use theoretical formulation to compare with the finite element analysis using ABAQUS software analysis result.

1.3 Objective of the study

General objective

- Surface Wear analysis of cam and follower

The specific objectives of this thesis work are

- Force analaysis
- Stress analysis
- Identifying the critical point of cam and follower wear
- Modeling of cam and follower system to study contact using ABAQUS software to study the stress analysis
- To estimate of stress in the contact condition in cam and follower by finite element method using ABAQUS finite element s software.
- Comparing the ABAQUS software results with theoretical formulation.

1.4 Organization of the thesis

This thesis is organized in to six chapters. In the first chapter, background and justification of this thesis work and the objectives to be achieved are discussed. In chapter two, a review of literature relevant to this thesis work, which has been investigated by different researchers, is given. Chapter three is about analytical method in contact stress analysis and force analysis and wear analysis. In chapter four, finite element method (FEM) is used, to develop cam and follower using abaqus, and 3D finite element analysis is done in abaqus. In addition numerical and analytical model using cam and follower in contact is developed and contact stress analysis is done. This chapter also presents detailed information about the analysis of surface wear in cam and follower mating,.

In chapter five results of the analysis are summarized and discussions are made based on the outputs of the FEM. In addition, comparison of analytical and numerical solutions is made. Finally, chapter six gives conclusion achieved from this thesis work and propose future work in this field of study.

CHAPTER TWO: LITERATURE REVIEW

2. LITERATURE REVIEW

Contact stresses between curved bodies in compression are often termed "Hertzian" contact stresses after the work on the subject by Hertz in 1881. This work was concerned primarily with the evaluation of the maximum compressive stresses set up at the mating surfaces for various geometries of contacting body but it formed the basis for subsequent extension of consideration by other workers of stress conditions within the whole contact zone both at the surface and beneath it. It has now been shown that the strength and load-carrying capacity of engineering components subjected to contact conditions is not completely explained by the Hertz equations by themselves, but that further consideration of the working conditions is an essential additional requirement[5].

There are great deal of researches and number literatures on cam and follower analysis that has been published. Generally their major concerns are on the analysis of cam and follower stresses, failure, contact presure,realization of follower motion cam and follower system,kinematic and dynamic analysis, modeling, which are very useful for optimal design of cam and follower system. They have used various approaches and means to attain their main intention.

Per Lindholm,Stefan Bjorklung[6] uses experimental method to study characterization of wear on a cam follower system in a diesel engine. their main concern is wear analysis of cam and follower. In their study includes investigation of the running in of the most important contact surfaces of a modern diesel cam follower system. The running in is investigated by analyzing the changes in topography of the roller, pin and rocker arm of the fuel injector arm. In their study, the surface is assumed to be smooth. The aim of the test was to observe the behavior of the wear of standard components of an automobile engine before performing a test with coated components. During the tests moulds of the surface of the cam and the shaft were created.

P.lindholm[7] the test equipment consists of an overhead camshaft with inlet, outlet and injector rocker arms. In this system there are several surface contacts of tribological interest, including rolling/sliding, full film, mixed and boundary lubricated regimes. The investigated surfaces in this study were the inner and outer surfaces of the roller, the pin and the rocker arm bushing all belonging to the injector pump cam follower mechanism. the tests shows that the roller pin contact moves up into the mixed regime since the wear of the surface of the inner bearing is measurable. The calculation model showed that the slip, in a limited region closed to where the rocker arm reaches its upper position, was significantly higher than on the remaining part of the cam cycle. Film thickness calculations showed that the cam roller contact works in the mixed lubricated regime, the slider bearing between the arm and the shaft moves over all regimes and that the roller pin contact works as a full film bearing with tendencies to enter the mixed lubricated regime.

Juliano Savoy and Carlos Coelho [5] investigates the characteristics of an cam and follower system including contact contact pressure. To estimate the effect of contact pressure , the characteristics of cam and follower system were analyzed by using the finite element method. In their study they summarized some simulations in order to define an acceptable model to run 3D finite elements analysis and calculated the contact pressure. This work also aimed to assess the results of the Hertz’s theory when using finite element analysis using a cylinder in contact with a flat surface and a real camshaft case. So, finite elements models allow one to assess situations that are outside the scope of the theory of Hertz in order to make a better judgment of real cases. The importance of this experiment is that it shows us how far we can be away from the correct result, if we use the wrong size, type and order of an element in a finite element analysis. The calculation of stresses in a body due to a mechanical contact for cases involving complex geometries can be evaluated only numerically. In complex situations other than, for instance, plane state of stress, there are no analytical solutions to the equations describing the stresses on the bodies in contact, so a deeper assessment of the finite element modeling is crucial for designing robust components.

J. Michalski), J. Marszalek, K. Kubiak[5] uses experimental method to invetgate the characterization of wear on a cam follower system in a diesel engine. The investigation focused on the effect of cam and follower materials and their thermal and thermochemical treatment on cam and follower functional properties. During the investigation The preferred material for a camshaft mating with a slide cam follower is grey metallurgically hardened cast iron. And The mating follower should be made of chilled cast iron, hardened and tempered The investigation was carried out on a laboratory bench equipped with an engine head with a camshaft, followers and systems creating the conditions necessary for a routine run of the valve gear. Cam wear was defined by comparing the profile lifts of the cams. It often concerns cams mating with a roller follower where the wear may be increased, particularly due to the occurrence and propagation of cracks. With respect to the geometry and kinematics of combustion engine valve gear, it is generally assumed that the value of Hertzian pressure has a direct effect on the pitting wear value. In the analysis Camshafts are made of nodular cast iron, surface hardened, ion nitrided and nitrosulphurized, and those made of grey chilled cast iron are mated with followers made of chilled grey cast iron and hardened steel. The investigation result shows The formation of a nitrided layer on the hard and thick martensitobainitic matrix of cams C5 ensures its small.radial wear . The cam undergoes smooth abrasive wear . Follower also undergoes smooth abrasive wear of a small value. The application of sulfonitriding is not recommended because of the increased wear onthe cam and follower despite smaller resistance to motion The explanation of the results obtained for the cam–follower kinematic pair wear arises from the Hertzian theory as well as from the occurrence of tensile stresses at slidecontact . At contact points, tensile stresses behind the contact surface of the C2 nodular cast iron cam and the grey cast iron follower, hardened during casting, may have attained a value exceeding their tensile strength. Internal stresses were also of tensile nature at small depths, which helped the process.

mohd hafiz bin ghazalli[8] Studied the contact stress analysis of cam and follower using finite element analysis . This project aim determines the stress concentration on the cam and followers during normal operation. More over, this project used the cam, rocker arms, valvelifter, exhaust valve and accessories used in 4G13 engine in type. The finite element analysis are done for determination of stress concentration during 30 degree of cam where the roller fully climbing the cam and during maximum exhaust valve lift. in the analysis, the typical values for coefficient of friction, materials, and spring rate are used. The result from finite element analysis showed that the maximum stress concentration occurred at rocker arm that leads to the failure of the component. Value for maximum stress is over the allowable stress for rocker arm material. Other components are approximately safe where the maximum stress is not over the allowable stress for components.

Vasin Paradorn, [9] investigatesan impact model for the industrial cam-follower system using simulation and experiment. From this model, an insight into proper design of systems with deliberate impact was developed through computer modeling. To attain more precise representations of a simplified industrial cam-follower system model was constructed in SolidWorks CAD software. A two mass, single-degree-of-freedom dynamic model was created in Simulink, a dynamic modeling tool. and validated by comparing to the model results from the cam design program, DYNACAM. After the model was validated, a controlled impact and over-travel mechanism was designed, manufactured, and assembled to a simplified industrial cam-follower system, the Cam Dynamic Test Machine (CDTM). Then, a new three-mass, two-degree-of-freedom dynamic model was created. Once the model was simulated, it was found that the magnitude and the frequency of the vibration, in acceleration comparison, of the dynamic model matched with the experimental results fairly well. By using this three-mass, two-DOF impact model, machine design engineers will be able to simulate and predict the behavior of the assembly machines prior to manufacturing. If the results found through the model are determined to be unsatisfactory, modifications to the design can be made and the simulation rerun until an acceptable design is obtained. During the invetigation An impact and over-travel mechanism was designed, manufactured, and assembled by the author on the CDTM. The most important requirement was to ensure that the impact and over-travel mechanism was implemented and the events were consistent and repeatable. The design must not exceed the available space or interfere with the existing machine as well as minimizing the modification made to the existing parts. Adjustability of the over-travel distance is highly desirable because this will allow the user to change the over-travel distance, impact velocity, and force exerted on the hard-stop. The over-travel spring must prevent the separation of the impact mass and the intermediate mass until impact occurs to ensure that the condition experienced in the actual system is maintained in the experiment. Lastly, the force experienced by the impact mechanism must be obtainable through experiment, which is done by placing the force sensor where the striking impact occurs. In the study the assumptions, detailed derivations of the dynamic model were performed for different possible conditions these condtions are no contact condition, intial contact that is impact condition.the result of comparison between the experimental and simulated showed that the experimental and simulated acceleration of Mass. The maximum magnitudes of the first and third impacts match relatively well, even though the shapes after impact do not. The main reason for this discrepancy is the lower degree-of-freedom of the simulated system versus the real system. By reducing DOF, higher modal characteristics of the links were excluded from the model. The maximum magnitudes of the simulated acceleration for the second and fourth impacts were 1.32 times smaller than the experimental acceleration.

Jose alejandro Escobar[10], Fatigue analysis of svi- tested cam lobe, the research objective is abetter understanding of the grinding process and its effect on the workpiece residual stress state.more important for design engineers however, is the vinculum between indiced residual stress andlife time of the cam lobe.This thesis addresses this problem by linking the results found experimentally to fatigue life analysis. The analysis includes the life prediction of cam lobes, failure of cam, crack analysis. From the analysis he concludes the following;

- Cracking occurred almost exclusively in the opening ramp of the most abusively ground lobes. It was induced by tensile residual stresses generated during grinding, and to a lesser extent by dynamic loading. Cracks were confirmed to propagate as deep as 300μm below the surface of the lobe
- Residual stress relaxation was measured and confirmed to occur in the most abusively ground lobes. It was induced by tensile residual stresses generated stresses were reduced and shifted deeper into the material. These residual stresses stabilized after a certain number of cycles, regardless of the level of grinding

Two types of cracks were identified after simulated engine testing: straight and pitted. Nucleation of straight cracks occurs during grinding by the localized thermal expansion of the steel. Pitted cracks are nucleated by the synergistic effect of subsurface tensile residual stresses and impact loads generated by the follower during testing. Propagation of straight cracks has fatigue characteristics, with “lava flow” fracture surfaces suggesting lateral rubbing of crack surfaces due to cyclic loading. Propagation of pitted cracks is closely related to the process of pit formation (pieces of material are flake off due to impact loading, leaving behind a pit). These cracks propagate by pits joining together.

H.van Leeuwen, H.meijer and M. schoten[11]”elastohydrodynamic film thickness and temprature messurement in dynamically loaded concentrated contacts:eccentric cam-flate folower”. This paper discribes some results of local film thickness and temprate measurement in an eccentric cam-flate follower contact by means of miniature vapour depositer thin layer transducers.Complex trasnduer paterns can be realized by employing photo lithography, allowing local measurments in axial direction and a fullfilm will be develop at relativelly low speeds.film thickness and temprature at both sides of contact differ appriciablly,thus invalidating the assumption of line contact**.** And the paper considers capacitance mesurment under missaalignment and temprature measuremen**t.**considering the above conditions they concluded the following: if the contact has some missallignment, mixed film condition may prevail at the heavy load side.The rise in film tempreture can become high, leading to chemical reaction layer formation.the capacity measuring technique allows an examination of the extent and thickness of these layers.and if good allignment was obtained affter incorporating sef alligning elastic hinge support.the measurement temprature variation are not high, but need to be included in the film thickness calculations. Probably starvation occurs, because calculated film thickness are still to o high. the transducers worked well and have satisfactory life expectancy

CHAPTER THREE: MATERIALS AND METHODS

3.1 Materials used for cams and followers

The preferred material for a cam mating with a slide cam follower at indirect valve drive is grey cast iron w10,11x. It has the following composition: 3.1–3.5% C, 1.8–2.5% Si, max. 0.15% S, max. 0.2% P, 0.5–0.8% Mn, 0.15–0.25% Ni, 0.7–1% Cu, 0.15–0.25% Mo, the rest is Fe. The cam surface gets hardened during primary crystallization. The minimum hardness should be 45 HRC with a 3-mm hardened layer at the lobe and the flanks.

The mating follower should be made of chilled cast iron, hardened and tempered, with an increased content of Ni, Cr, Mo, V. It has the following composition 3–3.6% C, 2–2.8% Si, max. 0.12% S, max. 0.25% P, 0.4–1% Mn, 0.4–0.7% Ni, 0.9–1.25% Cr, 0.4–0.7% Mo,the rest was Fe. The minimum hardness of the follower should be 54 HRC with the depth of the hardened layer being 3 mm

3.2 Analytical Method In Wear Analysis of Cam and Follower

3.2.1 Force analysis

The forces acting against the direction of motion of follower can be broken down in to four components, these are the frictional forces ,the inertial force, the spring forceand the normal load.

3.2.1.1 Frictional forces

$$Ff = \mu N1 + \mu N2 \qquad \text{……………………………..(1)}$$

Where μ = coefficient of friction

N1, N2 normal forces

3.2.1.2 Inertial force

$$Finertial = \frac{w*a}{g} \qquad \text{…………………………………………………(2)}$$

Where g is the gravitational acceleration

W= is the weight of follower

Where $a = \frac{dv}{dt}$

$$V = \frac{dy}{dt} = \frac{dy}{d\theta}\frac{d\theta}{dt} = \omega\frac{dy}{d\theta}$$

$$V = \frac{dy}{d\theta} * \omega$$

$$a = \frac{d(\frac{dy}{d\theta} * \omega)}{dt}$$

Where ω is the angular velocity of cam

v is the velocity of follower

θ is cam angle

a is the acceleration of follower

3.2.1.3 Spring Force

The functional role of the spring in cam anf follower designs engine is to ensure a constant contact between the cam and the follower. The presence of the spring complicates the design and results in increased contact stresses.

Fs = K(Δy)…………………………………………………………….(3)

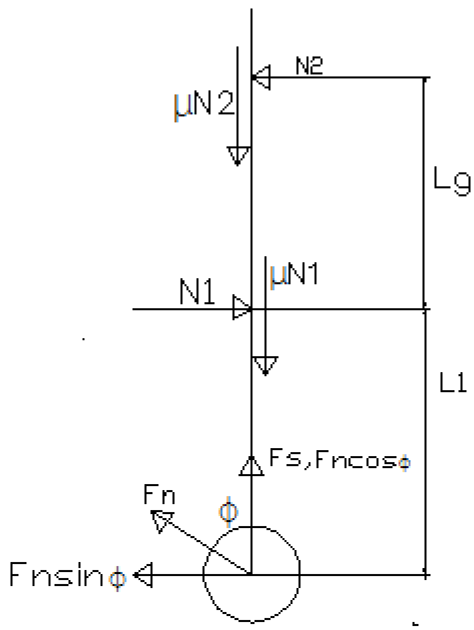
Δy = Y2-Y1

Where Y1 = spring compression at the start of lift

Y2=spring compression

K= spring constant

The normal force which determines the the contact stress that dermines the wear of cam and follower mating can be calculted using dynamic motion equation using normal tangential coordinate system.



The fig shows the forces on the follower . As shown above the figure the forces acting on the follower are the frictional forces on the push rod and the normal forces and the inertial forces. So inorder to investigate the expression of the normal force on the roller follower we can use dynamics i.e. the kinitics part in coopration of newtons second low.Taking the summation of moment about the center of roller to be zero

This gives N1*L1 - N2*(L1+Lg)=0

$$N1= N2(1+\frac{Lg}{L1}) \text{ …………………………..(4)}$$

Another equationis

$$N1-N2-Fnsin\phi=0$$

$$N1= N2+Fnsin\phi\text{………………………..(5)}$$

From these two equations we can derive

$$N1= Fnsin\phi(1+\frac{L1}{Lg})$$

If we consider the motion along the follower motion, is accelerates with acceleration

$$a = \frac{dv}{dt}$$

this gives

$$\mu\ Fn\cos\phi - \mu N1 - \mu N2 - Fs = ma \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(3.6)$$

where ma is inerial forces resulted from the motion cam

from eqauation (1)and(2)

$$N1 = Fn\sin\phi(1+\frac{L1}{Lg})$$

$$N2 = Fn\sin\phi(\frac{L1}{Lg})$$

This gives

$$Fn = \frac{Fs-Finertial}{\cos\phi + \mu\sin\phi(1+\frac{2L1}{Lg})} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(3.7)$$

Fn is the forces that determines the wear of cam and follower system by affect the contact stress.

Here the frictional forces are negligible as compared with the spring force and the inertail forces of follower .

Then equation (4) becomes

$$Fn = \frac{Fs-Finertia}{\cos\phi} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(3.8)$$

The severity of wear depends on this force.

This normal compressive force is dynamic load depending on the cam angle. We estimated normal compressive force applied on the cam surface and is plotted with cam angle.the distribution of thenormal forces drived here described as a function of cam rotational angle is depicted below

.

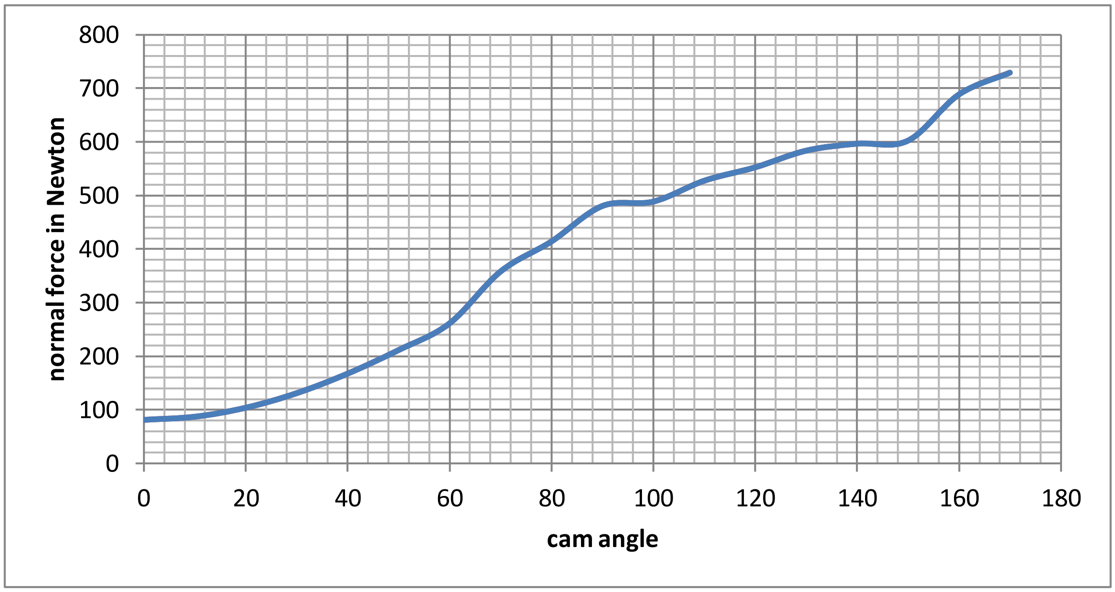


Fig 3.1 shows the distribution of normal force as a function of cam angle

this fig shows the normal load increase with the respect to the cam angle and the normal force is maximum at the cam lobe that is when the cam rotational angle is 180°.

