ADDIS ABABA UNIVERSITY

INSTITUTE OF TECHNOLOGY

SCHOOL OF MECHANICAL AND INDUSTRIAL ENGINEERING

Finite Element Analysis and Optimization of Single Cylinder Engine Crankshaft for Improving Fatigue Life

A Thesis Submitted to the Graduate School of Addis Ababa University in Partial Fulfillment of the Requirements for the Degree of Masters of Science

In

Mechanical Engineering

(Mechanical Design)

By

Tamana Dabasa

Advisor

Dr. Daniel Tilahun

June, 2016

Declaration

I, the under signed, declare that this thesis work is my original work and has not been presented for a degree in any other university, and that all sources of material are duly acknowledged.

Tamana Dabasa__________ ____________________ _____________________

Name Signature Date

Title of thesis: **Finite Element Analysis and Optimization of Single Cylinder Engine Crankshaft for Improving Fatigue Life**

This thesis has been presented for examination with my approval a university advisor

Dr. Daniel Tilahun__________ ____________________ _____________________

Advisor Signature Date

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ABSTRACT

Crankshaft is large volume production component with a complex geometry in the Internal Combustion (ICE) Engine. This converts the reciprocating displacement of the piston in to a rotary motion of the crank. An attempt is made in this paper to study improve fatigue life of single cylinder engine crankshaft with geometry optimization. The modeling of the original and optimized crankshaft is created using SOLIDWORK Software and was imported to ANSYS software for analysis. Finite element analysis (FEA) is performed to obtain maximum stress point or dangerous area, and life of original and optimized crank shaft using the ANSYS software with applying the boundary conditions. The maximum stress appears at the fillet areas between the crankshaft journal and crank web. The FE model of the crankshaft geometry is meshed with tetrahedral elements. Mesh refinement are done on the crank pin fillet and journal fillet, so that fine mesh is obtained on fillet areas, which are generally critical locations on crankshaft. The failure in the crankshaft initiated at the fillet region of the journal, and fatigue is the dominant mechanism of failure.

Geometry optimization resulted in 15% stress reduction of and life is optimized 62.55% crankshaft which was achieved by changing crankpin fillet radius and 25.88% stress reduction of and life is optimized 70.63% of crankpin diameter change. Then the results Von-misses stress, shear stress and life of crankshaft is done using ANSYS software results. From result of geometry optimization parameter like changing crankpin fillet radius and crankpin diameter are changes in model of crankshaft to improvement in fatigue life.

Keywords: Crankshaft, Fatigue, Finite Element Analysis (FEA), Optimization.

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Nomenclatures

FEM	Finite element method
3D	Three dimensions
LEFM	Linear elastic fracture mechanics
PRO/E	Pro Engineer
CrMov13	Crominium Molybdenum vanadium
FEA	Finite element analysis
SAE	Society of Automotive Engineers
AISI	American Iron and Steel Institute
cc	Capacity
$T_{1,}$ T_2	Belt pull
W	Weight of fly wheel
FP	Force of piston
K_b	Combine shock, fatigue factor for bending
K_t	Combine shock, fatigue factor for torsion
$R_{H1\,(F),}$ $R_{H2\,(F}$	Horizontal reactions
P_{max}	Maximum intensity of pressure on piston
D	Bore diameter
M_c	Bending momentum
σ_b	Bending stress
d_c	Diameter of crankpin
Ft	Tangential force
Fr	Radial force
Ø	Angle of inclination (degree)
θ	Angle of twist (degree)
I	Inertia
$R_{H1FT,}$ R_{H2FT}	Tangential horizontal reactions
$R_{H1FR,}$ R_{H2FR}	Radial horizontal reactions
l/r	Ratio of stroke length to crank radius
F_Q	Force of connecting rod

T_e	Twisting equivalent moment
J	Polar moment of inertia
τ	Shear stress
T_c	Twisting moment
M_{ev}	Equivalent bending moment
r	Fillet radius
h	Thickness
σ_{nom}	Nominal stress
σ_{max}	Maximum stress
σ_v	Von $mises$s stress
σ_a	Amplitude stress
σ_m	Mean stress
σ_y	Yield stress
σ_{ut}	Ultimate tensile stress
τ_{nom}	Nominal shear stress
IGES	Initial graphics exchange specification
σ_1	First principal stress
σ_2	Second principal stress
DCI	Diesel common rail-injection
ICE	Internal Combustion Engine

CHAPTER ONE

INTRODUCTION

1.1Background

Crankshaft is one of the most important moving parts in internal combustion engine and it is a large component with a complex geometry in the engine. In general it converts reciprocating motion of the piston is linear and is converted into rotary motion and vice versa with a four link mechanism [3]. The most common application of a crankshaft is in an automobile engine. However, there are many other applications of a crankshaft which range from small one cylinder lawnmower engines to very large multi cylinder marine crankshafts and everything in between [2]. A crankshaft consists of cylinders as bearings, plates as the crank webs and crank-pin. The main components of a crankshaft are shown in Figure 1.2.

Crankshaft experiences large forces from gas combustion. This force is applied to the top of the piston and since the connecting rod connects the piston to the crank shaft, the force will be transmitted to the crankshaft. It must be strong enough to take the downward force of the power stroke without excessive bending so the reliability and life of the internal combustion engine depend on the strength of the crankshaft largely.

Section geometry changes in the crankshaft cause stress concentration at fillet areas where bearings are connected to the webs of the crank. In addition, these component experiences both bending and torsional load during its life service. Therefore, areas of filleted portion are locations that experience the most critical stresses during the service life of the crankshaft [27]. The type of crankshaft depends on number of crankpin. The number of crank-pins depends on the type of engine and number of cylinders. A single cylinder engine will have only one crank-pin and two webs. Multi-cylinder engines will have one crank-pin per piston if the engine is a straight engine, meaning that all cylinders are in a line. If the engine is a V-engine, one bank of cylinders on each side of the crankshaft, two pistons will attach to the same crank-pin. Commonly a crankshaft will be classified by the number of “crank throws” or simply “throws”, which simply refers to the combination of the two webs and crank-pin. There are several different material options available for manufacturing crankshafts, with the two most popular being steel and iron. Crankshafts can be machined by forging, casting and billet. Machining a crankshaft from a billet is not typically done due to the prohibitively long machining times; however for low production custom pieces it is still done. The steel crankshaft is usually forged to near net shape and then finished by machining processes. Generally a crankshaft can be classified as forged steel or cast iron, however, within these two categories there are many options.

Production

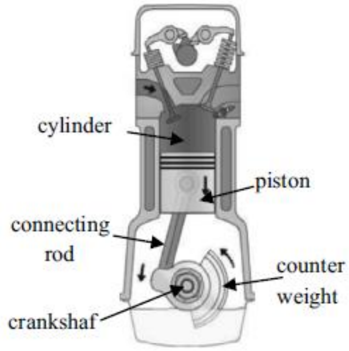
Figure 1.1 Production method of crankshaft

A single-cylinder engine is a basic piston engine configuration of an internal combustion engine. It has only one crank-pin and two webs. Multi cylinder engines will have one crankpin per piston. It is often seen on motorcycles, motor scooters, and has many uses in portable tools and garden machinery. It has been used in automobiles and tractors.

Figure 1.2 Crankshaft nomenclatures

Single-cylinder engines are simple, compact and economical in construction. The vibration they generate is acceptable in many applications. As a result, these engines are important auxiliary agricultural tool in rural areas for water pumping and lawnmower.

Figure 1.3 Three examples of auxiliary agricultural vehicle driven by the single cylinder diesel engine [9]



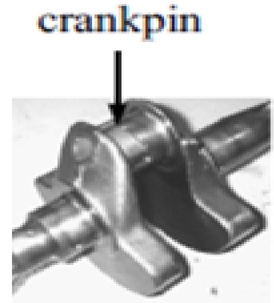


Figure 1.4 single cylinder engines

Figure 1.5 single cylinder engines crankshaft

The objective of this study was to analyzing stress in critical location for improving fatigue life with geometry optimization of a single cylinder engine typical to that used in a riding lawnmower. Rate failure of crankshaft is not limited to selecting a material, such as steel or iron, a process, such as forging or casting, and surface treatment.

Farzin H. Montazersadgh et al [5] suggested the modifications for improvement in fatigue life such as changing the main bearing radius, crank pin radius, fillet radius of crankpin pin and changing the type of material in crankshaft, which are very common modifications usually done in crankshaft geometry.

1.1.1 Failure of a crankshaft

Failure of the crankshaft makes an engine unworkable, resulting into costly procurement and replacement. There are many sources of failure in the engine one of the most common crankshaft failure is fatigue at the fillet areas due to the bending and torsional load caused by the combustion. The moment of combustion the load from the piston is transmitted to the crankpin, causing a large bending moment on the entire geometry of the crankshaft. At the root of the fillet areas stress concentrations exist and these high stress range locations are the points where cyclic loads could cause fatigue crack initiation leading to fracture.

The failure of crankshafts classified into three categories [12], [13].

Operating sources

- ✓ Oil absence
- ✓ Defective lubrication on journals
- ✓ High operating oil temperature

- ✓ Improper use of the engine

Mechanical sources

- ✓ Misalignments of the crankshaft on assembly
- ✓ Improper journal bearings (wrong size)
- ✓ Crankshaft vibrations (like misalignment of the crankshaft, incorrect fillet size)

Repairing sources

- ✓ Misalignments of the journals (due to improper grinding)
- ✓ Misalignments of the crankshaft (due to improper alignment of the crankshaft)
- ✓ High stress concentrations (due to improper grinding at the fillet radius areas)
- ✓ Improper welding or nitriding

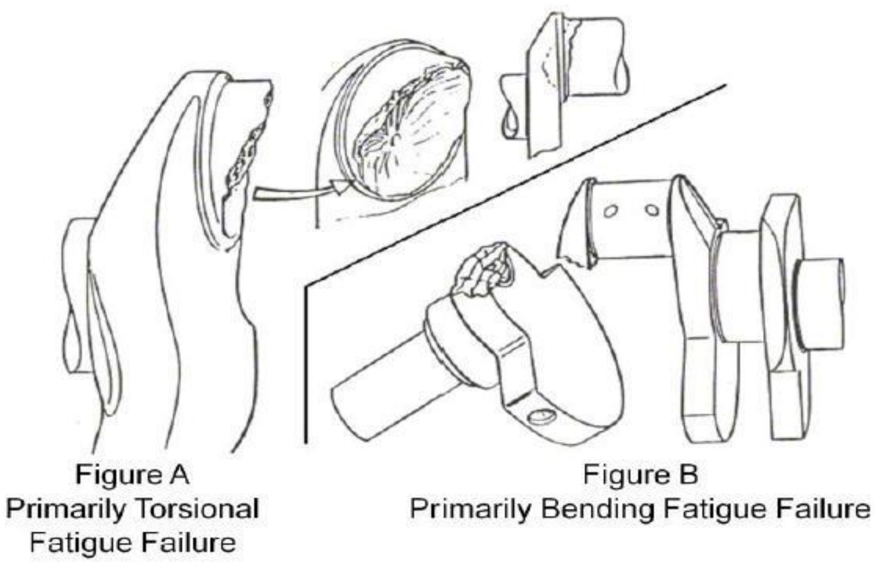


Figure 1.6 Different fatigue failures

1.2 PROBLEM STATEMENT

Crankshaft is one of the critically loaded components in engine. Fatigue is the primary cause of failure of crankshafts due to the cyclic loading and presence of stress concentrations at the fillet areas caused by the bending and torsional load [27]. It exists when cyclic stresses/deformations occur in an area on a component. The fillet regions in crankshafts have been identified as the highest stressed or critical location of a crankshaft and are often the sight of fatigue crack initiation. Therefore due to this, this paper was proposed to analyze the stresses acting on the crank shaft and to improve fatigue life using geometry optimization like increase crankpin fillet radius, increasing crankpin diameter will improve life of crankshaft.

1.3 Objectives

1.3.1 General objectives

- Finite Element Analysis and Optimization of Single Cylinder Engine Crankshaft for Improving Fatigue Life

1.3.2 Specific objectives

- To model single cylinder engine crank shaft using SOLID WORK software with different fillet radius and crankpin diameter
- Analysis of single cylinder engine crank shaft using ANSYS 16 software with different fillet radius and crankpin diameter.
- Optimizing the geometry for reducing the stress, maximize fatigue life of single cylinder engine crankshaft using ANSYS software.
- To evaluate the von-misses stress and shear stress and life at different fillet radius and diameter of crankpin using analytical and ANSYS solution.
- To identify critical portion where stress acting are maximum then reduce maximum stress to reduce rate of failure using geometry change.

1.4 Methodology

The methods employed to achieve the above objectives are

1.4.1 Literature survey

There is a vast amount of literature related to Finite Element Analysis and optimization of Crankshaft. Many research publications, journals, reference manuals, newspaper articles, handbooks; books are available of national and international editions dealing with basic concepts of FEA. The literature review presented here considers failure analysis, optimization and finite element modeling of crankshaft.

1.4.2 Analytical

In the design of the crankshafts, it is assumed that the crankshaft is a beam with two or more supports. The crankpin is designed by considering two crank positions, i.e. when the crank is at dead center and when the crank is at an angle at which the twisting moment is maximum.

In this by using both parameters optimized like increasing crankpin diameter by using distortion energy theory the stress induced in the crankpin is calculated. Shoulder fillet is used for increasing fillet radius to obtain stress induced in the crankshaft.

1.4.3 Modeling and FEM analysis `

The finite element method is numerical analysis technique for obtaining approximate solutions to a wide variety of engineering problems. Finite element modeling of any solid component consists of geometry generation, applying material properties, meshing the component, defining the boundary constraints, and applying the proper load type. These steps will lead to the stresses and displacements in the component. In this study, this component is analyzed using SOLID WORK software for 3D modeling and ANSYS 16.0 software work bench for analysis were performed for both parameters like different fillet radius and different crankpin diameter of crankshafts.

Procedures in FEA

a. Generation of the Geometry of Crankshafts

Finite element analysis of crankshaft is done by geometry generation using SOLIDWORK software and analysis is using through the ANSYS 16 WORKBENCH software. Using proper type of loading and boundary condition are very important in finite element analysis. The solid model generated for the crankshaft is shown in Figure 3.6 and the optimized is drawn by using increasing both parameters like increasing fillet radius of crankpin and increasing crankpin diameter.

b. Mesh Generation

FEA analysis was performed on crankshafts for improving fatigue life, and boundary conditions are applied according to engine mounting condition. Tetrahedral elements were used to mesh the crankshaft finite element geometry. FE models were done for each optimized to analyze stress and improve life of crankshaft with different mesh size or by using mesh optimization to obtain good result which is close to analytical result.

This step involves subdividing the domain into elements and nodes. For continuous system this step is very important and the answers obtained are only approximate. In this case, the accuracy of the solution depends on the discretization used. Import IGES format model in ANSYS

WORKBENCH simulation module. Tetrahedral elements are used to mesh the crankshaft finite element geometry.

c. Loading And Boundary Conditions

Boundary conditions are playing the important role in finite element analysis. Load applied on the component when the crank is at the position of maximum bending moment or is at the dead centre and also load data were taken from the calculated result. Boundary condition is based on under supporting condition of crankshaft then obtaining results.

1.5 Organization of the Paper

This thesis is organized into five main chapters. In the first chapter introduction, background, statement of the problem and objective to be achieved are discussed. In chapter two a review of literature appropriate to this thesis is stated. In chapter three material, dimension, conditions, and methods used were discussed. In chapter four results and discussions from FEM and analytical method at different fillet radius and crankpin diameter are analyzed and discussed. Finally in chapter five conclusions, recommendation and future works are presented.

CHAPTER TWO

LITERATURE REVIEW

2.1. Introduction

This chapter reviews the previous studies, which are assisting to introduce the current study. Some of them are direct and the others are indirectly related to the current study. There are many journals, books; thesis and dissertation are discussed on this paper. This study is mostly about optimization of crankshaft for improving fatigue life, finite element modeling research journal concerning these issues are also summarized and discussed in this chapter.

2.2 Failure analysis

The analysis of failed in service crankshafts is vital to laboratory crankshaft studies, as it allows the researcher to better adapt experiments to real life situations as well as validates results. Crankshaft studies, suggest that crankshaft failures often occur in the fillet areas due to fatigue failure and another failure sources.

Bayrakceken et al. (2007) [9] has presented his work on the single cylinder diesel engine crankshaft used in the agricultural vehicles. The study shows that two different cases of crankshaft failure have been investigated and in both cases slight design differences were mentioned. As the failure mechanism for both the crankshafts is due to fatigue only.

Xiao-lei Xu et al (2011) [3] had described the Truck Petrol Engine Crankshaft Failure Analysis. In that, he discovered the dominant failure mechanism of the crankshaft. During the study he realized that the fatigue crack initiated at the fillet region of the first crankpin-web. This crankpin is the one among the six crankpins which bear operational load. Absence of the induction hardening case in the fillet region decreased the fatigue strength and led to fatigue initiation and propagation the weakened region.

Silva (2003) [12] has investigated the crack produced in the two crankshafts of diesel van which were used for 30,000 km and then grinding is done. After machining both the crankshafts lasted only for 1000 km and resulting into the Journal damage. Due to wrong grinding process small fatigue crack developed along the centre of journal causing journal damage. The cracks are sharp and invisible in nature. The study was also supported by examining the various failure modes of crankshaft.

Jimenez Espadafor et al. (2009) [22] has studied failure of 4-stoke 18 V diesel generator crankshaft used in electrical power plant. The crankshaft is running at speed of 1500 r.p.m before failure it has worked for 20,000 hrs of life. Failure of crankshaft was seen along the web between second journal and second crank pin. The presence of the beach mark shows fatigue fracture was produced along with a thin and hard zone which was found in the template surface showing crack initiation.

Infante et al. (2013) [21] has also worked on finding out the causes of failure of aircraft engine crankshaft. The research was oriented about the failure of the crankpin journal due to fatigue. The investigation shows that the presence of beach mark and also striation marks. To analyse the crack initiation and crack propagation on the fractured surface optical and electronic microscopic technology was used.

Katari et al. (2011) [23] has considered 3 different crankshafts made up of same material, i.e., forged steel which is used in train. The investigation shows that crankshaft failure is due to fatigue which is supported through examination of mechanical properties of crankshaft material. The failure regions showing semi-elliptical fracture indicating progressive fracture growth. Cracks are produced due to combined effect of mechanical and thermal fatigue load (2003). Heat and stresses during contact bearing with repetition of heating and cooling would eventually create thermal fatigue cracks and it will propagate with time.

Osman Asi (2006) [4] had described failure analysis of a crankshaft made from ductile cast iron. In which he studied the failure analysis of a petrol engine crankshaft used in a truck, which is made from ductile cast iron. The crankshaft was found to break into two pieces at the crankpin portion before completion of warranty period. The crankshaft was induction hardened.

R.M. Metkar, et al [11], review on fatigue life of crankshaft is to investigate the behavior of crankshaft under complex loading conditions. Automobile industries are always interested to develop a new product which will be innovative and fulfill market expectations. Various other factors also play important role in mechanical fatigue failure of crankshaft, such as, misalignment of crankshaft on assembly, improper journal bearing, and vibration due to some problem with main bearing or due to high stress concentration caused by incorrect fillet size, oil leakage, and overloading, high operating oil temperature. Heat and stresses during contact bearing with repetition of heating and cooling would eventually create thermal fatigue cracks and it will propagate with time.

Alex.K.D, etal [27] modeled the crankshaft by CATIA and dynamic analysis was done in ANSYS in which von-misses stress and principal shear stress was found. Crankshaft material was En-9 medium carbon steel in which static analysis gives approximate results and dynamic analysis gives accurate results. They noticed that in dynamic analysis torsional loads has no effect so only bending load is applied. They also noticed that maximum stress occurs in fillet area.

S.Vigneshwaran T.M.Vigneshwaran [28] have been analyzed Failure Analysis and Optimization Crankshaft in Diesel Locomotive.

Deals various failures occurring in the crank shaft and to analyze the failure due to fatigue crack. to develop an optimized geometry, material which will reduce the manufacturing cost due to high volume production of this component and also to improve fatigue performance.

When the failure of this component is concluded crankshaft has wide application area like truck, agricultural, train, aircraft, and marine engine. The problem of their premature failure has attracted several investigators for over a century. The extensive studies have been made to identify the cause of failure and several have been listed.

For determining the cause of the crankshaft failures, the method involves like-Presentation or formulation of the problem, Visual analysis, Analysis of possible explanations for the problem based on crankshaft failure causes, Definition of laboratory tests and analysis using Softwares.

There are different source of failure like operating sources, mechanical sources and repairing sources. The most failure of crankshaft was fatigue fracture resulting from co-effect of bending and twisting; the failure in the crankshaft initiated at the region between the fillets of the crankpin and main journal, and fatigue was the dominant mechanism of failure.

An extensive research in the past clearly indicates that the problem has not yet been overcome completely and designers are facing lot of problems specially related with multiaxial loading (Bending and Torsion), stress concentration and stress gradient and effect of variable amplitude loading. The fatigue life and damage are within permissible limits but need to improve. The crankshaft show maximum stress at critical location the geometry simplification may reduce stress at particular location.

2.3 Finite Element Modeling

Since the crankshaft has a complex geometry for analysis, finite element models have been considered to give an accurate and reasonable solution whenever laboratory testing is not available.

R. Metkar et al [11] described finite element method as the most favorite method to solve stress and fatigue analysis and it is commonly used for analyzing engineering problems. He also studied stress life, strain life and LEFM methods to solve fatigue analysis. There are lots of softwares available for use in finite element analysis applications, such as, ANSYS, Abaqus, Nastran, and MSC.

Rinkle garg and Sunil Baghl [29] has been analyzed crankshaft model and crank throw were created by Pro/E Software and then imported to ANSYS software. The result shows that the improvement in the strength of the crankshaft as the maximum limits of stress, total deformation, and the strain is reduced. The weight of the crankshaft is reduced .There by, reduces the inertia force. As the weight of the crankshaft is decreased this will decrease the cost of the crankshaft and increase the I.C engine performance.

Uchida and Hara (1984) [20] used a single throw FEM model. In the extrapolation of the experimental equation in their study. In their study the web thickness of a 60° V-6 crankshaft was reduced while maintaining its fatigue performance and durability under turbo-charged gas pressure. In the study of the 60° V-6 engine crankshaft dimensions, the stress evaluation of the crankpin fillet part was extremely critical since, to reduce the crank's total length, it was necessary to reduce the thickness of the web between main journal and crankpin as much as possible while maintaining its strength. FEM was used to estimate the stress concentration factor at thicknesses where confirmed calculation was not available.

Jamin Brahmbhatt et al. [36] created a 3D model in SOLID WORKS and simulation was done in ANSYS. Dynamic analysis was done to calculate the stress in critical areas. The results showed that the edge of main journal was at maximum stress region and the center of crankpin shows the maximum deformation area.

Ms.Shweta Ambadas Naik [30] carried out the Failure Analysis of crankshaft by finite element method. In this paper, the stress analysis and modal analysis of a 4-cylinder crankshaft are discussed using finite element method. The 3D model of crankshaft was developed in PRO/E and imported to ANSYS for strength analysis. In this study, failure analysis of a crankshaft was carried out. Several causes are there to fail the crankshaft. All crankshafts were failed from the same region. Failures had occurred in the first crankpin, the nearest crankpin to the flywheel. Dynamic analysis and finite element modeling were carried out to determine the state of stress in the crankshaft. Finite element analysis was performed to obtain the variation of stress magnitude at critical locations.

Jian et al. [24] analyzed three dimensional model of 380 diesel engine crankshaft. They used ProE and ANSYS as FEA tools. First of all, the 380 diesel engine entity crankshaft model was created by Pro E software. Next, the model was imported to ANSYS software. Material properties, constraints boundary conditions and mechanical boundary conditions of the 380 diesel engine crankshaft were determined. Finally, the strain and the stress figures of the 380 diesel crankshaft were calculated combined with maximum stress point and dangerous area. This article checked the crankshaft's static strength and fatigue evaluations. That provided theoretical foundation for the optimization and improvement of engine design. The maximum deformation occurs in the end of the second cylinder balance weight.

Mr.J.E.Pachpande.et al [31] Have been analyzed Optimization of Crank Shaft using FEA.

The modal analysis of a 3-cylinder crankshaft of SWARAJ 855 XM tractor is being analyzed using finite element method. The crankshaft model was created by using PRO-E wildfire 4.0 for modeling the crank shaft. Then, the created model was import into ANSYS software for static structural analysis. The stresses induced on crankshaft was investigated through a systematic force analysis in ANSYS for different kind of two materials like forged steel, steel alloy CrMoV13 with same operating condition. In finite element analysis of different materials have done static structural analysis by putting the values of properties in ANSYS. In force analysis have calculated total deformations, maximum shear stress, equivalent (vonmisses) stress, shear stress, maximum principle stress. The comparison of analysis results of two different materials will show the effect of stresses on different materials and to evaluate the most suitable material for manufacturing crankshaft which gives good performance and to avoid possible structural damages on the crankshaft surface.

Gu Yingkui, Zhou Zhibo. [16] Crankshaft model of diesel engine was created in Pro-E and analyzed in ANSYS. In their paper they have studied high stress area which is the joint between the web and crankpin which is more likely to break. Using ANSYS analysis tool the crank stress change model and the crank stress biggest hazard point were found by using finite element analysis, and the improvement method for the crankshaft structure design was given.

Generally, Finite Element Method (FEM) is the numerical simulation of the behavior of mechanical components acquired by discrediting engineering components into finite elements, often just called elements, connected by nodes, and obtains an approximate solution. After the boundary conditions are applied, the nodal displacements are found by solving the matrix stiffness equation. Once nodal displacements are known, element stresses and strains can be calculated. Many researchers have been used finite element method for crankshaft due to this component has a complex geometry for analysis, finite element models have been considered to give an accurate and reasonable solution whenever laboratory testing is not available. Finite element analysis was performed to obtain the variation of stress magnitude at critical locations. The time and efforts required for analysis using software is very less and it is a good tool to reduce time consuming theoretical work.

