

WEIGHT OPTIMIZATION OF CRANKSHAFT FOR SINGLE CYLINDER 4- STROKE ENGINE



Tamerat Shewamare

A Thesis Submitted to Department of Mechanical Engineering
School of Mechanical, Chemical and Materials Engineering

Presented in partial fulfillment of the requirement for the degree of masters in
Automotive engineering

Office of Graduate Studies
Adama Science and Technology University

June, 2023
Adama Ethiopia

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Adviser Getachew alemayehu (PhD)

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DECLARATION

I hereby declare that this Master Thesis entitled " Weight optimization of crankshaft for single cylinder 4-stroke engine" is my original work. That is, it has not been submitted for the award of any academic degree, diploma or certificate in any other university. All sources of materials that are used for this thesis have been duly acknowledged through citation.

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LIST OF ABRIVATION

CAD	computer aided design software
BDC	Bottom dead center
B	diameter of cylinder bore
FEA	Finite Element Analysis
M	Material index
T DC	Top Dead Centre
CW	Counter Weight
b_c	Diameter of the crankpin
E	young's modulus
ε	strain
F_p	Piston Force
F_t	Tangential force
F_r	Radial force
IC	Internal Combustion
K_b	Combine shock, fatigue factor for bending
k_g	kilogram
K_t	Combine shock, fatigue factor for torsion
M	Bending moment
M_{EQU}	Equivalent bending moment
M_{pa}	Mega Pascal
mm	millimeter
NM	Newton Meter
P	Gas Pressure in the Max Bending Angle
ρ	density
RH1, RH2	Horizontal Reaction one and two
rR ,rR	Reaction at bearing 1&2 due to radial force
tR ,tR	Reaction at bearing 1&2 due to tangential force
T_e	equivalent twisting
x	

T_M	Twisting moment
V_d	displaced volume
σ_v	Von miss stress
σ_b	Bending Stress
σ_u	ultimate stress
σ_y	yield stress
τ	Shear stress
$\emptyset$	Angle of inclination (degree)
θ	Angle of twist (degree)
E	Strain

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ABSTRACT

Crankshaft is 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. The three dimensional crankshaft models was created by using the software SOLIDWORKS, was used for 3D modeling and FEA simulation analysis was performed by using ANSYS 2020 R2 SOFTWARE to obtain the total deformation magnitude in critical locations. The objective of the study focuses analysis and optimization of crankshaft to get lightweight and cost effective crankshaft. To achieve this objective, we have to use the methods to select materials and to optimize the crankshaft. The crankshaft would be analysis by theoretical, static finite element and modal analysis. The static finite element analysis result shows that the total deformation in all materials to be 0.0109-0.032mm, von-miss stress 109 -112.79Mpa and equivalent strain 0.00061- 0.015. FEA give the results of mode shape vibration and frequency of the modes. The lowest free mode vibration of the crankshaft was 173 Hz, which is much greater than natural frequency which was 27.2Hz. So, the possible action of resonance of the crankshaft is very low and crankshaft is not running in critical speed. The weight of the alloy steel (original) material is 13.6kg and the other three materials where between 4.79 kg- 13.58 kg. those materials the cast iron is the best crankshaft among the listed and 12.46kg which is 8 % weight reduce from the original material (alloy steel crank shaft), after topology The counter weight is optimized in 0.66 kg by weight, which 5.28 % from the cast iron crankshaft changing the material type and geometry optimization total of 13.15% from original alloy steel crankshaft.

Keywords: crankshaft, geometry optimization, FEA finite element Analysis, static finite element,modalanalys,frequency

CHAPTER ONE

INTRODUCTION

1.1 Background

(Dindore & Badiger, 2020) .The crankshaft is one of the complex parts of the engine which converts the reciprocating motion of the piston to the rotation motion of the crankshaft. It has three main part which are the webs, the journal parts and crankpin parts.

(Walter D. Pilkey) 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.

1.2 parts of crankshaft

Crank pin; the crankpin is a mechanical part of an engine. Which allows the connecting rod to be attached to the crankshaft very firmly? The surface of the crankpin is cylindrical, to give the rotate force to the large end of the connecting rod. These are also known as connecting rod journals.

Main Journal; have attached to the engine block. These bearings hold the crankshaft and provide it to rotate inside the engine block. This bearing is such as a plain bearing or journal bearings. The main bearings vary from engine to engine, often according to the forces given by the engine.