It is very obvious to explore whether this dynamic load is reduced or not with respect to the cam angle.this can be possible by analyzing the pressure angle with respect to the cam angle. Variation in pressure angle with cam rotation is given in Fig 3.6. It is possible by redesigning cam so that pressure angle remain lesser than 30^0 . Pressure angle Φ is angle between direction of motion(motion direction of follower) and the axis of force transmission. When ø = 0 Transmitted Force is Completely utilizeto move the follower.

if the pressure angle is more than 30^0 there will be jumming of the follower with the bushing. Then to avoid this jumming the pressure angle must kept below 30^0.the variation pressure angle with cam rotation is shown in fig below.

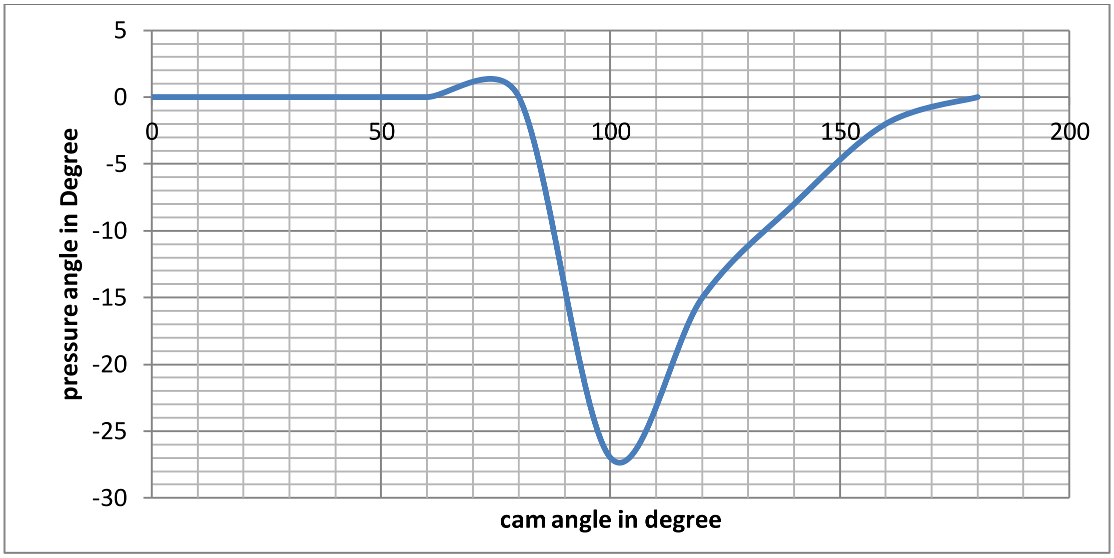


Fig 3.2 shows the distribution of pressure angle as a function of cam angle

As shown the depicted above in the fig, it sows the distribution of pressure angle with respect to the cam rotational angle. It also shows the effect is on the cam flank

3.2.2 Hertz Contact stress analysis

There are different types of rolling and /or sliding elements that are mating each other.these elements are cam and follower mating , chain and sprocket mating , wheel and rail mating, and different types of gear mating etc. when this mating elements are under high pressure they develop compressive stress that causes wear. Different types of wear will be experienced here that are pitting, polish and scuffing are developed. To avoid this and to increase the service life of these mating elements it is necessary to consider the surface strength of these contacting elements.

Hertz in 1882 was the first to develop the study of pressure fields acting on two surfaces and the stress field acting on the bodies in contact. There is a distinction between conforming and non-conforming contacts. When the surface of two bodies fit exactly or even closely together without deformation, can be said that the contact is conforming. For a conforming contact an example is journal bearings. Bodies which have dissimilar profiles are said to be non-conforming. When brought into contact without deformation they will touch first at a point - ‘point contact’- or along a line - ‘line contact’. For example, in a ball-bearing, the ball makes point contact with the races, whereas in a roller bearing the roller makes line contact. Line contact arises when the profiles of the bodies are conforming in one direction and nonconforming in the perpendicular direction. The contact area between non-conforming bodies is generally small compared with the dimensions of the bodies themselves; the stresses are highly concentrated in the region close to the contact zone and are not greatly influenced by the shape of the bodies at a distance from the contact area.

When two non-conforming elastic bodies are initially brought into contact, they touch at one point or along a line. When an external force is applied to these bodies, so that they remain in contact, the bodies are deformed in the vicinity of the point of initial contact, so that the contact point becomes a finite area of contact.

The Hertz's theory considers that the surfaces of bodies in contact are geometrically smooth. This formulation proposed by Hertz is valid for all surfaces of revolution in non-conforming contact. In case of contact between a surface of revolution and a plan,suggests that the radii of curvature of the functions that represent the surface in contact should be considered as tending to infinity.

Based on his observations, Hertz proposes that the problem of elasticity should be formulated as a semi-infinite body loaded into an elliptical region. For the proposal made by Hertz it is true that two conditions must be met:

- The dimensions of the contact area should be much smaller than the dimensions of each elastic body.
- The dimensions of the contact area should be much smaller than the smallest radius of curvature of elastic bodies in contact.

The first condition is related to the calculation of all stress fields and is based on the hypothesis of a semi-infinite body, i.e. the region of contact should be located away from the edges of bodies in contact, so that the result is not influenced by the large stress concentration caused at the edges of the elastic bodies. The second condition ensures that the surfaces of bodies can be approximated by a plane. Finally, the surfaces are assumed to be frictionless so that only a normal load is transmitted between them. Most machine components are designed on the basis of stress in the main body of the member, that is, in portions of the body not affected by the localized stresses. In other words, damage to most mechanical components is associated with stresses and strains in portions of the component far removed from the points of application of the loads. However, in certain situations the contact stresses between the surfaces of two externally loaded bodies that is cam and follower mating can be the significant stresses; that is, the stresses on or somewhat below the surface of the

contact are the major causes of damage to one or both of the bodies. Therefore, analysis Hertzian stresses provide important information about surface and subsurface stresses for static loading of concentrated contacts.

The contact between two surfaces gives rise to an area of contact, and a pressure distribution over this contact area. If the surfaces have a simple geometry, the contact theory of Hertz (1881) can be used for calculating the contact area and the pressure distribution and the compressive stress along the contacting zone. The shape of the contact area depends on the shape(the geometry) of the contacting bodies.

To express the contact between two elastic bodies contact hertiz proposed the contact between two parallel cylinders is presented in line contact. This contact analysis is used to analysis the contact pressure, contact conpresssive stress for line contact.

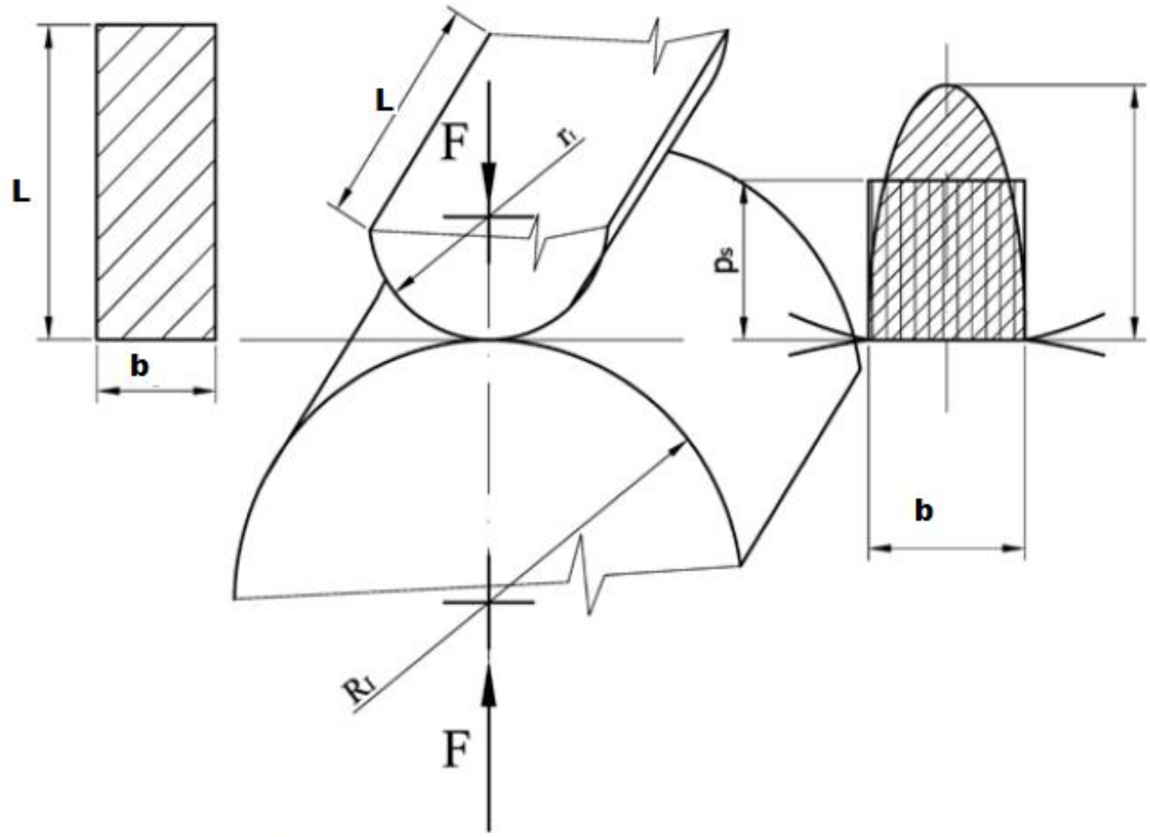


Fig 3.3 shows two cylinders in contact

The Hertzian contact theory was based on some simplifications as follow that made the development of the contact theory possible at the time

1. The displacements and stresses must satisfy the differential equations of equilibrium for elastic bodies, and the stresses must vanish at a great distance from the contact surface.
2. The bodies are in frictionless contact.
3. The contacting materials are elastically isotropic.
4. Young's modulus, *E* and Poisson's ratio υ remain unchanged under load.
5. The curvature of contacting bodies could be approximated by second-degree surfaces.
6. The contact radius is very much smaller than the radius of contacting bodies.

Hertz made his analysis general by attributing a quadratic function to represent the profile of the two opposing surfaces and gave particular attention to the case of contacting spheres. But for the contact between two cylinders with parallel axes subjected to a normal force, a rectangular contact area will be produced. The xy plane is specified coincident with the contact area with the x-axis parallel to the cylinder axes

Here for the analysis of two parallel cylinders in contact; the developed contact area is rectangular with L the length of contact and b which is half width of contact.

As depicted above in figure 3.3 the pressure developed is elliptical distribution with respect to the normal load and

It can be expressed as a function of normal load as

$$Po = \frac{2Fn}{\pi bL} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(3.9)$$

Where Fn is the normal load

p_o is the maximum pressure at the center of contact

b is half contact width

l is the contact length

the elliptical pressure distribution throughout the width of contact can be expressed as two dimensional version equation

$$P(y) = \mathrm{Po}\sqrt{1 - \frac{\mathrm{y2}}{\mathrm{b2}}} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(3.10)$$

The total load is then the volume of the prism

$$Fn = \int_0^b 2\pi LP(y)dy \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(3.11)$$

Where $P(y) = \sqrt{1 - \frac{y2}{b2}}$

The integration of the above equation will give

$$Fn = \frac{1}{2}\pi bl\mathrm{Po} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(3.12)$$

Then the maximum pressure at the center of contact can be expressed as

$$\mathrm{Po} = \frac{2Fn}{\pi bl}$$

the pressure at any point through out the contact length can be as a function of load, contact length, and contact width of contacting elements

$$Po = \frac{2F}{\pi bl}\sqrt{1 - \frac{y2}{b2}}$$

For a given contact load Fn it is necessary to determine the value of b before the maximum contact stress can be evaluated. these are found analytically from equation suggested by Timoshinko and Goodier.

The half width b depends on the geometry of contacting cylinders and for two cylinders of unequal diameter and unlike materials, the half width is:

$$b = 2\sqrt{(\frac{1-\nu 2}{\pi})\frac{p(\frac{1}{E1}+\frac{1}{E2})}{L\left(\frac{1}{R1}-\frac{1}{R2}\right)}} \quad \ldots\ldots(3.13)$$

then themaximum pressure can be given as

$$\text{po} = \sqrt{\frac{1}{\pi(1-\nu^2)}\frac{p\left(\frac{1}{R_1}+\frac{1}{R_2}\right)}{L(\frac{1}{E_1}+\frac{1}{E_2})}} \quad \ldots\ldots(3.14)$$

this maximum pressure is along the load axis(y=0)
where $R1$ and $R2$ are the radiuses of two cylinders
The stress conditions at the surfaces on the load axis are :-The stress along the z axis which is equal to the maximum compressive stress or the maximum pressure,the stress along the y-axis, the stress along the x-axis thatis the stress along the cylinder length.The contact stresses studied herein pertains to compressive stresses developed at the contact surfaces between the roller and cam due to tangential and normal loads of the valve train. The contact stresses can be expressed as reported by smith and liu,
Wear due to inelastic deformation is associated with the maximum shear stress. This can be expressed as

$$\tau\text{max} = \frac{\sigma 1-\sigma 2}{2} \quad \ldots\ldots(3.15)$$

Where

$\sigma 1$ is themaximum principal stress at the point of maximum shear stress

$\sigma 2$ is the minimum principal stress at the point of maximum shear stress

The analytic model for subsurface contact stress proposed by giradin as a function of y, z ,p_o and b and can be calculated using the following relation by Assuming the plane strain case at the contact point, the stresses at yz plane.

$$\sigma y = -\frac{po}{\pi}\left(\frac{b^2 + 2y^2 + 2z^2}{b}\psi - \frac{2\pi}{b} - 3y\varphi\right) \quad \ldots\ldots(3.16)$$

$$\sigma z = -\frac{po}{\pi} z(b\psi - y\varphi) \quad \ldots\ldots(3.17)$$

$$\tau yz = -\frac{po}{\pi} z^2\psi \quad \ldots\ldots(3.18)$$

Where ψ, φ are the functions that depends on y, z, and b. and can be expressed as

$$\varphi = \frac{\pi}{k_1}\,\frac{1-\sqrt{\frac{k_2}{k_1}}}{\sqrt{\frac{k_2}{k_1}}\sqrt{2\sqrt{\frac{k_2}{k_1}}\;+\left(\frac{k_1+k_2-4b^2}{k_1}\right)}} \quad \ldots\ldots(3.19)$$

$$\psi = \frac{\pi}{k_1}\,\frac{1+\sqrt{\frac{k_2}{k_1}}}{\sqrt{\frac{k_2}{k_1}}\sqrt{2\sqrt{\frac{k_2}{k_1}}+\left(\frac{k_1+k_2-4b^2}{k_1}\right)}} \quad \ldots\ldots(3.20)$$

Where k_1, k_2 also can be expressed as follows:

$$k_1 = ((b+y)^2 + z^2) \quad \ldots\ldots(3.21)$$

$$k_2 = ((b-y)^2 + z^2) \quad \ldots\ldots(3.22)$$

The normal and shear plane stresses can be calculated adopting the method of smith and liu in conjunction with the assumptions that no sliding between the two parallel cylinders in contact ie tangential loads are neglected. At the surface, the plane stress adopt the following values

$$\sigma_y = -p_o \quad \ldots\ldots(3.23)$$

$\sigma_z = -p_o$ ……………………………………………………………………..(3.24)

$\tau_{yz} = 0$ …………………………………………………..…………..(3.25)

3.2.3 Non hertizian contact

Even though, Hertz's theory forms the basis for classical contact mechanics, most of the assumptions have been removed by different authors, to take in to account the real working conditions. These conditions are, when there is friction, non linear material properties, complex geometries, when there is a lubricant, etc. Now a days, numerical methods has got attention to solve these types of nonlinear problems. Generally nonlinear structural behavior arises for a number of reasons, as cited in Wei [9] which can be categorized as:

1. Geometric Nonlinearities (Large Strains, Large Deflections)
2. Material Nonlinearities (Plasticity)
3. Change in Status (boundary) Nonlinearities (Contact).

Contact is considered as a "changing status" type of non-linearity. Depending on whether the contact is open or closed, and if closed, sticking or sliding, the system's stiffness changes accordingly. In addition, the area over which contact occurs is typically not known at the beginning of analysis. Hence the contact between two bodies categorized to the third group. Therefore, in this thesis contact non-linearity will be examined closely, because the cam and follower in mesh are behaving in this fashion.

The contact is highly non-linear because one or both of the following are present, Stachowiak

- The actual regions of contact are unknown until the problem has been solved. Depending on the load, material, boundary conditions and other factors, and surfaces can come into and go out of contact with each other in a largely unpredictable and abrupt manner.
- Most contact problems need to account for friction. Contact and friction between solids is a complex problem which involves the mechanical properties of the materials, the geometrical characteristics of the rough surfaces, the properties ofany lubricant that might be present in the motion, and the temperature of the contacting surfaces. There are several friction laws and models to choose from, and all are nonlinear. Frictional response can be chaotic, making solution convergence difficult. In addition to those difficulties, many contact problems must also address multi-field effects, such as the conductance of heat

and electrical currents in the areas of contact. However with the rapid development of computational mechanics, great progress has been made in numerical analysis nonlinear problem. Using the finite element method, many contact problems, ranging from relatively simple ones to quite complicated ones, can be solved with high accuracy.