2.4 Optimization of Crankshafts with Geometry, Material

Crankshaft is among large volume production components in the internal combustion engine industry. Optimizations like geometry, material and manufacturing of this component will result in high cost saving increase the fuel efficiency of the engine and improve fatigue life.

2.4.1 Geometry optimization

The modifications for improvement in fatigue life such as changing the main bearing radius, crank pin radius, fillet radius of crankpin pin and changing the type of material in crankshaft, which are very common modifications usually done in crankshaft geometry [5].

Amitpal Singh Punewale et al [15] studied torsional vibration with modified crankshaft geometry by adding some material at face inclination on crankshaft and found out improvement in results as compared to original geometry of crankshaft. Thus, Addition of material in Face-inclination area results in an increased torsional stiffness. Also, the fatigue life of crankshaft increases by this change.

Development of the DCI crankshaft for the Nissan 60° V-6 engine was studied by **Uchida** and **Hara (1984) [20].** It was aimed to reduce the web thickness while maintaining the performance of the crankshaft used before. This resulted in shortening the engine length. They used the finite element method to perform structural analyses. The analyses were necessary to set the absolute minimum dimensions for the cylinder pitch as well as each of the parts. The objective of their study was to reduce available length for various engine installation types, reduction of friction for better fuel economy, and good stiffness for lower noise.

Henry et al. (1992) [13] used the durability assessment tool developed by RENAULT to optimize the crank web design of a turbocharged diesel engine crankshaft. The geometry of the web design was changed according to reasonable geometries in order to reduce the maximum stress. This constraint allowed changes in the web width and pin side web profile.

C M. Balamurgan et al [1]. He has been studied the Computer aided Modeling and Optimization of crankshaft and compared the fatigue performance of two competing materials of crankshaft namely forged steel and ductile cast iron. The 3D models of crankshaft were created by solid edge software and then imported to ANSYS software. The optimization process included geometry changes compatible with the current engine, fillet rolling and results in increased fatigue strength and reduced the cost and mass of the crankshaft.

Farzin H.Montazersadgh and Ali Fatemi (2007) [14] conducted a study on stress analysis and optimization of crankshafts subject to dynamic load. In this study a dynamic simulation was conducted on a crankshaft from a single cylinder four stroke engine. Finite element analysis was performed to obtain the variation of stress magnitude at critical locations. Geometry optimization resulted in 18% weight reduction of the forged steel crankshaft, which was achieved by changing the dimensions.

Momin Muhammad Zia Muhammad Idris, etal [26] investigated Optimization of Crankshaft using Strength Analysis. This paper presents results of strength analysis done on crankshaft of a single cylinder two stroke petrol engine, to optimize its design, using PRO/E and ANSYS software. The paper also proposes a design modification in the crankshaft to reduce its mass. Therefore web thickness was reduced from 13 mm to 10 mm. The reduction in mass obtained by this design modification is 16.98 %.

Mr. Suraj K. Kolhe [31] carried out Diesel Engine Crankshaft High Cycle Fatigue Life Estimation and Improvement through FEA, The main objective of this paper is to improve fatigue life of crankshaft. In this study a detailed stress analysis is done. This paper describes the exact nature and magnitude of stresses at crankpin fillet and journal fillet during crankshaft operation. Maximum stress location is observed at the same location as that of actual failure location for baseline crankshaft, with maximum tensile stress at crank pin fillet and minimum compressive stress at journal fillet. Slight addition of material on crankshaft face inclination gives 20% reduction in stresses and 15 times increment in fatigue life for modified configuration.

Burrell (1985) [18] studied the effect of controlled shot peening on the fatigue properties of different automotive components, including crankshafts. Six different crankshafts from different engines were investigated. The critically stressed location of each crankshaft was the journal bearing fillet. The high stress point was the bottom side of the fillet when the journal is in the top dead center position during the firing cycle. Each fatigue test was performed on single journal bearing of crankshafts. The result obtained is the fatigue strength of either forged steel or cast iron crankshafts increases as a result of shot controlled peening process, compared with the same un peened component.

Park et al. (2001) [19] showed that without any dimensional modification, fatigue life of crankshaft could be improved significantly by applying various surface treatments. Fillet rolling and nitriding were the surface treatment processes that were studied in their research.

As mentioned above by many researchers geometry optimization like addition of material on face inclination, fillet rolling, reducing web thickness and so on are done for weight reduction and to reduce rate of failure. Generally when geometry optimization is concluded Crankshaft is among large volume production components in the internal combustion engine industry and it is necessary to improve its reliability. The modifications for improvement in fatigue life such as changing the main bearing radius, crank pin radius, fillet radius of crankpin pin and changing the type of material in crankshaft, which are very common modifications usually done in crankshaft geometry.

2.4.2 Material optimization

An extensive study was performed by **Nallicheri et al. (1991) [32]** on material alternatives for the automotive crankshaft based on manufacturing economics. They considered steel forging, nodular cast iron, micro-alloy forging, and austempered ductile iron casting as manufacturing options to evaluate the cost effectiveness of using these alternatives for crankshafts.

Ashwani Kumar Singh, et al. [17] the modeling of crankshaft was done in Pro-E and simulation in ANSYS. They used nickel chrome steel and structural steel for the material of crank shaft, and concluded that nickel chrome is more reliable than structural steel. In a literature survey by **Zoroufi and Fatemi [8],** they have discussed the fatigue performance and compared forged steel and cast iron crankshafts. In their study failure sources of crankshaft were discussed and also different methods of crack forming in fillet region. They also compared nodular cast iron, forged steel, austempered ductile iron, which concluded that fatigue properties of forged steel are better than cast iron. They also added the cost analysis and geometry optimization of crankshaft. Adding fillet rolling was considered in the manufacturing process. Fillet rolling induces compressive residual stress in the fillet areas, which results in 165% increase in fatigue strength of the crankshaft and increases the life of the component significantly [8]. Since fatigue fracture initiated near the fillets is one of the primary failure mechanisms of automotive crankshafts, fillet rolling process has been used to improve the fatigue lives of crankshafts in many applications. Fillet rolling manufacturing process is good in cost and making save time.

Bhumesh J. Bagde etal [25] carried out finite element analysis of single cylinder engine crank shaft. In this paper, the crankshaft model was created by Pro-E Wildfire 4.0 software. Then, the model created by Pro-E Wildfire 4.0 was imported to ANSYS software. The analysis of the crank shaft will be done using five different materials. These materials are EN9, SAE 1046, SAE 1137, SAE 3140 & Nickel Cast iron. The comparison of analysis results of all five materials will show the effect of stresses on different materials and this will help to select suitable material.

Material optimization is concluded that considered based on manufacturing economics, application, and fatigue properties of material. During using alternative material they have used only software analysis not only this is satisfactory the other issue like mechanical properties and carbon content of material should be considered. Generally optimizations like geometry, material and manufacturing of this component will result in high cost saving increase the fuel efficiency of the engine and improve fatigue life.

CHAPTER THREE

METHODS AND CONDITIONS

3.1 Material

For the proper functioning, the crankshaft should full fill the following conditions:

- ✓ Enough strength to withstand the forces to which it is subjected i.e. the bending and twisting moments.
- ✓ Enough rigidity to keep the distortion a minimum.
- ✓ Stiffness to minimize, and strength to resist, the stresses due to torsional vibrations of the
- ✓ Minimum weight, especially in aero engines.

Generally, crankshafts materials should be readily shaped, machined and heat-treated, and have adequate strength, toughness, hardness, and high fatigue strength.

The crankshaft material used in this study is forged steel (AISI 1045) steel which is medium carbon steel and the type of engine is Honda engine.

Typical Uses of Medium Carbon Steel:-

- ✓ 0.3 - 0.4: Lead screws, Gears, Worms, Spindles, Shafts, and Machine parts.
- ✓ 0.4 - 0.5: Crankshafts, Gears, Axles, Mandrels, Tool shanks, and Heat-treated machine parts.

Since the carbon content 0.4% - 0.5% is better for crankshaft the material AISI 1045steel is between this percent and the best one for crankshaft production.

This study is focuses on single cylinder diesel engine crankshaft used in agricultural areas. This single cylinder engine is used for off roads in rural areas for agricultural purpose. Due to its application is off road its rate of failure is maximum rather than on road vehicles, so it is necessary to improve its life.

Table3.1 Material properties of the crankshaft

Material Type	Forged steel (AISI 1045steel)	Unit
Density	7833	kg/m^3
Young’s modulus	221	GPa
Poisson’s ratio	0.3	-
Yield stress	625	MPa
Ultimate tensile strength	827	MPa

Ref. [10]

3.2 Dimension

Table3.2 Specifications: Honda Engine

Capacity	395cc
Number of cylinder	1
Bore x stroke	86x68mm
Compression ratio	18.1
Maximum power	8.1hp@3600rpm
Maximum torque	16.7Nm@2200rpm
Maximum gas pressure	25bar

Table3.3 Dimension of single cylinder engine crankshaft

S. No	Parameters	Symbol	Values
1	Diameter of the Crank Pin	d_c	37mm
2	Length of the Crank Pin	l_c	32mm
3	Crankpin oil hole diameter	c_{oh}	18mm
4	Diameter of the shaft/journal	d_s	35mm
5	Web Thickness (Both Left and Right Hand)	w_t	18mm
6	Web Width (Both Left and Right Hand)	w_w	65mm
7	Length of the Crank shaft	L	341mm
8	Crankpin fillet radius	R_h	3mm

Ref. [10]

Table3.4 Parameters for optimization geometry

S. No	Parameters	Original crankshaft	Optimized crankshaft		
1	Crankpin fillet radius	3	3.5	4	4.5
2	Crankpin diameter	37	38	39	40

Crankpin fillet radius increment is depending on the r/d ratio (fillet radius to crankpin diameter) which exists between $0.03 \leq r/d \leq 0.13$ [40]. $0.03 \leq 3/37, 3.5/37, 4/37, 4.5/37 \leq 0.13$ or $0.03 \leq 0.08, 0.09, 0.011, 0.012, \leq 0.13$. It is impossible optimize greater than 4.5mm because does not exist between this range.

If diameter of crankpin is 25mm to 50mm 1mm increment is will be done [39]. Since the original crankpin diameter is 37mm 1mm is increased due to exists between this ranges.

3.3 Conditions

- ✓ Crankshaft is assumed to a beam with two or more supports [7].
- ✓ Force is acting on the center of crankpin or at dead center [37]
- ✓ Assume that the distance (b) between the bearing 1 and 2 is equal to twice the piston diameter (D) [7]. Due to Piston force will act at the middle of the crankpin, and it will be balanced by the reactions.
- ✓ The crankpin is subjected to various forces and analysed in two positions which is at the position of maximum bending (when the crank is at dead center and twisting [37].
- ✓ The maximum value of the tangential force lies when the crank is at an angle of 25° to $30^\circ$from the dead center for petrol engines and 30° to 40° for diesel engines [36].

In this study since it is diesel engine 30° to 40°so 35° is taken for this analytical calculation.

- ✓ In order to find the thrust in the connecting rod (F_Q), angle of inclination of the connecting rod with the line of stroke (i.e. angle Ø), Take l/r =4 [29]
- ✓ When the crank is at dead center the bending moment on the shaft is maximum [7]

- ✓ Belt pull (T_1+T_2), acting horizontally and Weight of the fly wheel (W), acting vertically [7].
- ✓ Geometry optimization [10]
- ✓ Piston force F_p is calculated using maximum cylinder pressure, and bore diameter of engine cylinder [31].
- ✓ K_b= Combine shock, fatigue factor for bending = 1 and K_t= Combine shock, fatigue factor for torsion = 1 [34]
- ✓ Considering tangential and radial forces during crankshaft is an angle of twisting [7]
- ✓ In terms of fillet radius calculation shoulder fillet and Gerber Criteria is used [2]

3.4 Methods

3.4.1 Analytical method of bending and shear stress of crankshaft

There are two types of stresses are in the crankpin.

- ✓ Bending stress
- ✓ Shear stress due to torsional moment on the shaft

The crankpin is subjected to various forces but generally needs to be analysed in two positions. Firstly, failure may occur at the position of maximum bending (when the crank is at dead center). In such a condition the failure is due to bending and the pressure in the cylinder is maximum. Second the crank may fail due to twisting, so the crankpin needs to be checked for shear at the position of maximum twisting. The failure of a crankpin is likely to cause serious engine destruction.

In the design of the crankshafts, it is assumed that the crankshaft is a beam with two or more supports. The crankpin is designed by considering two crank positions, i.e. when the crank is at dead center (or when the crankpin is subjected to maximum bending moment) and when the crank is at an angle at which the twisting moment is maximum. These two cases are discussed in detail as follows.

3.4.1.1 Crank at Dead Center

At this position of the crank, the maximum gas pressure on the piston transmits maximum force on the crankpin in the plane of the crank causing only bending. The crankpin is only subjected to bending moment. Thus, when the crank is at dead center the bending moment on the shaft is maximum and the twisting moment is zero.

When the crank is on dead center, maximum bending moment will act in the crankshaft. The thrust in the connecting rod will be equal to the piston gas load (F), W is the weight of the flywheel acting downward and T_1 and T_2 is the belt pull acting horizontally [7].

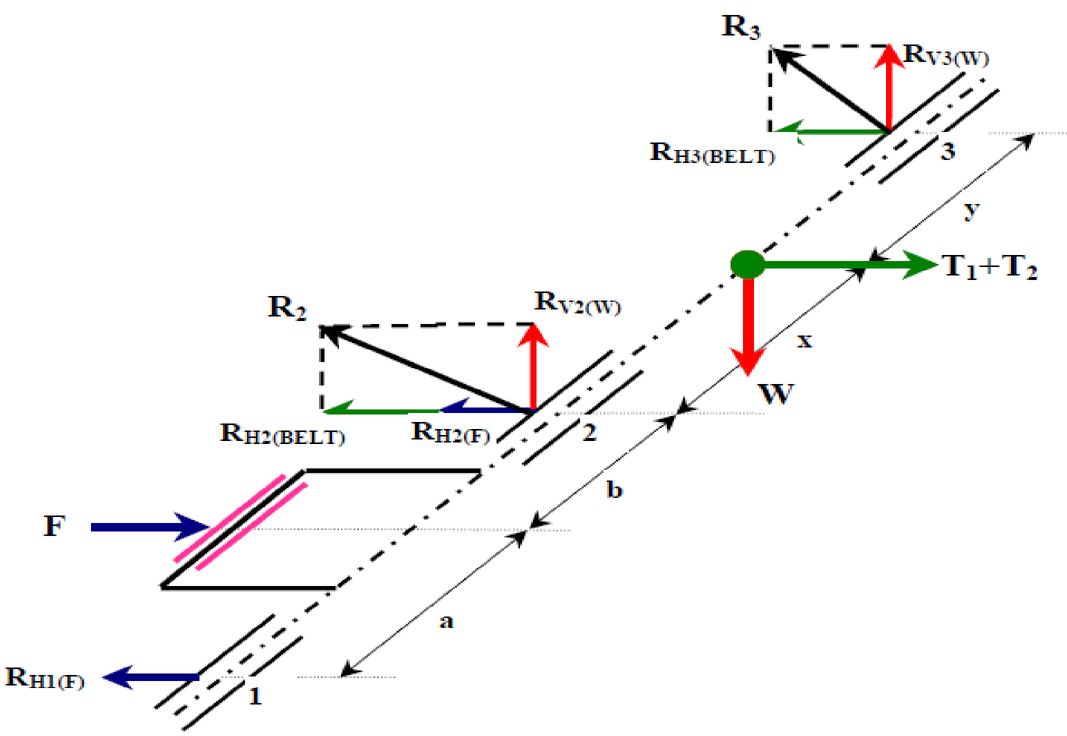


Figure 3.1 Force Analysis of Crank at Dead Center

In between bearings 1 and 2, Gas Load F, acts as shown

Now Gas Force F is calculated using maximum cylinder pressure, and bore diameter of engine cylinder [7].

$F_p = \frac{\pi}{4} D^2 P_{max}$

Where,

D= is the diameter of the piston in mm

P_{max} = Maximum intensity of pressure on piston in N/mm^2

Due to this there will be two horizontal reactions, $R_{H1\ (F)}$ at bearing 1, and $R_{H2\ (F),}$ at bearing 2,

So that, to find the reactions $R_{H1\ (F)}$ and $R_{H2\ (F)}$

$\Sigma M_1 = 0$…..3.1

Clock Wise direction is taken as positive bending moment and Counter Clockwise as negative bending moment.

$F_P(a) - R_{H2}(a+b) = 0$…...3.2

$R_{H2} = \frac{F_p(a)}{a+b}$…...3.3

$\Sigma F_y = 0$…..3.4

Upward force is taken as positive and downward is taken as negative.

$R_{H1\ (F)} + R_{H2\ (F)} - Fp = 0$…..3.5

By substituting equation 3.3 in equation 3.5 we get

$R_{H1} = Fp - \frac{F(a)}{a+b} = \frac{F(b)}{a+b}$…...3.6

If $a = b$, then $R_{H1} = R_{H2} = \frac{Fp}{2}$…..3.7

In between bearing 2 and 3, we have two loads

i) Belt pull(T1+T2), acting horizontally

ii) Weight of the fly wheel (W), acting vertically.

We know that

Bore diameter (D) =86mm,

Maximum gas pressure =25bar = 2.5N/mm^2

Fp = Area of the bore ×P max

$$Fp = \frac{\pi}{4} D^2 * Pmax$$

$Fp = \frac{\pi}{4}(86mm)^2 * 2.5\frac{N}{mm^2}$ =14.52KN

We know that due to piston gas load, there will be two horizontal reactions H_1 and H_2 at bearings 1 and 2 respectively, such that

$R_{H1} = R_{H2} = \frac{Fp}{2}$

Assume that the distance (b) between the bearing 1 and 2 is equal to twice the piston diameter (D) [7].

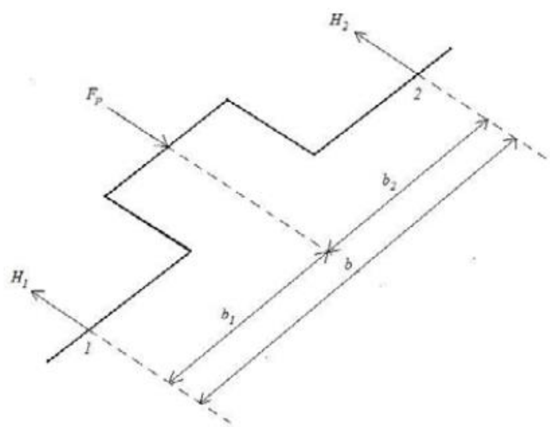


Figure 3.2: Crank at Dead Centre

$b = 2D$

$b = 2 \times 86 = 172\text{mm}$

$\text{Therefore } b1 = b2 = \frac{b}{2} = 86\text{mm}$

$$R_{H1} = R_{H2} = \frac{Fp}{2}$$

$$R_{H1} = R_{H2} = \frac{14.52\text{KN}}{2} = 7.26\text{KN}$$

Bending moment at the centre of the crankpin will be

$M_c = R_{H1} \text{x } b1$

M_c= 7.26KN x86

=624.36KN

We know that,

$\frac{M}{I} = \frac{\sigma}{c}$……………………………………………………………………………………………………3.8

Substituting the values of $c = \frac{dc}{2}$ and $I = \frac{\pi(dc)^4}{64}$ in Equation 3.8 and solving for M we get,

$$M = \frac{\sigma_b}{c}(I)$$

$$M = \frac{\sigma_b}{\frac{dc}{2}}\left(\frac{\pi(dc)^4}{64}\right)$$

$M = \frac{\pi}{32}{d_c}^3\sigma_b$……………………………………………………………………………………………3.9

$\sigma_b = \frac{32M}{\pi {d_c}^3}$…………………………………………………………………………………………………3.10

We know that,

M is Bending moment

σ_b is allowable stress in bending, and

d_c is diameter of the crankpin

3.4.1.2 Crank at an angle of maximum twisting moment

The twisting moment on the crankshaft is maximum when the tangential force on the crank (F_T) is maximum. The maximum value of the tangential force lies when the crank is at an angle of 25° to 30° from the dead center for petrol engines and 30° to 40° for diesel engines [7].

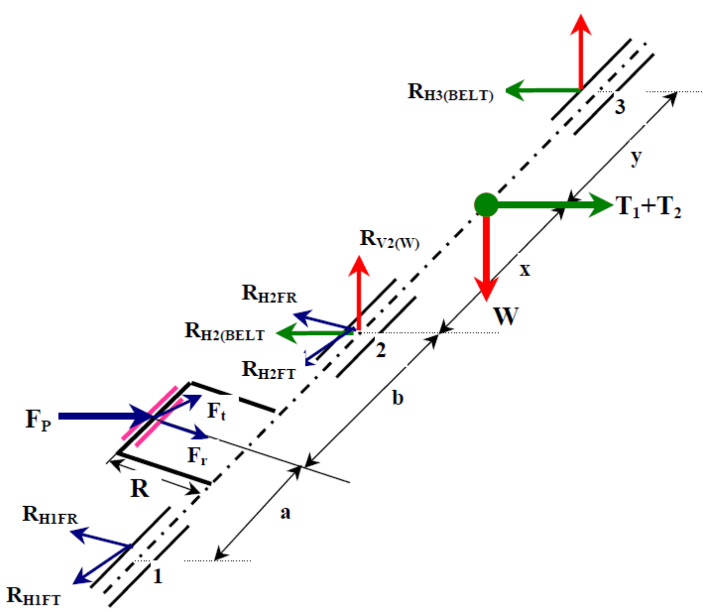


Figure 3.3 Force Analysis of Crank at angle of maximum twisting Moment

Tangential force F_t will have two reactions R_{H1FT} and R_{H2FT} at bearing 1 and 2 respectively. Radial force Fr will have two reactions R_{H1FR} and R_{H2FR} at bearing 1 and 2 respectively. The reactions at the bearings 2 and 3 due to belt pull (T_1+T_2) and Flywheel W will be same as before. In this position of the crankshaft, the different sections will be subjected to both bending and torsional moments and these must be checked for combined stress. At this point, Shear stress is taken as failure criteria for crankshaft.

The reactions due Radial force (Fr)

To find the reactions R_{H1FR} and R_{H2FR}

$\Sigma M_2 = 0$……………………………………………………………………………3.11

Clock Wise direction is taken as positive bending moment and Counter Clockwise as negative bending moment.

$-F_r(b) + R_{H1FR}(a + b) = 0$……………………………………………………………3.12

$R_{H1FR} = \frac{F_r(b)}{(a+b)}$……………………………………………………………………3.13

$\Sigma F_y = 0$……………………………………………………………………………3.14

$-F_r + R_{H1FR} + R_{H2FR} = 0$……………………………………………………………3.15

$R_{H2FR} = \frac{F_r(a)}{(a+b)}$……………………………………………………………………3.16

if a = b, $R_{H1FR} = R_{H2FR} = \frac{F_r}{2}$……………………………………………………3.17

The reactions due tangential force (Ft)

To find the reactions R_{H1FT} and R_{H2FT}

$\Sigma M_2 = 0$……………………………………………………………………………3.18

$-F_T(b) + R_{H1FT}(a + b) = 0$……………………………………………………………3.19

$R_{H1FT}=\frac{F_t(b)}{a+b}$……………………………………………………………………………3.20

$\Sigma F_y = 0$……………………………………………………………………………3.21

$-F_T + R_{H1FT} + R_{H2FT} = 0$……………………………………………………………3.22

$R_{H2FT} = \frac{F_T(a)}{(a+b)}$……………………………………………………………………………3.23

if $a = b, R_{H1FT} = R_{H2FT}=\frac{F_t}{2}$……………………………………………………………3.24

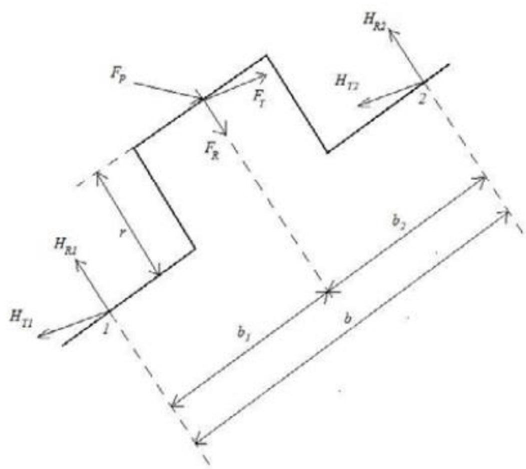


Figure 3.4 Cranks at Position of Maximum Twisting Moment.

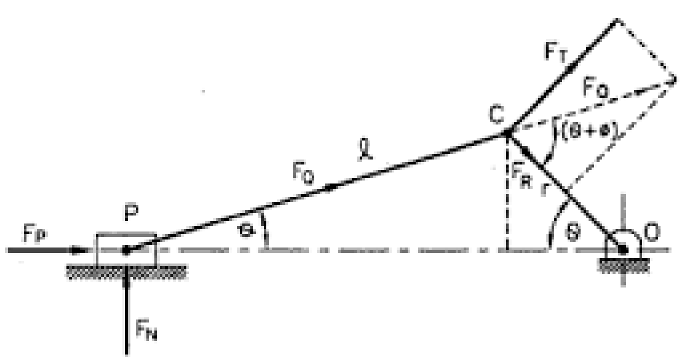


Figure 3.5 Forces on crank Arm

In order to find the thrust in the connecting rod (F_Q), angle of inclination of the connecting rod with the line of stroke (i.e. angle Ø), Take l/r = 4, the maximum value of tangential force lies when the crank is at an angle of θ=35° (from 30^0 to 40^0 for diesel engines [7].

$Sin\ Ø = (Sin\ \Theta / (l/r)) = sin\ 35°/4$

Which implies Ø = 8.24° we know that thrust in the connecting rod or the force exerted on the crankpin by the piston:

$$F_Q = \frac{F_P}{cosØ}$$

$$F_Q = \frac{F_P}{cosØ}$$

$$F_Q = \frac{14.52KN}{\cos 8.24^{\circ}} = 14.67\text{KN}$$

Thrust on the crank shaft can be split into Tangential Component and the radial component.