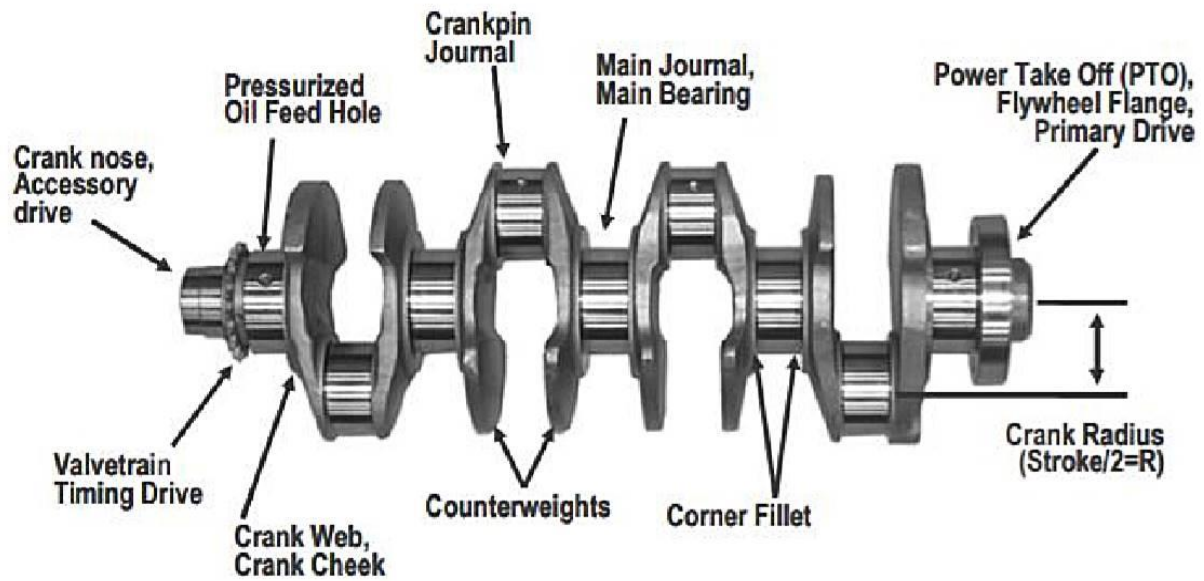


Figure1.1: parts of crank shaft posting comeantar komentar February 16, 2021

Crank Web; the Crank web is the most essential part of the crankshaft. Crank web connects the crankshaft to the main bearing journals. Last Updated on: lyas April 24, 2023 by

Counterweights; the counterweights are a type of weight that applies opposite force, which provides balance and stability to the crankshaft. These are mounted on the crank web. The reason for installing counterweights in the crankshaft is, they can eliminate the reaction caused by rotation. And it is very helpful for achieving the higher RPM and makes the engine run easily.

Thrust Washers; at some points, two or more thrust washers are provided to stop the crankshaft from moving lengthways. These thrust washers assemble among the machined surfaces in the web and the crankshaft saddle. With the help of thrust washers, it can be easily maintained the gap and helps to reduce the lateral movement of the crankshaft. In many engines, these are made as part of the main bearings, usually, older types .David brick January 15, 2023 at

Oil Passage and Oil Seals; Crankshaft oil passage passes oil from main bearing journals to the big end journals. Normally the hole is drilled on the crank web. When the crankpin is in an

upward position and combustion forces push the connecting rod to a downward position, it allows oil to enter between journal and bearing. The crankshaft has some extend beyond the crankcase on both ends. This causes oil to leak from these ends. So, preventing oil from these openings, oil seals are provided. There are two main oil seals are connected at the front end and rear end.

Front End Oil Seals; they are very similar to the rear end oil seals. However, its failure is less destructive and it is more easily accessible. The front oil seal will be set behind the pulley

Rear End Oil Seals; they are placed within the main journals and flywheels. This is pushed into hole between the engine block and the oil pan. The oil seal has a shaped lip that is kept tightly into the crankshaft by a spring called a garter spring.

Flywheel Mounting Flange; in most cases, the crankshaft attaches to the flywheel through the flanges. The diameter of the crankshaft wheel end is larger than the other end. This gives a flange face to mount the flywheel

1.3 Types of crankshaft

Following are the types of crankshafts:

- Fully built shaft
- Semi built shaft
- Welded shafts
- Solid single piece shaft
- Forged crankshaft
- Cast crankshaft

1.4 manufacturing process of crankshaft

(Angelopoulos2015). The manufacturing processes of crankshaft have three main modern processes which are sand casting, forging and powder metallurgy. Each manufacturing method has its own process and use different material for making the shaft. The casting techniques

Have its advantage and disadvantage. The advantages are, Cast iron crankshafts are cheaper and less time consuming to manufacture than those made from forged steel. Metal may be more economically used in the casting process when the design of the crankshaft becomes more complex. The wear resistance of the main and crankpin bearings increases due to the presence of graphite in cast iron. It requires less skill and low cost. The disadvantages of this process are which require further heat treatment and external machining.

There are mainly three processes which are used for crankshaft manufacturing for metals: forging, casting, and machining.

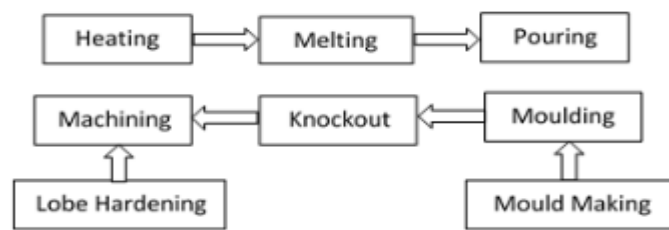


Figure1.2 casting fatemi Technical Conference, Chicago, 2005

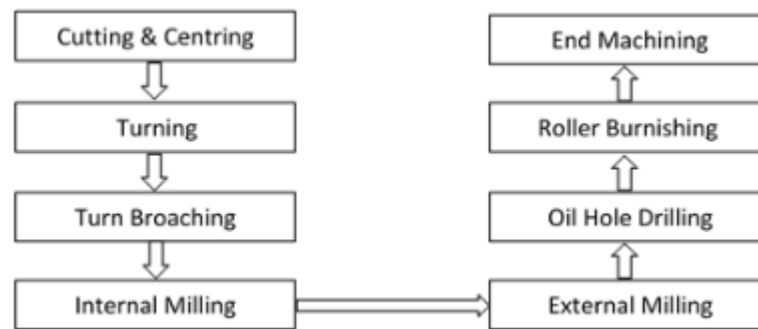


Figure 1.3 Machining Hoffmann and Turonek 1992

(Montazersadgh Fatemi, 2007). The materials used for production of crankshafts are depending on the function, availability and the cost of the materials. The crankshaft for automotive engine like motorcycles, agricultural vehicles, water pumps and generators are manufactured from different materials which have the materials property of enough strength, enough rigidity, enough stiffness, Minimum weight and less costs. Cast iron, forged steel, carbon steel cast steel; alloy steel and aluminum alloy are some of the materials at which the crankshafts are

manufactured. Weight and cost reduction will conduct during material and geometry optimization of the crankshafts. The selection of the best alternative material will give the most cost effective high performance crankshaft. The material selection of the crankshaft depends on the objective, function and constraints

(Vijaykumar, K., Manoj 2015) . There are casting and closed die forging for an automobile engine. Die forging is mainly used for engine that needs the high rigidity and strength. However, in recent years, the forces to act on crankshaft become higher, because the high-performance required by engines.