3.2.4 Application of hertz theory for cam and follower system

Contact width

$$b = 2\sqrt{\left(\frac{1-\nu 2}{\pi}\right)\frac{\mathrm{Fn}\left(\frac{1}{E1}+\frac{1}{E2}\right)}{L\left(\frac{1}{\mathrm{R1}}-\frac{1}{\mathrm{R2}}\right)}} \quad \ldots\ldots(3.26)$$

from equation (3.8) is the normal load

$$Fn = \frac{Fs-Finertial}{\cos\phi} = \frac{Fs-Fin}{\cos\phi}$$

From this the contact width can be written as

$$\mathrm{b} = 2\sqrt{\left(\frac{1-\nu 2}{\pi}\right)\frac{(F_S-F_{in})\left(\frac{1}{E1}+\frac{1}{E2}\right)}{L\cos\phi\left(\frac{1}{\mathrm{R1}}-\frac{1}{\mathrm{R2}}\right)}} \quad \ldots\ldots(3.27)$$

the maximum pressure can also expressed as

$$\mathrm{po} = \sqrt{\frac{1}{\pi(1-\nu^2)}\frac{(\mathrm{Fn}\left(\frac{1}{R_1}+\frac{1}{R_2}\right)}{L\left(\frac{1}{E_1}+\frac{1}{E_2}\right)}} \quad \ldots\ldots(3.28)$$

then all the stress values depends on the pressure angle and the contact with from the center of contact.Thesubsurface stresses can beexpressed as

$$\sigma\mathrm{y} = -\frac{\sqrt{\frac{1}{\pi(1-\nu^2)}\frac{(\mathrm{Fn})\left(\frac{1}{R_1}+\frac{1}{R_2}\right)}{L\left(\frac{1}{E_1}+\frac{1}{E_2}\right)}}}{\pi} z\left(\frac{b^2+2y^2+2z^2}{b}\psi - \frac{2\pi}{b} - 3y\varphi\right) \ldots\ldots(3.29)$$

$$\sigma\mathrm{z} = -\frac{\sqrt{\frac{1}{\pi(1-\nu^2)}\frac{(\mathrm{Fn})\left(\frac{1}{R_1}+\frac{1}{R_2}\right)}{L\left(\frac{1}{E_1}+\frac{1}{E_2}\right)}}}{\pi} z(b\psi - y\varphi) \ldots\ldots(3.30)$$

$$\tau yz = -\frac{\sqrt{\frac{1}{\pi(1-\nu^2)}\frac{(Fn)\left(\frac{1}{R_1}+\frac{1}{R_2}\right)}{L\left(\frac{1}{E_1}+\frac{1}{E_2}\right)}}}{\pi} z^2\psi \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(3.31)$$

Where

R_1, R_2 are radius of cam and follower respectively

E_1, E_2, and ν are material properties of cam and follower. The cam and roller follower are made of the same material of different hardness.

Then the modulus of elasticity is $E_1 = E_2$ = 83.5Gpa and

The poisons ratio is v_1 = v_1= 0.29

The mathematics term pi is π = 3.14

Substitute this values and simplifying gives:the contact width b is

$$b = 0.0000332\sqrt{\frac{(F_n)}{L\left(\frac{1}{R1}-\frac{1}{R2}\right)}} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots....(3.32)$$

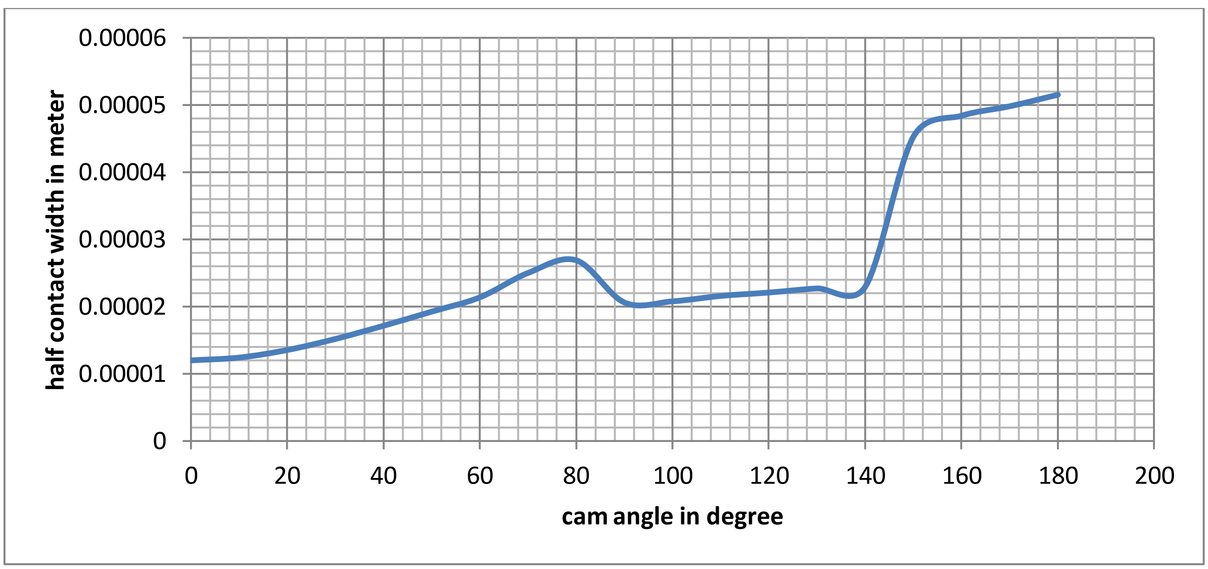


Fig 3.4 shows the distribution of half width as a function cam rotational angle

The maximum compressive pressure can be expressed as

$$po = 23780.32\sqrt{\frac{(F_{s}+F_{in})\left(\frac{1}{R_1}+\frac{1}{R_2}\right)}{Lcos\phi}} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(3.33)$$

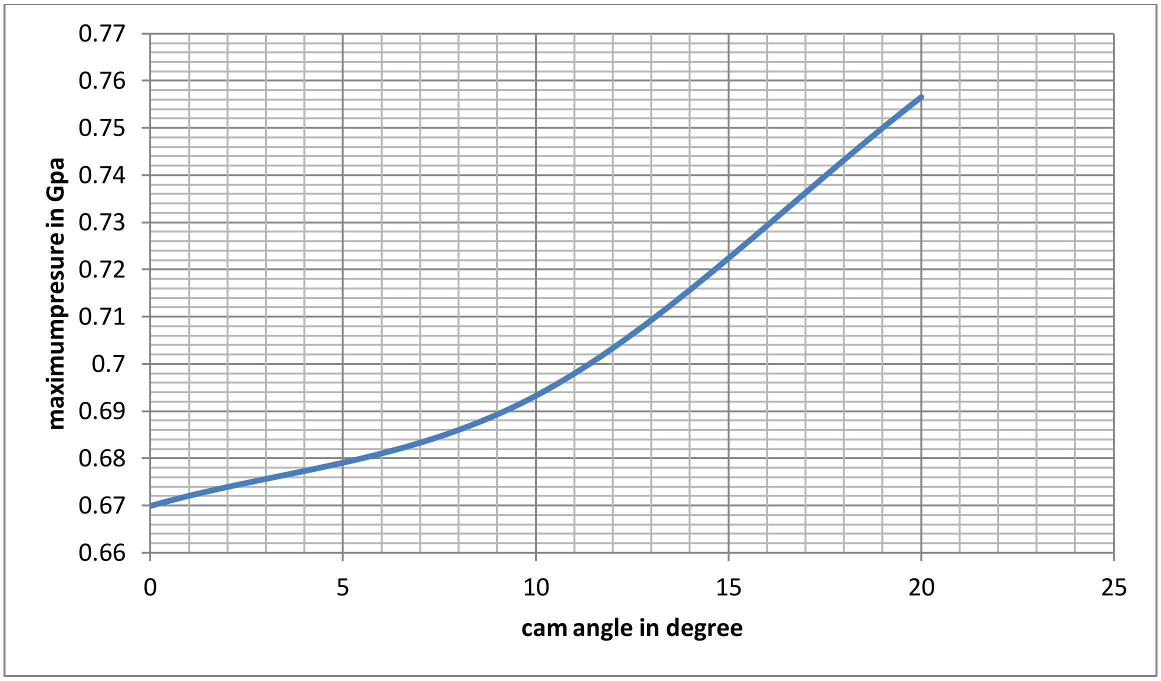


Fig 3.5maximum pressure distribution

$$\sigma y = -7569.51\sqrt{\frac{(F_{s+} F_{in})\left(\frac{1}{R_1}+\frac{1}{R_2}\right)}{L\cos\phi}}\, z \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(3.34)$$

$$\sigma z = -7569.51\sqrt{\frac{(F_{s-} F_{in})\left(\frac{1}{R_1}+\frac{1}{R_2}\right)}{L\cos\phi}}\, z(b\psi - y\varphi) \quad \ldots\ldots\ldots\ldots\ldots\ldots(3.35)$$

$$\tau yz = -7569.51\sqrt{\frac{(F_{s-} F_{in})\left(\frac{1}{R_1}+\frac{1}{R_2}\right)}{L\cos\phi}}\, z^2\psi \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots(3.36)$$

The maximum pressure angle is 30^0 . if the pressure angle is more than 30^0 there will be jumming that affects the motion of follower totally. To avoid this jumming of follower thepressure angle should be kept less than 30°.

The hertizian contact stress can be calculated as

$$\sigma h = 0.564\sqrt{\frac{Fn*E(\frac{1}{R_1}+\frac{1}{R_2})}{2(1-\nu^2)}} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(3.37)$$

The cam stresses analysis can be seen as symmetrical way.that is the rising action and the returning action.

The stress analysis can be numerical calculated using the above derived formulas. The contact pressure, the von misses stress and the shear stress are calculated. the maximum pressure, the von misses stress and the shear stress are is attained at the cam lobe surface when the cam rotational angle is 180° and the pressure angle is 0°. This results is expected due to spring force is high and the contact area is small this combination of high normal load and smaller area results these variables to be maximum. The maximum contact pressure is 1025.15Mpa at the cam lobe.the maximum von misses stress is also at the cam lobe and is value is 520.1 Mpa and the maximu shear is 175.9Mpa.

3.2.5 Wear Analysis

When two bodies are in contact and are in relative motion with respect to each other, wear is expected to develop on the regions of contact. The type of contact that the bodies experience is dependent on how the bodies move relative to each other. One type of contact condition that is of interest is the oscillatory contact. This type of contact condition is characterized by an oscillatory relative motion between the bodies that are in contact. The contact between a cam and folllower is an example of this type contact.

Generally, wear does not involve a single mechanism, therefore it is advisable to take an integrated wear analysis approach assuming the wear behavior as a system property. In other words wear analysis is not limited to the evaluation of the effects of materials on wear behavior, but recommends changes in contact geometry, roughness, tolerance, and so on so that overall favorable results can be achieved. it is probably accurate to say that there is little incentive for a designer to use any of the wear-equations available in the literature. A scan of many wear models shows considerable incongruity. Equation have either too many undefined variables or too few variables to adequately describe the system". Most of available equations are derived/made for mild wear rate of components. Therefore; it can be said that to estimate wear theoretical equations, experimental coefficients are required.

Most wear models assume linearity, and they often also assume that the wear is directly proportional to the local contact pressure. The most common wear model is named Archard's Wear Law , although Holm formulated the same model much earlier than Archard. However, Archard and Holm interpreted the model differently. There are different wear mechanisms in rolling sliding contact i.e

A) Delamation
B) Abrasion
C) Adhesion
D) Oxidation etc

In developing the wear-prediction methodology it is assumed that all the wear cases tobe predicted fall within the plastically dominated wear regime, where slide velocities are small and surface heating can be considered negligible. Archard's wear law would thus serve as the appropriate wear model to describe the wear, so the above listed wear mechanisms can be described by archad's wear equation. The model is expressed mathematically as follows;

$$\mathrm{V} = \mathrm{k} * \frac{F_n}{H} * s \quad \ldots\ldots(3.38)$$

The normal force can be expressed interms of spring force and inertial force

That is: $\mathrm{Fn} = \frac{Fs - Fin}{\cos\phi}$

Then the wear volume can be expressed as follows:

$$V = \frac{K * (Fs - Fin) * S}{H\cos\phi} \ldots\ldots(3.39)$$

There is small sliding in case of cam and follower contact. So the wear volume can be expressed as the rate of sliding distance. The wear volume per sliding can be expressed as follows;

$$\mathrm{Q} = \frac{K * F_n}{H} \quad \ldots\ldots(3.40)$$

V is the wear volume

Q is the wear volume per unit sliding

F_n is normal load

H is the hardness of the material

where *V* is the volume lost, *s* the sliding distance, *K* the dimensionless wear coefficient, *H*the Brinell hardness of the softer material, and F_n the normal force. The non-dimensioned wear coefficient *K* and the hardness are bundled up into a single dimensioned wear coefficient *k* (Pa-1). It should be noted that the wear coefficient *k* is not an intrinsic material property but is also dependent on the operating condition. The value of *k* for a specific operating condition and given pair of materials may be obtained by experiments. Also worth noting, is that measured values of wear coefficients usually have large scatter and may affect wear predictions significantly. Care should thus be taken in obtaining these values K is the wear coefficient which distigush between the wear mechanisms By considering the area subjected to wear it is possible to express the wear displacement interms of the contact pressure.

Dividing both sides by the subjected area

$$\frac{Q}{A} = \frac{K * F_n}{HA} \quad \text{……………………………………………………………(3.41)}$$

Where K is a dimensionless quantity which is usually called the wear coefficient and which provides a valuable means of comparing the severity of different wear processes.

For engineering applications, and especially for the wear of materials whose hardness cannot readily be defined the wear rate is commonly stated as:

$$\frac{K}{H} = k_i = \frac{V}{F_n}$$

k_i is often called the specific wear rate and quoted in units of $mm^3\ N^{-1}\ m^{-1}$. For a material with a hardness H of 1 GPa (a soft steel, or a hard aluminium alloy, for example), the numerical value of k expressed in $mm^3\ N^{-1}\ m^{-1}$ is exactly the same as the value of K.

Under unlubricated sliding conditions (so-called dry sliding), k_i can be as high as 10^{-2}, although it can also be as low as 10^{-6}. Often two distinct regimes of wear are distinguished, termed 'severe' and 'mild'. Not only do these correspond to quite different wear rates (with k_i often above and below 10^{-4}, respectively), but they also involve significantly different mechanisms of material loss. In metals, 'severe' sliding wear is associated with relatively large particles of metallic debris, while in 'mild' wear the debris is finer and formed of oxide particles. In the case of ceramics, the 'severe' wear regime is associated with brittle fracture, whereas 'mild' wear results from the removal of reacted (often hydrated) surface material.

When hard particles are present and the wear process involves abrasion (by sliding or rolling particles) or erosion (by the impact of particles), then the highest values of K occur.

We can show the range of k_i exihibited under different condtions of wear

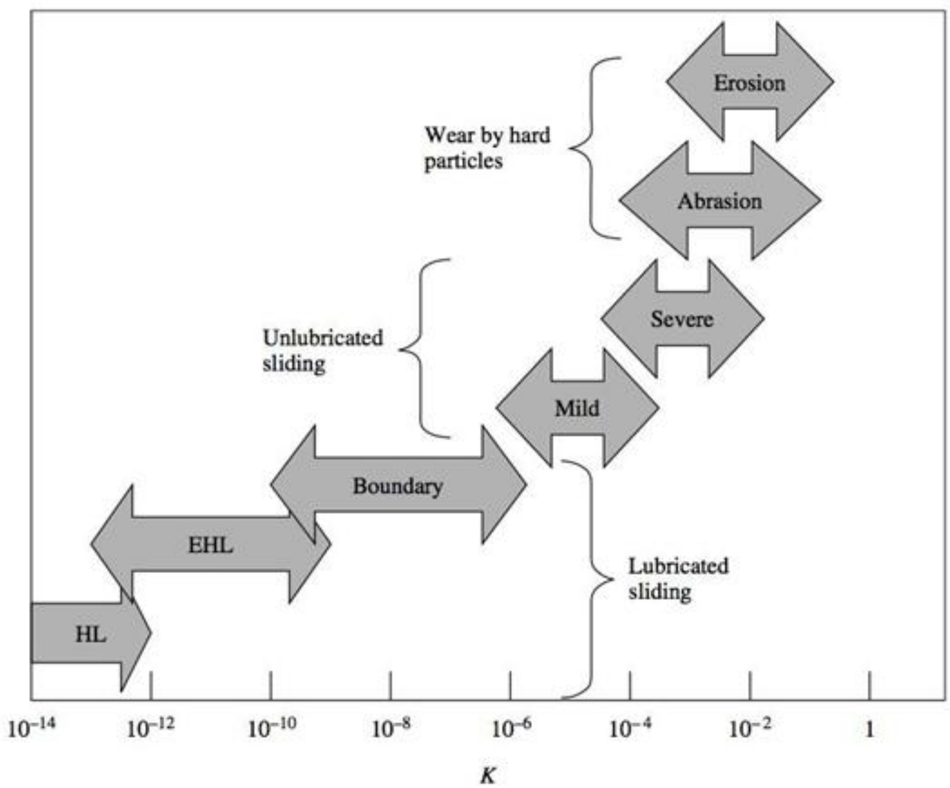


Fig 3.6 wear coefficient values

Wear mechanisms may be briefly classified as mechanical, chemical and thermal wear.

- Mechanical wear describes wear governed mainly by the processes of deformation and fracturing. The deformation process has a substantial role in the overall wear of ductile materials and the fracturing process has a major role in the wear of brittle materials.
- Chemical wear describes wear governed mainly by the growth rate of a chemical reaction film. The growth rate of the film is accelerated mechanically by friction. Therefore, chemical wear is called 'tribochemical wear'
- Thermal wear describes wear governed mainly by local surface melting due to frictional heating. The wear of brittle materials caused by fractures following thermal shocks may also be included in thermal wear.
- Diffusive wear is also included in the term 'thermal wear', since it becomes noticeable only at high temperatures.

Here I am concerned with more descriptive expressions for mechanical wear on cam and follower contacts which has different parts i.e roller, camlobe, cam flank, and basic surface. These parts are damaged by different types of wear depending on the pressure angle, the contact pressure and cam angle.

The cam and follower contact exposed for different types of wear

- Abrasive
- Adhesive
- Plastic deformation
- Fatigue wears

3.2.4.1 Wear of different parts of cam and follower

3.2.4.1.1 Cam lobe wear analysis

This part is exposed too high load i.e the spring compressed extrimly and the normal load becomes high and this force is apllied on small area this produces high pressure that results high temperature and high sliding-rolling action. This high temperature and high sliding- rolling action causes plastic deformation, abrasion and sever scratch.the cam nose cam be stated as cam angle that is from 147 to 180 and from 180 to 212 of cam angle. Then the nose is characterized by the widest wear track of the cam. In fact, these area is affected by the biggest deformations, implying an increased area of contact between cam and roller. Furthermore, fluctuation of the contact width can be observed caused by vibrations and contact pressure variations as well as bumping marks, due to roller bouncing.

3.4.1.1.1 Plastic deformation is a considerable amount of material has been displaced from the top of the lobe surface to the edge where a borr is formed. There is high compression at the cam nose this is dueto the high load is applied on smaller area that results high temperature and pressure. Under such conditions plastic deformation occurs on the rolling sliding surfaces. It can also be verified by calculating the total deformation of the contacting bodies according to Hertz theory

$$\delta = \frac{b^2}{R}$$

Where b is half contact width

R is radius of curvature

The plastic deformation is inversely proportional to the radius of curvature.

3.4.1.1.2 Abrasive wear:

In abrasive wear, the wear volume, V , is given by the following expression

$$V = \frac{\alpha\beta * F_n * s}{H} \quad \text{..(3.42)}$$

where $\boldsymbol{F_n}$ is the load, s the sliding distance, **α** the shape factor of an asperity and **β** the degree of wear by abrasive asperity and H is the hardness. Experimentally, **α** takes a value of about 0.1 and **β** varies between 0 and 1.0, depending on the value of the degree of penetration of an abrasive asperity, the shear strength at the contact interface and the mechanical properties of the wearing material.