The tangential force or the rotative effort on the crank

$$\text{FT} = \text{FQ} \sin(\theta + Ø) = 10.05\text{ KN}$$

The radial force along the crank

$$\text{FR} = \text{FQ} \cos(\theta + Ø) = 10.68\text{ KN}$$

Reactions at bearings (1 & 2) due to tangential and radial force is given by

$$\text{HT1} = \text{HT2} = \frac{F_t}{2}\ (\text{Since b1} = \text{b2} = \frac{b}{2})$$

$$\text{HT1} = \text{HT2} = \frac{10.05\text{KN}}{2} = 5.02\text{KN}\ (\text{Since b1} = \text{b2} = \frac{b}{2})$$

$$\text{HR1} = \text{HR2} = \frac{F_r}{2}\ (\text{Since b1} = \text{b2} = \frac{b}{2})$$

$$\text{HR1} = \text{HR2} = \frac{10.68\text{ KN}}{2} = 5.34\text{KN}\ (\text{Since b1} = \text{b2} = \frac{b}{2})$$

The bending moment at the centre of the crankpin is

$M_b = R_{H1FR}(a)$ ……………………………………………………………………………………3.25

The Twisting moment is $T = R_{H1FT}(R)$ …………………………………………………………3.26

Equivalent twisting moment, $T_e = \sqrt{Mc^2 + Tc^2}$……………………………………………………3.27

We know that

$\frac{T_e}{J} = \frac{\tau}{r}$………………………………………………………………………………………3.28

Where T_e is torque or Torsional moment, N-mm

J=polar moment of inertia, mm^4

$=\frac{\pi}{32} {d_c}^4$, where d_c is the solid shaft diameter.

τ = allowable shear stress, MPa

r = Distance from the Neutral axis to the top most fibre, mm $=\frac{d_c}{2}$

Substituting the values of J and r in equation 3.28 and simplifying we get,

$T_e = \frac{\pi}{16} x (dc)^3 x\tau$……………………………………………………………………………3.29

$\tau = \frac{16T_e}{\pi {d_c}^3}$…………………………………………………………………………………………3.30

Design of crankpin:

Let d_c= Diameter of crankpin in mm.

We know that the bending moment at the center of the crankpin,

M_c= HR1 x b2 = 5.34 x 86 = 459.24 KN-mm

Twisting moment on the crankpin

$T_c = H_{T1}$*r=5.02 x 34=170.68 KN-mm

Twisting moment on the crankpin=170.68 KN-mm

From this we have the equivalent twisting moment

$T_e = \sqrt{Mc^2 + Tc^2}$

T_e $=\sqrt{459.24^2 + 170.68^2}$

= 489.93 KN-mm

We know that equivalent twisting moment (T_e)

T_e $=\frac{\pi}{16} x(dc)^3 x\tau$

According to distortion energy theory, the Von-Misses stress induced in the crank-pin is [34],

$M_{ev} = \sqrt{(kbxMc)^2 + \frac{3}{4}(ktxTc)^2}$

Where, Kb = combined shock and fatigue factor for bending (Take Kb=1)

K_t = combined shock and fatigue factor for torsion (Take Kt =1)

$M_{ev} = \sqrt{(1x459.24)^2 + \frac{3}{4}(1x170.68)^2}$

=482.44 KN-mm

σ_v Equivalent (Von-Misses) stress is

$M_{ev} = \frac{\pi}{32} x(dc)^3 \sigma_v$ and

T_e $=\frac{\pi}{16} x(dc)^3 x\tau$

$$\sigma_v = \frac{Mev}{\frac{\pi}{32} x(dc)^3}$$

For dc =37mm

$$\sigma_v = \frac{482.44\text{KN\_mm}}{\frac{\pi}{32} x(37^3 - 18^3)}$$

= 109.69N/mm^2

T_e $=\frac{\pi}{16} x(dc)^3 x\tau$

$$\tau = \frac{Te}{\frac{\pi}{16} x (dc)^3}$$

$$\tau = \frac{489.93 \text{ KN\_mm}}{\frac{\pi}{16} x (37^3 - 18^3)}$$

τ = 55.69N/mm^2

For dc =38mm

$$\sigma_v = \frac{482.44\text{KN\_mm}}{\frac{\pi}{32} x (38^3 - 18^3)}$$

σ_v = 100.25N/mm^2

$$\tau = \frac{489.93 \text{ KN\_mm}}{\frac{\pi}{16} x (38^3 - 18^3)}$$

τ = 50.91N/mm^2

For dc =39mm

$$\sigma_v = \frac{482.44\text{KN\_mm}}{\frac{\pi}{32} x (39^3 - 18^3)}$$

σ_v = 91.92N/mm^2

$$\tau = \frac{489.93 \text{ KN\_mm}}{\frac{\pi}{16} x (37^3 - 18^3)}$$

τ = 46.47N/mm^2

For dc =40mm

$$\sigma_v = \frac{482.44\text{KN\_mm}}{\frac{\pi}{32} x (40^3 - 18^3)}$$

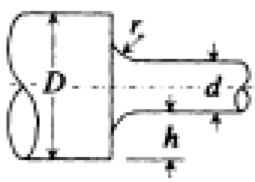
$=84.52 N/mm^2$

$\tau = \frac{489.93\ KN\_mm}{\frac{\pi}{16}x(40^3-18^3)} = 42.92 N/mm^2$

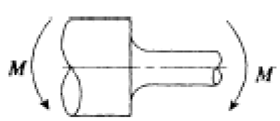
3.4.1.3 Von mises stresses and shear stress calculation related to crankpin fillet radius

Shoulder fillet

The shoulder fillet is the type of stress concentration that is more frequently encountered in machine design. Examples of filleted members are engine crankshaft, turbine rotor, motor shaft, railway axle usually involve a number of diameters connected by shoulders with rounded fillets [2].



Bending



For 3mm fillet radius

$D = 50mm$

$d = 37mm$

$h = 6.5mm$

$r = 3mm$

$\frac{h}{r} = 2.16mm$

$\frac{2h}{D} = 0.26$

$\left(\frac{2h}{D}\right)^2 = 0.067mm$

$\left(\frac{2h}{D}\right)^3 = 0.017mm$

$$\sqrt{\frac{h}{r}} = 1.46mm$$

$$\sigma_{\boldsymbol{max}} = K_t\, \sigma_{nom}$$

$$\sigma_{nom} = \frac{32M}{\pi d^3}$$

$\sigma_{nom} = \frac{32M}{\pi {d_o}^3 - {d_i}^3}$, Since the crankpin is hollow

$$\sigma_{nom} = \frac{32 * 459.24KN/mm^2}{\pi * (37^3 - 18^3)} = 104.42N/mm^2$$

To obtain stress concentration factor (K_t)

$$K_t = C_1 + C_2\left(\frac{2h}{D}\right) + C_3\left(\frac{2h}{D}\right)^2 + C_4\left(\frac{2h}{D}\right)^3$$

$2.0 \leq \frac{h}{r} \leq 20.0$, where $\frac{h}{r} = 2.16mm$

$$C_1 = 1.232 + 0.832\sqrt{\frac{h}{r}} - 0.008(\, h/r)$$

$$C_1 = 1.232 + 0.832\sqrt{\frac{6.5}{3}} - 0.008(\, 6.5/3) = 2.42$$

$$C_2 = -3.813 + 0.968\sqrt{\frac{h}{r}} - 0.260(\, h/r)$$

$$C_2 = -3.813 + 0.968\sqrt{\frac{6.5}{3}} - 0.260(\, 6.5/3) = -2.95$$

$$C_3 = 7.423 - 4.868\sqrt{\frac{h}{r}} + 0.869(\, h/r)$$

$$C_3 = 7.423 - 4.868\sqrt{\frac{6.5}{3}} + 0.869(\, 6.5/3) = 2.19$$

$$C_4 = -3.839 + 3.070\sqrt{\frac{h}{r}} - 0.600(\,h/r)$$

$$C_4 = -3.839 + 3.070\sqrt{\frac{6.5}{3}} - 0.600(\,6.5/3) = -0.65$$

$$K_t = C_1 + C_2\left(\frac{2h}{D}\right) + C_3\left(\frac{2h}{D}\right)^2 + C_4\left(\frac{2h}{D}\right)^3$$

$$K_t = 2.42 - 2.95\left(\frac{2*6.5}{50}\right) + 2.19\left(\frac{2*6.5}{50}\right)^2 - 0.65\left(\frac{2*6.5}{50}\right)^3$$

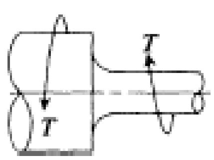
$$K_t = 1.78$$

$$\sigma_{\boldsymbol{max}} = K_t\,\sigma_{nom}$$

$$\sigma_{\boldsymbol{max}} = 1.78 * 104.42N/mm^2$$

$$\sigma_{\boldsymbol{max}} = 185.86N/mm^2$$

Torsion



$$\tau_{max} = K_t\,\tau_{nom}$$

$$\tau_{nom} = \frac{16T}{\pi d^3}$$

Where T is twisting momentum=170.68KN-mm

$$\tau_{nom} = \frac{16 * 170680}{\pi(37^3 - 18^3)} = 19.40N/mm^2$$

To obtain stress concentration factor (K_t)

$$K_t = C_1 + C_2\left(\frac{2h}{D}\right) + C_3\left(\frac{2h}{D}\right)^2 + C_4\left(\frac{2h}{D}\right)^3$$

$0.25 \leq \frac{h}{r} \leq 4.0$, where $\frac{h}{r} = 2.16mm$

$$C_1 = 0.905 + 0.783\sqrt{\frac{h}{r}} - 0.075(\, h/r)$$

$$C_1 = 0.905 + 0.783\sqrt{\frac{6.5}{3}} - 0.075(\, 6.5/3) = 1.89$$

$$C_2 = -0.437 - 1.969\sqrt{\frac{h}{r}} + 0.553(\, h/r)$$

$$C_2 = -0.437 - 1.969\sqrt{\frac{6.5}{3}} + 0.553\left(\frac{6.5}{3}\right) = -2.14$$

$$C_3 = 1.557 + 1.073\sqrt{\frac{h}{r}} - 0.578(\, h/r)$$

$$C_3 = 1.557 + 1.073\sqrt{\frac{6.5}{3}} - 0.578(\, 6.5/3) = 1.88$$

$$C_4 = -1.061 + 0.171\sqrt{\frac{h}{r}} + 0.086(\, h/r)$$

$$C_4 = -1.061 + 0.171\sqrt{\frac{6.5}{3}} + 0.086(\, 6.5/3) = -\, 0.63$$

$$K_t = C_1 + C_2\left(\frac{2h}{D}\right) + C_3\left(\frac{2h}{D}\right)^2 + C_4\left(\frac{2h}{D}\right)^3$$

$$K_t = 1.89 - 2.14\left(\frac{2*6.5}{50}\right) + 1.88\left(\frac{2*6.5}{50}\right)^2 - 0.63\left(\frac{2*6.5}{50}\right)^3$$

$$K_t = 1.44$$

$$\tau_{max} = K_t\, \tau_{nom}$$

$$\tau_{max} = 1.44 * 19.40 = 27.94N/mm^2$$

The maximum principal stresses are

$$\sigma_1 = \frac{\sigma_{max}}{2} + \sqrt{\left(\frac{\sigma_{max}}{2}\right)^2 + \tau_{max}{}^2}$$

$$\sigma_1 = \frac{185.86}{2} + \sqrt{\left(\frac{185.86}{2}\right)^2 + 27.94^2}$$

$$\sigma_1 = 189.96N/mm^2$$

$$\sigma_2 = \frac{\sigma_{max}}{2} - \sqrt{\left(\frac{\sigma_{max}}{2}\right)^2 + \tau_{max}{}^2}$$

$$\sigma_2 = \frac{185.86}{2} - \sqrt{\left(\frac{185.86}{2}\right)^2 + 27.94^2} = -4.1N/mm^2$$

$$Here\ \ \sigma_1 = \sigma_{max}\ and\ \ \sigma_2 = \sigma_{min}$$

$$\sigma_{equ} = \frac{\sigma_a}{1 - \left(\frac{\sigma_m}{\sigma_{ut}}\right)^n}$$

Where

σ_{equ} is equivalent alternating stress,

σ_a is alternating stress, $\frac{\sigma_{max}-\sigma_{min}}{2}$

σ_m is mean stress, $\frac{\sigma_{max}+\sigma_{min}}{2}$

σ_y is yield stress for soderberg criteria and ultimate stress for goodman, gerber criteria.

n = 1 for Soderberg and Goodman criteria and 2 for Gerber criteria

$\sigma_{ut} = 827MPa, \qquad \sigma_y = 625MPa$

$$\sigma_a = \frac{\sigma_{max} - \sigma_{min}}{2}$$

$$\sigma_a = \frac{189.96 - (-4.1)}{2} = 97.03N/mm^2$$

$$\sigma_m = \frac{\sigma_{max} + \sigma_{min}}{2}$$

$$\sigma_m = \frac{189.96 + (-4.1)}{2} = 92.94N/mm^2$$

$$\sigma_{equ} = \frac{\sigma_a}{1 - \left(\frac{\sigma_m}{\sigma_{ut}}\right)^n} \qquad \sigma_{equ} = \frac{97.03}{1 - \left(\frac{92.93}{827}\right)^2} = 98.27N/mm^2$$

For Gerber criteria n=2

Thus the corresponding maximum shear stress is

(Westinghouse code formula) [39]

$$\tau_{\boldsymbol{eq}} = \sqrt{\left(\frac{\sigma_{equ}}{2}\right)^2 + \tau_{max}{}^2}$$

$$\tau_{\boldsymbol{eq}} = \sqrt{\left(\frac{\sigma_{equ}}{2}\right)^2 + \tau_{max}{}^2}$$

$$\tau_{\boldsymbol{eq}} = \sqrt{\left(\frac{98.27}{2}\right)^2 + 27.94^2} = 56.52N/mm^2$$

For 3.5mm fillet radius

$D = 50mm$

$d = 37mm$ $\qquad \frac{2h}{D} = 0.26$

$h = 6.5mm$ $\qquad \left(\frac{2h}{D}\right)^2 = 0.067mm$

$r = 3.5mm$ $\qquad \left(\frac{2h}{D}\right)^3 = 0.017mm$

$\frac{h}{r} = 1.86mm$

$$\sqrt{\frac{h}{r}} = 1.36mm$$

For bending

$\sigma_{\boldsymbol{max}} = \boldsymbol{K_t}\, \sigma_{nom}$

σ_{nom}=104.42N/mm^2

To obtain stress concentration factor (K_t)

$$K_t = C_1 + C_2\left(\frac{2h}{D}\right) + C_3\left(\frac{2h}{D}\right)^2 + C_4\left(\frac{2h}{D}\right)^3$$

$0.1 \le \frac{h}{r} \le 2.0,\ where\ \frac{h}{r} = 1.86mm$

$$C_1 = 0.947 + 1.206\sqrt{\frac{h}{r}} - 0.131(\ h/r)$$

$$C_1 = 0.947 + 1.206\sqrt{\frac{6.5}{3.5}} - 0.131(\ 6.5/3.5) = 2.34$$

$$C_2 = -0.022 -\ 3.405\sqrt{\frac{h}{r}} + 0.915(\ h/r)$$

$$C_2 = -0.022 -\ 3.405\sqrt{\frac{6.5}{3.5}} + 0.915(\ 6.5/3.5) = -2.91$$

$$C_3 = 7.423 - 4.868\sqrt{\frac{h}{r}} + 0.869(\ h/r)$$

$$C_3 = 0.869 + 1.777\sqrt{\frac{6.5}{3.5}} - 0.555(\ 6.5/3) = 2.25$$

$$C_4 = -0.810 + 0.422\sqrt{\frac{h}{r}} - 0.260(\ h/r)$$

$C_4 = -0.810 + 0.422\sqrt{\frac{6.5}{3.5}} - 0.260(\ 6.5/3.5) = -0.72$

$$K_t = C_1 + C_2\left(\frac{2h}{D}\right) + C_3\left(\frac{2h}{D}\right)^2 + C_4\left(\frac{2h}{D}\right)^3$$

$$K_t = 2.34 - 2.91\left(\frac{2*6.5}{50}\right) + 2.25\left(\frac{2*6.5}{50}\right)^2 - 0.72\left(\frac{2*6.5}{50}\right)^3$$

$K_t = 1.72$

$\sigma_{max} = K_t\,\sigma_{nom}$

$\sigma_{max} = 1.72 * 104.42 N/mm^2$

$\sigma_{max} = 179.60 N/mm^2$

For Torsion

$\tau_{max} = K_t\,\tau_{nom}$

Where T is twisting momentum=170.68KN-mm

$$\tau_{nom} = \frac{16 * 170680}{\pi(37^3 - 18^3)} = 19.40 N/mm^2$$

To obtain stress concentration factor (K_t)

$$K_t = C_1 + C_2\left(\frac{2h}{D}\right) + C_3\left(\frac{2h}{D}\right)^2 + C_4\left(\frac{2h}{D}\right)^3$$

$0.25 \leq \frac{h}{r} \leq 4.0$, where $\frac{h}{r} = 1.86mm$

$$C_1 = 0.905 + 0.783\sqrt{\frac{h}{r}} - 0.075(\ h/r)$$

$C_1 = 0.905 + 0.783\sqrt{\frac{6.5}{3.5}} - 0.075(\ 6.5/3.5)$ =1.83

$$C_2 = -0.437 - 1.969\sqrt{\frac{h}{r}} + 0.553(\,h/r)$$

$$C_2 = -0.437 - 1.969\sqrt{\tfrac{6.5}{3.5}} + 0.553\left(\tfrac{6.5}{3.5}\right) = -2.08$$

$$C_3 = 1.557 + 1.073\sqrt{\frac{h}{r}} - 0.578(\,h/r)$$

$$C_3 = 1.557 + 1.073\sqrt{\tfrac{6.5}{3.5}} - 0.578(\,6.5/3.5) = 1.94$$

$$C_4 = -1.061 + 0.171\sqrt{\frac{h}{r}} + 0.086(\,h/r)$$

$$C_4 = -1.061 + 0.171\sqrt{\tfrac{6.5}{3.5}} + 0.086(\,6.5/3.5) = -0.67$$

$$K_t = C_1 + C_2\left(\frac{2h}{D}\right) + C_3\left(\frac{2h}{D}\right)^2 + C_4\left(\frac{2h}{D}\right)^3$$

$$K_t = 1.83 - 2.08\left(\frac{2*6.5}{50}\right) + 1.94\left(\frac{2*6.5}{50}\right)^2 - 0.67\left(\frac{2*6.5}{50}\right)^3$$

$$K_t = 1.45$$

$$\tau_{max} = K_t\,\tau_{nom}$$

$$\tau_{max} = 1.45 * 19.40 = 28.13N/mm^2$$

The maximum principal stresses are

$$\sigma_1 = \frac{\sigma_{max}}{2} + \sqrt{\left(\frac{\sigma_{max}}{2}\right)^2 + \tau_{max}{}^2}$$

$$\sigma_1 = \frac{179.60}{2} + \sqrt{\left(\frac{179.60}{2}\right)^2 + 28.13^2}$$

$\sigma_1 = 183.9N/mm^2$

$$\sigma_2 = \frac{\sigma_{max}}{2} - \sqrt{\left(\frac{\sigma_{max}}{2}\right)^2 + \tau_{max}{}^2}$$

$$= \frac{179.60}{2} - \sqrt{\left(\frac{179.60}{2}\right)^2 + 28.13^2} = -4.3N/mm^2$$

$$\sigma_a = \frac{\sigma_{max} - \sigma_{min}}{2}$$

$$\sigma_a = \frac{183.9 - (-4.3)}{2} = 94.10N/mm^2$$

$$\sigma_m = \frac{\sigma_{max} + \sigma_{min}}{2}$$

$$\sigma_m = \frac{183.9 + (-4.3)}{2} = 89.9N/mm^2$$

$$\sigma_{equ} = \frac{\sigma_a}{1 - \left(\frac{\sigma_m}{\sigma_{ut}}\right)^n}$$

$$\sigma_{equ} = \frac{94.10}{1 - \left(\frac{89.8}{827}\right)^2}$$

$=95.22N/mm^2$

Thus the corresponding maximum shear stress is

$$\tau_{\boldsymbol{eq}} = \sqrt{\left(\frac{\sigma_{equ}}{2}\right)^2 + \tau_{max}{}^2}$$

$$\tau_{\boldsymbol{eq}} = \sqrt{\left(\frac{95.22}{2}\right)^2 + 28.13^2}$$

$= 55.29N/mm^2$

For 4mm fillet radius

$D = 50mm$

$d = 37mm$

$h = 6.5mm$

$r = 4mm$

$\frac{h}{r} = 1.63mm$

$\sqrt{\frac{h}{r}} = 1.27mm$

$\frac{2h}{D} = 0.26$

$\left(\frac{2h}{D}\right)^2 = 0.067mm$

$\left(\frac{2h}{D}\right)^3 = 0.017mm$

For bending

$\sigma_{\boldsymbol{max}} = \boldsymbol{K_t}\, \sigma_{nom}$

$$\sigma_{nom} = \frac{32M}{\pi d^3}$$

σ_{nom}=104.42N/mm^2

To obtain stress concentration factor (K_t)

$$K_t = C_1 + C_2 \left(\frac{2h}{D}\right) + C_3 \left(\frac{2h}{D}\right)^2 + C_4 \left(\frac{2h}{D}\right)^3$$

$0.1 \le \frac{h}{r} \le 2.0$, where $\frac{h}{r} = 1.63mm$

$$C_1 = 0.947 + 1.206\sqrt{\frac{h}{r}} - 0.131(\,h/r)$$

$C_1 = 0.947 + 1.206\sqrt{\frac{6.5}{4}} - 0.131(\,6.5/4)$ =2.27

$$C_2 = -0.022 - 3.405\sqrt{\frac{h}{r}} + 0.915(h/r)$$

$$C_2 = -0.022 - 3.405\sqrt{\frac{6.5}{4}} + 0.915(6.5/4) = -2.81$$

$$C_3 = 7.423 - 4.868\sqrt{\frac{h}{r}} + 0.869(h/r)$$

$$C_3 = 0.869 + 1.777\sqrt{\frac{6.5}{4}} - 0.555(6.5/4 = 2.22$$

$$C_4 = -0.810 + 0.422\sqrt{\frac{h}{r}} - 0.260(h/r)$$

$$C_4 = -0.810 + 0.422\sqrt{\frac{6.5}{4}} - 0.260(6.5/4) = -0.16$$

$$K_t = C_1 + C_2\left(\frac{2h}{D}\right) + C_3\left(\frac{2h}{D}\right)^2 + C_4\left(\frac{2h}{D}\right)^3$$

$$K_t = 2.27 - 2.81\left(\frac{2*6.5}{50}\right) + 2.22\left(\frac{2*6.5}{50}\right)^2 - 0.16\left(\frac{2*6.5}{50}\right)^3$$

$$K_t = 1.68$$

$$\sigma_{max} = K_t\,\sigma_{nom}$$

$$\sigma_{max} = 1.68 * 104.42 N/mm^2$$

$$\sigma_{max} = 175.42 N/mm^2$$

For Torsion

$$\tau_{max} = K_t\,\tau_{nom}$$

$$\tau_{nom} = \frac{16T}{\pi d^3}$$

$$\tau_{nom} = \frac{16 * 170680}{\pi * 37^3 - 18^3} = 19.40 N/mm^2$$

To obtain stress concentration factor (K_t)

$$K_t = C_1 + C_2 \left(\frac{2h}{D}\right) + C_3 \left(\frac{2h}{D}\right)^2 + C_4 \left(\frac{2h}{D}\right)^3$$

$0.25 \leq \frac{h}{r} \leq 4.0$, where $\frac{h}{r} = 1.63mm$

$$C_1 = 0.905 + 0.783 \sqrt{\frac{h}{r}} - 0.075(h/r)$$

$$C_1 = 0.905 + 0.783 \sqrt{\frac{6.5}{4}} - 0.075(6.5/4) = 1.78$$

$$C_2 = -0.437 - 1.969 \sqrt{\frac{h}{r}} + 0.553(h/r)$$

$$C_2 = -0.437 - 1.969 \sqrt{\frac{6.5}{4}} + 0.553 \left(\frac{6.5}{4}\right) = -2.03$$

$$C_3 = 1.557 + 1.073 \sqrt{\frac{h}{r}} - 0.578(h/r)$$

$$C_3 = 1.557 + 1.073 \sqrt{\frac{6.5}{4}} - 0.578(6.5/4) = 1.98$$

$$C_4 = -1.061 + 0.171 \sqrt{\frac{h}{r}} + 0.086(h/r)$$

$$C_4 = -1.061 + 0.171 \sqrt{\frac{6.5}{4}} + 0.086(6.5/4) = -0.68$$

$$K_t = C_1 + C_2 \left(\frac{2h}{D}\right) + C_3 \left(\frac{2h}{D}\right)^2 + C_4 \left(\frac{2h}{D}\right)^3$$

$$K_t = 1.78 - 2.03\left(\frac{2*6.5}{50}\right) + 1.98\left(\frac{2*6.5}{50}\right)^2 - 0.68\left(\frac{2*6.5}{50}\right)^3$$

$$K_t = 1.37$$

$$\tau_{max} = K_t\,\tau_{nom}$$

$$\tau_{max} = 1.37 * 19.40 = 26.57N/mm^2$$

The maximum principal stress

$$\sigma_1 = \frac{\sigma_{max}}{2} + \sqrt{\left(\frac{\sigma_{max}}{2}\right)^2 + \tau_{max}{}^2}$$

$$\sigma_1 = \frac{175.42}{2} + \sqrt{\left(\frac{175.42}{2}\right)^2 + 26.57^2}$$

$$\sigma_1 = 179.5N/mm^2$$

$$\sigma_2 = \frac{\sigma_{max}}{2} - \sqrt{\left(\frac{\sigma_{max}}{2}\right)^2 + \tau_{max}{}^2}$$

$$\sigma_2 = \frac{175.42}{2} - \sqrt{\left(\frac{175.42}{2}\right)^2 + 26.57^2}$$

$$\sigma_2 = -3.94N/mm^2$$

$$\sigma_a = \frac{\sigma_{max} - \sigma_{min}}{2}$$

$$\sigma_a = \frac{179.5 - (-3.94)}{2} = 91.72N/mm^2$$

$$\sigma_m = \frac{\sigma_{max} + \sigma_{min}}{2}$$

$$\sigma_m = \frac{179.5 + (-3.94)}{2} = 87.78N/mm^2$$

$$\sigma_{equ} = \frac{\sigma_a}{1 - \left(\frac{\sigma_m}{\sigma_{ut}}\right)^n}$$

$$\sigma_{equ} = \frac{91.72}{1-\left(\frac{87.78}{827}\right)^2} = 92.76N/mm^2$$