(Durga K., Prasad 2018) .Crankshafts can be cast in ductile steel or forged from a steel bar usually through roll forging. Due to lighter weight, better inherent damping and compact dimension, most of the manufacturers tend to use forged crankshafts.

1.5 Advantages of lightened crankshaft

Reduced the weight of rotational engine parts you get performance boost less weight means it spins and down faster responding more quickly to engine power changes it also engine more efficient using less power to alter the speed of rotational mass that is the engine lighter piston lighter rods and lighter counter weights all need to be fitted to maintain the engine balance.

(Hoffmann and Turonek). analyzed the cost reduction opportunities for forged steel crankshaft, with raw material cost, fatigue strength and machinability as a primary factors evaluated. In this study, a cost model supported by fatigue and machinability area and was utilized to select the lowest cost among all materials.

(Anbul, T., Govindaraji). Durability of this component and fatigue performance has to be considered in the design process, since the crankshaft subjected to a large number of load cycles during its service life Design and development of crankshaft have always been an important issue in the crankshaft industry, for manufacturing a less expensive part with the minimum weight and proper fatigue strength along with other functional requirements.

So the crankshaft should have the capacity to take the maximum downward force during power stroke without excessive bending. To have this kind of capacity the material property (strength)

and the geometry should analyzing and optimizing. To investigate either the crankshaft is in optimum design or not, we should analysis the crankshafts in the theoretical; the experimental and in software analysis. This analysis should verify or should validate each other. To achieve the objective, different tools and machines are used such as, the measuring tools. The software is one of the tools which are used for design, modeling and analysis of the objects. The solid work is CAD software used for sketching, 3D modeling of the objects and the manufacturing drawings.

1.7 Application of single Cylinder engine

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



Figure1.5 Agricultural vehicle driven by the single cylinder diesel engine

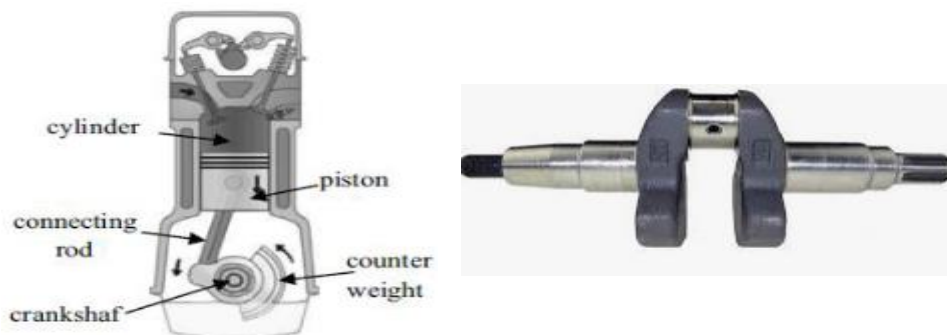


Figure1. 6 single cylinder and crankshaft

(Farzin H. Montazersadgh et al.). all 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.

1.8 Statement of the problem

Non-automotive engines are used as in many machines performing various jobs. They are majorly used in farm machinery's, water pumping, generator, tractor and others many factories and workshops. The weight, cost and the consumption of fuel of engine were the critical issue of the design of engine components (Arshad, Cong, et.al 2021)

The JSD 1800 engine is non-automotive engine which has the overall weight of 275kgs. From engine weights, the crankshaft itself has the weight of 13.6kg which is 5% of total weight

A linear regression analysis of curb weight versus carbon dioxide (CO₂) emissions (a measure of efficiency that is correlated with fuel consumption) for the model year 2008 vehicle fleet suggests that a 10% reduction in vehicle weight is associated with an 8% reduction of CO₂ emissions (Davis, 2010).

A model that combines curb weight and fuel consumption data with a technique for normalizing vehicle performance indicates that a 10% reduction in vehicle weight yields a 5.6% reduction in fuel consumption for cars and a 6.3% reduction in fuel consumption for light trucks. Massachusetts Institute of Technology, 2010

Reducing weight without compromising safety and comfort is generally achieved by moving from a heavier material to a lighter material to the extent for which it is both financially and technically viable

Reducing the weight of the crankshafts is delivering superior value to the customer. As the crankshaft weights reduce, the consumption of the fuel of the engine also reduces. It is logical that reduction of minimum weight in a single crankshaft will result in large saving during

large volume of production. ,JSD 1800 Non-automotive machinery the research will propose to analysis and optimize the material and geometry of JSD 1800 crankshaft to reduce weight and cost.

1.9 Objective

1.9.1 General objective

- The general objective of this research is to analysis and weight optimization of a crankshaft for a single-cylinder 4-stroke engine FEA .without affecting its performance

1.9.2 Specific objective

- To optimize the material used in crankshaft manufacturing for weight reduction.
- To model single cylinder engine crank shaft using SOLID WORK for different materials
- To analyze the models of crankshaft for the different modes of vibration of the crankshaft
- To compare von-miss stress, shear stress and deformation of the three candidate materials and recommend the light weight and with minimum stress.

1. 10 Significance of the study

In order to protect natural resources and economic energy weight redaction has been the main N. Lutsey, Review A model that combines curb weight and fuel consumption data with a technique for normalizing vehicle performance indicates that a10% reduction in vehicle weight yields a 5.6% reduction in fuel consumption for cars and a 6.3% reduction in fuel consumption for light trucks.