Also, If the wear rate is given by a specific wear rate, (w_s = wear volume/load × sliding distance), or a wear coefficient, (K =w_s *H , where H is the hardness), they are derived from equation as follows:

$$K = \boldsymbol{\alpha\beta}$$

$$w_s = \frac{\boldsymbol{\alpha\beta}}{\boldsymbol{H}}$$

The sliding distance can be given as

$$dS = X^*d\theta$$

Where X is the distance from the center of cam to the point of contact

now the wear volume can be expressed as

$$dV = \frac{\alpha\beta * F_n * ds}{H} \quad \text{……(3.43)}$$

the wear can also be expressed as a function of sliding velocity

$$V = k * p * v_s * t \quad \text{……(3.44)}$$

Where p is the contact pressure

v_s is the sliding velocity

This sliding velocity is the part of tangential velocity at the contact point

$$vt = X\dot{\theta}\ \text{cam} \quad \text{……(3.45)}$$

where X is the distance from the center of cam to the contact point that is the position of follower along the cam profile

$$D(\theta)=r_b + u(\theta)$$

u(θ) the displacement of follower

$$u(\theta) = 0.5h(1 - \cos\pi\frac{\theta}{\theta_A})$$

for the rise action and

$$u(\theta) = 0.5h(1 - \cos\pi\frac{\theta - \theta_c}{\theta_b - \theta_c}) \text{ for returning action}$$

Where

θ is the cam angle

$\theta_b, \theta_a, \theta_c$ are transition points

$\dot{\theta}_{cam}$ is rotational speed of cam

Then the sliding velocity can be expressed as

$$Vs = Vt\cos\alpha$$

$$Vs = X\dot{\theta}\ \text{cam}\cos\alpha$$

Where α is the pressure angle

Then

$$V = k * p * \mathrm{x}\dot{\theta}_{\mathrm{cam}}\cos\alpha * t$$

The wear can be expressed as

for the rising action

$$V = k * p * (\mathrm{rb} * 0.5\mathrm{h}(1 - \cos\pi\frac{\theta}{\theta_A}))\ \dot{\theta}_{\mathrm{cam}}\cos\alpha * t \text{..........................(3.46)}$$

for the returning action

$$V = k * p * (\mathrm{rb} * 0.5\mathrm{h}(1 - \cos\pi\frac{\theta-\theta_c}{\theta_b-\theta_c}))\dot{\theta}_{\mathrm{cam}}\cos\alpha * t \text{.........................(3.47)}$$

Under the cam lobe there is two cases

- The rising action
- The returning action

For therise action the roller is in contact with the cam from 147° to 180. The pressure, the distance from the center of cam to the contact point, and the cosine value of the pressure angle are maximum when the pressure angle is 180.this results the wear is sever at the top of the camlobe.the the wear value can be expressed as

$$V = k * p_{max} * \mathrm{x}_{\mathrm{max}}\dot{\theta}_{\mathrm{cam}}\cos\phi * t \text{ ....(3.48)}$$

$V = k * p_{max} * \mathrm{x}_{\mathrm{max}}\dot{\theta}_{\mathrm{cam}} * \cos\phi * t$ is the maximum wear at the the top of the camlobe. So for the lifting action the wear decreases with the increase of cam angle. For the lowering action the roller moves from 180to 212^0. The maximum wear attains at the top i.e at 180 deg. for this case the wear increases with the increase of the cam angle.

So the maximum wear is generally attains at the top the cam i.e at the cam lobe that is at the 180°.

The instantaneous rate of wear i(x,t)

$$i(x,t) = k * p * h * G * \mathrm{Vs} \text{ ....(3.49)}$$

For G inthis equation is a semi elliptical pressure distribution is assumed in theoretical approach . Then the wear occurring during a single passage of contact cam surface is given by

$$W = \int_{t1}^{t2} i(x,t)dt$$

$$W = \int_{t1}^{t2} k * p * h * G * Vs\, dt \quad \text{……………………………………….……(3.50)}$$

Where t_1 and t_2 are the time at the start and end of contact point on cam surface.

3.4.1.2 Basic circle surface wear

The wear rate can be expressed

$$V = \frac{K * Fn * S}{H}$$

The sliding distance s can be given as

$$d\,S = r*d\theta$$

$$V = \frac{K * Fn * r\theta}{H} \quad \text{…………………………………………………………..(3.51)}$$

The angular displacement θ can be expressed as:

$$\Theta = \omega * t$$

$$V = \frac{K * Fn * r\omega t}{H}$$

The rate of wear is

$$Q = \frac{K * Fn * r\omega}{H} \quad \text{………..……………………………………………………(3.52)}$$

Here the wear coefficient , the radius of basic circle, the hardness are constants through out the the analysis, and for the analysis the angular speed is taken to be 1200rpm. Then the wear rate is directly proportional to the normal force applied on the cam basic circle surface.the pressure angle is constant throughout this surfaces.

Where

r is basic circle radius

Θ is cam rotation angle(angular position of cam)

At the basic circle surfaces the pressure angle is 0°. The effect is constant throughout the basic circle surfaces and spring force is small and the contact area is high this shows low pressure is produced this results low wear rate(volume). So the wear rates is not sever and negligible plastic deformation due low pressure produced and temperature is produced on this surface.

3.2.4.1.3 Cam flank wear

The change in the shape of the surface during wear leads to a redistribution of the contact stresses. The shape of cam flank is flat surface. That means it is exposed for sliding tand rolling action. Here it exposed for more sliding action.

This sliding causes different types of wear mechanisms i.e pitting, scuffing, and scoring . Archard's law stays at the basis of different wear mechanisms and the following relation between the rate of wear, the contact pressure and the rate of slippage was proposed:

$$\frac{\partial w}{\partial t} = kpv$$

Where k is wear coefficient

P is contact pressure

v is rate of slippage

$$v = \frac{\partial s}{\partial t}$$

$$\frac{\partial w}{\partial t} = kp\frac{\partial s}{\partial t}$$

$$w = \int_{s1}^{s2} kp\partial s \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(3.53)$$

Where s is slippage

This can be generalized as

$w = kps$

Here the sliding distance can be measured directly. The flank length can be divided into different parts. Each part is exposed to different wear amount.the flank part covers from 82° to 147° of the cam rotation angle. If we consider the wear analysis of this part is looks as follows

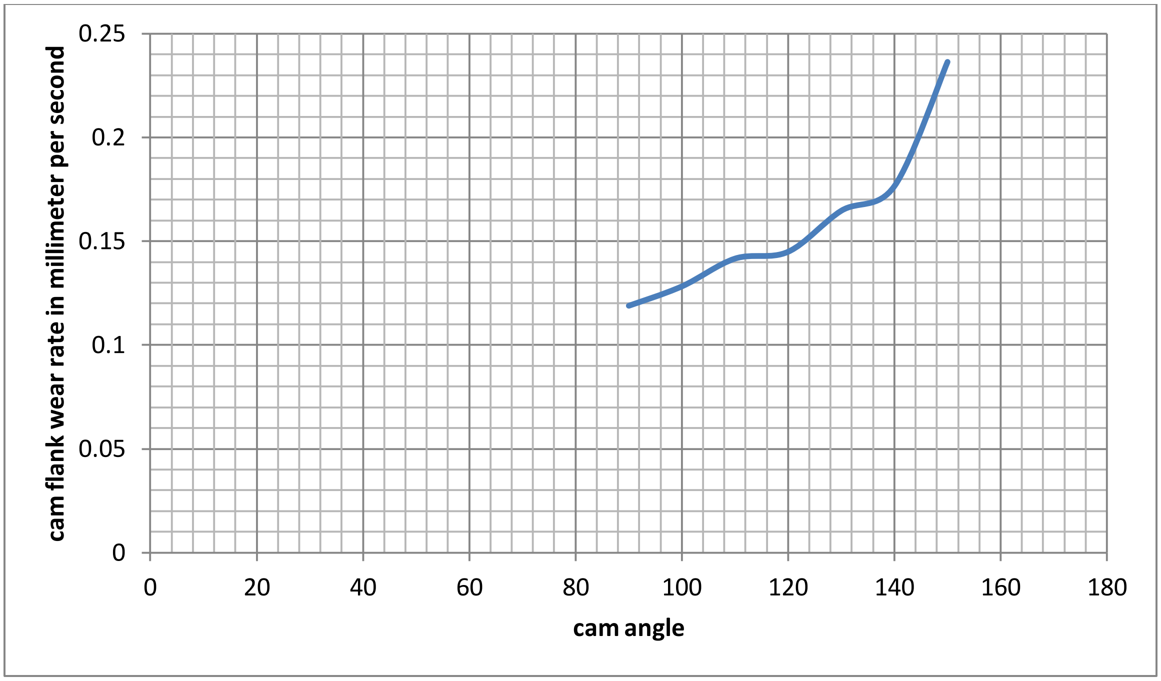


Fig 3.7 cam flank wear rate distribution

As the graph shows the wear rate increases with the increase of cam rotation angle. For the rising action of valve train system. .

This Wear equation can be expressed interms of normal load

$$w = w_s * F_n * s \qquad (3.54)$$

$$w = K * \frac{F_n}{H} * s$$

Where H is the hardness

Fn is the normal load

Generally cam wear rate with respect to can rotation angle and with respect to contact pressure can be analyzed as follows;

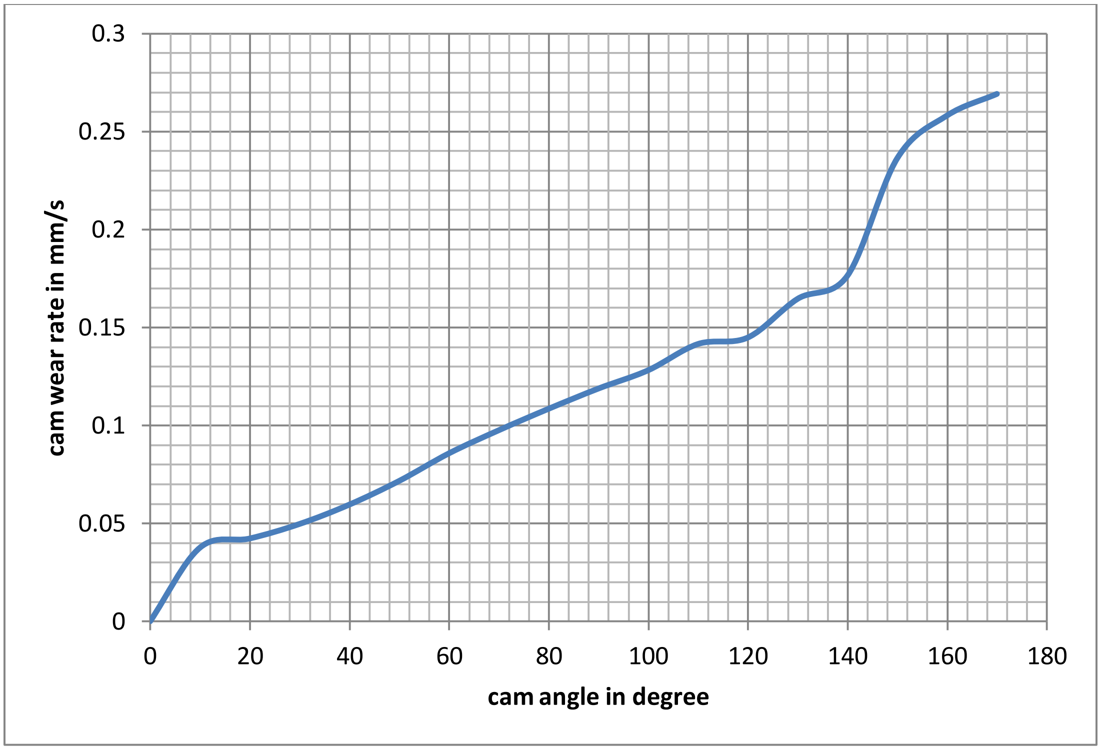


Fig 3.8 cam wear rate distribution

Fig. shows wear rate distribution as a function cam rotation angle. As the graph shows the wear rate for the rising action increases with the increase of cam angle and it is maximum at the camlobe that is when the rotation angle is 180°.

The cam and follower are made of the same material with different hardness. their ability to withstand the wear is different as their hardness. The material used for cam and follower is gray cast iron.the cam and themating follower is made from the same material but different hardness.the minimum hardness for cam surfaces should be 240–300 HB. with 3mm hardened layer at the lobe and cam flank. The minimum hardness of follower is 310–380 HB. with the hardened layer of 3mm. the follower is more hardned than cam.then the above derived wear model is the wear of the softer material. The wear of the the harder material(follower) is can be described as

$$V_2 = \left(\frac{H}{H_2}\right)^2 * \mathrm{V} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(3.55)$$

Where V_2 is the wear volume or wear rates of the follower

H_2 identation hardness of the follower

V is wear volume of the cam

H is the hardness of cam

These formulas merely determine the amount of wear which will be formed. But does not predict the effect of wear particle on a given system.

Substitute the hardness of cam anf roller follower and simplify it the wear rate of follower can be calculated as

V_2 = 0.623 V

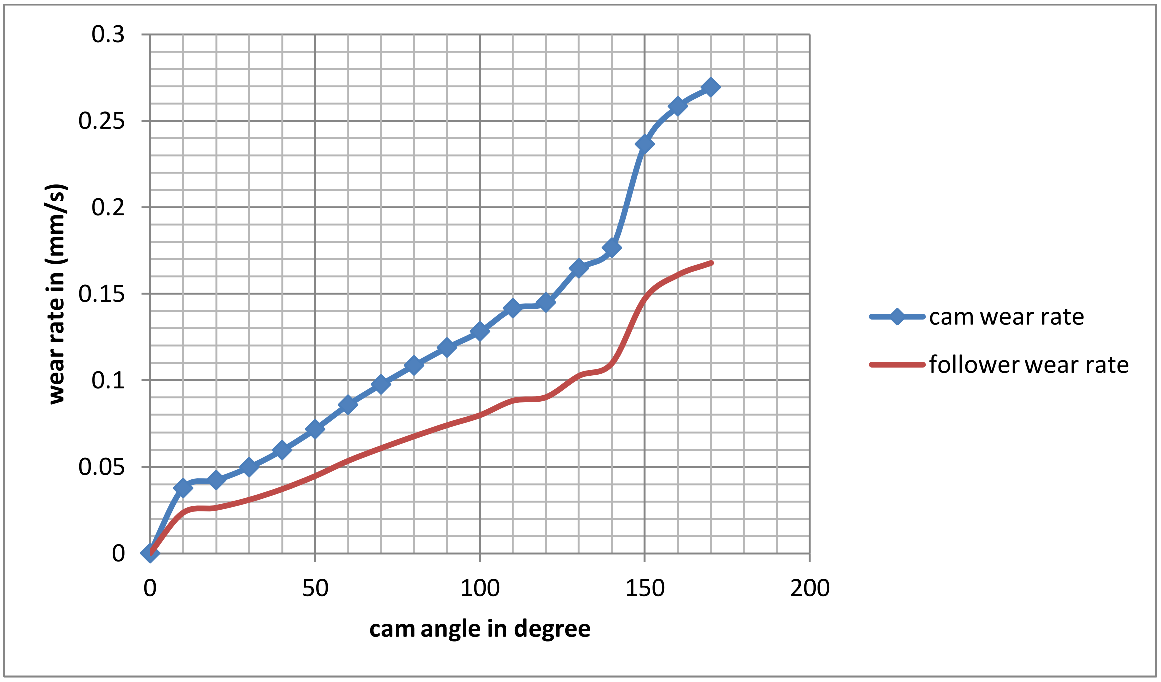


Fig 3.9 wear rate distribution of cam and follower as a function cam angle

As the fig shows the both the wear rate of cam and follower increases with respect to the cam rotational angle and the wear rate is sever at the cam lobe .

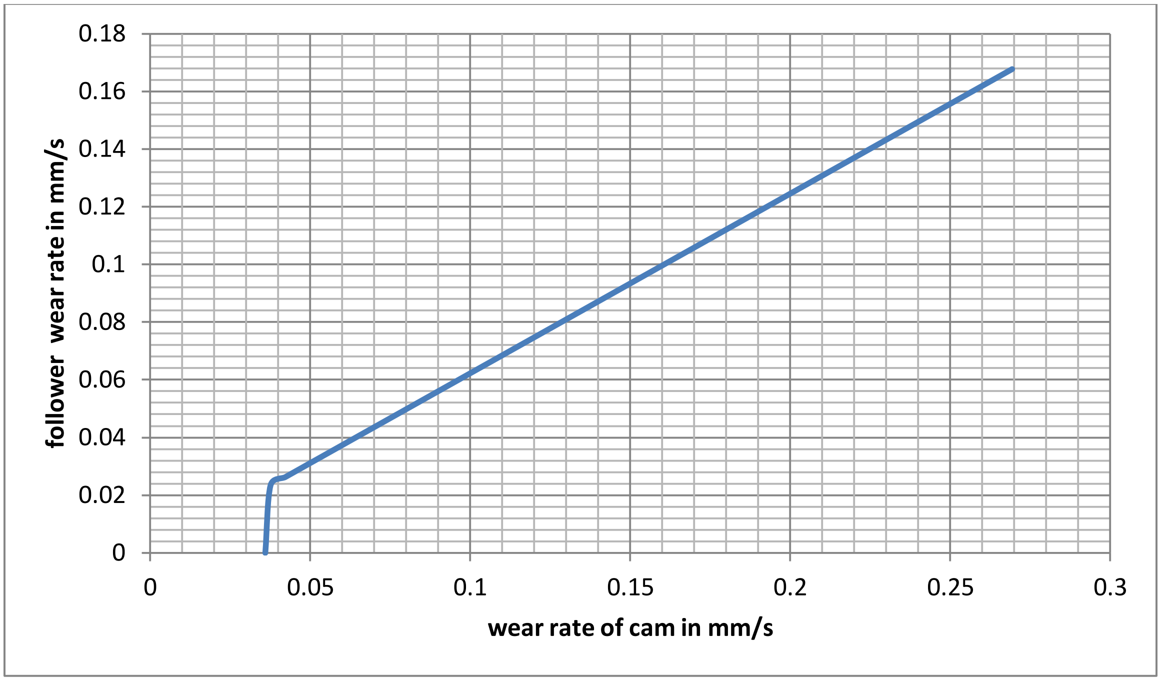


Fig 3.10 follower wear rate distribution as a function cam wear rate

The wear rate ofroller follower can be also interpret withrespect to the cam wear rate and it is depicted above . as shown the wear rate of roller follower increase with the increase of cam wear rate.

3.2.6 Wear displacement

The wear displacement can be drived from the wear volume

Let h is the wear depth

A is the contact area and

V is the wear volume

The wear volume/ the wear rate (V) = hA

$V = k\frac{F_n}{Hs}$ the can be expressed as

$$hA = k\frac{F_n}{Hs}$$

this can be simplified as

$$h = k\frac{F_n}{HsA} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots..(3.56)$$

but

$\frac{F_n}{A}$ is the pressure p

then $h = k\frac{P}{Hs}$ ……………………………………………………….(3.57)

for rolling contact there is small sliding then the wear displacement can be expressed as

$\frac{dh}{ds} = k\frac{P}{H}$ with respect to infinitesimal sliding displacement.