Thus the corresponding maximum shear stress is

$$\tau_{eq} = \sqrt{\left(\frac{\sigma_{equ}}{2}\right)^2 + {\tau_{max}}^2}$$

$$\tau_{eq} = \sqrt{\left(\frac{92.76}{2}\right)^2 + 26.57^2} = 53.45N/mm^2$$

For 4.5mm fillet radius

$D = 50mm$

$d = 37mm$

$\frac{2h}{D} = 0.26$

$h = 6.5mm$

$\left(\frac{2h}{D}\right)^2 = 0.067mm$

$r = 4.5mm$

$\left(\frac{2h}{D}\right)^3 = 0.017mm$

$\frac{h}{r} = 1.44mm$

$\sqrt{\frac{h}{r}} = 1.20mm$

For bending

$\sigma_{max} = K_t\, \sigma_{nom}$

To obtain stress concentration factor (K_t)

$$K_t = C_1 + C_2\left(\frac{2h}{D}\right) + C_3\left(\frac{2h}{D}\right)^2 + C_4\left(\frac{2h}{D}\right)^3$$

$0.1 \leq \frac{h}{r} \leq 2.0$, where $\frac{h}{r} = 1.44mm$

$$C_1 = 0.947 + 1.206\sqrt{\frac{h}{r}} - 0.131(\,h/r)$$

$$C_1 = 0.947 + 1.206\sqrt{\frac{6.5}{4.5}} - 0.131(\,6.5/4.5) = 2.21$$

$$C_2 = -0.022 - \ 3.405\sqrt{\frac{h}{r}} + 0.915(\,h/r)$$

$$C_2 = -0.022 - \ 3.405\sqrt{\frac{6.5}{4.5}} + 0.915(\,6.5/4.5) \ = -2.75$$

$$C_3 = 7.423 - 4.868\sqrt{\frac{h}{r}} + 0.869(\,h/r)$$

$$C_3 = 0.869 + 1.777\sqrt{\frac{6.5}{4.5}} - 0.555(\,6.5/4.5) = 2.20$$

$$C_4 = -0.810 + 0.422\sqrt{\frac{h}{r}} - 0.260(\,h/r)$$

$$C_4 = -0.810 + 0.422\sqrt{\frac{6.5}{4.5}} - 0.260(\,6.5/4.5) \ = -0.68$$

$$K_t = C_1 \ + C_2\left(\frac{2h}{D}\right) + C_3\left(\frac{2h}{D}\right)^2 + C_4\left(\frac{2h}{D}\right)^3$$

$$K_t = 2.21 - 2.75\left(\frac{2*6.5}{50}\right) + 2.20\left(\frac{2*6.5}{50}\right)^2 - 0.68\left(\frac{2*6.5}{50}\right)^3$$

$$K_t = 1.63$$

$$\sigma_{max} = K_t\,\sigma_{nom}$$

$$\sigma_{max} = 1.63 * 104.42N/mm^2$$

$\sigma_{max} = 170.20N/mm^2$

For Torsion

$\tau_{max} = K_t\, \tau_{nom}$

To obtain stress concentration factor (K_t)

$$K_t = C_1 + C_2\left(\frac{2h}{D}\right) + C_3\left(\frac{2h}{D}\right)^2 + C_4\left(\frac{2h}{D}\right)^3$$

$0.25 \leq \frac{h}{r} \leq 4.0$, where $\frac{h}{r} = 1.44mm$

$$C_1 = 0.905 + 0.783\sqrt{\frac{h}{r}} - 0.075(\,h/r)$$

$$C_1 = 0.905 + 0.783\sqrt{\frac{6.5}{4.5}} - 0.075(\,6.5/4.5) = 1.74$$

$$C_2 = -0.437 - 1.969\sqrt{\frac{h}{r}} + 0.553(\,h/r)$$

$$C_2 = -0.437 - 1.969\sqrt{\frac{6.5}{4.5}} + 0.553\left(\frac{6.5}{4.5}\right) = -2$$

$$C_3 = 1.557 + 1.073\sqrt{\frac{h}{r}} - 0.578(\,h/r)$$

$$C_3 = 1.557 + 1.073\sqrt{\frac{6.5}{4.5}} - 0.578(\,6.5/4.5) = 2.01$$

$$C_4 = -1.061 + 0.171\sqrt{\frac{h}{r}} + 0.086(\,h/r)$$

$$C_4 = -1.061 + 0.171\sqrt{\frac{6.5}{4.5}} + 0.086(\,6.5/4.5) = -0.73$$

$$K_t = C_1 + C_2\left(\frac{2h}{D}\right) + C_3\left(\frac{2h}{D}\right)^2 + C_4\left(\frac{2h}{D}\right)^3$$

$$K_t = 1.74 - 2\left(\frac{2*6.5}{50}\right) + 2.01\left(\frac{2*6.5}{50}\right)^2 - 0.73\left(\frac{2*6.5}{50}\right)^3$$

$$K_t = 1.34$$

$$\tau_{max} = K_t\,\tau_{nom}$$

$$\tau_{max} = 1.34 * 19.40 = 25.99 N/mm^2$$

The maximum principal stress is

$$\sigma_1 = \frac{\sigma_{max}}{2} + \sqrt{\left(\frac{\sigma_{max}}{2}\right)^2 + \tau_{max}{}^2}$$

$$\sigma_1 = \frac{170.20}{2} + \sqrt{\left(\frac{170.20}{2}\right)^2 + 25.99^2}$$

$$\sigma_1 = 174.08 N/mm^2$$

$$\sigma_2 = \frac{\sigma_{max}}{2} - \sqrt{\left(\frac{\sigma_{max}}{2}\right)^2 + \tau_{max}{}^2}$$

$$\sigma_2 = \frac{170.20}{2} + \sqrt{\left(\frac{170.20}{2}\right)^2 + 25.99^2}$$

$$\sigma_2 = -3.88 N/mm^2$$

$$\sigma_a = \frac{\sigma_{max} - \sigma_{min}}{2}$$

$$\sigma_a = \frac{174.08 - (-3.88)}{2} = 88.98 N/mm^2$$

$$\sigma_m = \frac{\sigma_{max} + \sigma_{min}}{2}$$

$$\sigma_m = \frac{174.08 + (-3.88)}{2} = 85.1 N/mm^2$$

$$\sigma_{equ} = \frac{\sigma_a}{1 - \left(\frac{\sigma_m}{\sigma_{ut}}\right)^n}$$

$$\sigma_{equ} = \frac{88.98}{1 - \left(\frac{85.1}{827}\right)^2} = 89.93 N/mm^2$$

Thus the corresponding maximum shear stress is

$$\tau_{\boldsymbol{eq}} = \sqrt{\left(\frac{\sigma_{equ}}{2}\right)^2 + {\tau_{max}}^2}$$

$$\tau_{\boldsymbol{eq}} = \sqrt{\left(\frac{89.93}{2}\right)^2 + 25.99^2} = 51.93 N/mm^2$$

3.4.2 Finite element method

The basis of FEA relies on the decomposition of the domain into a finite number of subdomains (elements) for which the systematic approximate solution is constructed by applying the variational or weighted residual methods. It is not always possible to obtain the exact analytical solution at any location in the body, especially for those elements having complex shapes or geometries. Always the most important are the boundary conditions and material properties. In such cases, the analytical solution that satisfies the governing equation or gives extreme values for the governing functional is difficult to obtain. Hence for most of the practical problems, the engineers resort to numerical methods like the finite element method to obtain approximate but most probable solutions. Finite element procedures are at present very widely used in engineering analysis.

a. Discretising the domain: This step involves subdividing the domain into elements and nodes. For continuous system this step is very important and the answers obtained are only approximate. In this case, the accuracy of the solution depends on the descritisation used.

b. Writing the element stiffness matrices: The element stiffness equations need to be written for each element in the domain.

c. Assembling the global stiffness matrix: This is done using the direct stiffness approach to obtain the stiffness matrix for the entire system.

d. Applying the boundary conditions: This involves specifying the load conditions and restraints like supports and applied loads and displacements. In this study this step is performed manually.

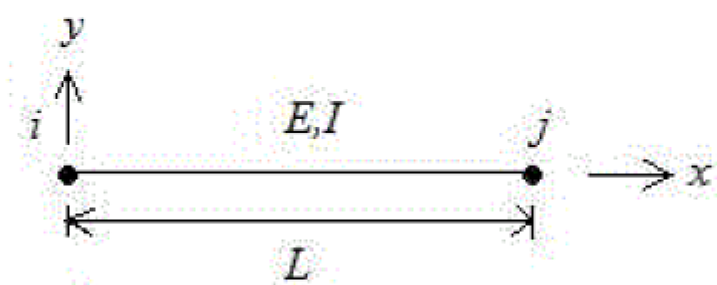
e. Solving the equation: This is done by partitioning the global stiffness matrix and then solving the resulting equation using Gaussian elimination and the reactions and element stresses and visualization of the resulting solution.

Types of elements used

- ✓ Linear bar element
- ✓ Truss element
- ✓ Beam element

In this study beam element is used since the crankshaft is assumed to abeam element with two or more supports. The beam element is a two-dimensional finite element where the local and global coordinates coincide. It is characterized by linear shape functions.

In the case of transverse loading, the beam element has modulus of elasticity E, moment of inertia I, and the length L. Each beam element has two nodes and is assumed to be horizontal as shown in Figure below.



In this case the element stiffness matrix is given by the following matrix,

$$K = EI/L^3 \begin{pmatrix} 12 & 6L & -12 & 6L \\ 6L & 4L^2 & -6L & 2L^2 \\ -12 & -6L & 12 & -6L \\ 6L & 2L^2 & -6L & 4L^2 \end{pmatrix} \begin{Bmatrix} v1 \\ \theta 1 \\ v2 \\ \theta 2 \end{Bmatrix} = \begin{Bmatrix} F1 \\ M1 \\ F2 \\ M2 \end{Bmatrix}$$

It is clear that the beam element has four degrees of freedom -two at each node (a transverse displacement and a rotation. The global stiffness matrix K is obtained, the structure equation is as follows:

$\{F\} = \{K\}\ \{U\}$

Where $\{U\}$ is the global nodal displacement vector and $\{F\}$ is the global nodal force vector. At this step the boundary conditions are applied manually to the vectors $\{U\}$ and $\{F\}$. Then the matrix equation is solved by partitioning and Gaussian elimination.

Finite element analysis of crankpin

The crank pin is like a built in beam with a distributed load along its length that varies with crank positions. Each web is like a cantilever beam subjected to bending and twisting

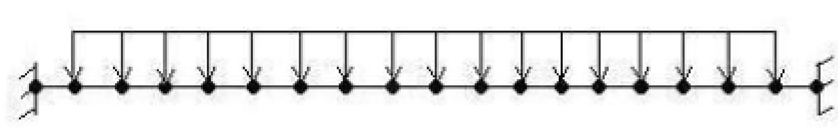
(A) Load analysis: When the crank is at dead center, the maximum gas load on the piston is transmitted to the crankpin and is assumed to act as uniformly distributed load acting along the length of the crankpin.

The gas load on the piston obtained during conventional design is 14.52 KN. When the crank is at the angle of maximum twisting moment, the equivalent twisting moment due to the bending and twisting loads is assumed to act at the center of the crankpin. The equivalent twisting moment acting at the center of crankpin obtained during conventional design is 489.93KN-mm.

(B) Boundary conditions: The crankpin is assumed to be acting as a fixed-fixed beam under all conditions. When at the dead center, it is assumed that a uniformly distributed load is acting on all the nodes except the end nodes which are fixed.

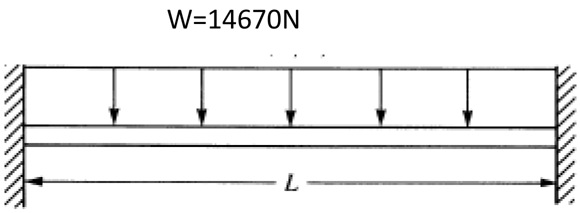
When at the position of maximum twisting moment, it is assumed that the equivalent twisting moment is acting on the central node and the end nodes are fixed.

(C) Finite element analysis: The conventional deign method gives the geometry of the crankpin, and the load conditions are obtained afterload analysis. These results are finally used to carry out the finite element analysis of the crankpin on the basis of the assumed boundary conditions.

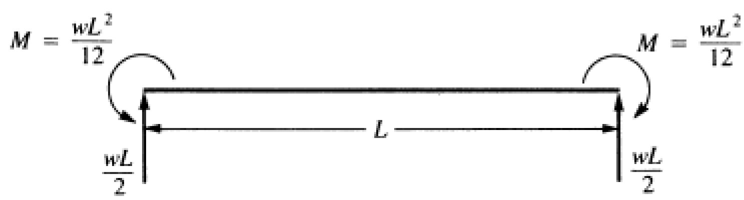


Crankpin at dead center

First, the analysis is carried out at the dead center. The crankpin is discretized using beam elements. Beam members can support distributed loading as well as concentrated nodal loading. Therefore, we must be able to account for distributed loading. Consider the fixed-fixed beam subjected to a uniformly distributed. These reactions are called fixed-end reactions. In general, fixed-end reactions are those reactions at the ends of an element if the ends of the element are assumed to be fixed that is, if displacements and rotations are prevented.



Fixed-fixed beam subjected to a uniformly distributed load



The work equivalent nodal force system

Where L = 32mm

E= $221000N/mm^2$

$I = \frac{\pi(dc)^4}{64}$=91951mm^4

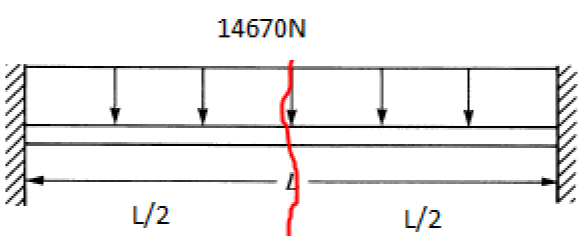
$I = \frac{\pi(do^4-di^4)}{64}$=8680064$mm^4$

W= 14670N/mm

W= 14670N/32mm

Unit load w=458.43N/mm

Total load W= 14670N



Work-equivalent nodal forces

Shape functions

$N_1(s) = 1-3s^2+2s^3$

$N_2(s) = L(s-2s^2+s^3)$

$N_3(s) = 3s^2-2s^3$

$N_4(s) = L\,(-s^2+s^3)$

Uniformly distributed load

$$F_1 = WL\int_0^1 N_1(s)\,ds = WL\int_0^1 (1-3s^2+2s^2)\,ds = \frac{WL}{2}$$

$$M_1 = WL\int_0^1 N_2(s)\,ds = WL^2\int_0^1 (s-2s^2+s^3)\,ds = \frac{WL^2}{12}$$

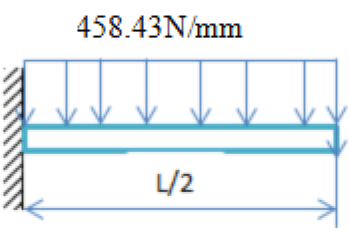
$$F_2 = WL\int_0^1 N_3(s)\,ds = WL\int_0^1 (3s^2-2s^3)\,ds = \frac{WL}{2}$$

$$M_2 = WL\int_0^1 N_4(s)\,ds = WL^2\int_0^1 (-s^2+s^3)\,ds = -\frac{WL^2}{12}$$

Finite element equation for beam

$$K = EI/L^3\begin{bmatrix} 12 & 6L & -12 & 6L \\ 6L & 4L^2 & -6L & 2L^2 \\ -12 & -6L & 12 & -6L \\ 6L & 2L^2 & -6L & 4L^2 \end{bmatrix}\begin{Bmatrix} v1 \\ \theta1 \\ v2 \\ \theta2 \end{Bmatrix} = \begin{Bmatrix} WL/2 \\ WL^2/12 \\ WL/2 \\ -WL^2/12 \end{Bmatrix} + \begin{Bmatrix} F1 \\ M1 \\ F2 \\ M2 \end{Bmatrix}$$

For element 1



L=16mm

Unit load w=458.43N/mm

Total load W= 7335N

WL=7335N

$$K = EI/L^3\;\begin{array}{c} \begin{array}{cccc} v1 & \theta1 & v2 & \theta2 \end{array} \\ \begin{bmatrix} 12 & 6L & -12 & 6L \\ 6L & 4L^2 & -6L & 2L^2 \\ -12 & -6L & 12 & -6L \\ 6L & 2L^2 & -6L & 4L^2 \end{bmatrix} \end{array}\begin{array}{c} \\ v1 \\ \theta1 \\ v2 \\ \theta2 \end{array}$$

$$K = (221000N/mm^2 * 86800.64mm^4)/16^3 \begin{array}{c} \begin{bmatrix} 12 & 6L & -12 & 6L \\ 6L & 4L^2 & -6L & 2L^2 \\ -12 & -6L & 12 & -6L \\ 6L & 2L^2 & -6L & 4L^2 \end{bmatrix} \end{array} \begin{array}{c} v1 \\ \theta 1 \\ v2 \\ \theta 2 \end{array}$$

$$K = 468\text{x}10^4 \begin{bmatrix} 12 & 96 & -12 & 96 \\ 96 & 1024 & -96 & 512 \\ -12 & -96 & 12 & -96 \\ 96 & 512 & -96 & 1024 \end{bmatrix} \begin{array}{c} v1 \\ \theta 1 \\ v2 \\ \theta 2 \end{array}$$

- We can only apply loads to nodes in FE analyses
- Hence, distributed loads must be converted to equivalent nodal loads

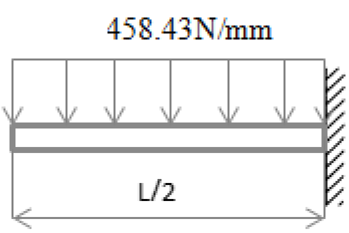
Work-equivalent nodal forces

$$\begin{Bmatrix} F_{1e} \\ M_{1e} \\ F_{2e} \\ M_{2e} \end{Bmatrix} = \text{WL} \int_0^1 \begin{Bmatrix} 1 - 3s^2 + 2s^2 \\ s - 2s^2 + s^3 \\ 3s^2 - 2s^3 \\ -s^2 + s^3 \end{Bmatrix} \text{ds} = \text{WL} \begin{Bmatrix} L/2 \\ L/12 \\ L/2 \\ -L/12 \end{Bmatrix} = \begin{Bmatrix} 3667.5 \\ 611.25 \\ 3667.5 \\ -611.25 \end{Bmatrix}$$

FE matrix equation

$$K = 468\text{x}10^4 \begin{vmatrix} 12 & 96 & -12 & 96 \\ 96 & 1024 & -96 & 512 \\ -12 & -96 & 12 & -96 \\ 96 & 512 & -96 & 1024 \end{vmatrix} \begin{Bmatrix} v1 \\ \theta 1 \\ v2 \\ \theta 2 \end{Bmatrix} = \begin{Bmatrix} F1 + 3667.5 \\ M1 + 611.25 \\ 3667.5 \\ -611.25 \end{Bmatrix}$$

For element 2



$$K = EI/L^3 \begin{array}{c} \begin{array}{cccc} v2 & \theta 2 & v3 & \theta 3 \end{array} \\ \begin{bmatrix} 12 & 6L & -12 & 6L \\ 6L & 4L^2 & -6L & 2L^2 \\ -12 & -6L & 12 & -6L \\ 6L & 2L^2 & -6L & 4L^2 \end{bmatrix} \end{array} \begin{array}{c} v2 \\ \theta 2 \\ v3 \\ \theta 3 \end{array}$$

$$K = 468\text{x}10^4 \begin{bmatrix} 12 & 6L & -12 & 6L \\ 6L & 4L^2 & -6L & 2L^2 \\ -12 & -6L & 12 & -6L \\ 6L & 2L^2 & -6L & 4L^2 \end{bmatrix} \begin{array}{c} v2 \\ \theta 2 \\ v3 \\ \theta 3 \end{array}$$

$$K = 468\text{x}10^4 \begin{bmatrix} 12 & 96 & -12 & 96 \\ 96 & 1024 & -96 & 512 \\ -12 & -96 & 12 & -96 \\ 96 & 512 & -96 & 1024 \end{bmatrix} \begin{array}{c} v2 \\ \theta 2 \\ v3 \\ \theta 3 \end{array}$$

Work-equivalent nodal forces

$$\begin{Bmatrix} F_{2e} \\ M_{2e} \\ F_{3e} \\ M_{3e} \end{Bmatrix} = \text{WL} \int_0^1 \begin{Bmatrix} 1 - 3s^2 + 2s^2 \\ s - 2s^2 + s^3 \\ 3s^2 - 2s^3 \\ -s^2 + s^3 \end{Bmatrix} \text{ds} = \text{WL} \begin{Bmatrix} L/2 \\ L/12 \\ L/2 \\ -L/12 \end{Bmatrix} = \begin{Bmatrix} 3667.5 \\ 611.25 \\ 3667.5 \\ -611.25 \end{Bmatrix}$$

FE matrix equation

$$K = 468\text{x}10^4 \begin{vmatrix} 12 & 96 & -12 & 96 \\ 96 & 1024 & -96 & 512 \\ -12 & -96 & 12 & -96 \\ 96 & 512 & -96 & 1024 \end{vmatrix} \begin{Bmatrix} v2 \\ \theta 2 \\ v3 \\ \theta 3 \end{Bmatrix} = \left\{ \begin{Bmatrix} 3667.5 \\ 611.25 \\ 3667.5 \\ -611.25 \end{Bmatrix} \right\}$$

The governing equations for the beam are thus given by

$$468\text{x}10^4 \begin{array}{c} \begin{array}{cccccc} v1 & \theta 1 & v2 & \theta 2 & v3 & \theta 3 \end{array} \\ \begin{bmatrix} 12 & 6L & -12 & 6L & 0 & 0 \\ 6L & 4L^2 & -6L & 2L^2 & 0 & 0 \\ -12 & -6L & 24 & 0 & -12 & 6L \\ 6L & 2L^2 & 0 & 8L^2 & -6L & 2L^2 \\ 0 & 0 & -12 & -6L & 12 & -6L \\ 0 & 0 & 6L & 2L^2 & -6L & 4L^2 \end{bmatrix} \end{array} \begin{Bmatrix} v1 \\ \theta 1 \\ v2 \\ \theta 2 \\ v3 \\ \theta 3 \end{Bmatrix} = \begin{Bmatrix} F1 \\ M1 \\ F2 \\ M2 \\ F3 \\ M3 \end{Bmatrix}$$

$$468\text{x}10^4 \begin{array}{c} \begin{array}{cccccc} v1 & \theta 1 & v2 & \theta 2 & v3 & \theta 3 \end{array} \\ \begin{bmatrix} 12 & 96 & -12 & 96 & 0 & 0 \\ 96 & 1024 & -96 & 512 & 0 & 0 \\ -12 & -96 & 24 & 0 & -12 & 96 \\ 96 & 512 & 0 & 2048 & -96 & 512 \\ 0 & 0 & -12 & -96 & 12 & -96 \\ 0 & 0 & 96 & 512 & -96 & 1024 \end{bmatrix} \end{array} \begin{Bmatrix} v1 \\ \theta 1 \\ v2 \\ \theta 2 \\ v3 \\ \theta 3 \end{Bmatrix} = \begin{Bmatrix} F1 \\ M1 \\ F2 \\ M2 \\ F3 \\ M3 \end{Bmatrix}$$

Apply boundary condition

$$
\begin{array}{c}
\begin{array}{cccccc} v1 & \theta 1 & v2 & \theta 2 & v3 & \theta 3 \end{array} \\
468\text{x}10^4 \begin{bmatrix} 12 & 96 & -12 & 96 & 0 & 0 \\ 96 & 1024 & -96 & 512 & 0 & 0 \\ -12 & -96 & 24 & 0 & -12 & 96 \\ 96 & 512 & 0 & 2048 & -96 & 512 \\ 0 & 0 & -12 & -96 & 12 & -96 \\ 0 & 0 & 96 & 512 & -96 & 1024 \end{bmatrix} \begin{Bmatrix} 0 \\ 0 \\ v2 \\ \theta 2 \\ 0 \\ 0 \end{Bmatrix} = \begin{Bmatrix} F1 \\ M1 \\ 3667.5 \\ -611.25 \\ F3 \\ M3 \end{Bmatrix}
\end{array}
$$

K= $468\text{x}10^4 \begin{bmatrix} 24 & 0 \\ 0 & 2048 \end{bmatrix} \begin{Bmatrix} v2 \\ \theta 2 \end{Bmatrix} = \begin{Bmatrix} 3667.5 \\ -611.25 \end{Bmatrix}$

$468\text{x}10^4 x 24 v2 = 3667.5$

$v2$=0.0000326mm

$\theta 2$ = - 6.37x10^{-8}rad

We will now illustrate the procedure for obtaining the global nodal forces and where $F^{(e)}$ are called the effective global nodal forces.