1.11 Methodology

The methods employed to achieve the above objectives are

1.11.1 Literature survey

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.

1.11.2 Analytical

In the design of the crankshafts, it is assumed that the crankshaft is a beam with two 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 using distortion energy theory the stress induced in the crankpin is calculated.

1.11.3 Modeling and FEM analysis`

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 2020R2 software work bench for analysis were performed for both parameters like total different, von-misses stress, shear stress and equivalent strain and of crankshafts.

1.12 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 2020R2 WORKBENCH software. Using proper type of loading and boundary condition are very important in finite element analysis..

b. Mesh Generation

Tetrahedral elements were used to mesh the crankshaft finite element geometry. FE models were done for each optimized to analyze stress 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.

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 center and also load data were taken from the calculated result. Boundary condition is based on under supporting condition of crankshaft then obtaining results.

1.13 Scope of the study

The study of this research covers modeling and analysis of existing crankshaft depending on the existing material and dimension. The analysis conduct in mathematical method and in software. The modeling of crankshaft created by SOLIDWORK and the analysis conduct in ANSYS 2020R2 software. This study conduct until evaluation and optimization of crankshaft in material and geometry depending on the specification of the engine

1.14 Organization of the thesis

This thesis is organized into five chapters. Chapter one is an introduction that comprises the background of the study, statement of the problem, objectives, and scopes, significance of the study, and organization of the thesis. Chapter two presents a review of literature related to the study that emphasizes the study of design, modeling, and analysis of crank shaft. Chapter three describes material, dimension, conditions, and methods Used to complete this work. Chapter four describes analytical result and FEA analysis, Chapter five finally in chapter five conclusions, recommendation and future works are presented.

CHAPTER TWO

LITERATURE REVIEW

2.1 Theoretical Background

B. Vijaya Ramnath et al. (2018). In this paper, an attempt has been made to derive all the parameters such as designing, manufacturing, maintaining of products, service and structures. The crankshaft is reverse engineered using CMM device for unknown geometrical data's. Computer aided modeling using CATIA and optimization analysis of crankshaft is used to study and compare the fatigue performance of three different materials of automotive crankshaft, namely forged steel, ductile cast iron and aluminum alloy. The dynamic analysis was done and verified by simulations in ANSYS. The maximum stress region and dangerous areas are found by the deformation analysis of crankshaft. Based on the finite element analysis results, possible weight reduction is determined to the feasible material forged steel after weight reduction.

(Anbu T.2014) conducted analysis in three modules: modal, static, and transit. The research was carried out on crankshaft of a multi-cylinder engine in order to determine the stresses and deformation. They took values from the load characteristics of the engine as boundary values. In the end, they presented a proposal to reduce the cost of production of materials that could be used

(Fonte, M., Duarte. 2015) Nowadays, the more common crankshafts manufactured as:

- a. assembled crankshafts, where the main journal, crankpin journal and crank webs are manufactured separately and then fitted together using a shrink fitting method;
- b. Semi-built crankshaft where main journals and/or crank webs and crankpin journals are forged as one piece and shrink fitted together.

(Solanki, et. Al 2011) During its service life, it experience a large number of load cycles, the fatigue performance and durability of the component has to be considered. In order to produce

less expensive and minimum weight component, the design development has to be an important issue which considers the material and geometry of the component.

(Farzin H. Montazersadgh et al. (2007). The crankshaft undergoes a large number of load cycles during its fatigue performance, durability and service life of this component has to be taken in the design process. Design and development of the crankshaft is an important issue in the manufacturing industry, in order to produce a less expensive component with the light weight, good fatigue strength with better fuel efficiency and higher power output.

Katari et al. (2011) 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 .

(Solanki 2011) .present the research work on crankshaft design and optimization. Selection of materials, manufacturing process for crankshaft, failure analysis, design consideration etc. are studied. In design of crankshaft considers the dynamic loading condition and optimization of shaft satisfying the requirements of the automobile specification with minimize cost and size effectiveness. Their conclusion is the crack grows faster on the free surface while the central part of the crack front becomes straighter. Fatigue is the main factor for failure of the crankshaft.

2.2 Material optimization

(Sravanthi, Pooja et al 2020) .they were studying on weight optimization of crankshaft in two different materials which were carbon steel and Kevlar 49. They were design and analysis the crankshaft to investigate the deformation and the von misses stress. The weight of the material of the carbon steel is 128.9 kgs and the Kevlar 49 is 52.5 kgs. 76.5 kgs weight was reduced by changing the carbon steel material with Kevlar 49 material. By doing this the deformation of the cranks has no significant change on the critical location of the crankshaft.

(Soderhjelm, 2017) study on Reducing weight without compromising safety and comfort is generally achieved by moving from a heavier material to a lighter material to the extent for which it is both financially and technically viable. For many of the metal components, weight reduction can be achieved by switching from ferrous alloys to aluminum. It is lightweight, corrosion resistant, superior specific strength, and cheap relative to other lightweight metals. However, aluminum does not have the necessary stiffness for certain components, and its strength properties deteriorate at elevated temperatures. This mean that two strategies for decreasing the carbon footprint of the car collision. To increase the efficiency of the powertrain requires higher temperatures and pressures, and we want the engine lightweight to reduce its fuel consumption even more. A powerful technique is to create hybrid or bimetallic structures, so the majority of the component is manufactured out of aluminum and locally reinforce a crucial section of the component using steel. The component will act as a bimetallic composite which allows for a lightweight component with the required strength at the necessary locations. An efficient production method to produce this bimetallic structure for cast components is to place the reinforcing insert in the mold and cast the aluminum around it creating a multi material metal cast component.