So the rate of wear displacement with respect to the sliding distance is directly proportional to the contact pressure and inversely proportional to the hardnes of the material.the sliding distance cam be expressed as afunction of time and sliding velocity this depicted below

Let us consider rollers follower and cam of radius R1 and R2 respectively. The follower and cam are pressed together by force *FN* and rotate at angular velocities ω1 and ω2 , respectively. The peripheral velocities of the contact surfaces are $v1 = \omega 1 \cdot R1$ and $v2 = \omega 2 \cdot R2$. The wear of the contact surfaces is assumed to be properly described by the following wear model:

$$\frac{dh_i}{dt} = k_i * \mathrm{p} * V_{i,s} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(3.58)$$

where i = 1 for cam and i = 2 for roller follower; *hi* is the wear depth at a point on surface *I* when it rubs against the opposite contact surface, *ki* is the wear coefficient for a point on surface *i* when it rubs against the opposite contact surface, *p* is the local contact pressure, and *vs,i* is the sliding velocity at a point on surface *i* sliding against the opposite interacting surface. The sliding velocity, *vs,i* , for points on both contact surfaces equals

$$vs,i = v1 - v2$$

then the above equation can be rewritten as

$$\frac{dh_i}{dt} = k_i * P * |V_1 - V_2|$$

$$dh_i = (k_i * \mathrm{p} * |V_1 - V_2|)\, dt$$

$$\int_0^{h_i} dh_i = \int_0^{t} (k_i * \mathrm{p} * |V_1 - V_2|)\, dt \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(3.59)$$

this is for the starting action. if the wear of the next duration is considered the can develop this expression

$$\int_{h_{iold}}^{h_{inew}} dh_i = \int_{t_1}^{t_2} (k_i * \mathrm{p} * |V_1 - V_2|)\, dt$$

This integral equation thus can be reformulated as follows

$$h_{inew} = h_{iold} + \int_{t_1}^{t_2} (k_i * \mathrm{p} * |V_1 - V_2|)\, dt \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(3.60)$$

But $|V_1 - V_2| = V_{i,s}$

and $V_{i,s}$ can be expressed interms of the cam angle

$$V_{i,s} = \omega * \mathrm{R}$$

ω is the angular velocity of cam

R is the radius of curvature

Now the above equation can be reformulated as follows

$$h_{inew} = h_{iold} + \int_{t_1}^{t_2} \left(k_i * \mathrm{p} * V_{i,s}\right) dt$$

Then substitute for $V_{i,s}$

$$h_{inew} = h_{iold} + \int_{t_1}^{t_2} (k_i * \mathrm{p} * \omega * \mathrm{R})\, dt$$

From this equation k_i, and R are constants and $\omega * dt$ is the infinitesimal angular position of the ratating cam

$$h_{inew} = h_{iold} + k_i * R \int_{\theta_1}^{\theta_2} P\, d\theta \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(3.61)$$

This can describe the wear displacement can be estimated in terms of the cam angle

CHAPTER FOUR: FINITE ELEMENT METHOD IN SURFACE WEAR ANALYSIS

4.1 Introduction

The finite element method is a numerical method for solving problems of engineering and mathematical physics. Typical problems areas of interest in engineering that are solvable by the use of finite element method include structural analysis, heat transfer, fluid flow, mass, transport and electromagnetic potential.

For problems involving complicated geometry, loading , and material properties, it is generally not possible to obtain analytical matimatical solutions. Hence we need to rely on numerical methods such as finite element method, for acceptable solutions. The finite element formulation of the problem results in a system of simultinous algebraic equations for solution rather than requiring the solution of differential equations. These numerical methods yield approximate values of the unknowns at the discrete number of points in a continuum. Hence this process of modeling A body by diving it into an equivalent system of smaller bodies or units(finite elements) interconnected at points common to two or more elements (nodal points or nodes) and/or boundary lines or surfaces is called discretization. In finite element method instead of solving the problem for the entire body in one operation on formulates the equation each finite elements and combines them to obtain the solution of the whole body.

Mechanical component is a continuous elastic structure. Then FEA methods divide the structure into small but finite, well-defined, elastic substructures called elements. By using a system of simultinous algebraic equations polynomial functions, together with matrix operations, the continuous elastic behavior of each element is developed in terms of the element's material and geometric properties. Loads can be applied within the element, on the surface of the element, or at the *nodes* of the element. The element's nodes are the fundamental governing entities of the element, as it is the node where the element connects to other elements, where elastic properties of the element are eventually established, where boundary conditions are assigned, and where forces (contact or body) are ultimately applied. A node possesses *degrees of freedom*. Degrees of freedom are the independent translational and

rotational motions that can exist at a node. At most, a node can possess three translational and three rotational degrees of freedom. Once each element within a structure is defined *locally* in a matrix form, the elements are then *globally* assembled through their common nodes into an overall system matrix. Applied loads and boundary conditions are then specified through matrix operations the values of all unknown displacement degrees of freedom are determined.
Once this is done, it is a simple matter to use these displacements to determine strains and stresses through the constitutive equations of elasticity.

The establishment of the finite element model and the finite element calculation of this thesis work were conducted through the software package, **ABAQUS**. **Abaqus** is a suite of powerful engineering simulation programs based on the finite element method,. **Abaqus** is designed as a general-purpose simulation tool, Abaqus can be used to study more than just structural (stress/displacement) problems. It can simulate problems in such diverse areas as **heat transfer**, **mass diffusion**, **thermal management of electrical components**(**coupled thermal-electrical analyses**), **acoustics**, **soil mechanics** (coupled pore fluidstress analyses), and **piezoelectric analysis**. Abaqus offers a wide range of capabilities for simulation of **linear and nonlinear applications**. Problems with multiple components are modeled by associating the geometry defining each component with the appropriate material models and specifying component interactions. In a nonlinear analysis Abaqus automatically chooses appropriate load increments and convergence tolerances and continually adjusts them during the analysis to ensure that an accurate solution is obtained efficiently.

Abaqus/Standard provides several contact fomulations. Each formulation is based on a choice of a contact discretization, a tracking approach, and assignment of “master” and “slave” roles to the contact surfaces. For general contact interactions, the discretization, tracking approach, and surface role assignments are selected automatically by Abaqus/Standard.

Your choice of a tracking approach will have a considerable impact on how contact surfaces interact. In Abaqus/Standard there are two tracking approaches to account for the relative motion of two interacting surfaces in mechanical contact simulations:

- finite sliding, which is the most general and allows any arbitrary motion of the surfaces

- small sliding, which assumes that although two bodies may undergo large motions, there will be relatively little sliding of one surface along the other

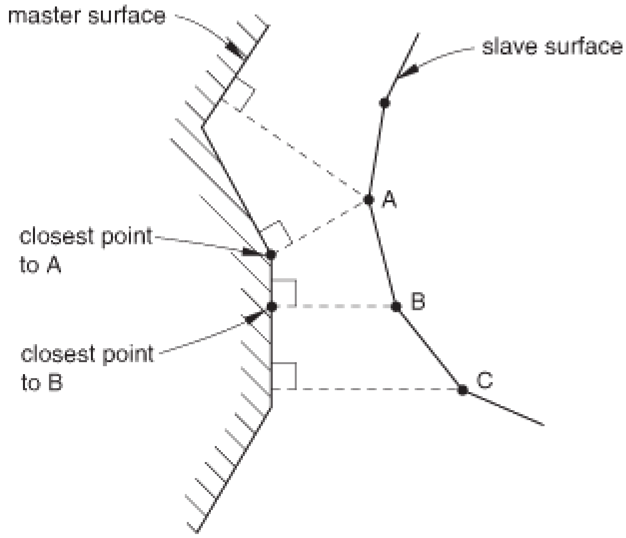
generally abaqus uses two types of contact classification. These are node to surface discritization and true surface to surface discritization.so we can choose between node-to-surface contact discretization and true surface-to-surface contact discretization for each of the above tracking approaches.

4.1.1 Discretization of contact pair surfaces

Abaqus/Standard applies conditional constraints at various locations on interacting surfaces to simulate contact conditions. The locations and conditions of these constraints depend on the contact discretization used in the overall contact formulation. Abaqus/Standard offers two contact discretization options: a traditional “node-to-surface” discretization and a true “surface-to-surface” discretization

4.1.1.1 Node-to-surface contact discretization

With traditional node-to-surface discretization the contact conditions are established such that each “slave” node on one side of a contact interface effectively interacts with a point of projection on the “master” surface on the opposite side of the contact interface Thus, each contact condition involves a single slave node and a group of nearby master nodes from which values are interpolated to the projection point.



Node-to-surface contact discretization.

Traditional node-to-surface discretization has the following characteristics

- The slave nodes are constrained not to penetrate into the master surface; however, the nodes of the master surface can, in principle, penetrate into the slave surface
- The contact direction is based on the normal of the master surface.
- The only information needed for the slave surface is the location and surface area associated with each node; the direction of the slave surface normal and slave surface curvature are not relevant. Thus, the slave surface can be defined as a group of nodes—a node-based surface.
- Node-to-surface discretization is available even if a node-based surface is not used in a contact pair definition.

4.1.1.2 Surface-to-surface contact Discretization:

Surface-to-surface discretization considers the shape of both the slave and master surfaces inthe region of contact constraints. Surface-to-surface discretization has the following key characteristics:

- The surface-to-surface formulation enforces contact conditions in an average sense over regions nearby slave nodes rather than only at individual slave nodes. The averaging regions are approximately centered on slave nodes, so each contact constraint will predominantly consider one slave node but will also consider adjacent slave nodes. Some penetration may be observed at individual nodes; however, large, undetected penetrations of master nodes into the slave surface do not occur with this discretization. compares contact enforcement for node-to-surface and surface-to-surface contact for an example with dissimilar mesh refinement on the contacting bodies.

- The contact direction is based on an average normal of the slave surface in the region surrounding a slave node.
- Surface-to-surface discretization is not applicable if a node-based surface is used in the contact pair definition.

4.1.1.3 Choosing a contact Discretization

In general, surface-to-surface discretization provides more accurate stress and pressure results than node-to-surface discretization if the surface geometry is reasonably well represented by the contact surfaces. Since node-to-surface discretization simply resists penetrations of slave nodes into the master surface, forces tend to concentrate at these slave nodes. This concentration leads to spikes and valleys in the distribution of pressure across the surface. Surface-to-surface discretization resists penetrations in an average sense over finite regions of the slave surface, which has a smoothing effect. As the mesh is refined, the discrepancies between the discretizations lessen, but for a given mesh refinement the surface-to-surface approach tends to provide more accurate stresses

4.1.2 Contact tracking approaches

In Abaqus/Standard there are two tracking approaches to account for the relative motion of two interacting surfaces in mechanical contact simulations.these are finite sliding and small sliding tracking approaches.

The finite-sliding tracking approach: Finite-sliding contact is the most general tracking approach and allows for arbitrary relative separation, sliding, and rotation of the contacting surfaces. For finite-sliding contact the connectivity of the currently active contact constraints changes upon relative tangential motion of the contacting surfaces.

The small-sliding tracking approach: Small-sliding contact assumes that there will be relatively little sliding of one surface along the other and is based on linearized approximations of the master surface per constraint. The groups of nodes involved with individual contact constraints are fixed throughout the analysis for small-sliding contact, although the active/inactive status of these constraints typically can change during the analysis. You should consider using small-sliding contact when the approximations are reasonable, due to computational savings and added robustness

4.1.3 Choosing the master and slave roles in a two-surface contact pair

- Analytical rigid surfaces and rigid-element-based surfaces must always be the master surface.
- A node-based surface can act only as a slave surface and always uses node-to-surface contact.
- Slave surfaces must always be attached to deformable bodies or deformable bodies defined as rigid.
- Both surfaces in a contact pair cannot be rigid surfaces with the exception of deformable surfaces defined as rigid

When both surfaces in a contact pair are element-based and attached to either deformable bodies or deformable bodies defined as rigid, you have to choose which surface will be the slave surface and which will be the master surface. This choice is particularly important for node-to-surface contact. Generally, if a smaller surface contacts a larger surface, it is best to choose the smaller surface as the slave surface. If that distinction cannot be made, the master surface should be chosen as the surface of the stiffer body or as the surface with the coarser mesh if the two surfaces are on structures with comparable stiffnesses. The stiffness of the structure and not just the material should be considered when choosing the master and slave surface. For example, a thin sheet of metal may be less stiff than a larger block of rubber even though the steel has a larger modulus than the rubber material. If the stiffness and mesh density are the same on both surfaces, the preferred choice is not always obvious.

The choice of master and slave roles typically has much less effect on the results with a surface-to-surface contact formulation than with a node-to-surface contact formulation. However, the assignment of master and slave roles can have a significant effect on performance with surface-to-surface contact if the two surfaces have dissimilar mesh refinement; the solution can become quite expensive if the slave surface is much coarser than the master surface.

4.1.4 Fundamental choices affecting the contact formulation: our choice of contact discretization and tracking approach have considerable impact on an analysis. This can be summarized below

Characteristic	Contact formulation			
	Node-to-surface		Surface-to-surface	
	Finite-sliding	Small-sliding	Finite-sliding	Small-sliding
Account for shell thickness by default	No	Yes	Yes	Yes
Allow self-contact	Yes	No	Yes	No
Allow double-sided surfaces	No	No	Yes[1]	Yes
Surface smoothing by default	Some smoothing of master surface	Yes for anchor points; each constraint uses flat approximation of master surface	No	No for anchor points; each constraint uses flat approximation of master surface
Default constraint enforcement method	Augmented Lagrange method for 3-D self-contact; otherwise, direct method	Direct method	Penalty method	Direct method

<table>
<tr><th rowspan="3">Characteristic</th><th colspan="4">Contact formulation</th></tr>
<tr><th colspan="2">Node-to-surface</th><th colspan="2">Surface-to-surface</th></tr>
<tr><th>Finite-sliding</th><th>Small-sliding</th><th>Finite-sliding</th><th>Small-sliding</th></tr>
<tr><td>Ensure moment equilibrium for offset reference surfaces with friction</td><td>No</td><td>No</td><td>Yes</td><td>Yes</td></tr>
<tr><td colspan="5">[1]Double-sided master surfaces are allowed with the finite-sliding, surface-to-surface formulation only if the path-based tracking algorithm is used .Double-sided slave surfaces are allowed with both tracking algorithms if the master surface is not user defined.</td></tr>
</table>

4.2. Modeling of 3D Cam and Follower

In this section finite element approach is used for contact stress analysis in cam and follower system. The approach is based on application of ABAQUS software version 6.10CAE computer programs.

General steps of abaqus software:

1..Modeling the three dimentional cam and follower:

To model the system the geometry cam and follower must be known that is the basic circle radius of cam, radius of cam lobe, radius of follower and the the maximum lift must be specified These parameters are depicted below:

Elements name	value
1.Cam basic circle radius	24mm
2.cam lobe radius	13mm
3.roller radius	9mm

Table 4.1. The geometrical values of cam and follower

The model can be generated using the abaove values

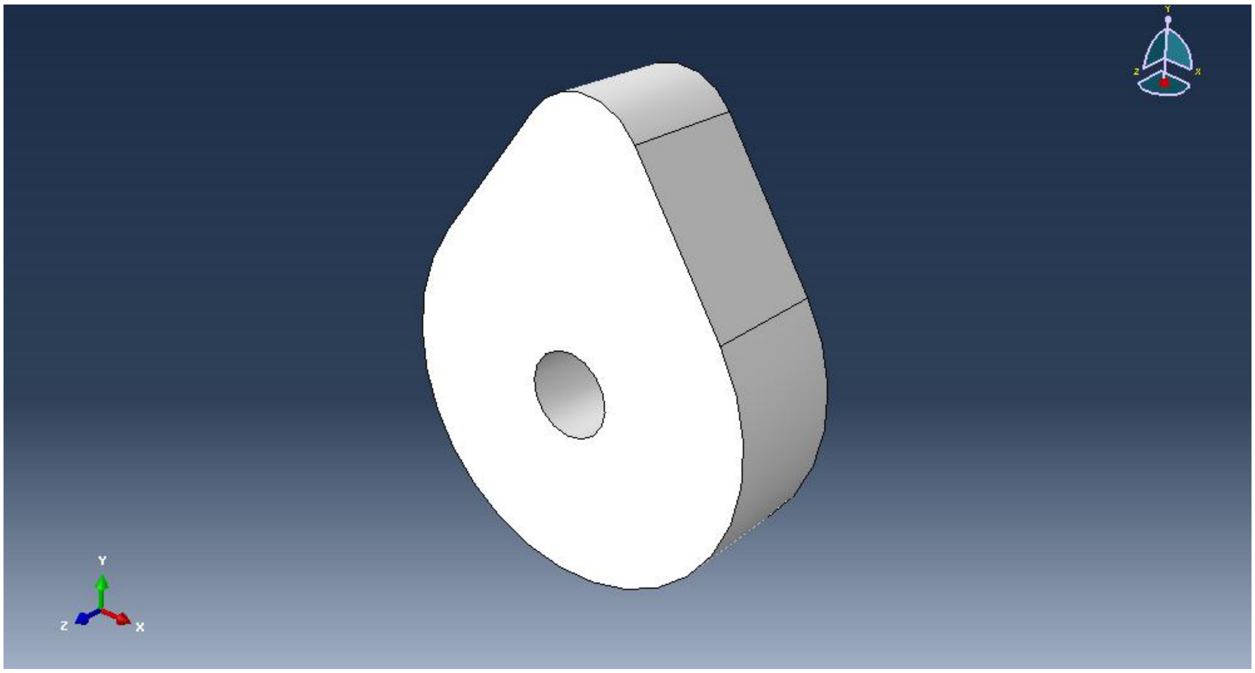


Fig 4.1 3D cam model

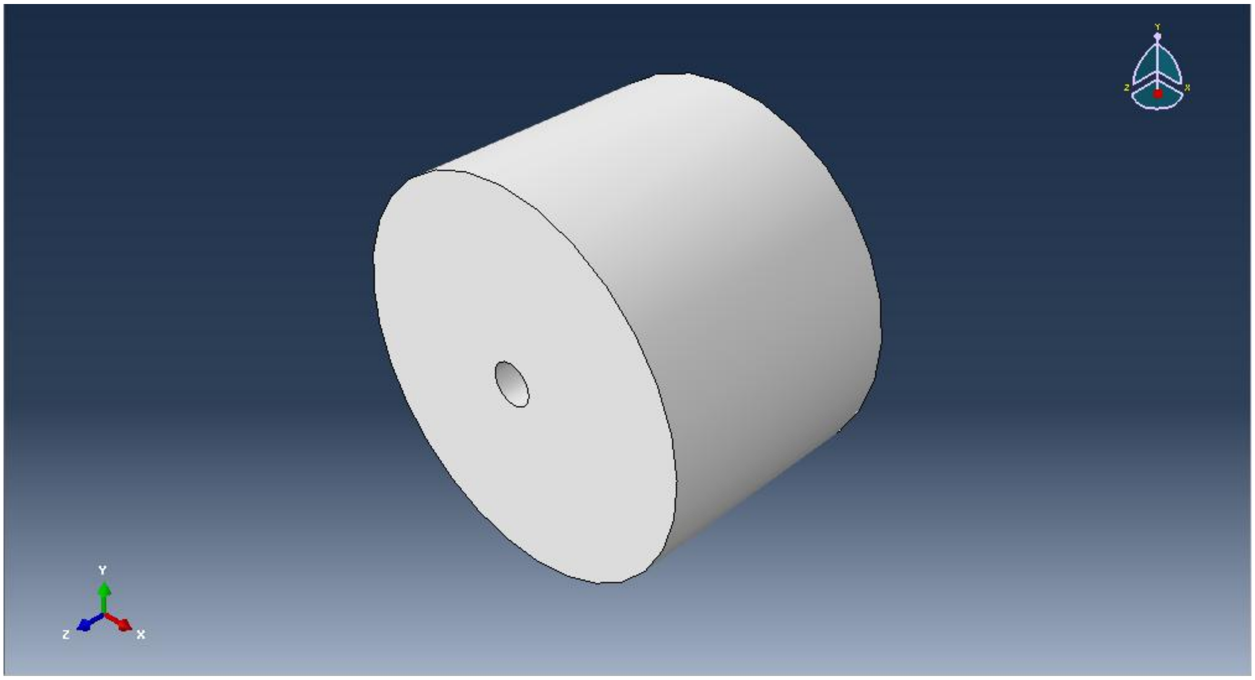


Fig 4.2 3D roller model

2.Thematerials used :

Cam and follower are made from the same material of different hardness. The material used is cast iron so the material properties of cast iron must be specified i.e the modulus of elasticity, poisons ratio, density.