$\{F\} = [K\{d\}$

$$
\begin{array}{c}
\begin{array}{cccccc} v1 & \theta 1 & v2 & \theta 2 & v3 & \theta 3 \end{array} \\
\begin{Bmatrix} {F_{1y}}^{(e)} \\ {M_1}^{(e)} \\ {F_{2y}}^{(e)} \\ {M_2}^{(e)} \\ {F_{3y}}^{(e)} \\ {M_3}^{(e)} \end{Bmatrix} = 4683335.3 \begin{bmatrix} 12 & 96 & -12 & 96 & 0 & 0 \\ 96 & 1024 & -96 & 512 & 0 & 0 \\ -12 & -96 & 24 & 0 & -12 & 96 \\ 96 & 512 & 0 & 2048 & -96 & 512 \\ 0 & 0 & -12 & -96 & 12 & -96 \\ 0 & 0 & 96 & 512 & -96 & 1024 \end{bmatrix} \begin{Bmatrix} 0 \\ 0 \\ 0.0000326 \\ -6.37\text{x}10^{-8} \\ 0 \\ 0 \end{Bmatrix}
\end{array}
$$

$\{F\} = [K\{d\}$

$$
\begin{Bmatrix} {F_{1y}}^{(e)} \\ {M_1}^{(e)} \\ {F_{2y}}^{(e)} \\ {M_2}^{(e)} \\ {F_{3y}}^{(e)} \\ {M_3}^{(e)} \end{Bmatrix} = \begin{Bmatrix} -1860.67 \\ -14809.7 \\ 3664.24 \\ -610.97 \\ -1803.38 \\ 14504.22 \end{Bmatrix}
$$

$\{F\} = [K\{d\} - \{F_o\}$

Where $\{F\}$ are the concentrated nodal forces and $\{F_o\}$ are called the equivalent nodal forces, now expressed in terms of global-coordinate components, which are of such magnitude that they yield the same displacements at the nodes as would the distributed load.

$$\{F_o\} = \begin{Bmatrix} WL/2 \\ WL^2/12 \\ WL/2 \\ -\frac{WL^2}{12} \\ \frac{WL}{2} \\ WL^2/12 \end{Bmatrix} = \begin{Bmatrix} 3667.5 \\ 611.25 \\ 3667.5 \\ -611.25 \\ 3667.5 \\ 611.25 \end{Bmatrix}$$

$$\{F\} = [K\{d\} - \{F_o\}$$

$$\begin{Bmatrix} F_{1y} \\ M_1 \\ F_{2y} \\ M_2 \\ F_{3y} \\ M_3 \end{Bmatrix} = \begin{Bmatrix} -1860.67 \\ -14809.7 \\ 3664.24 \\ -610.97 \\ -1803.38 \\ 14504.22 \end{Bmatrix} - \begin{Bmatrix} 3667.5 \\ 611.25 \\ 3667.5 \\ -611.25 \\ 3667.5 \\ 611.25 \end{Bmatrix}$$

$$\begin{Bmatrix} F_{1y} \\ M_1 \\ F_{2y} \\ M_2 \\ F_{3y} \\ M_3 \end{Bmatrix} = \begin{Bmatrix} -5528.17 \\ -15420.95 \\ -3.26 \\ 0.28 \\ 5470.88 \\ 13892.97 \end{Bmatrix}$$

Bending moment

$$M(s) = \frac{EI}{L^2}[B]\{q\}$$

$$M(s) = \frac{EI}{L^2}[(-6 + 12s)v1 + L(-4 + 6s)\theta 1 + (6 - 12s)v2 + L(-2 + 6s)\theta 2]$$

$$M(s) = 468x10^4[0 + (6 - 12s)0.0000326 + 16(-2 + 6s)(-6.37x10^{-8})]$$

M(s) = -1859s+924.95 N.m

Shear force

$$V_y = \frac{EI}{L^3}[12v1 + 6L\theta 1 - 12v2 + 6L\theta 2]$$

$$V_y = 468x10^4[0 - 12 * 0.0000326 - 6 * 16 * 6.37x10^{-8}]$$

V_y = -1859.44N

Bending stress $\sigma_b = \frac{Mc}{I}$

Substituting the values of $c = \frac{dc}{2}$ and $I = \frac{\pi(dc)^4}{64}$

$\sigma_b = \frac{32M}{\pi {d_c}^3}$

$$\sigma_b = \frac{32M}{\pi {d_o}^3 - {d_i}^3}$$

$\sigma_b = \frac{32*459.24KN/mm^2}{\pi*(37^3-18^3)}$ =104.42N/mm^2

Finite element modeling of any solid component consists of geometry generation, applying material Properties, meshing the component, defining the boundary constraints, and applying the proper load type. In this section developing the 3D of original and optimized crankshaft model and the FEA technique are described in details. Von mises stress, shear stress and life of original crankshaft and optimized crankshaft are done.

3.4.2.1 Procedure of static Analysis

First, I prepare a model of crankshaft in SOLID WORK software and save as IGES file format for Analysis of crankshaft in ANSYS WORKBENCH 16 import IGES model in ANSYS Workbench simulation module.

A. Applying material for crankshaft

Material type: - Forged steel (AISI 1045steel)

Density: - 7833 Kg/m^3

Young's modulus: - 221GPa

Poisson ratio: - 0.3

Yield stress 625 MP

Ultimate tensile strength 827MPa

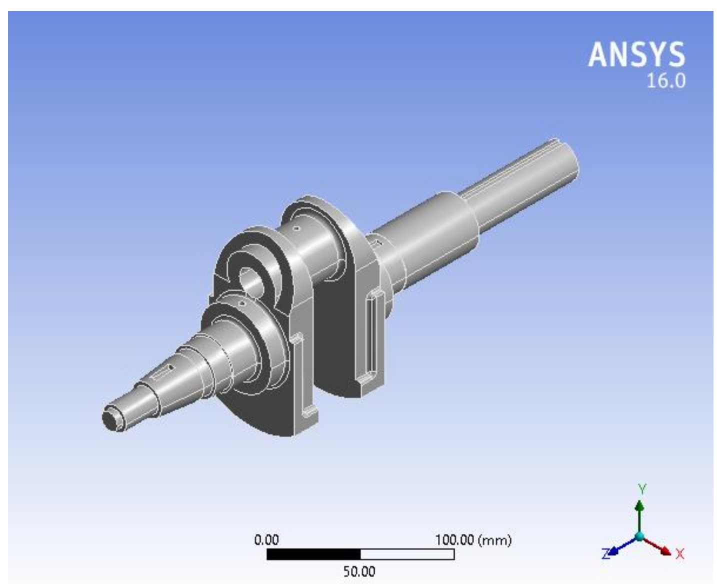


Figure 3.6: Single cylinder engine crankshaft model with SOLIDWORK

B. Meshing of Crankshaft:

The FE model of the crankshaft geometry is meshed with tetrahedral elements. Mesh refinement are done on the crank pin fillet and journal fillet, so that fine mesh is obtained on fillet areas, which are generally critical locations on crankshaft.

Tetrahedral shape of element is used for meshing the imported complex geometries to the ANSYSWORKBENCH software. Three dimensional model (3D) of crankshafts is performed in Solid work. After that, model is exported by *ANSYS* and profile is subdivided into nodes and elements. Collection of elements is called mesh and it is necessary to make mesh optimization to get more accuracy results. Mesh optimization is carried out until the *FEA* results and analytical solutions are close to each other. Meshing of Model: We discritized the solid model into small elements. Depending upon the requirement of the accuracy of results the fineness of meshing varies. This meshing varies used is 1.5mm, 2mm, 2.5mm and 3mm. Finer is the meshing more we are closer to the actual results and when mesh size increases maximum stress on the component become decreased.

Mesh statics:

Type of Element: Tetrahedral shape

Number of nodes: 244685

Number of elements: 165077

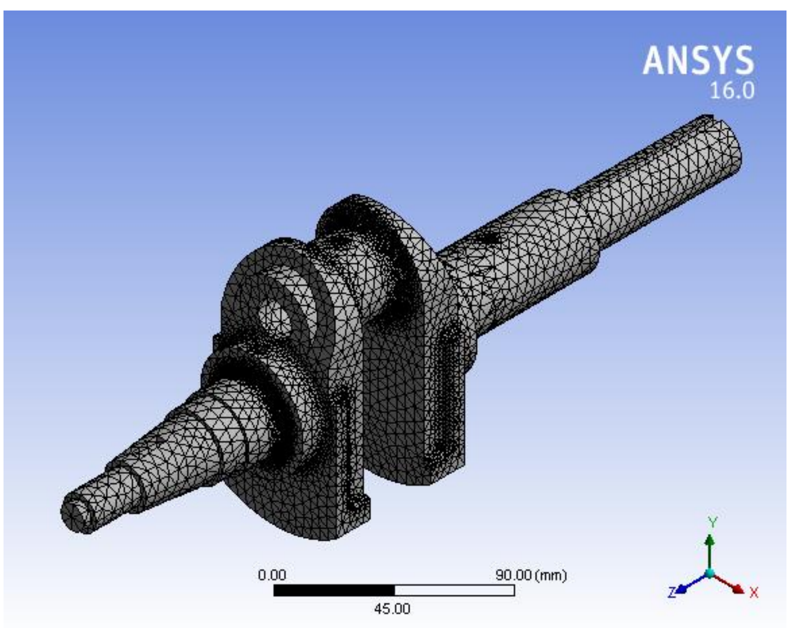


Figure 3.7: Single cylinder engine crankshaft meshing with triangular shape of elements

The most commonly used finite element shapes are triangular and quadrilateral for two dimensional (2-D) problems and tetrahedral for three dimensional (3-D) problems. The triangular and tetrahedral edge elements have the advantage of being able to model very complex geometries. This meshing size 1.5mm for original crankshaft and for optimized 2mm, 2.5mm and 3mm. Finer is the meshing more we are closer to the actual results and when mesh size increases maximum stress on the component become decreased.

C. Defining boundary condition and applied load for analysis:

Load applied on the component when the crank is at the position of maximum bending moment or is at the dead centre and also load data were taken from the calculated result. Boundary condition is based on under supporting condition of crankshaft [33].

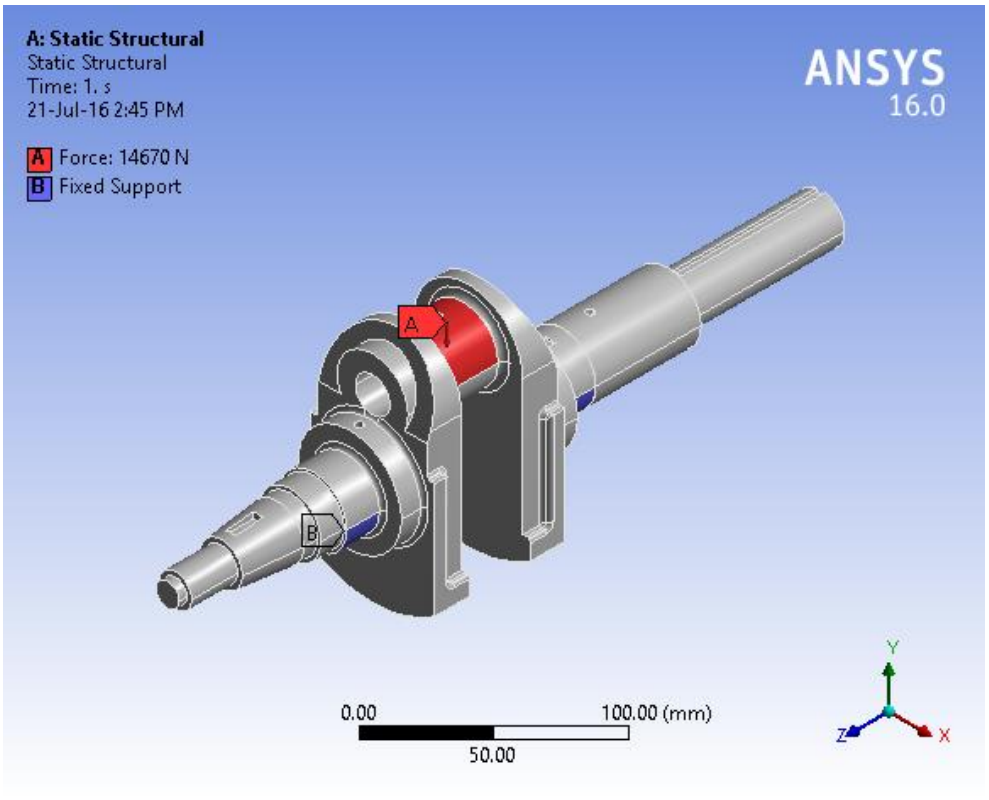


Figure 3.8: single cylinder engine crankshaft boundary conditions and input data

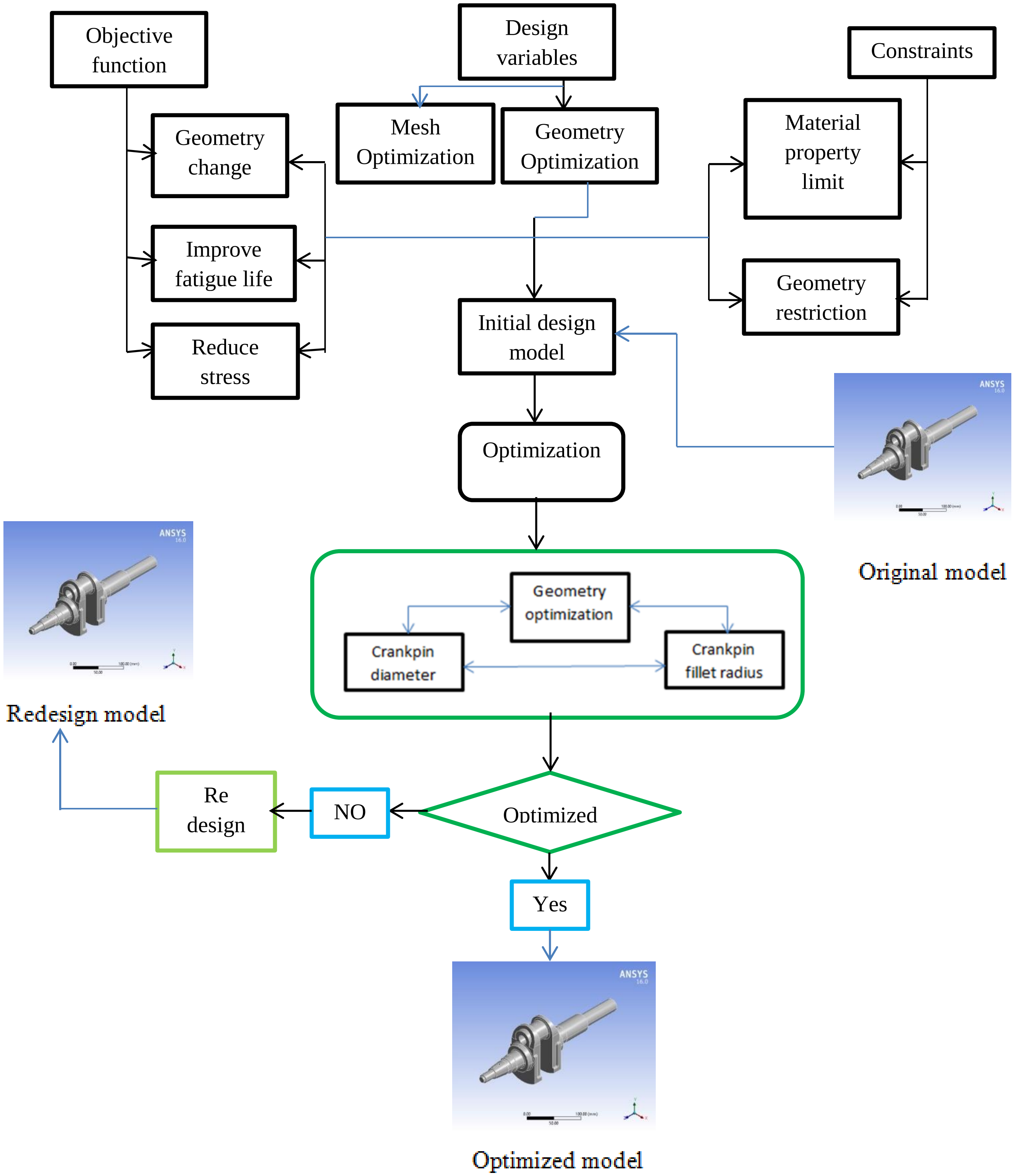


Figure 3.9 General flow chart of crankshaft optimization procedure.

CHAPTER FOUR

RESULTS AND DISCUSSIONS

4.1 Results

In this present work, the single cylinder engine crankshaft model was created by Solid work software. Then, the model created by solid work was imported to ANSYS software for analysis.