(K.Sandya, and M Keerthi, 2016) analyze the crankshaft in several positions of the crank. The static analysis is conducted on the crankshaft with three different materials such as Structural steel, Al 6061, Inconel x750 in different orientations. The results are validated with theoretical calculations for two crank positions for all materials and it is observed that Al6061 is subjected to high von-miss stresses compared to remaining two materials. Incone x750 is subjected to little deformation when compared to remaining two materials

Nallicheri et al. Performed on material alternatives for the automotive crankshaft based on manufacturing economics. They considered steel forging, nodular cast iron, micro-alloy forging, and au tempered ductile iron casting as manufacturing options to evaluate the cost effectiveness of using these alternatives for crankshafts

(Amit Solanki, et al2011) studied the design considerations and concluded that to apply the selection of material and manufacturing process need comparative study to have cost

effectiveness and defects free shape respectively. In crankshaft the crack grows faster on the free surface while the central part of the crack front becomes straighter. The very prevailing mechanism of failure of crankshaft is fatigue. So the accurate stresses are critical input for fatigue. Residual imbalances along with the length of crankshaft are crucial to performance. Also the stress and strain analysis must be conducted due to the nature of load applied on crankshaft.

(Kartik D. Kothari 2009) Studied the optimization options, their combination under a set of defined constraints and a comparison between the Original forged steel crankshaft and the final optimized Forged steel component. The main objective of this analysis was to optimize the weight and manufacturing cost of the forged steel crankshaft, which not only reduces the final production cost of the component, but also results in a lighter weight crankshaft which will increase the fuel efficiency of the engine. As the main objective of this analysis, it was attempted to reduce the weight and final cost of the component by changing the crankpin geometry, Increasing the oil hole diameter and fillet radius, increasing the oil hole depth and Changing the crank web geometry.

(Khurmi & Gupta, 2005) the machine parts should be made from the material that the properties of which would be suitable for the condition of operation. Understanding of the material properties, treatments, and manufacturing process is essential to good design. Based on the property of the material, the designed parts could be resisting the applied load that would be imposed on the component during the operation. The material to be selected for the designed parts should have strong in bending, excellent fatigue resistance and easily machine.

1.6 Stress in crank shaft

(Raut & Mestry, 2017) .There are many sources of failure in the engine. One of the most common crankshaft failures is fatigue at the fillet area due to the bending loads. The moment of pressure force from plunger rod 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 concentration exist

and these high stress range locations are the points where cyclic loads could cause fatigue crack initiation leading to fracture.

(Montazersadgh, F. H. and Fatemi, A., 2007). The crankshaft is subjected to various forces but generally needs to be analyzed in two positions. Firstly, failure may occur at the position of maximum bending; this may be at the center of the crank or at either end. In such a condition the failure is due to bending causes the pressure in the cylinder is maximal. Second, the crank may fail due to twisting. The pressure at this position is the maximum. Since the crankshaft is subjected to millions of repetitive cyclic loading, there are two different load sources in the crankshaft. The first load source is the inertia of rotating components (e.g. connecting rod) which increases with an increase in engine speed. This force is directly related to the rotating speed and acceleration of rotating components. The second load source is the force applied to the crankshaft due to gas combustion in the cylinder. 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 crankshaft, the force will be transmitted to the crankshaft. This magnitude of the force transmitted depends on many factors which consist of crank radius, connecting rod dimensions, and weight of the connecting rod

The crankpin is like a built in beam with a distributed load along its length that varies with different crank angle position. Each web like a cantilever beam subjected to bending & twisting. Journals would be principally subjected to twisting as shown in Figure 1.4 Following stresses are developed in the crankshaft:

- Bending causes tensile and compressive stresses.
- Twisting causes shear stress.
- Due to shrinkage of the web onto the journals, compressive stresses are set up in journals & tensile hoop stresses in the webs (Anthony P. Sime, B.Eng)

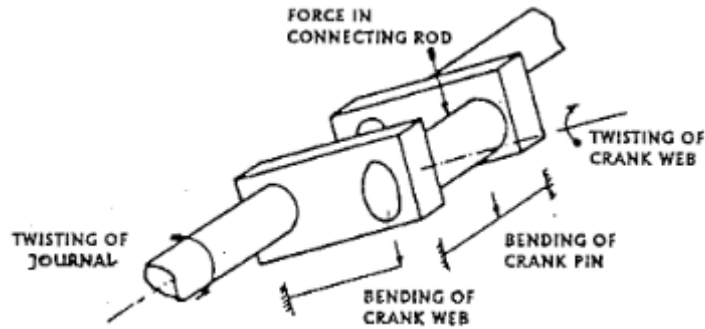


Figure1. 4 crank shaft subjected force B.V.M. Engineering College, V.V. Nagar, Gujarat, India

2.3 Failure analysis of crank shaft

(Aliakbari, ande Safarzadeh 2018) the study focuses on failure analysis of the crankshaft of the wheel loader diesel engine. The analysis is done to investigate the cause of failed crankshaft by analyzing the chemical composition of the material, by testing the mechanical properties of a material including hardness and tensile stress and also examining the micro structure of the material by using optical micrograph. The result of the study concluded that the failure of the crankshaft had happened after 4800 hours in services. The examination shows that the steel structure is martensitic type. The fracture of the structure is happened due to the rotational bending stress. Also the crack initiation occurs on the crank pin due to oil impurity, inappropriate machining, sever wear due to lack of lubrication and impurities of the surface of the crank pin.

(Farrahi, etl ,2011) Studies on an improper manufacturing and operation or mechanical fatigue may tend to a catastrophic failure in crankshafts. A failure investigation of a nitride crankshaft revealed that the partial absence of the nitride layer in the fillet region decreased the fatigue strength and in fatigue initiation and propagation in weaker region. In the case of a cast crankshaft that a break into two pieces had been occurring in the crankpin region, examination of the failure zone proved the absence of hardening case in the fillet region and the presence of free graphite in the microstructure, led to the initiation and propagation of fatigue cracks.