Material property	value	units
Modulus of elasticity	83	Gpa
Poisons ratio	0.29	unitless
density	7200	Kg/m^3

Table 4.2 material properties of cast iron

Abaqus devided these properties as general property and elastic property i.e density is the general property of material and the modulus of elasticity and poisons ratio are the elastic properties of material.

3.Creating an assembly

An assembly contains all the geometry included in the finite element model. Each ABAQUS/CAE model contains a single assembly. The assembly is initially empty, even though we have already created the parts. We will create an instance of the parts in the Assembly module to include it in our model. In order to create an instance we have different options i.e. parallel face, face to face, parallel edge, edge to edge, coaxial, coincident points.in this thesis an instace is created by coincident points and selecting parallel faces to label the face direction.

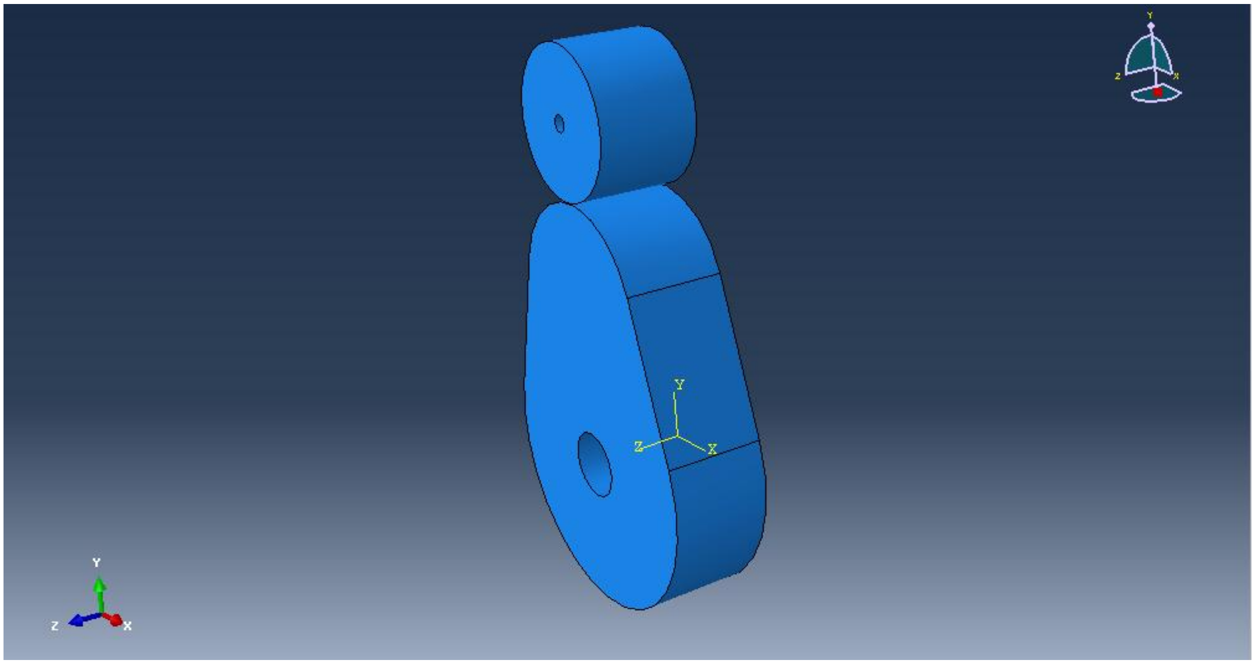


Fig 4.3 assembly of cam and follower system

4.Interaction :

In this step the interacting surfaces should be identified i.e. the targeted surfaces to make contact.

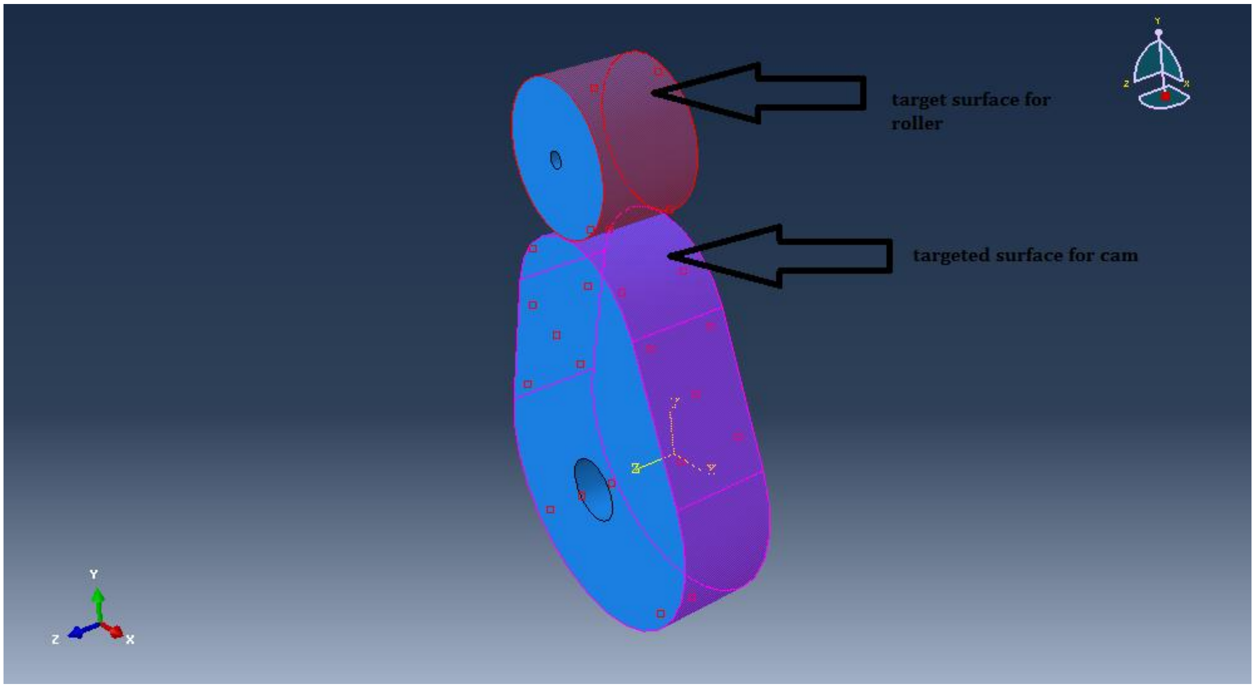


Fig 4.4 interacting surfaces

5.Prescribing boundary conditions and applied loads

In ABAQUS/CAE boundary conditions are applied to geometric regions of a part rather than to the finite element mesh itself. This association between boundary conditions and part geometry makes it very easy to vary the mesh without having to respecify the boundary conditions. In this model, for the first analysis (using TIE between the cam and roller) the part of the cam which is in contact with cam shaft part (the annular region) needs to be constrained in all three directions. This is a simplification for carrying out this analysis.

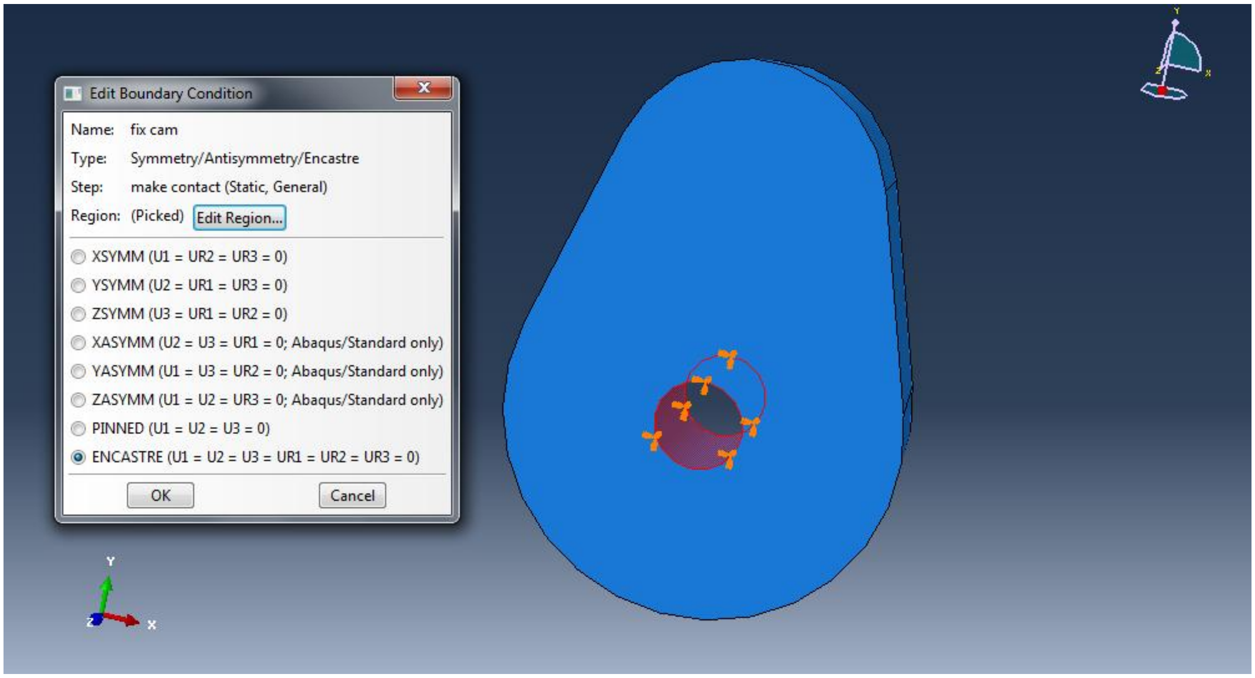


Fig 4.5 boundary conditions of cam

The loads applied on the system are the the normal load which is 729.0344N in magnitude and the weight of roller follower follower.

6.Creating the mesh assembly

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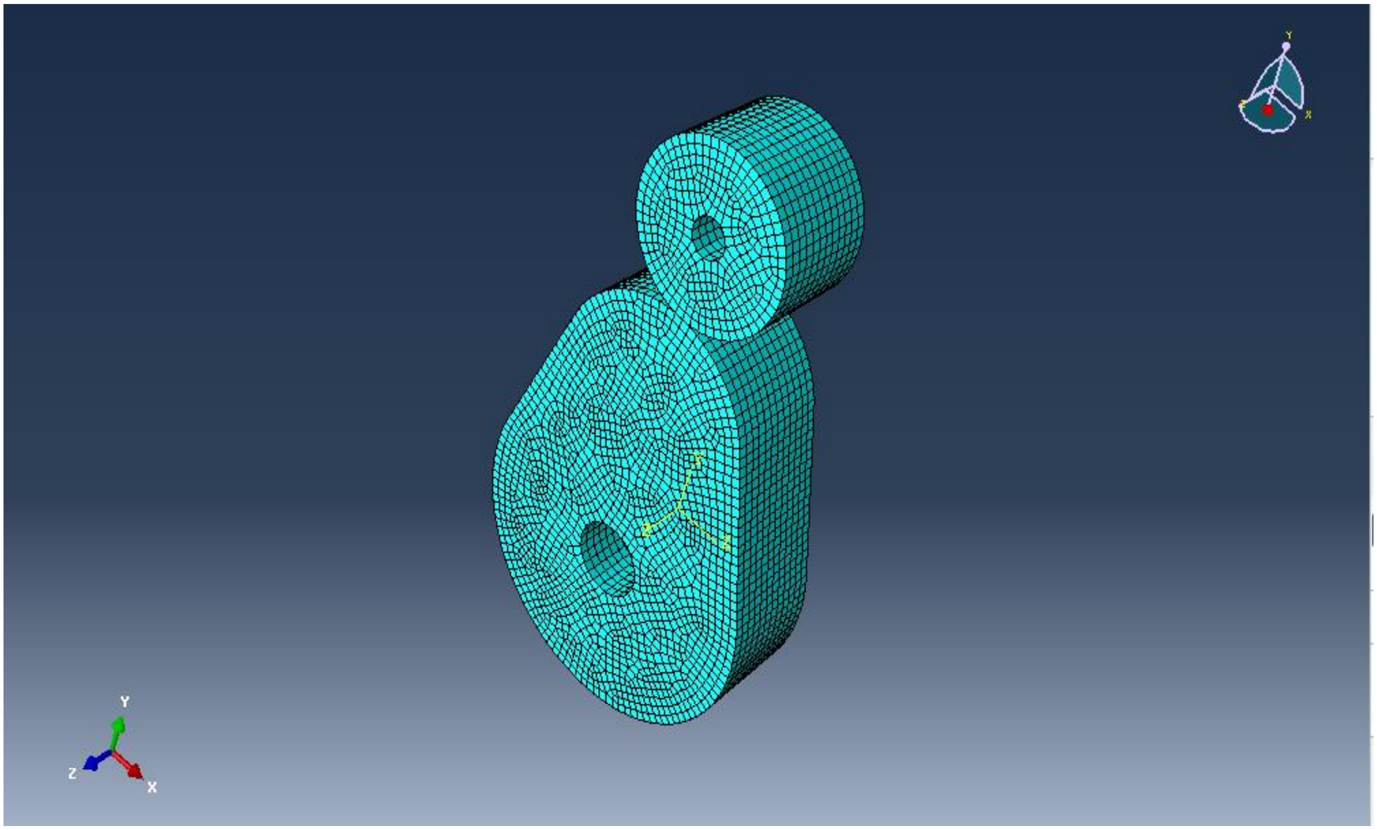

Fig 4.6 meshed assembly

The summary of the general procedure can be described as:

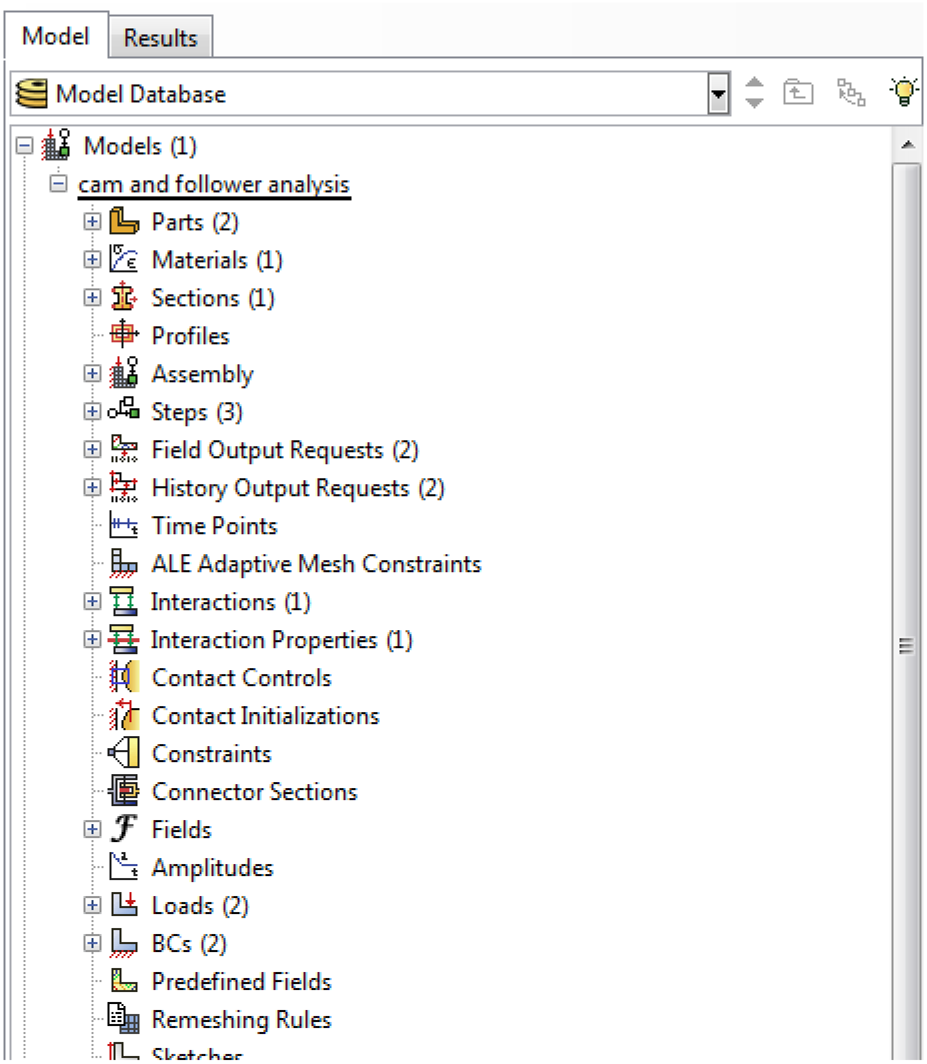

Fig 4.7 shows the summary of cam and follower assembly

CHAPTER FIVE: RESULTS AND DISCUSSION

5.1 Results

In this section results from finite element method i.e. the results using abaqus for contact stress and the theoretical analysis with the application of hertz contact theory for stress analysis will be described and the results will be compared.