4.1.1 Results interms of fillet radius

4.1.1.1 Stress at different fillet radius

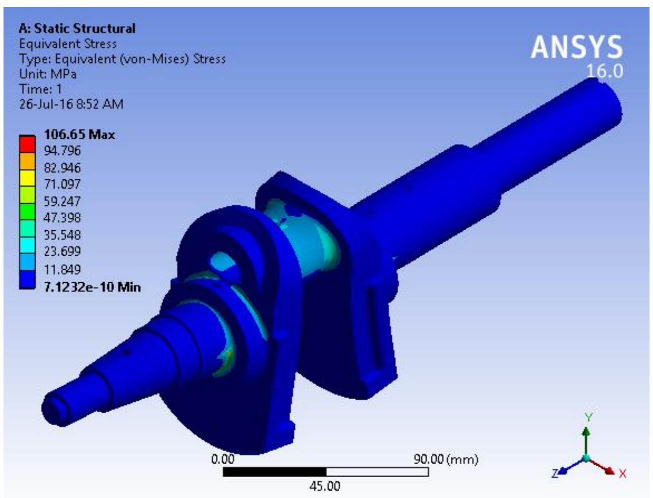


Figure 4.1a Von mises stress at 3 mm fillet radius

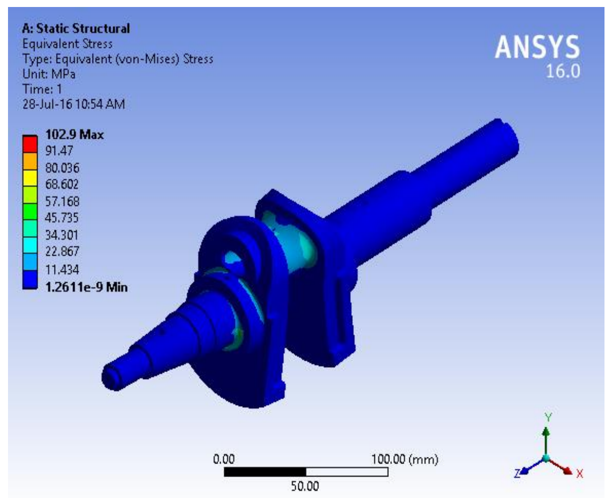


Figure 4.1b Von mises stress at 3.5 mm fillet radius

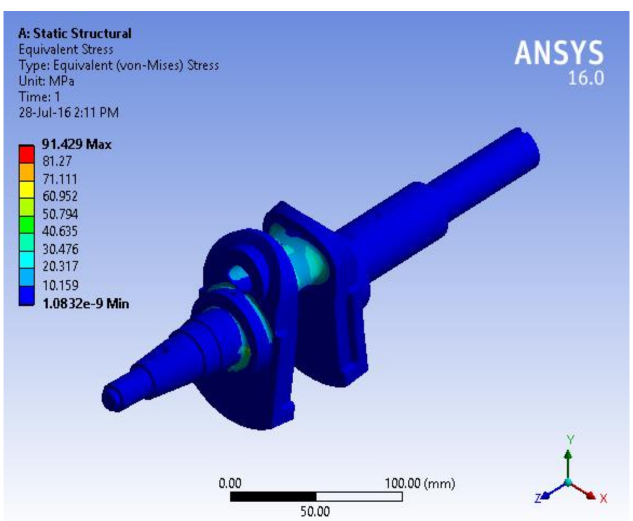


Figure 4.1c Von mises stress at 4 mm fillet radius

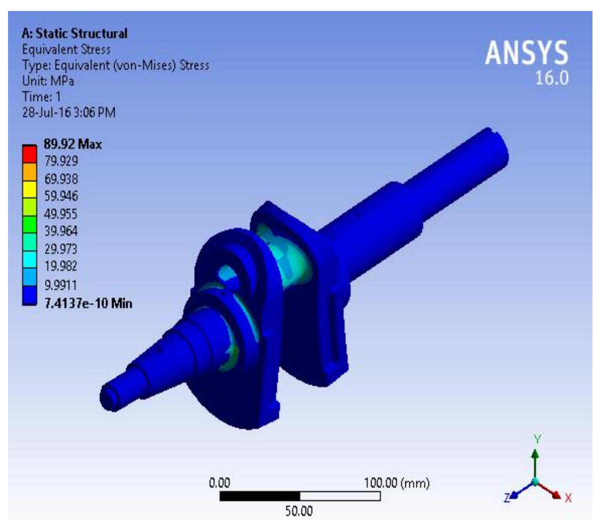


Figure 4.1d Von mises stress at 4.5 mm fillet radius

4.1.1.2 Shear stress at different fillet radius

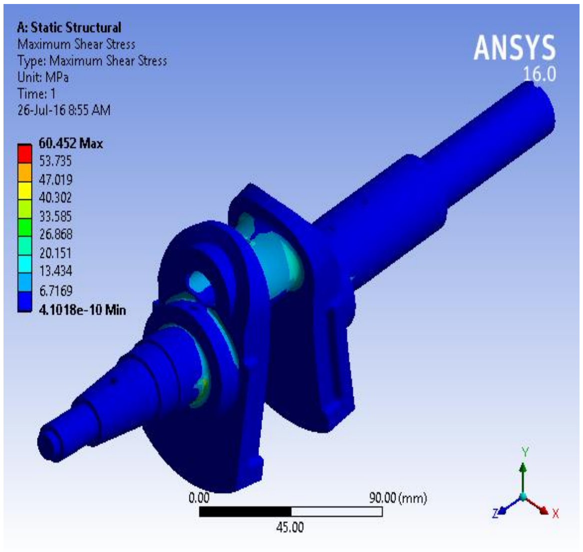


Figure 4.2a Shear stress at 3 mm fillet radius

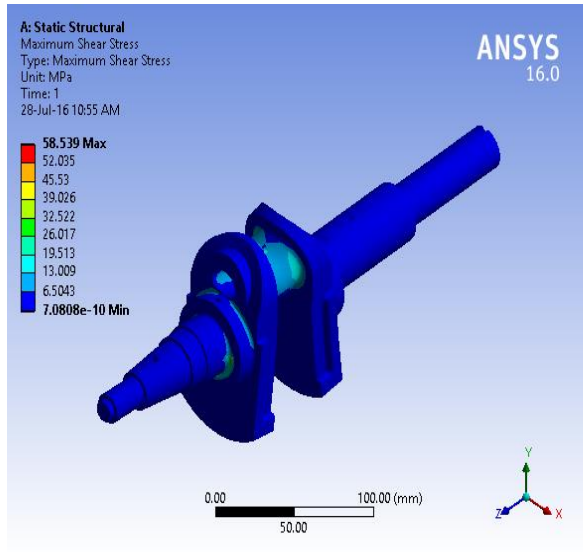


Figure 4.2b Shear stress at 3.5 mm fillet radius

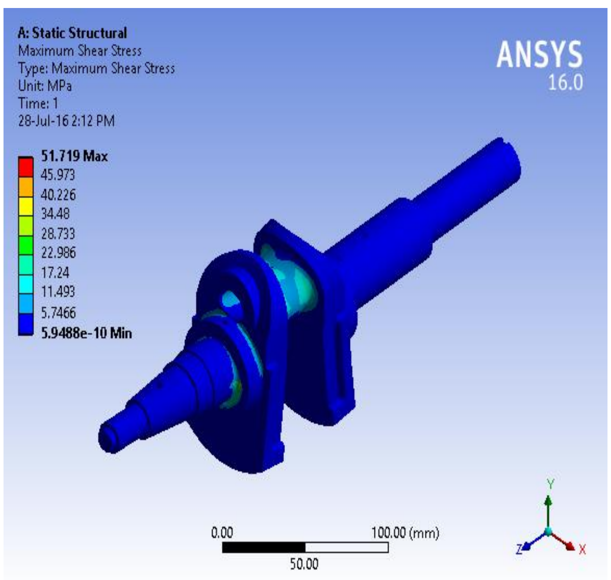


Figure 4.2c Shear stress at 4 mm fillet radius

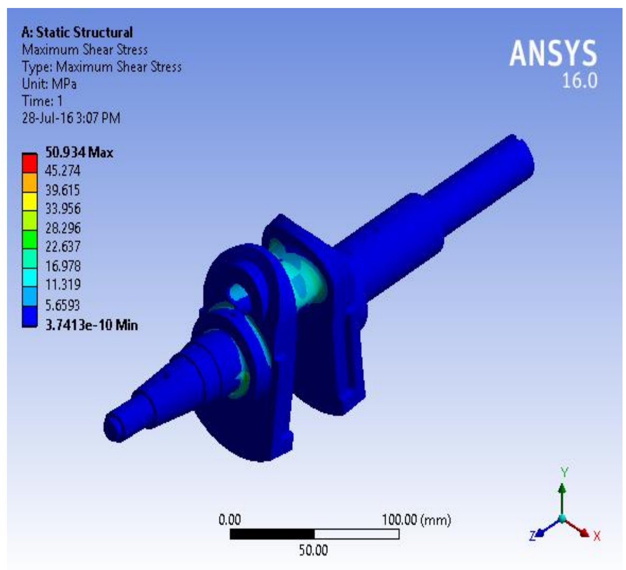


Figure 4.2d Shear stress at 4.5 mm fillet radius

4.1.1.3 Life at different fillet radius

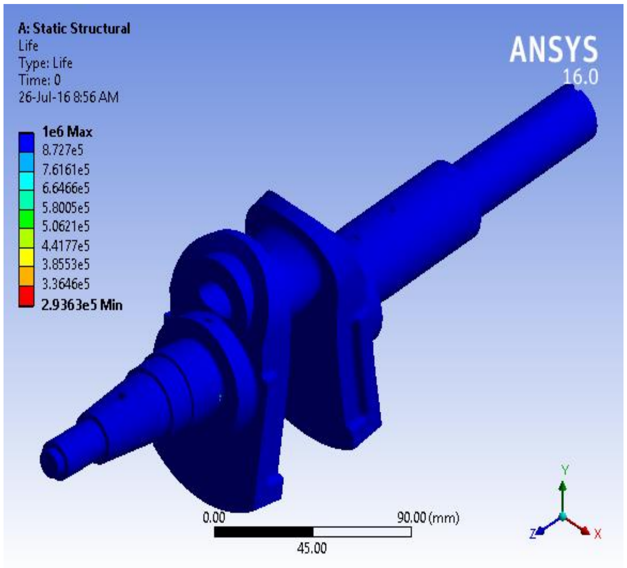


Figure 4.3a Life at 3 mm fillet radius

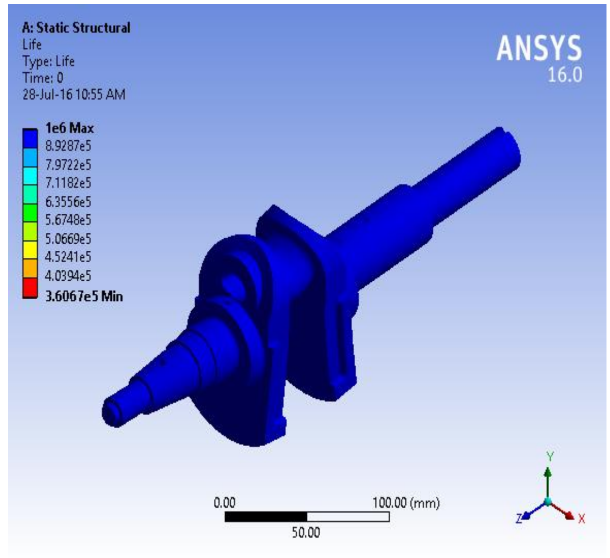


Figure 4.3b Life at 3.5 mm fillet radius

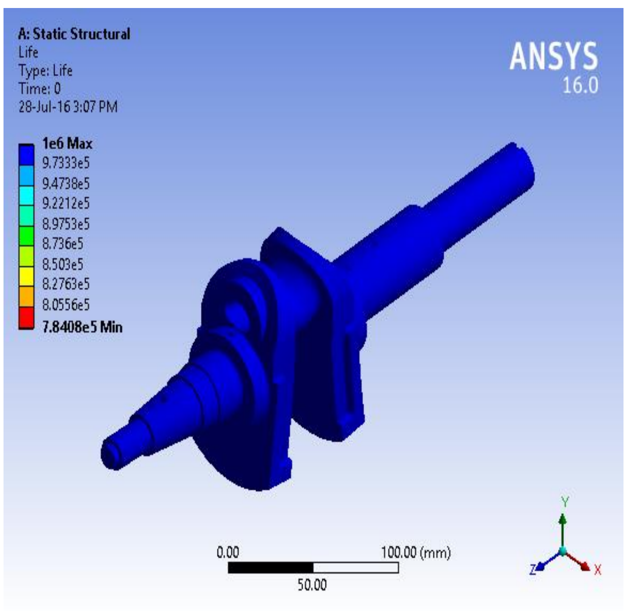


Figure 4.3c Life at 4mm fillet radius

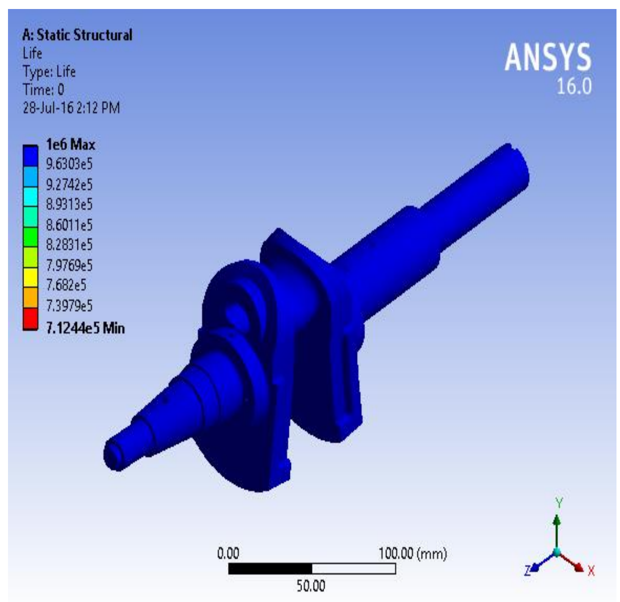


Figure 4.3d Life at 4.5 mm fillet radius

4.5mm is optimized one due to its stress is minimum and its life is maximum rather than the others. So it can be replaced the original one.

4.1.2 Results interms of crankpin diameter

4.1.2.1 Stress at different crankpin diameter

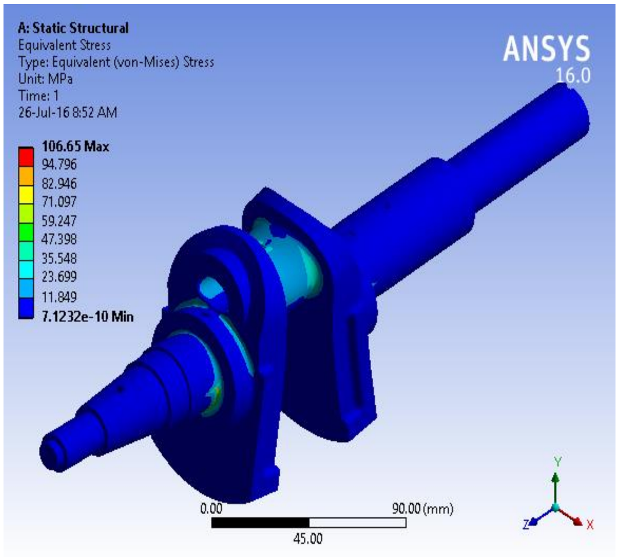


Figure 4.4aVon mises stress at 37mm d_c

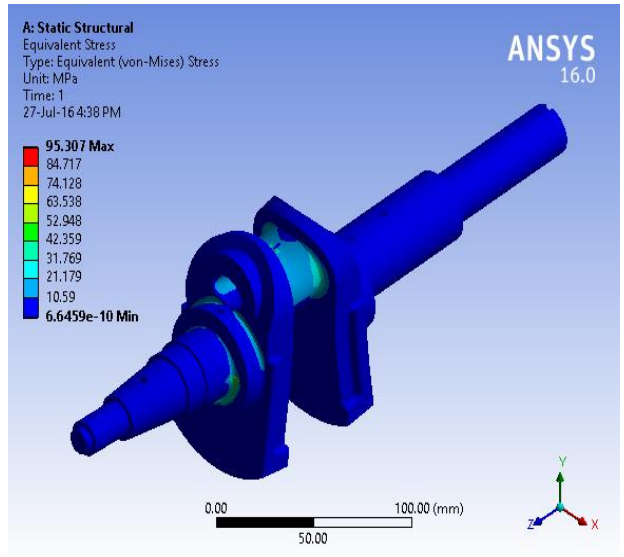


Figure 4.4b Von mises stress at 38 mm d_c

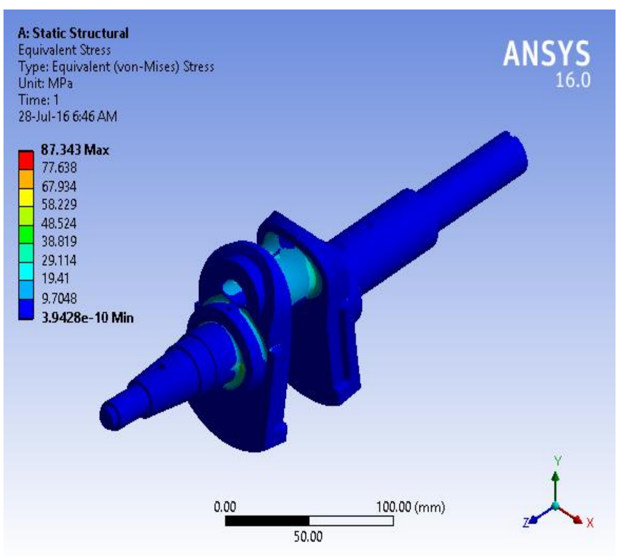


Figure 4.4c Von mises stress at 39 mm d_c

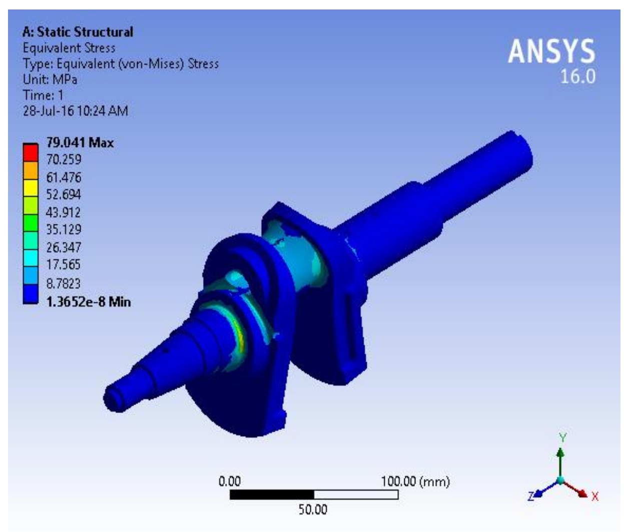


Figure 4.4d Von mises stress at 40 mm d_c

4.1.2.2 Shear stress at different crankpin diameter

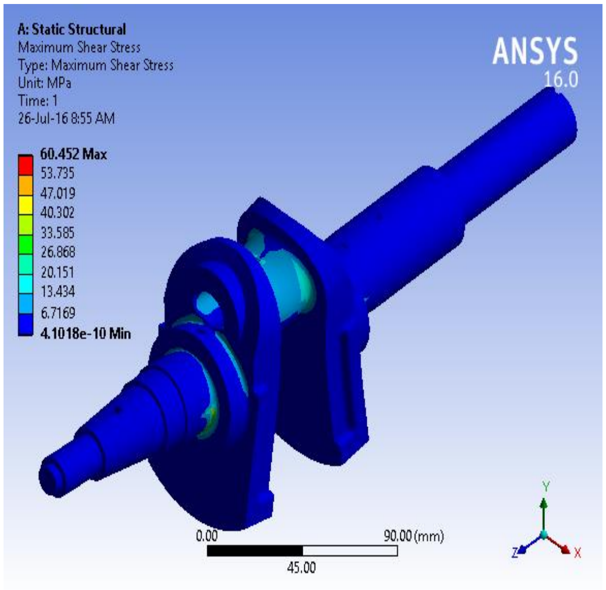


Figure 4.5a Shear stress at 37 mm d_c

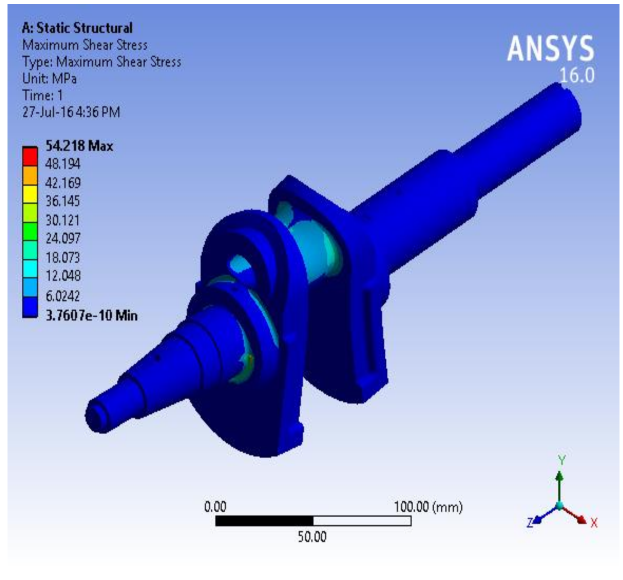


Figure 4.5b Shear stress at 38 mm d_c

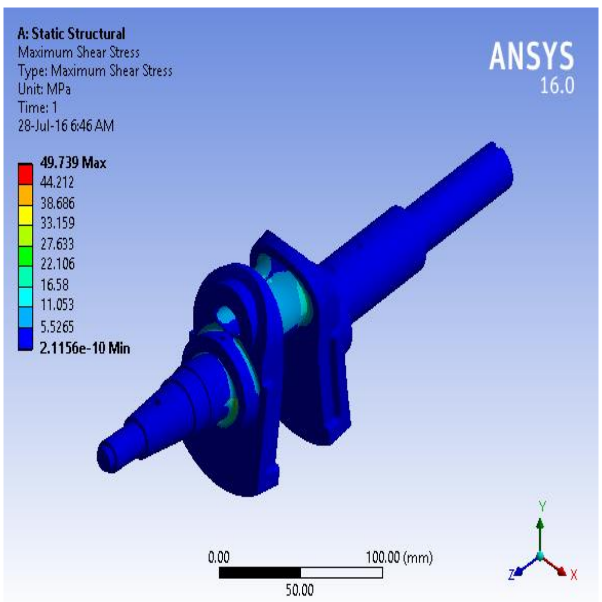


Figure 4.5c Shear stress at 39 mm d_c

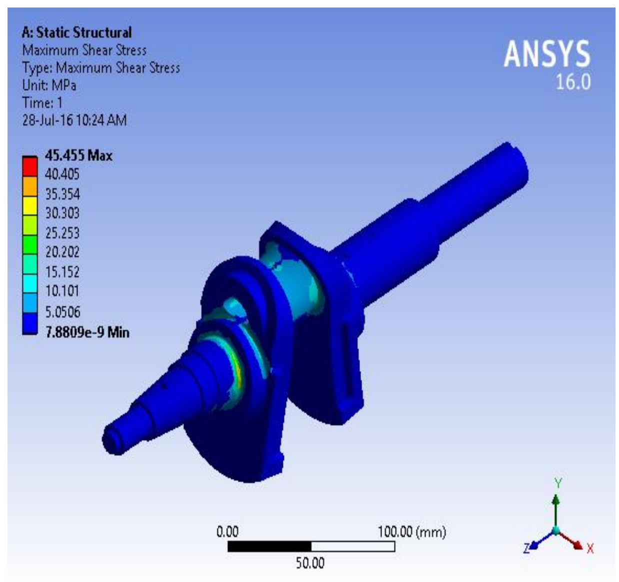


Figure 4.5d Shear stress at 40 mm d_c

4.1.2.3 Life at different crankpin diameter

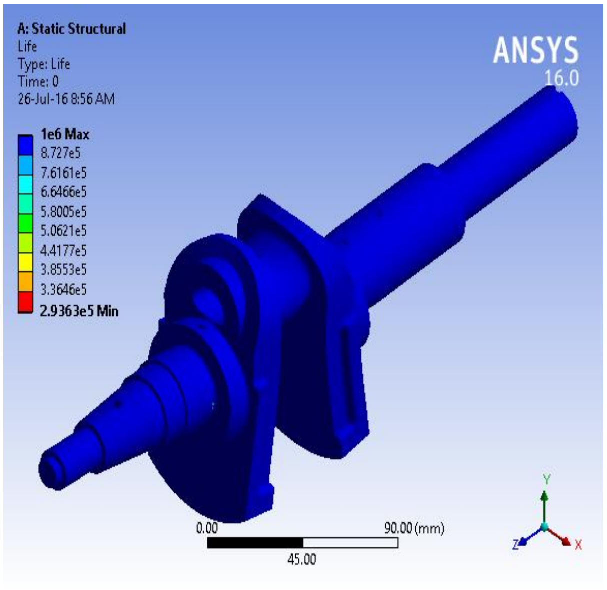


Figure 4.6 a Life at 37 mm d_c

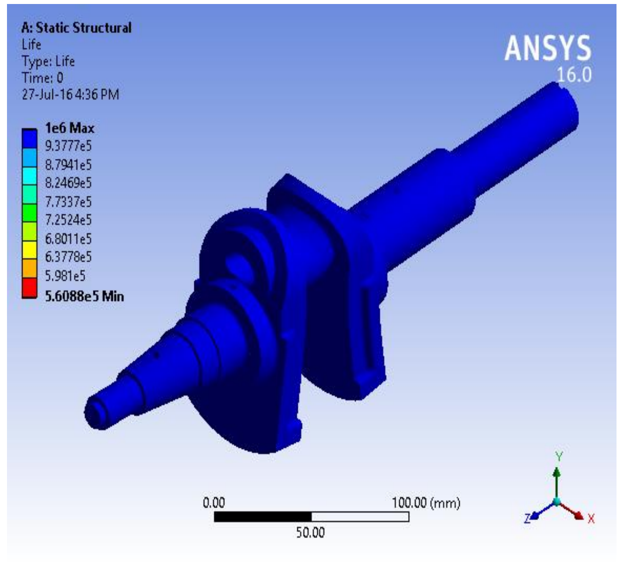


Figure 4.6 b Life at 38mm d_c

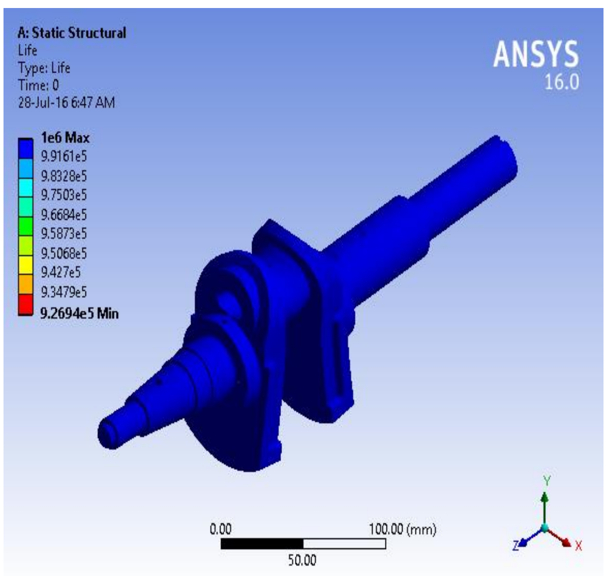


Figure 4.6 c Life at 39 mm d_c

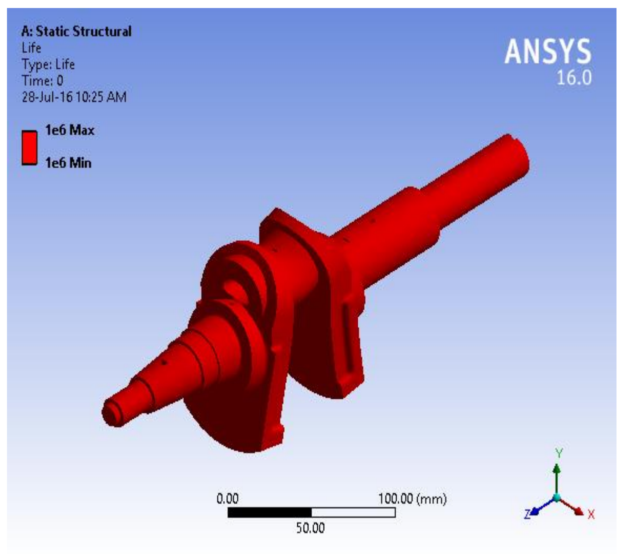


Figure 4.6 d Life at 40 mm d_c

According to FE analysis, shown in Figure above in both parameters red location shows maximum stress and blue locations have low stresses during service life and have the potential for material removal and weight reduction

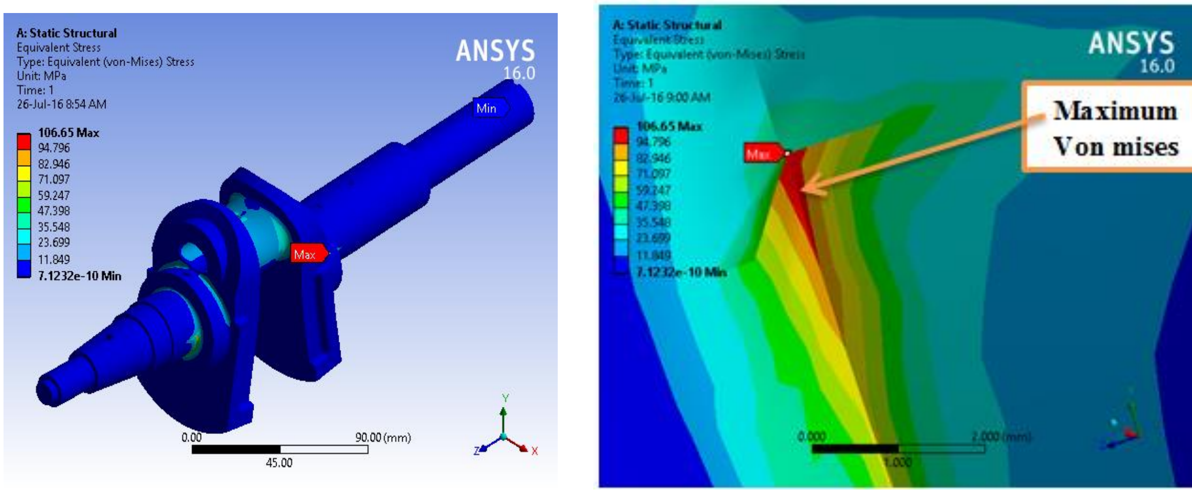


Figure 4.7 Maximum von mises stress of original cranksahaft

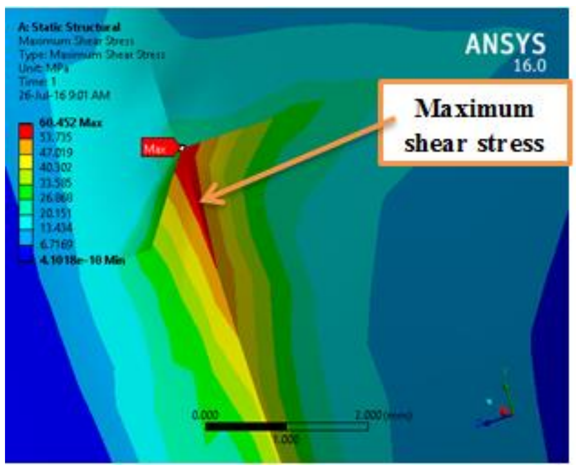


Figure 4.8 Maximum shear stress of original cranksahaft

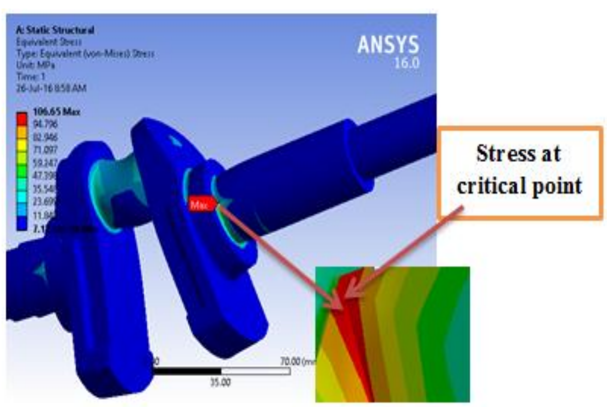


Figure 4.9: FE model showing stresses at the critical fillet

4.2 DISCUSSIONS

This paper focuses on the single cylinder engine crankshaft for improving fatigue life. Single cylinder engine crankshaft model in ANSYS predicts that the maximum value of the equivalent alternating stress decreases, and fatigue life increases.

Alternating Stress (MPa)	Cycles
3999	10
2827	20
1896	50
1413	100
1069	200
441	2000
262	10000
214	20000
138	1.e+005
114	2.e+005
86.2	1.e+006

Table 4.1 Alternating stress vs. cycles

Fatigue Sensitivity Figure 4.10 shows fatigue sensitivity curve and how the fatigue results change as a function of the loading at the critical location on the model.