2.4 Finite Element Modeling

(Bagde, 2013) They have been conducted on FEM in Static structural analysis and fatigue analysis of the crankshaft by applying different materials, EN-9, SAE – 1045, SAE-1137, SAE-3140 and NI – CI (Nickel Cast Iron). Based on the stress distribution and maximum deformation results, we get to know which material suits better for the crankshaft and thus, fatigue life can be calculated. From results of analysis the von misses stress of SAE 1137 has to be minimum from the other materials. The von-misses stress in all crank shaft materials found to be 1.863×10^8 Pa. to 1.864×10^8 Pa. the maximum deformation value is 0.0006 mm to 0.0010 mm for all materials. Hence, fatigue life will be more for SAE 1137 than that of the other.

(Rinkle Garg et al. 2012)Conducted a static analysis on a cast iron crankshaft from a single cylinder four stroke engine. Finite element analysis was performed to obtain the variation of the stress magnitude at critical locations. Three dimensional model of the crankshaft was created in Pro-E software. The load was then applied to the FE model and boundary conditions were applied as per the mounting conditions of the engine in the ANSYS. Results obtained from the analysis were then used in optimization of the cast iron crankshaft.

(Abhishekchoubey, and Jamin Brahmhatt.). Crankshaft model and 3-dimentional model of the crankshaft were created by SOLID WORKS Software and imported to ANSYS software. The crankshaft maximum deformation appears at the center of crankpin neck surface. The maximum stress appears at the fillets between the crankshaft journals and crank cheeks and near the central point journal. The edge of main journal is high stress area.

(C.M. Balamurugan et al2011) has been studied the Computer aided Modeling and Optimization of crankshaft and compare the fatigue performance of two competing manufacturing technologies for automotive crankshafts, namely forged steel and ductile cast iron. The Three dimensional model 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 cost of the crankshaft, without changing connecting rod and engine block.

(Balamurugan et al.2011)done computer aided modeling and optimization analysis of crankshaft to evaluate and compare the fatigue performance. They compared two crankshafts, cast iron and forged steel, from a single cylinder four stroke engines. Finite element analysis was performed to obtain the variation of stress magnitude at critical locations. The dynamic analysis was done analytically and was verified by simulation in ANSYS.

(Jaimin Brahmbhatt et al.) conducted a dynamic simulation on a crankshaft from a single cylinder 4- stroke diesel engine. A three-dimension model of diesel engine crankshaft is created using SOLID WORKS software. Finite element analysis (FEA) is performed to obtain the variation of stress magnitude at critical locations of crankshaft. Simulation inputs are taken from the engine specification chart. The dynamic analysis is done using FEA Software ANSYS which resulted in the load spectrum applied to crank pin bearing. This load is applied to the FE model in ANSYS, and boundary conditions are applied according to the engine mounting conditions. The analysis is done for finding critical location in crankshaft

(Pandiyar) studies on the crankshaft for single cylinder engines by using reverse engineering which focuses on parametric optimization to replace the existing crankshaft without changing the engine block and the connecting rod. The ANSYS analysis, investigation shows that the counterweight, crank web and connecting rod journal are experiencing less stress. During parametric optimization the most significant way is the removal and addition of material must symmetric to the central axis. The study focuses on AISI 1040, AISI 1045, AISI 4140 and AISI 4615 material and conducted the stress and deformation analysis on each material. From those the AISI 1045 is the best material for crankshaft which can withstand and sustain the engine load cycles. Factor of safety for AISI 1045 is more as compared to other three materials, means that design is much stronger.

(Rinkle Garg, Sunil Baghla) they work on a cast iron single cylinder engine crankshaft. A static analysis is done on a cast iron crankshaft. 3D model of crankshaft is created in Proe software. FEA is done and obtain result shows the variation of the stress magnitude at critical location. Obtained results are used in optimization of crankshaft. They conclude the strength of

the crankshaft as the maximum limits of stresses, total deformation and strain is decreased. The weight of the crankshaft is also decreased by 3934 gram also decrease the inertia force. Weight reduced hence the cost of crankshaft is reduced and increase the engine performance.

(Solanki et al, 2011) study improving fatigue life of the single cylinder crankshaft by modeling The shaft in SOLIDWORKS to predict the maximum value of the equivalent alternative stress which decreases as fatigue life increase. The analysis is done on the ANSYS software using geometry optimization like crankpin fillet radius and crankpin diameter which the maximum stress is found at the fillet area between the journal and web.

2.5 Optimization of crank shaft with geometry

(Shahane & Pawar, 2017) The geometry optimization of the crankshaft will conduct in the crankshaft of different portion. Depending on its function, the crankshaft is as match as possible compatible to minimize the weight. Changing the crankpin geometry, fillet radius, oil hole diameter, counterweight, and webs modification are the alternative portion of the parts.

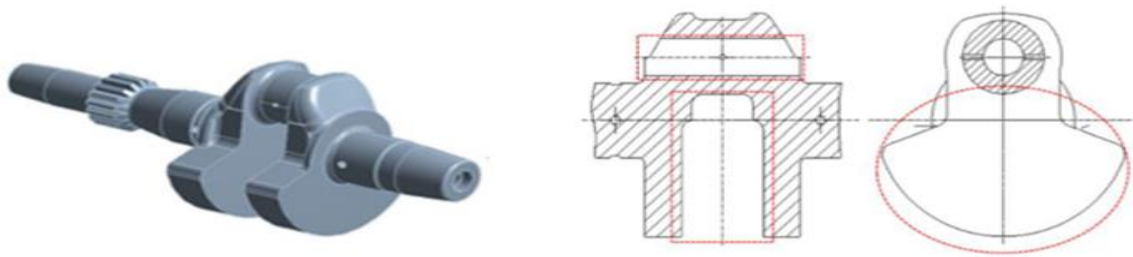


Figure 2.1: the original and the optimized portion of crankshaft (shahane and pawar, 2017)

(Biradar & Savanur, 2014) studies on static stress analysis and optimization of diesel engine crank shaft by FEA. In this study the objective is weight optimization. The methodology used to optimize it is, measuring the dimension of the crankshaft and modeling it by CATIA. Material and weight optimization of the crankshaft was done by analysis of ANSYS software. Four materials were used and by comparing all the determinants like Von miss stress, shear stress, and deformation and mass on all type of

materials crankshaft, they would show Gray cast iron crankshaft is best suitable from the study by reducing 64 grams of material.