The finite element representation of cam and follower systems have three position the at the follower will in contact with the cam surfaces. These are the cam basic surface, the cam flank and the cam lobe. The results of these three conditions are depicted below:

5.1.1 ABAQUS results

5.1.1.1 Cam Basic Circle Results

Contact pressure

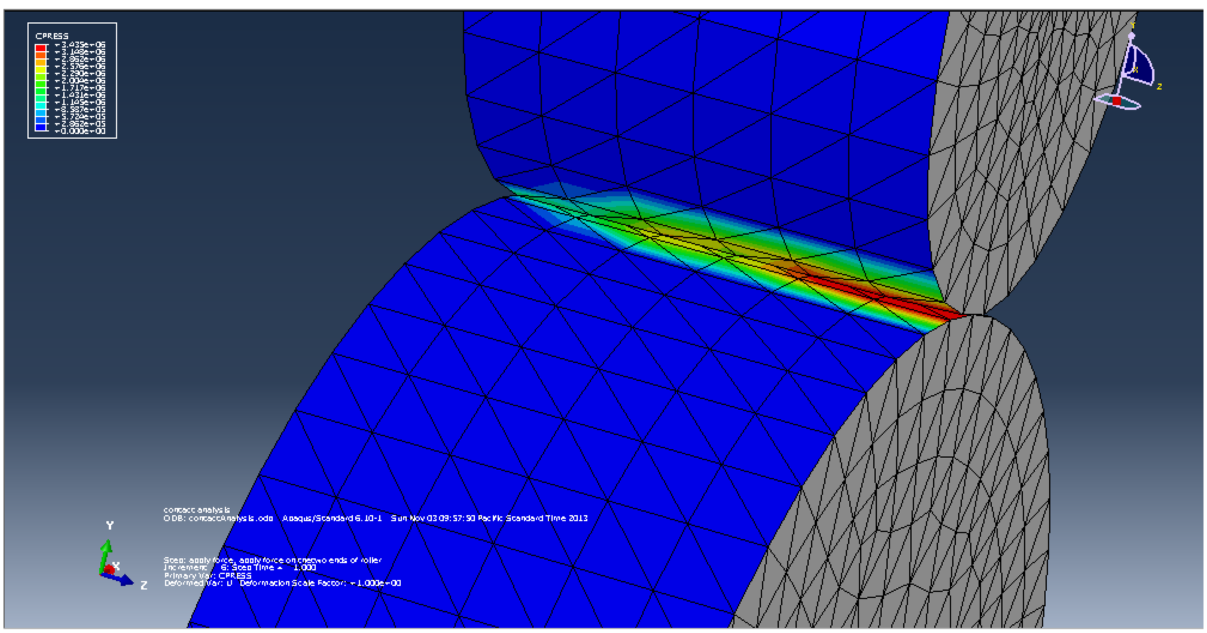

Fig 5.1 the distribution of contact stress between the roller and the cam basic surfaces.

This figure shows the contour plot of distribution of contact pressure along the cam basic circle surfaces when the roller is in contact with the cam basic circle surfaces.

The numerical values of the contact pressure resulted from abaqus are depicted below:

Contact pressure

Minimum 0.

Maximum 552.856Mpa

The contact pressure of the system on the cam flank is 552.856Mpa

Von misses stress

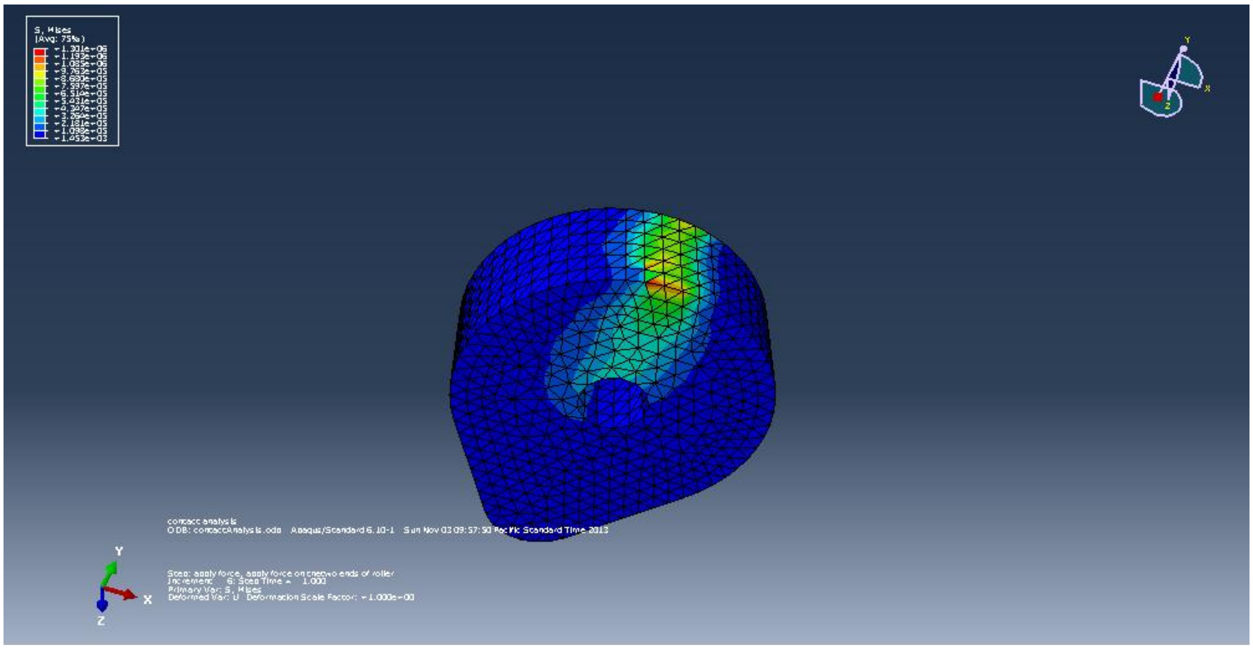

Fig 5.2 the distribution of von misses stress of cam basic circle surface

the resulted numerical values of von misses

Minimum 340.389E+03

Maximum 387.759E+06

The maximum von misses stress is 387.759EMpa .

5.1.1.2 Cam Flank Results

Contact pressure and stresses on the cam flank can be discussed below

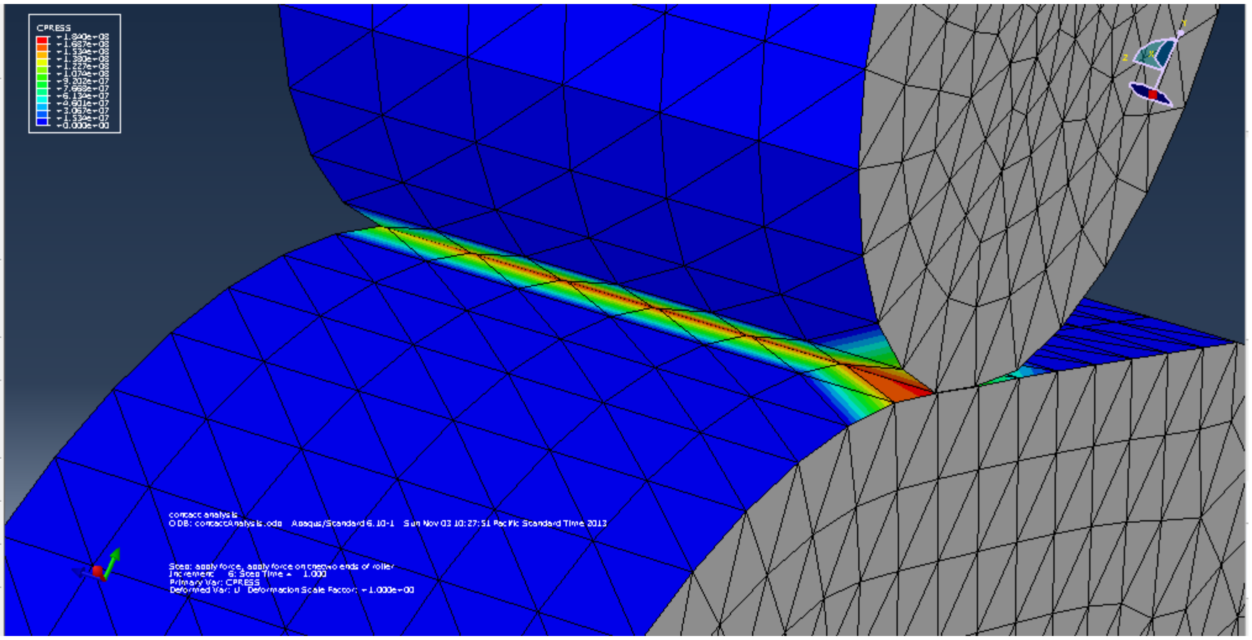

Fig 5.3 the distribution of contact pressure distribution along the cam flank and roller

the numerical values of Contact pressure

Minimum	0.
Maximum	844.985E+06

The maximum contact pressure on the cam flank is 844.985 Mpa .

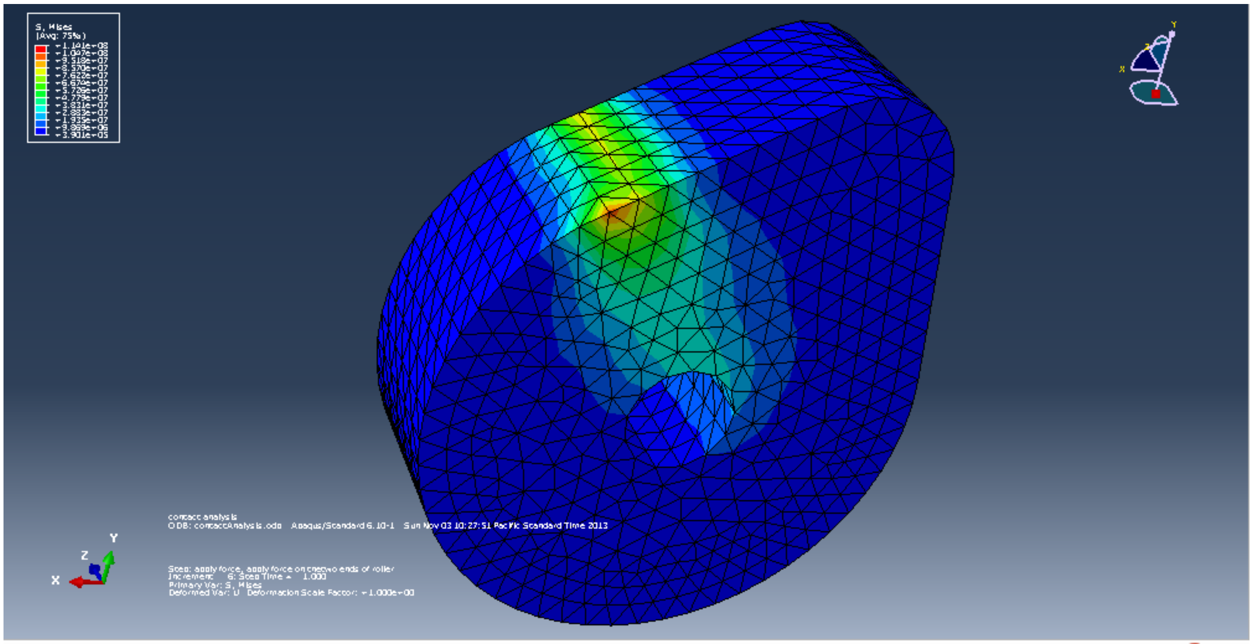

Fig 5.4 von misses stress distribution

	Von misses
Minimum	1.00701E+06
Maximum	527.79E+06

These are the numerical stress values resulted from abaqus for the cam flank part.

5.1.1.3 Camlobe Results

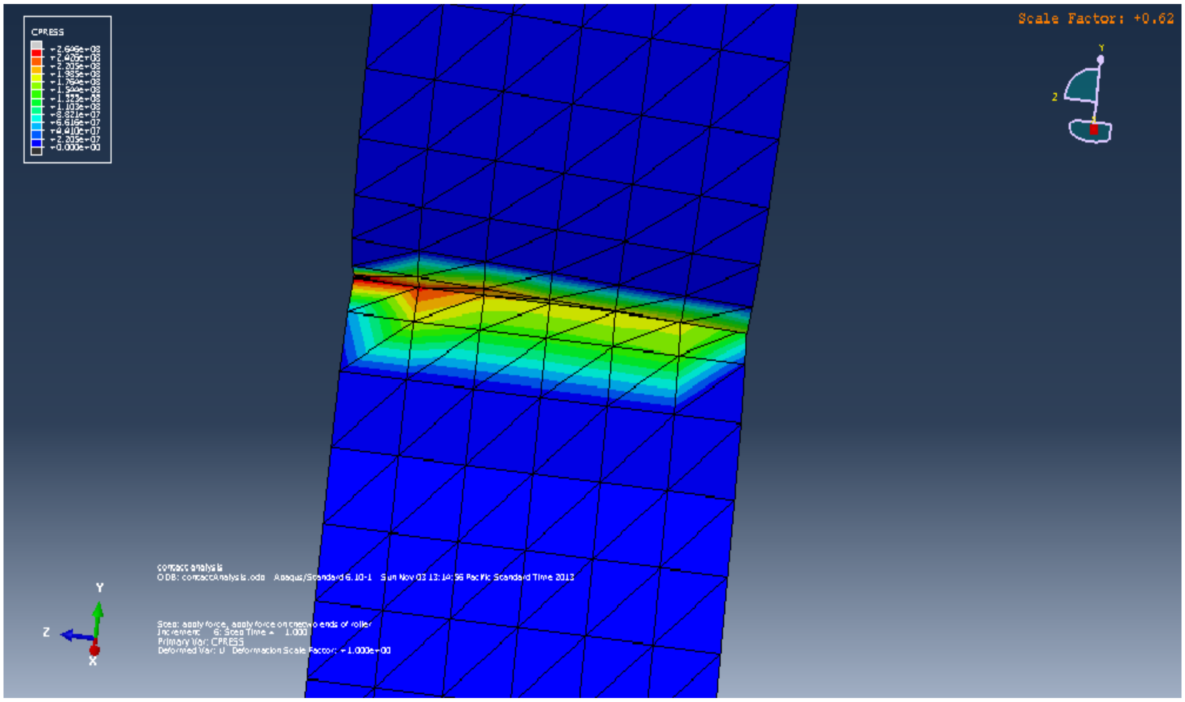

Fig 5.5 contact pressure distribution

	Contact pressure
Minimum	0.
Maximum	1.44134E+09

The maximum contact pressure is at the the node of 3 and at the element of with the magnitude of 1.4413Gpa

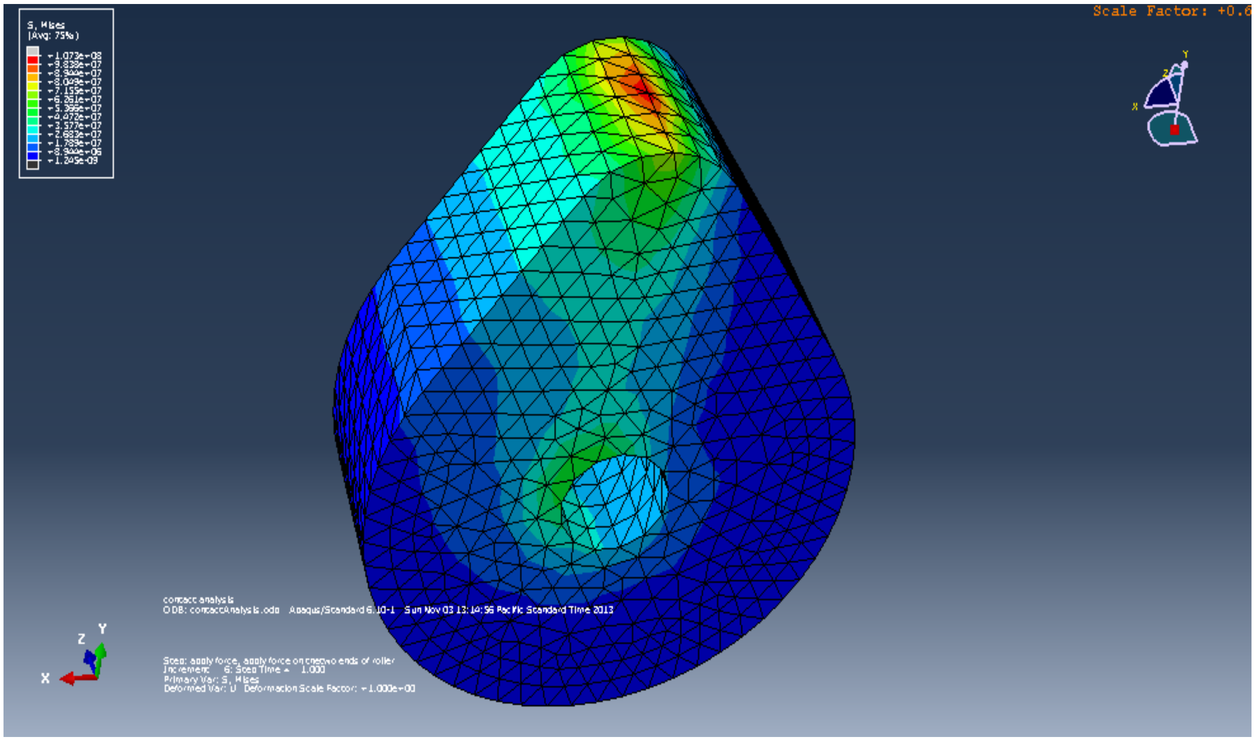

Fig 5.6 von misses stress distribution

	v.misses
Minimum	248.807E+03
Maximum	773.955E+06

The maximum von misses stress resulted from abaqus is 773.955Mpa.

5.1.2 Theoretical contact pressure and von misses stress

The theoretical results calculated from the analytical formulas are discussed below. These theoretical values are calculated based on the analytical analysis. The analytical analysis is based on the hertz contact stress nalysis. based on this results the maximum contact pressure and vonmisses stress are stated below:

5.1.2.1 Cam Basic Circle

Maximum contact pressure 701.825Mpa

Von misses stress	356.2Mpa

5.1.2.2 Cam Flank

Maximum contact pressure	716.975Mpa
Von misses stress	363.2Mpa

5.1.2.3 Cam Lobe

Maximum contact pressure	1025.15Mpa
Von misses stress	520.1Mpa

5.2 discussion

A series of comparisons have been made between the theoretical predictions andABAQUS software analysis results to examine the validity of the present theoretical model. These comparisons include; quantitative comparisons of contact pressure, von misses stresses , shear stresses and wear prediction calculated from the contact stress results of both theoretically as well as ABAQUS results.

5.2.1 Contact pressure

The maximum contact pressure value for the cam when the roller follower is in contact with the cam basic surface obtained analytically was 701.825MPa. The contact pressure analysis has been done by ABAQUS software, where the results have been presented by contours and numerical values. The maximum contact pressure resulted from ABAQUS software is 552.856MPa.

The maximum contact pressure of cam when the roller follower is in contact with the cam flank region is716.975 Mpa and the maximum contact pressure using ABAQUS is 844.985Mpa. and when the roller is in contact with the cam lobe the theoretically calculated maximum contact pressure is 1025.15Mpa and the ABAQUS result is 1441.34Mpa.