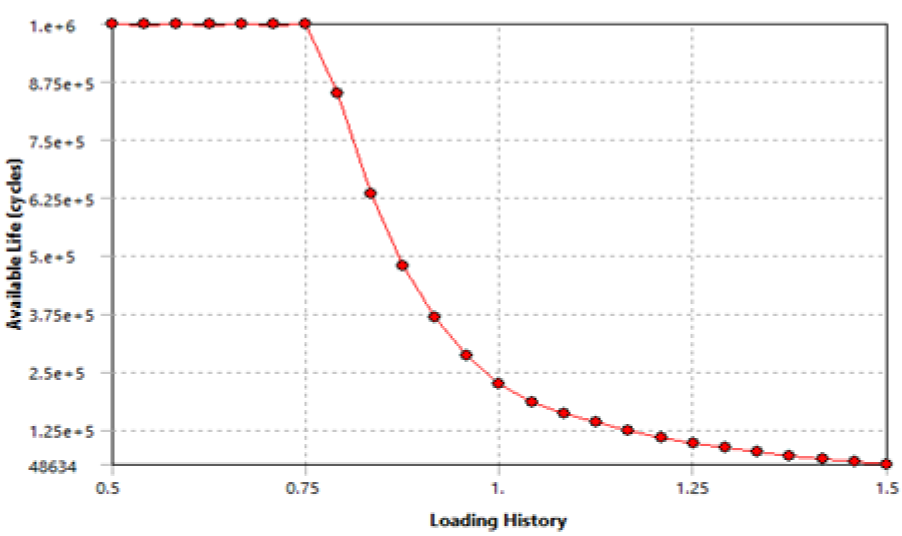


Figure 4.10: Fatigue Sensitivity Curve

Figure shows fatigue sensitivity curve and between 1.25 and 1.5 is dangerous region as load going to increase material get failure at critical location

Table 4.2 Number of Cycles to Failure related to crankpin fillet radius

Fillet radius	Von misses stresses(MPa)		Shear stresses(MPa)		Number of Cycles to Failure (N)
	Theoretical solution	Ansys solution	Theoretical solution	Ansys solution	Ansys solution
3	98.27 N/mm^2	106.65 N/mm^2	56.52 N/mm^2	60.45 N/mm^2	$2.9363x10^5$
3.5	95.22 N/mm^2	102.9 N/mm^2	55.29 N/mm^2	58.53 N/mm^2	$3.6067x10^5$
4	92.76 N/mm^2	91.42 N/mm^2	53.54 N/mm^2	51.71 N/mm^2	$7.1244x10^5$
4.5	89.93 N/mm^2	89.92 N/mm^2	51.93 N/mm^2	50.93N/mm^2	$7.8408x10^5$

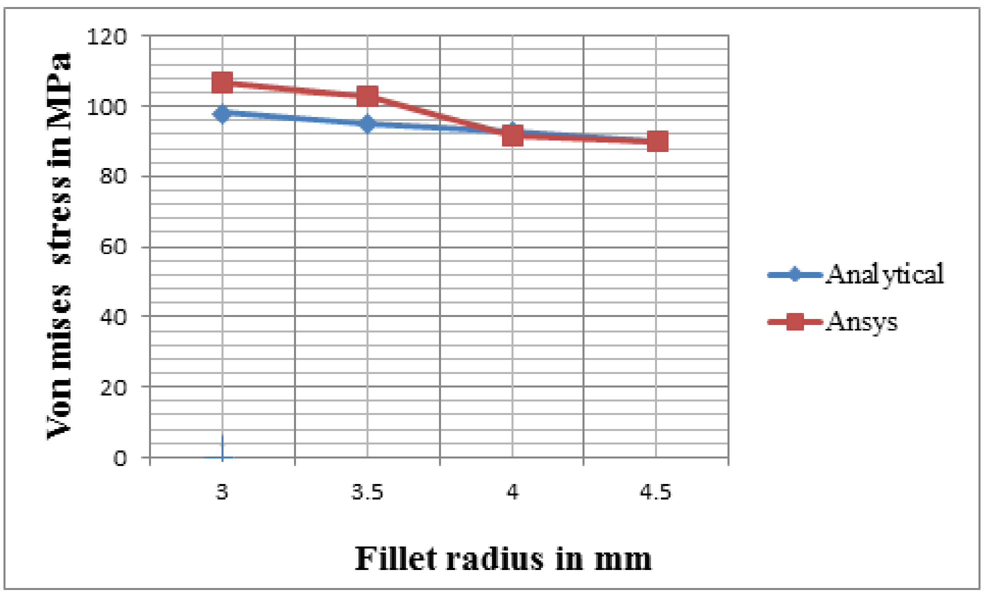


Figure 4.11 Graphical representation of fillet radius vs. von mises stress

Fillet radius	**Von misses stresses(MPa)**		**Error %**
	Theoretical solution	Ansys solution	
3	98.27 N/mm^2	106.65 N/mm^2	7.85
3.5	95.22 N/mm^2	102.9 N/mm^2	7.46
4	92.76 N/mm^2	91.42 N/mm^2	1.44
4.5	89.93 N/mm^2	89.92 N/mm^2	0.01

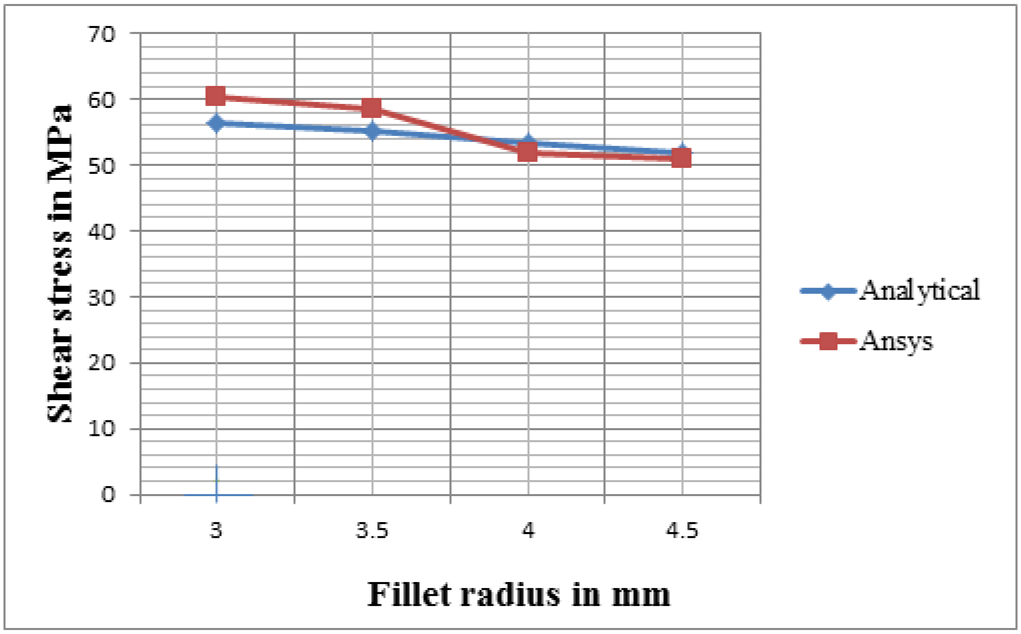


Figure 4.12 Graphical representation of fillet radius vs. shear stress

Fillet radius	**Shear stresses**(**MPa**)		**Error %**
	Theoretical solution	Ansys solution	
3	56.52 N/mm^2	60.45 N/mm^2	6.50
3.5	55.29 N/mm^2	58.53 N/mm^2	5.53
4	53.45 N/mm^2	51.71 N/mm^2	3.25
4.5	51.93 N/mm^2	50.93 N/mm^2	1.96

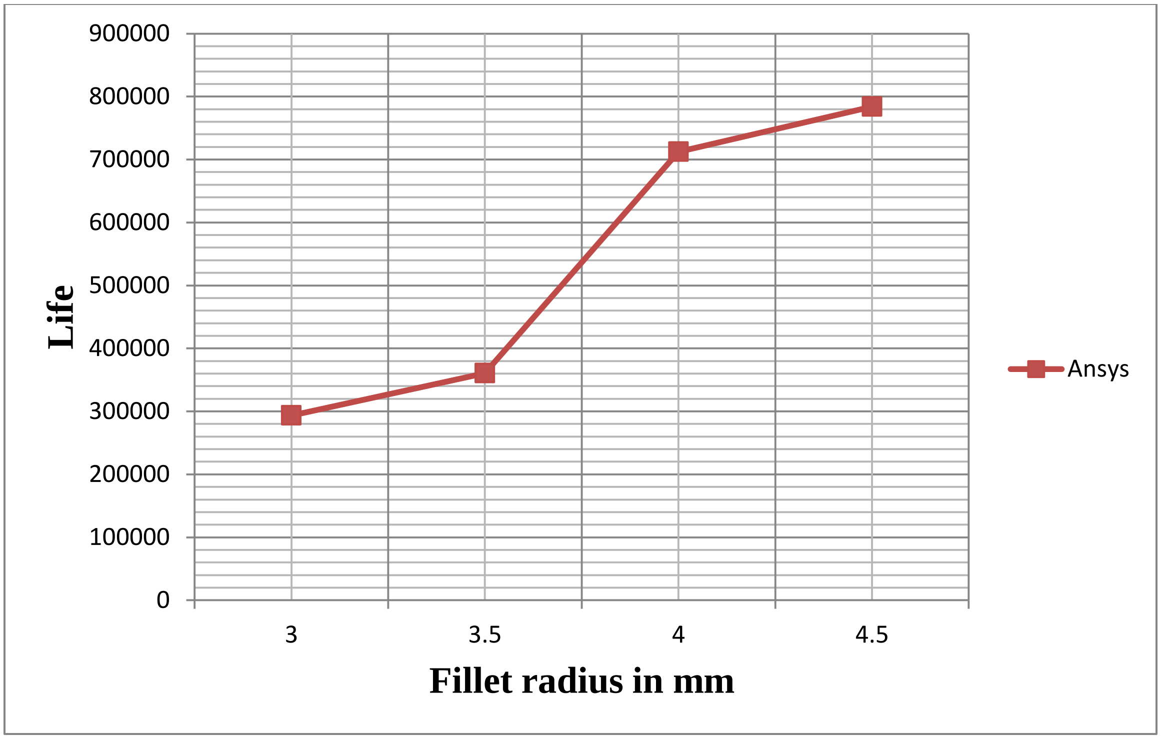


Figure 4.13 Graphical representation of fillet radius vs. Life

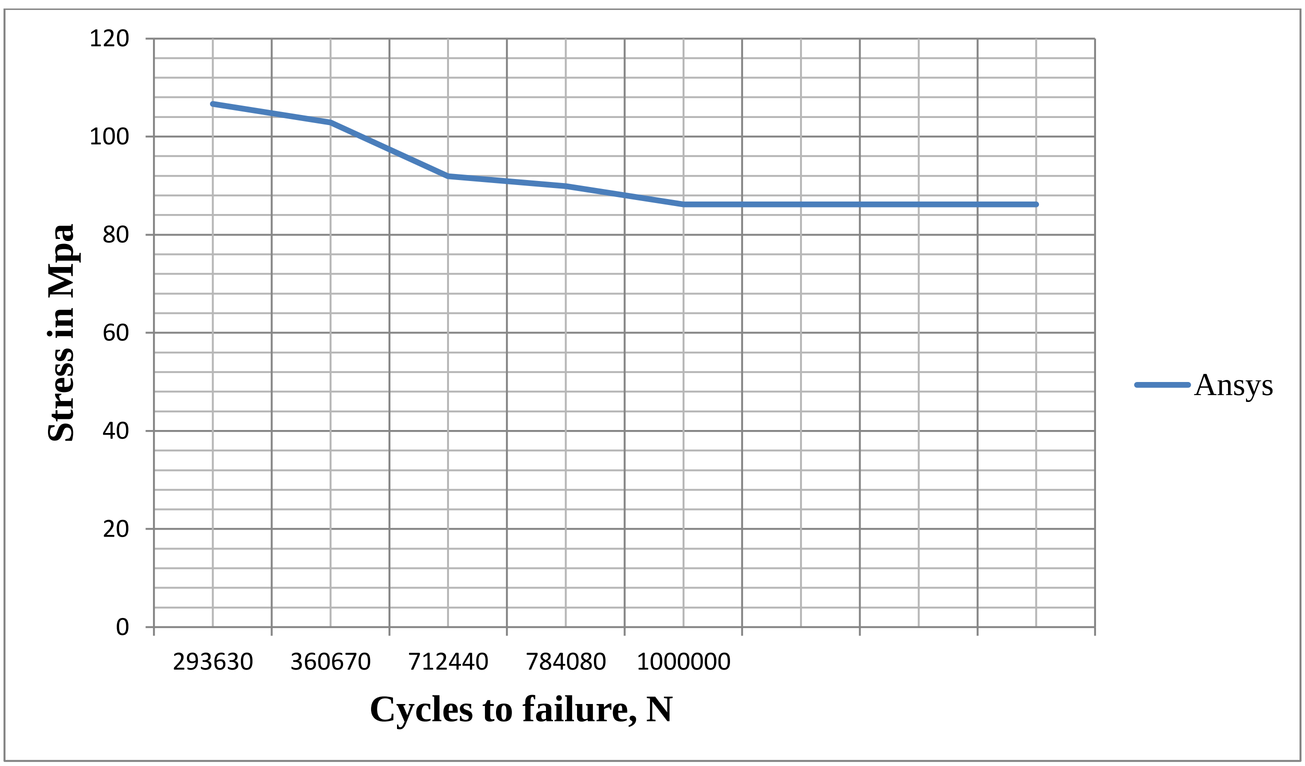


Figure 4.14 Graphical representations of S-N

Table 4.3 Number of Cycles Failure related to diameter of crankpin (d_c)

Crankpin diameter	Von misses stresses(MPa)		Shear stresses(MPa)		Number of Cycles to Failure
	Theoretical solution	Ansys solution	Theoretical solution	Ansys solution	Ansys solution
37	109.69N/mm^2	106.65 N/mm^2	55.69N/mm^2	60.45 N/mm^2	2.9363x10^5
38	100.25N/mm^2	95.22 N/mm^2	50.91N/mm^2	54.21N/mm^2	5.6088x10^5
39	91.92N/mm^2	87.34 N/mm^2	46.47N/mm^2	49.73N/mm^2	9.2694x10^5
40	84.52N/mm^2	79.04 N/mm^2	42.92N/mm^2	45.45N/mm^2	1x10^6

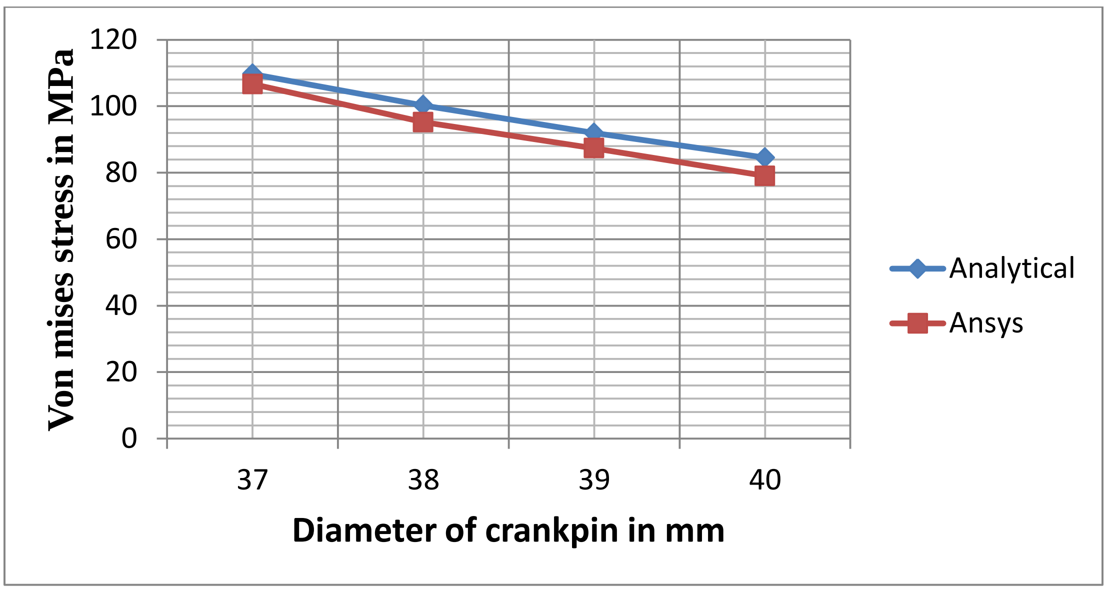


Figure 4.15 Graphical representation of crankpin diameter vs. von mises stress

Crankpin diameter	**Von misses stresses(MPa)**		**Error %**
	Theoretical solution	Ansys solution	
37	109.69N/mm2	106.65 N/mm2	2.77
38	100.25N/mm2	95.22 N/mm2	4.93
39	91.92N/mm2	87.34 N/mm2	4.98
40	84.52N/mm2	79.04 N/mm2	6.48

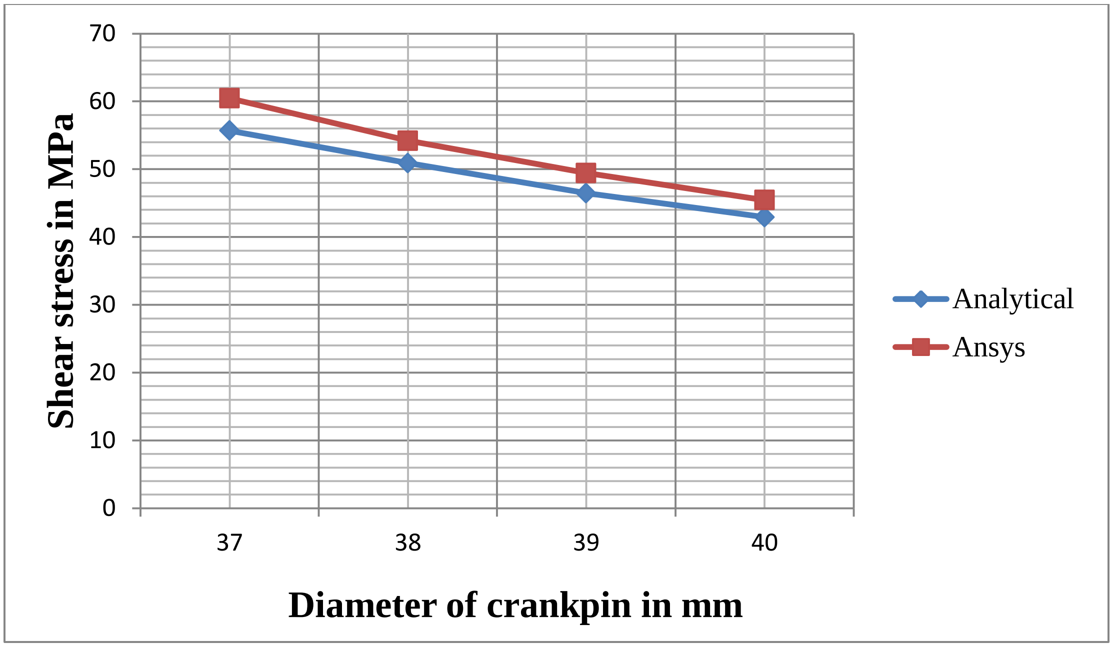


Figure 4.16 Graphical representation of crankpin diameter vs. shear stress

Crankpin diameter	**Shear stresses(MPa)**		**Error %**
	Theoretical solution	Ansys solution	
37	55.69N/mm^2	63.4 N/mm^2	7.87
38	50.91N/mm^2	54.21N/mm^2	6.08
39	46.47N/mm^2	49.73N/mm^2	6.55
40	42.92N/mm^2	45.45N/mm^2	6.51

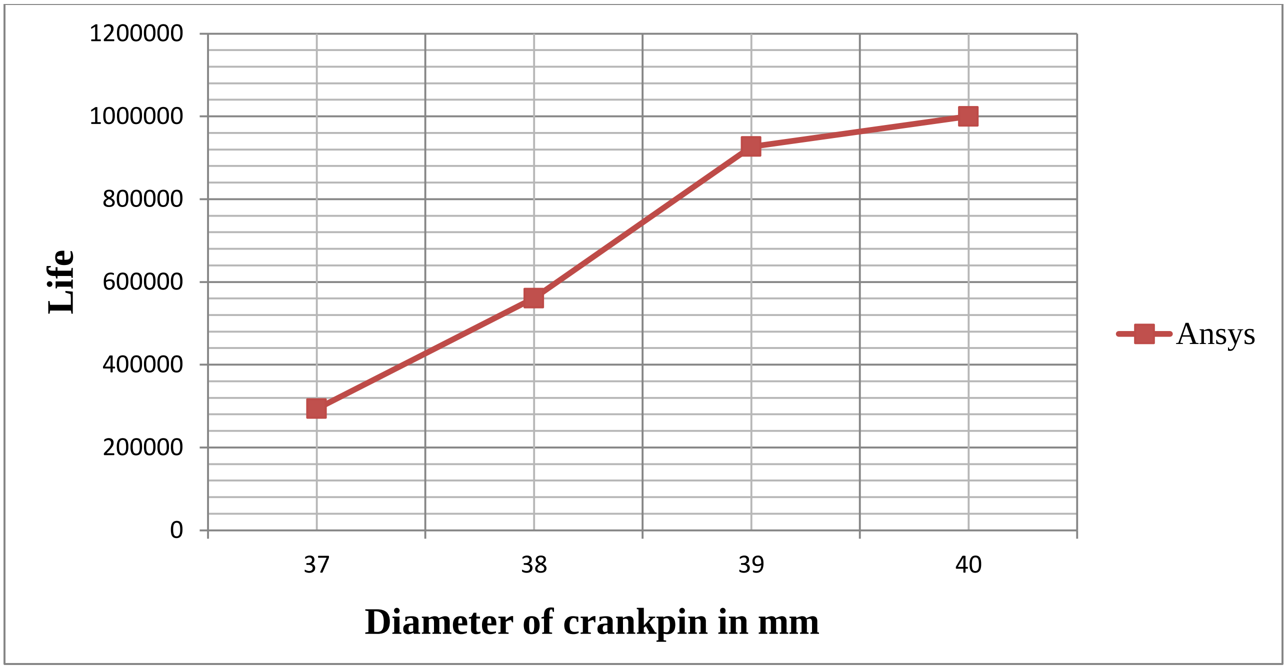


Figure 4.17 Graphical representation of crankpin diameter vs. Life

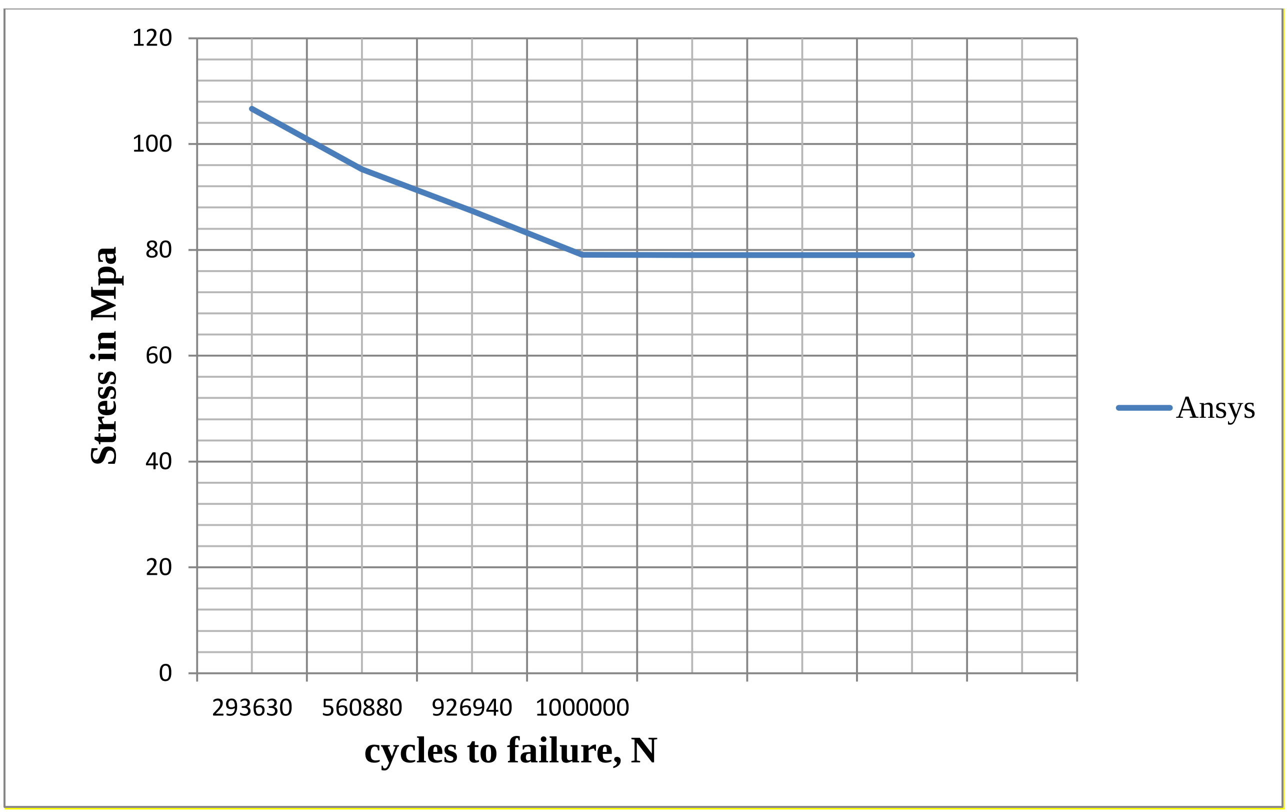
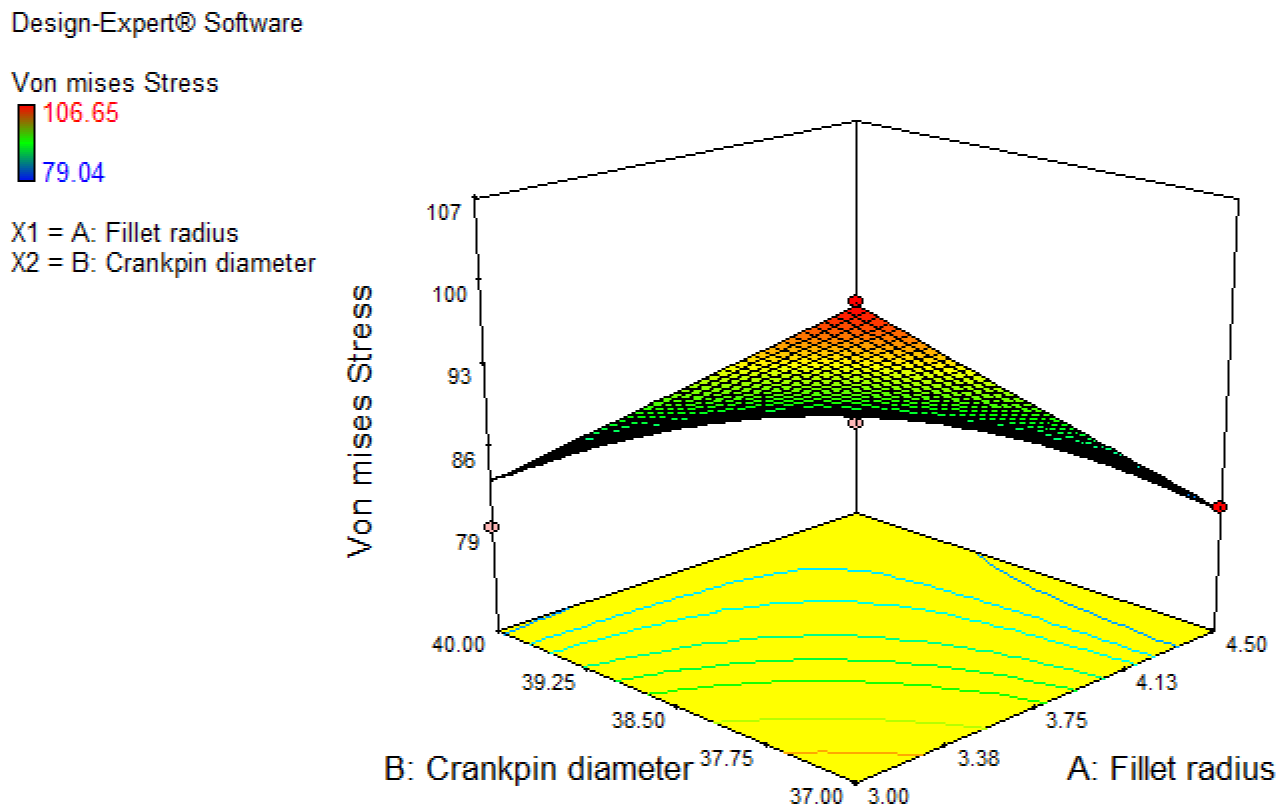