(MominMuhammad Zia Muhammad Idris, etal) 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 %

2.6 Summary from literature survey

Review From literature survey we came to know that In order to produce less expensive, minimum weight and manufacturing economics. Various authors had done analysis for different materials given their results and conclusions Based on their work the following review used to identify the research areas which will need further studies.

- Different researcher wear done crank shaft analysis using different material in terms of cost effectiveness.
- Static analysis is done on crankshaft is created ANSYS software. FEA is done and obtain result shows the variation of the stress magnitude at critical location. Obtained results are used in optimization of crankshaft.
- Based on the stress distribution and maximum deformation results; we get to know which material suits better for the crankshaft
- In order to produce less expensive and minimum weight component, the design development has to be an important issue which considers the material and geometry of the component.
- The research work on crankshaft design and optimization. Selection of materials, manufacturing process for crankshaft, failure analysis, design consideration etc. are studied

- Optimization options, their combination under a set of defined constraints and a comparison between the Original forged steel crankshaft and the final optimized Forged steel component.
- there is no study which design and analysis of JSD 1800 engine presence advanced FEA tool
- After reviewing for the existing literature.it is observed that there is no enough researcher work done on design Bothe material optimization and topology optimization on JSD 1800 engine crank shaft.

This analysis was to optimize the weight and manufacturing cost of the forged steel crankshaft, which not only reduces the final production cost of the component, but also results in a lighter weight crankshaft which will increase the fuel efficiency of the engine

Weight was reduced by changing the carbon steel material with Kevlar 49 material. By doing this the deformation of the cranks has no significant change on the critical location of the crankshaft.

2.7 Research gape

From the above literature review it can be seen most researcher study of crank shaft material static fatigue analysis to optimize less expensive and minimum weight component material of crankshaft for weight reduction improve vehicle performance.

The previous researcher did not considered modal analysis (natural frequency of material) that system teds to vibrate in absence of driving or damping forces of materials

CHAPTER THREE

Material and methods

3.1 Material and methods

For proper functioning the crankshaft material 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, 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 alloy steel which is medium carbon steel and the type of engine is JSD1800

3.1.1 Material

The study stands from the crankshaft material and dimension of JSD 1800 model engine. The specification of the engine described in table 3.2, the full dimension of the crankshaft taken by measuring tool of caliper. The dimension listed in table 3.3



Figure 3.1 Images of JSD1800 engine

Table 3.1 Material of original crank shaft

NO	MATERIAL TYPE	ALLOY STEEL	UNITE
1	Young's modules	200	Gpa
2	Density	7850	Kg/m
3	Poisson ratio	0.2-0.3	-
4	Yield stress	590	Mpa
5	Hardness	<217/19.8	HB/HRC
6	Ultimate tensile	750	Mpa

Table 3.2 Table specification of engine

NO	Parameter	Value
1	GSD 1800	Diesel engine horizontal 4stroke single cylinder
2	Cylinder diameter	135 mm
3	Stroke length	130mm
4	Piston Displacement (v_d)	1.86mm
5	Starting methods	Electric starting
6	Compression ratio	20±1
7	Power (RPM)	1600
8	Cooling methods	Water
9	Lubrication type	Force and splash
10	Net Wight (KG)	300



Figure 3.2 Photo of measuring crank shaft of JSD 1800

Table 3.3 Dimension of JSD 1800 crank shaft

NO	Parameter	Dimension in millimeters
1	Crank radius	40
2	Web thickness	37
3	Web width	90
4	Diameter of crank pin	75
5	Crankpin length	77
6	Right and left journal dia.	75
7	Length of left journal dia.	41
8	Length of right journal	41
9	Shaft length	436.7
10	Oil hole diameter	19
11	Weight of crankshaft (KG)	13.6

3.2 Alternative material selection

Material Selection In material selection of the components, the design of the structural elements is specified by three things which are the functional requirements, geometry and the properties of the material. The first step in relating design requirements to material property is a clear statement of function, objective, constraints (Fernandez,2012) .

1. Function: Crankshaft for power transmission
2. Objective: To reduce weight and cost
3. Constraint: the density of the material and the gas force F.

Density of original crankshaft of alloy steel

$$\rho \leq 7850$$

$$\text{The functional equation} \quad \rho = \frac{m}{v} \quad 3.1$$

Where m mass, v volume, ρ density of object

$$\text{Mass of object } m = \rho v \quad 3.2$$

Where m mass, v volume, ρ density of object

$$V = AL \quad 3.3$$

Where $A = \text{area}$ $L = \text{length}$

$$m = AL\rho$$

We can reduce the mass by reducing the cross section, but there is a constraint the area of the crankshaft must be sufficient to carry the load F .

$$\frac{F}{A} \leq \sigma_f \quad 3.5$$

Where σ_f is the failure strength

$$m \geq (F)(L) \left(\frac{\sigma_f}{\rho} \right) \quad 3.6$$

In the equations, the first bracket contains the specified load F , the second bracket contains the specified geometry, and the last bracket contains the material property. The lightest crankshaft which will carry the load F safely is that made of the material with the smallest value of $\frac{\sigma_f}{\rho}$

We therefore invert the material property and define the material index

$$M = \frac{\rho}{\sigma_f} \quad 3.7$$

The lightest crankshaft which carries the load F safely without failing is that the largest value of this index.