The comparison of contact pressure between the theoretical and ABAQUS results is done and results are summarized in table below.

region	Theoretical result	Abaqus result	Error pecentage
Cam basic surface	701.825.1Mpa	552.856Mpa	-11.87
Cam flank	716.975Mpa	844.985Mpa	8.19
Cam lobe	1025.15Mpa	1441.31Mpa	16.87

Table 4.3 ABAQUS and theoretical results numerically

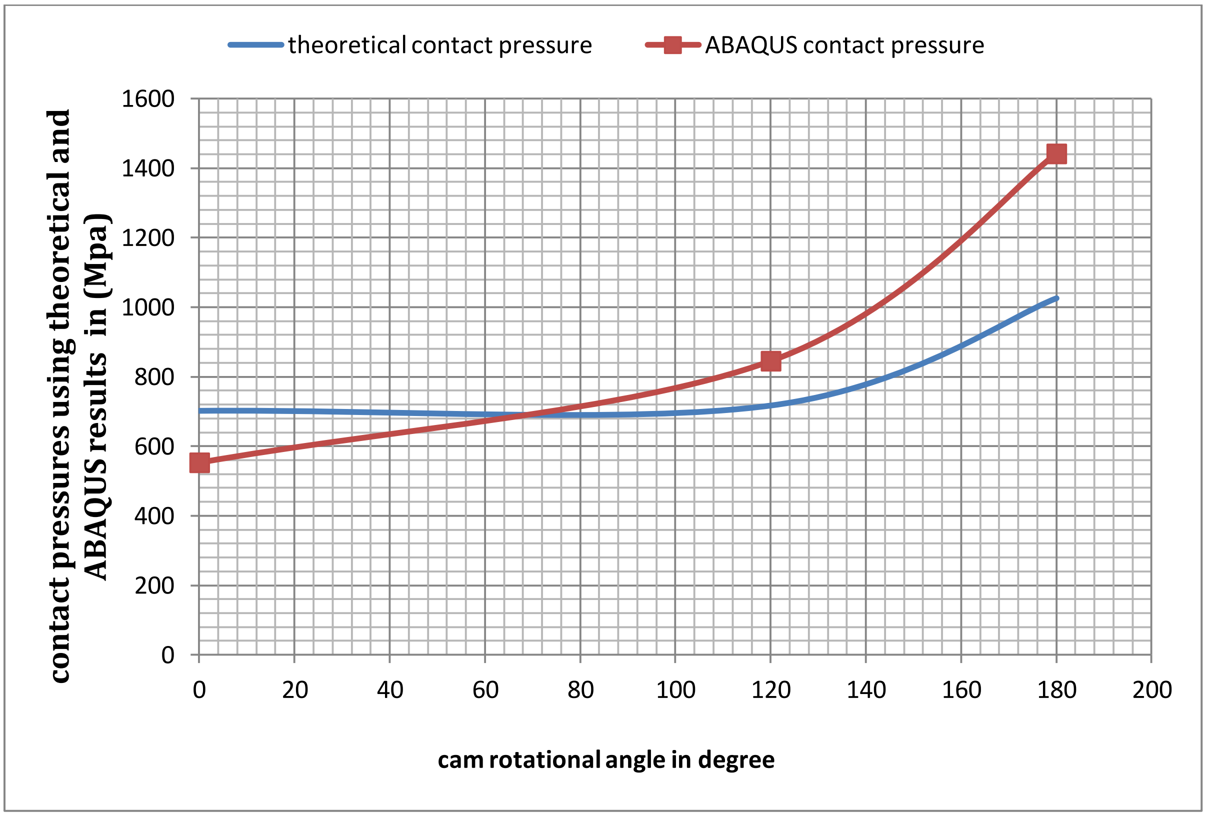

Fig 5.6contact pressure distribution

When we analyze the comparison between the theoretical and ABAQUS software results, both the theoretical and ABAQUS results shows that the contact pressure increases with the increases of cam rotational angle for the rising action of cam. The follower is assumed to be started at the cam basic circle region that is 0° and increases to 180° at the top of cam lobe. The theoretical result shows that the contact pressure increases form 701.825Mpa when $\theta = 0°$ and 1025.15Mpa when $\theta = 180°$. And the ABAQUS software result shows the contact pressure increases from

552.856 to 1441.31Mpa. the variation of this compared values also increase with the increase of cam rotational angle.so the comparison shows these values are in good agreement that the percentage error is less 20%.

5.2.2 Von misses stresses

The maximum von misses stress value for the cam when the roller follower is in contact with the cam basic surface obtained analytically was 356.2Mpa. The von misses analysis has been done by ABAQUS software and t he maximum von misses stress resulted from ABAQUS software is 387.759 MPa.

The maximum von misses stress of cam when the roller follower is in contact with the cam flank region is 363.2Mpa and the maximum contact pressure using ABAQUS is 527.79Mpa. and when the roller is in contact with the cam lobe the theoretically calculated maximum contact pressure is 520.1Mpa and the ABAQUS result is 773.95Mpa.

The comparison of von misses between the theoretical and ABAQUS results is done and results are summarized in table below.

	Theoretical results	ABAQUS results	Error %
Basic circle region	356.2Mpa	387.759 Mpa	4.24
Cam flank	363.2 Mpa	527.79 Mpa	18.47
Cam lobe	520.1 Mpa	773.95 Mpa	19.62

This can be summarized using graph as follows:

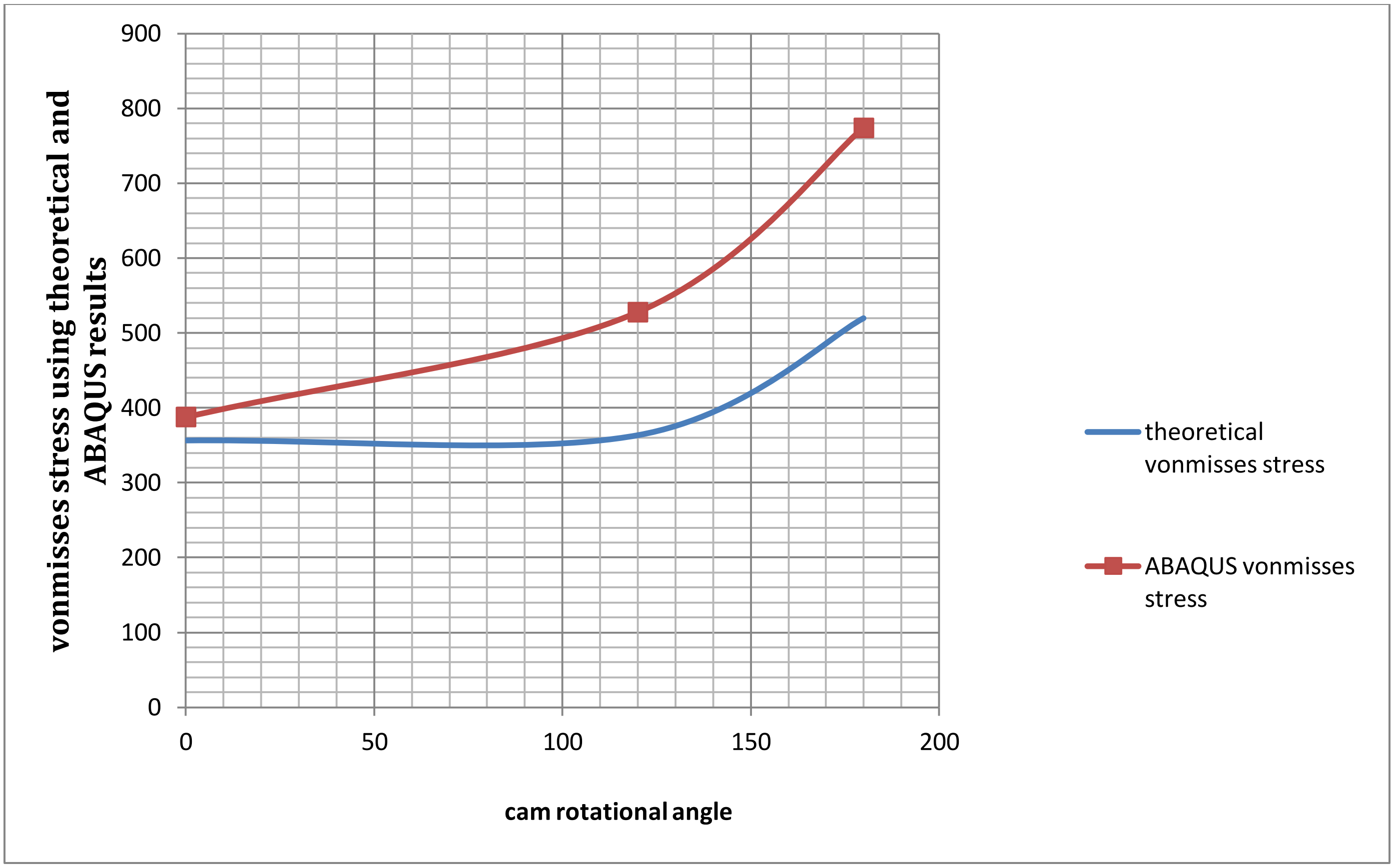

Fig 5.8 von misses stress distribution 1

When we analyze the comparison between the theoretical and ABAQUS software results of the von misses stress , both the theoretical and ABAQUS results shows that the von misses stress increases with the increases of cam rotational angle for the rising action of cam. The theoretical result shows that the the von misses stress increases form 356.2Mpa when θ = 0° and 520.1Mpa when θ = 180°. And the ABAQUS software result shows the von misses stress increases from 387.756Mpa to 773.95Mpa. the variation of this compared values also increase with the increase of cam rotational angle. so the comparison shows these values is 4.2% at the cam lobe and 19.012% at the cam lobe. From this it can be possible to conclude the results are in good agreement that the percentage error is less 20%.

5.2.3 cam and follower wear

The wear of cam and follower can be also calculated using the the results of ABAQUS analysis software. The wear rate can be summarized using graph:

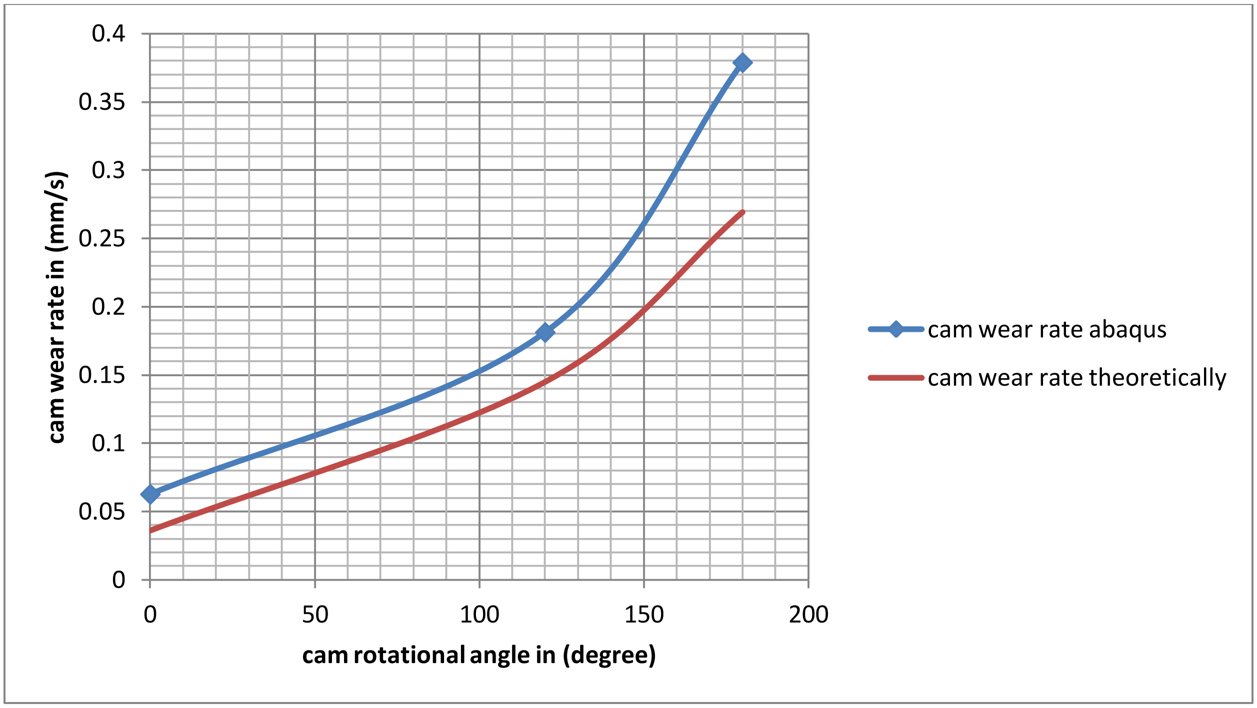

Fig 5.9 cam wears distribution

If we analyze the comparison of wear calculated from the contact result of ABAQUS software and theoretical contact pressure results, both theoretical and ABAQUS results of wear is in good agreement the the wear increases with the increase of cam rotation angle, and the wear is also increases linearly with the contact pressure.

The distribution of follower wear is depicted below it shows that in the roller follower wear rate increases linearly with the cam wear rate. The cam rotation angle had an effect on follower wear rate i.e. the roller follower wear rate increases with the increase of cam angle

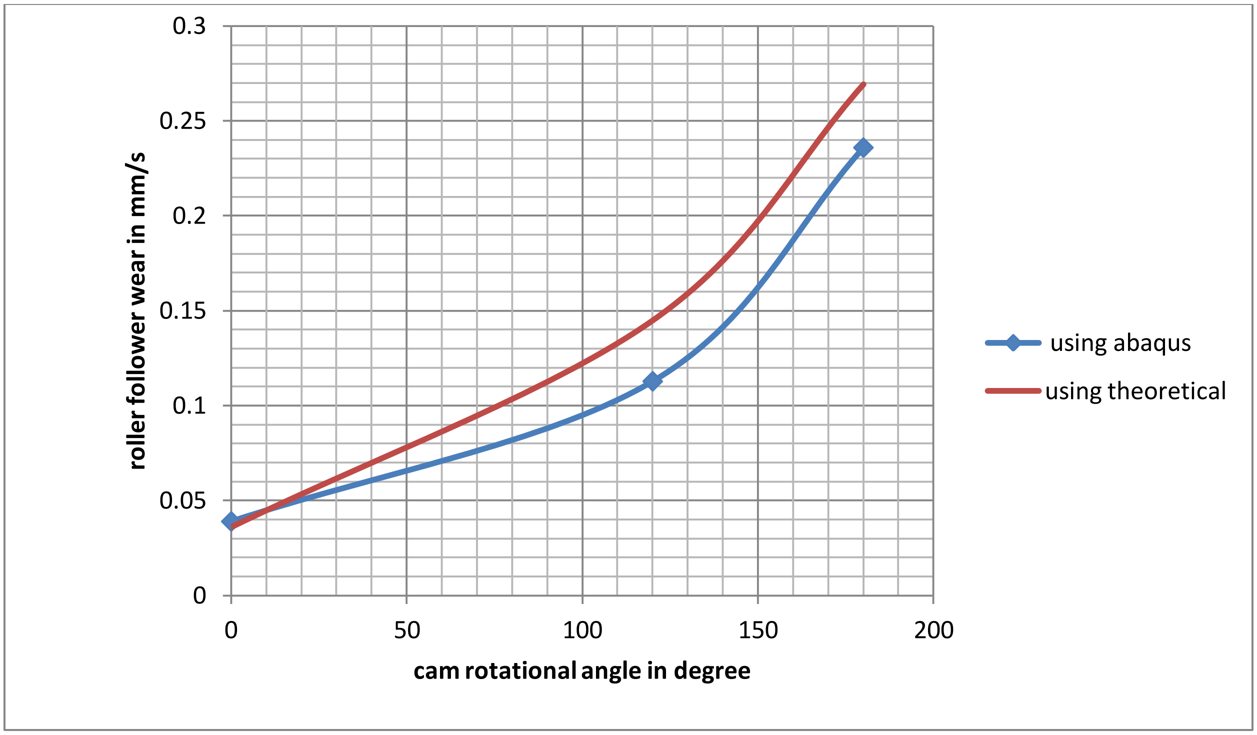

fig 5.10 roller wear distribution

CHAPTER SIX: CONCLUSION AND FUTURE WORK

6.1 Conclusion

A theoretical model has been developed for predicting the valve train tribological performance of cam and follower which includes the contact pressure, von misses stress and surface wear analysis on cam/follower contact. It is the first time that a theoretical model for evaluating the surface wear from the follower and cam contact by taking into account the follower displacement, has been presented .

Furthermore, a multi-aspect comparison between the theoretical predictions and the ABAQUS results has been undertaken. The main conclusions which can be obtained from these theoretical and ABAQUS results and theoretical comparisons are summarized below;

(1) The good agreement between the theoretical predictions and ABAQUS results in many aspects shows that the theoretical model developed in this chapter provides a good basis for the prediction of tribological characteristics of the cam/roller follower and associated Mechanism.

(2) Both ABAQUS observation and the theoretical analysis predict that the cam rotational angle has a significant influence not only on the actual load on the cam but also on the the surface wear of the contacting element. Whilst the pressure angle has a major influence on the follower motion even creating jamm unless it is specified so the effects of the follower/guide contact can also be important and must be taken into consideration.

(3) both the theoretical and ABAQUS software results show that the contact pressure increases with the increase of cam angle for the rising action of the valve train. by comparing the von misses predicted by the theoretical analysis is in good agreement with the ABAQUS analysis results in the case of increasing the von misses stress with respect to the cam angle

(4) The difference in the theoretical and finite element result is due to their basis of derivation that is the finite element formulation of the problem results in a system of simultaneous algebraic equations for solution rather than requiring the solution of differential equations and analytical formulation are based on the differential equations .

(5) The critical position of cam for wear is the cam lobe . this is due to the application of highly compressed spring tha results high normal load on smaller contact area. The cam angle and

pressure angle are an important geometrical parameters during the design of cam and follower system. As it is expected, in this work wear increases with increasing of cam angle with pressure angle of zero degree and it will belower on cam ofsmaller cam angle with the pressure angle of zero degree.

As a result, based on this finding if the pressure angle and the cam angle are the criterias for wear of cam and follower mating then a Cam and follower mating with gray cast iron of desired hardness with relatively smaller pressure angle with maximum limit 30 degree is preferred for good function of engine valve train system.

6.2 future work

In this thesis work contact pressure, von misses stress and wear is studied for different portions of cam at dry condition. Other influencing factors are not studied. So this work is restricted to the specified cases. However, this paper can be extended to other situation listed below.

- numerical method investigations should be conducted on Effect of lubricants in cam and follower contact.
- The wear of cam and follower system should be conducted experimentally and the results should be compared with the theoretical and finite element analysis software i.e. ABAQUS results

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36. ABAQUS 6.10 documentation

Appendix

Car model	Toyota HZJ 80
Engine(motor) Type	1HZ six cylinder
Cam diameter	46mm
Material used	Grey cast iron

Table car model