Figure 4.18 Graphical representations of S-N

Finding optimum design points for Von mises stress, shear stress and number of cycles to failure

Table 4.4 input data to obtain optimum points using Dx7 design expert

Constraints name	Goal	Lower limit	Upper limit
Fillet radius (mm)	is in range	3	4.5
Crankpin diameter(mm)	is in range	37	40
Von mises Stress	Minimize	79.04	106.65
Shear stress	Minimize	45.45	60.45
Cycles to failure (N)	Maximize	293630	1000000



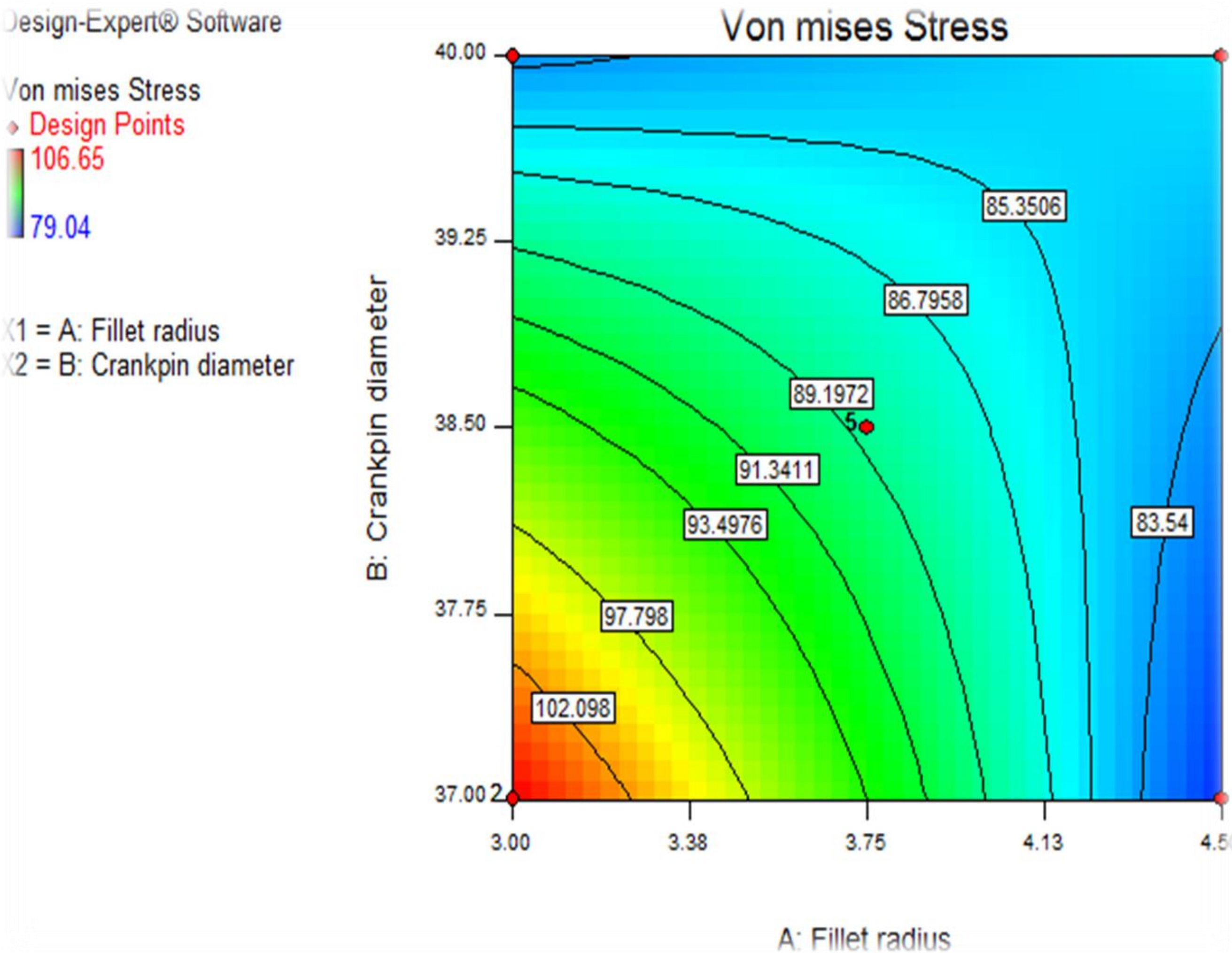
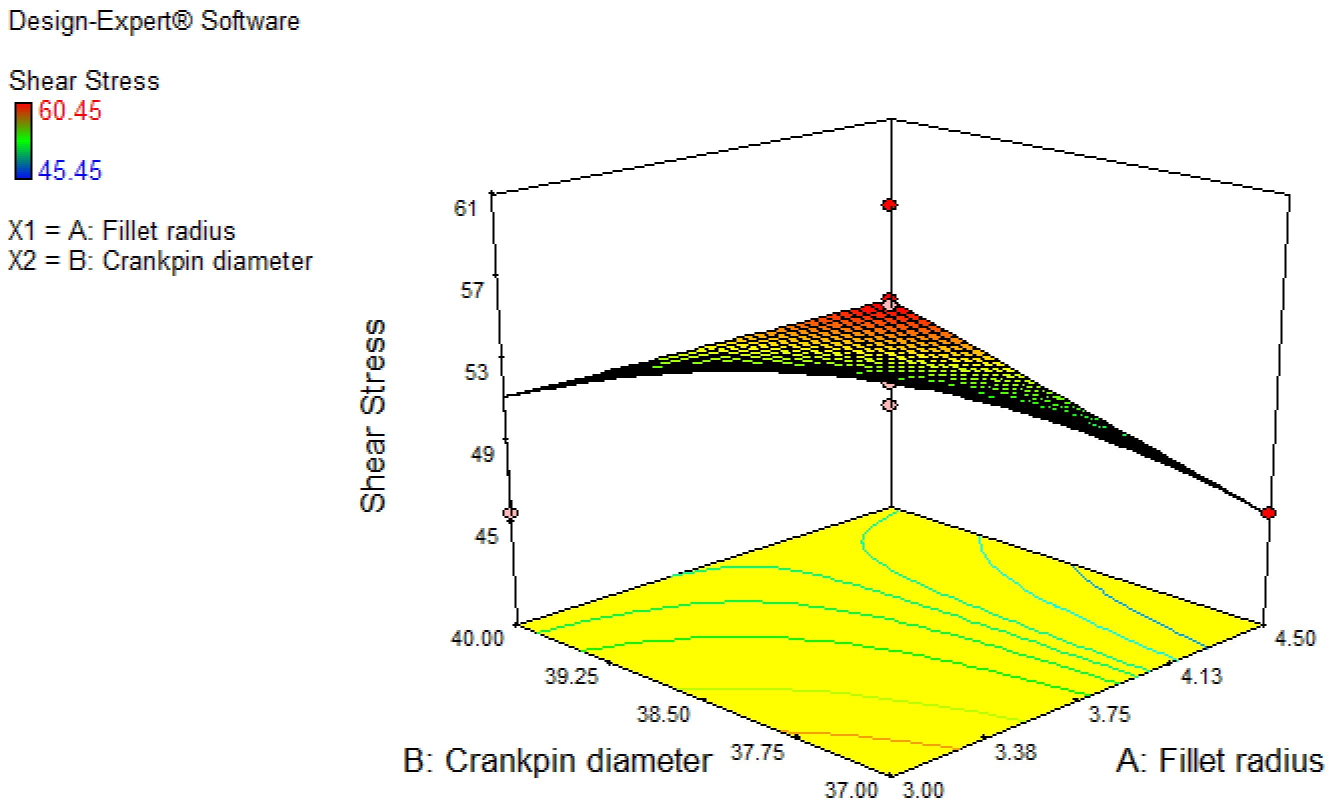


Figure 4.19 Von mises stress at different design points



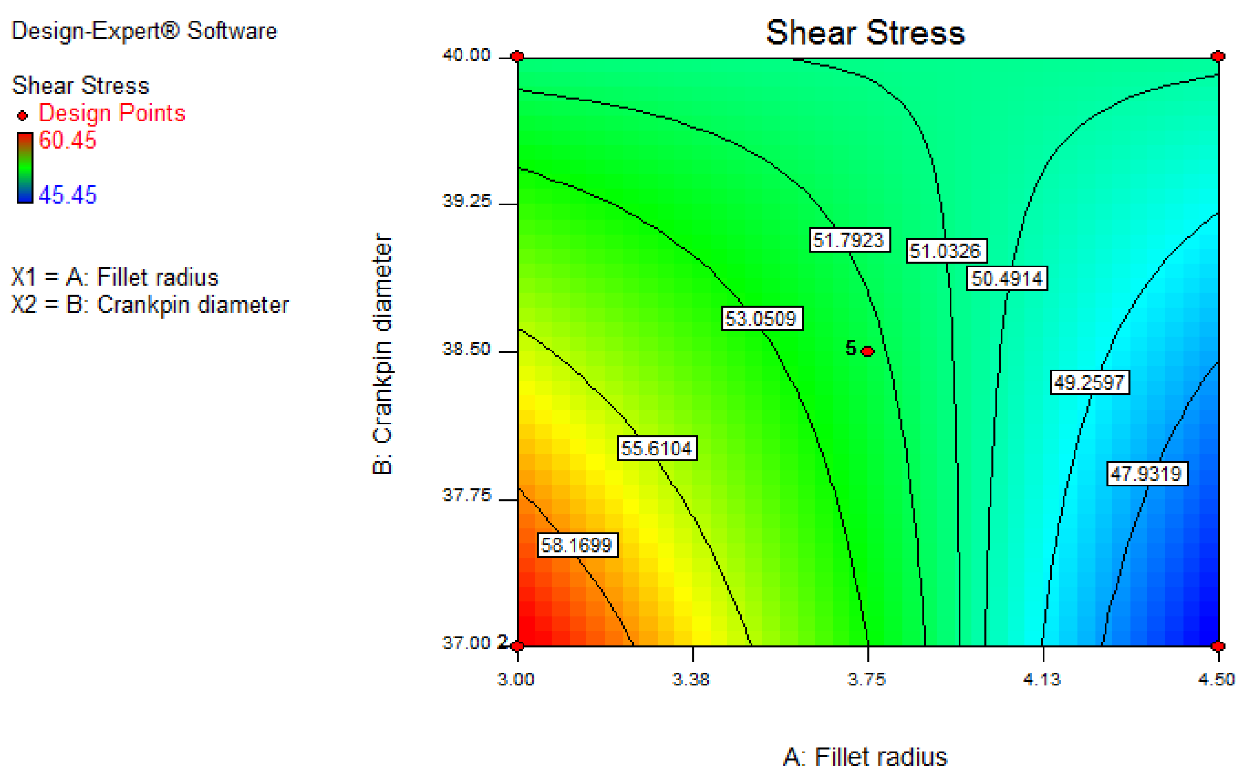
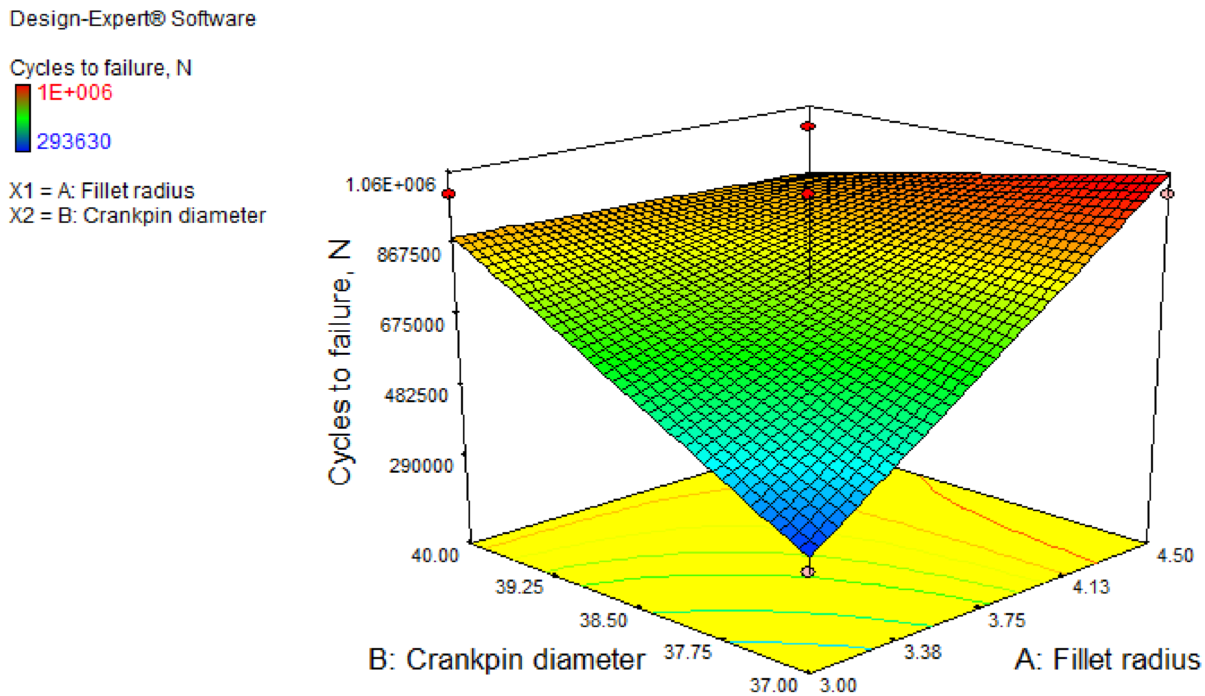


Figure 4.20 Shear stress at different design points



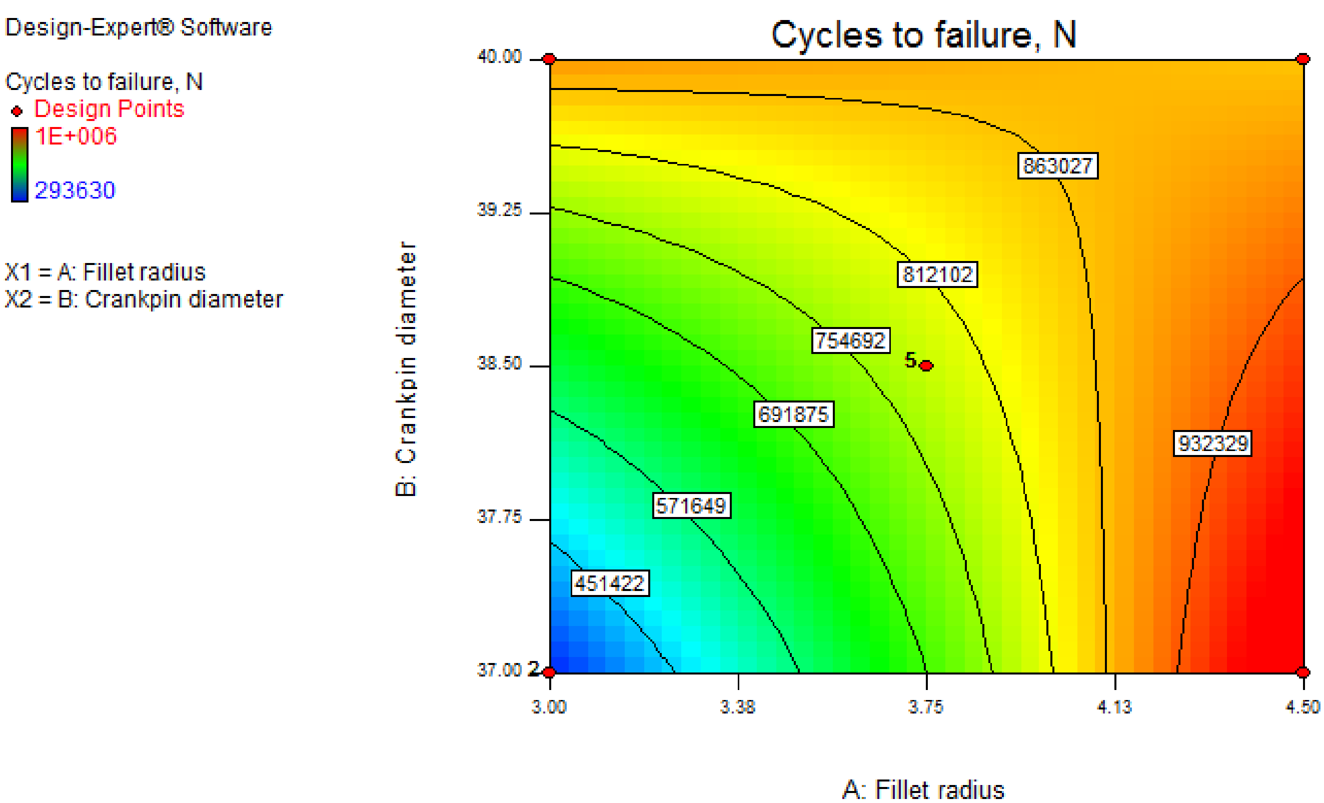


Figure 4.21 Number of cycles to failure at different design points

CHAPTER FIVE

CONCLUSION, RECOMMENDATION, AND FUTURE WORK

5.1. CONCLUSION

In this paper, the crankshaft model was created by SOLIDWORK software. Then, the model created by SOLIDWORK was imported to ANSYS software. The analysis of the crank shaft will be done using geometry optimization like different crankpin fillet radius and crankpin diameter. Finite element analysis using ANSYS and Analytical methods are done. The maximum stress appears at the fillet areas between the crankshaft journal and crank web. The edge of main journal is maximum stress area in the crankshaft. The FE model of the crankshaft geometry is meshed with tetrahedral elements. Mesh refinement are done on the crank pin fillet and journal fillet, so that fine mesh is obtained on fillet areas, which are generally critical locations on crankshaft. The failure in the crankshaft initiated at the fillet region of the journal, and fatigue is the dominant mechanism of failure. The comparison results of all different crankpin fillet radius and crankpin diameter will show the effect of stresses on crankshaft and this will help to select optimized one.

Geometry optimization resulted in 15% stress reduction of and life is optimized 62.55% crankshaft which was achieved by changing crankpin fillet radius and 25.88% stress reduction of and life is optimized 70.63% of crankpin diameter change. As the stress of the crankshaft is decreased this will increase fatigue life of the crankshaft. The crankpin fillet radius and crankpin diameter increases then Von misses stress and shear stresses are decreases as well as number of cycles to failure in increases. Analytical stress and FEA results showed close agreement and Fatigue life of modified crankshaft is improved rather than original and the optimized one can be replaced the original one.

5.2. RECOMMENDATION

From the result found from analytical and ANSYS result, the best way of analyzing fatigue life is in ANSYS result. In this paper better fatigue life of crankshaft is done due to geometry size optimization. We can see that as the stress decreases the life of the crankshaft increase. In this manner there are some recommendations given for the crankshaft. Predicting and preventive maintenance will help to decrease the crack initiation and crack propagation and minimize the cost of maintenance of crankshaft change due to damage. Additionally by using better strength material property the fatigue life of the crankshaft will be improved.

5.3. FUTURE WORK

By considering the aspects presented in this study more work can also be done in the same area. The considerations for future work are as follows,

- In Future parametric optimization can be done by changing other parameters (changing the main bearing radius, type of material etc.)
- According to FE analysis, blue locations shown in Figure have low stresses during service life and have the potential for material removal and weight reduction.
- Crankshaft has to be dynamically balanced, optimization process fulfill by changing the dimensions, geometry and removing or adding material in the crank web.

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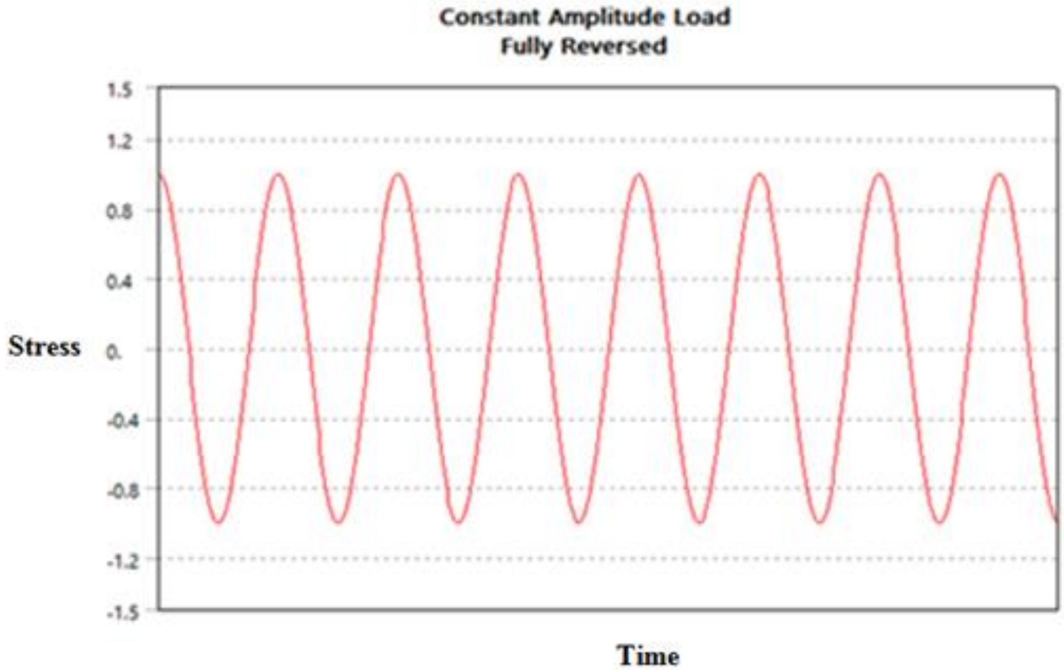
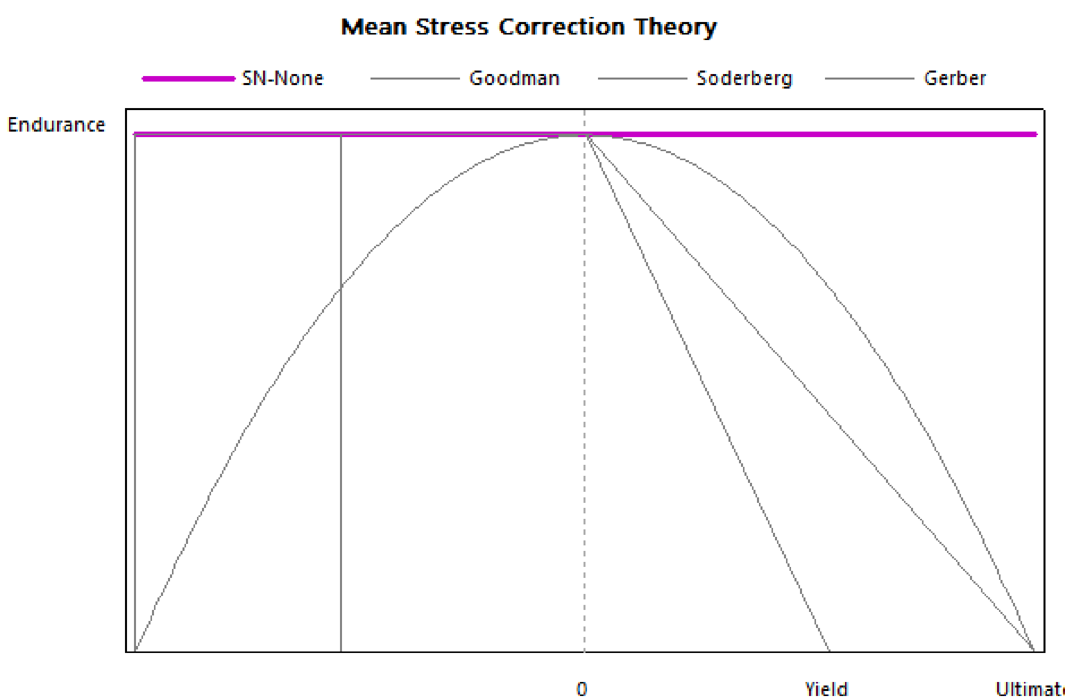
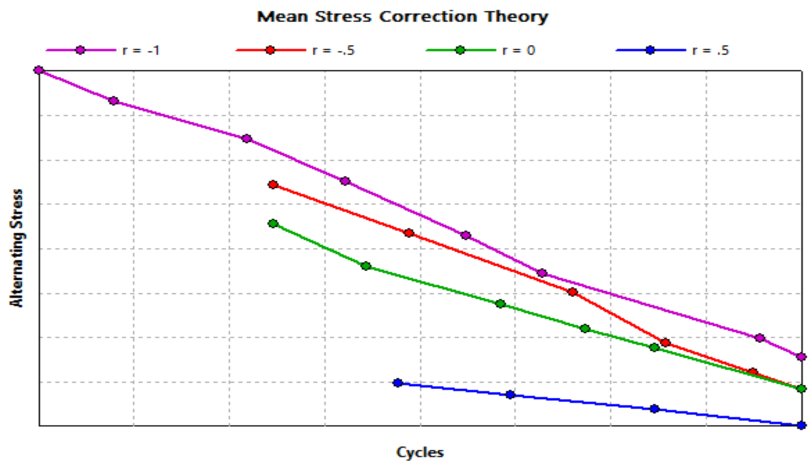
Appendix

1. Constant amplitude load



2. Mean stress correction





3. 2D representation of single cylinder engine Crankshaft