The alternative materials were selected in the material indexes (Fernandez, 2012) by using the strength and the density.

Table 3.4 alternative material

Name of material	Alloy steel	Cast iron	Aluminum alloy	Cast alloy steel	Unit
Elastic modulus	2.1e+11	6.6e+10	7.24×10	1.9e+11	N/m^2
Poisson ratio	0.28	0.27	0.33	0.26	N/m^2
Mass density	7750	7200	2680	7300	N/m^2
Tensile strength	7.2×10^8	1.5×10^8	1.4×10^8	4.4×10^8	N/m^2
Compressive strength	-	5.7×10^8	-	-	
Yield strengthen	6.2×10^8	- 4.5×10^8	-	2.4×10^8	N/m^2

3.3 Generation of the Geometry of Crankshafts

The crankshaft is modeled in SOLIDWORK by the use of the dimension which was measured in the original crankshaft by caliper. The solid model generated shown in fig. has two parts which include counter weight as one parts and journal, web and the crankpin as the other parts. Once the physical model was created, the model will import into ANSYS software for conduct FEA to determine the von misses stress, shear stress, strain and deformation of the existing crank shaft.

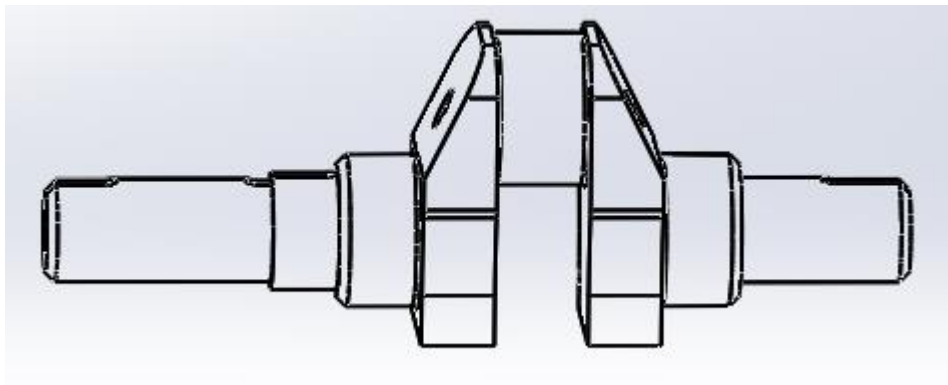


Figure3.4 Original crank shaft 3D

3.4 Meshing of Crankshaft

The mesh generation is the first step of finite element method for dividing the part into smaller parts to allow a localized element size. As the number of the element increases the stress values

predicted by FEM approximates to exact values. The Purpose of discretization is to perform the analysis on each small division separately. The part is subdivided into nodes and elements. The collection of elements is called mesh and it is necessary to make mesh optimization to get more accurate results. Mesh optimization is carried out until the FEA results and analytical solution close to each other.

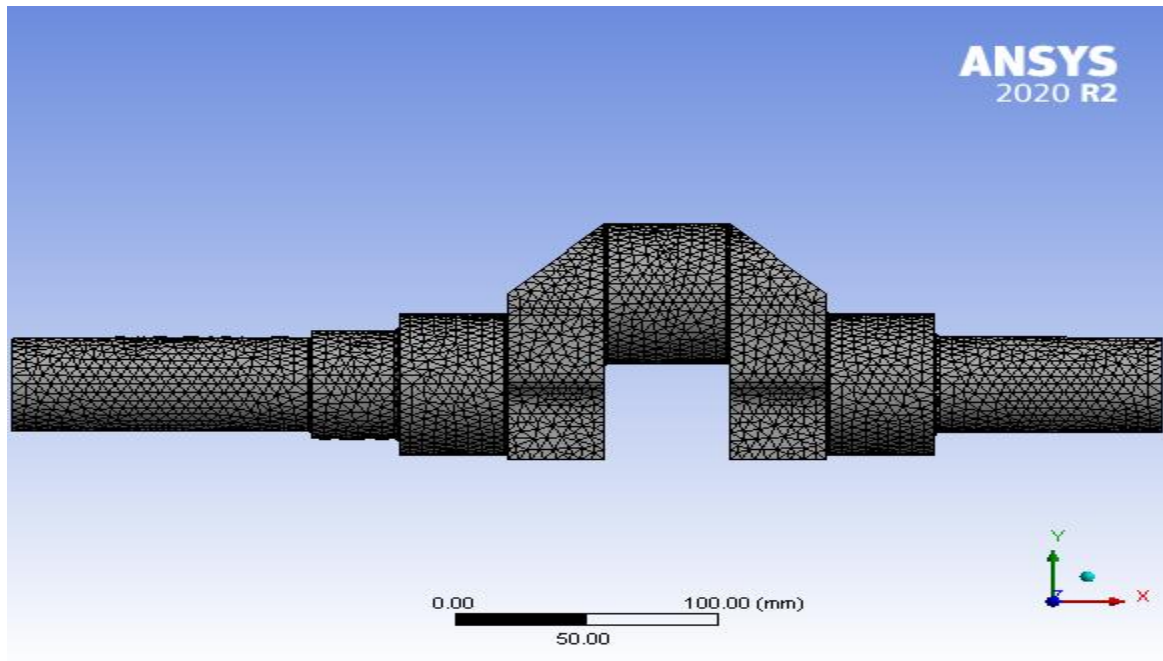


Figure 3.5 meshes of original crank shaft

3.5 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 center and also load data were taken from the calculated result. Boundary condition is based on under supporting condition of crankshaft

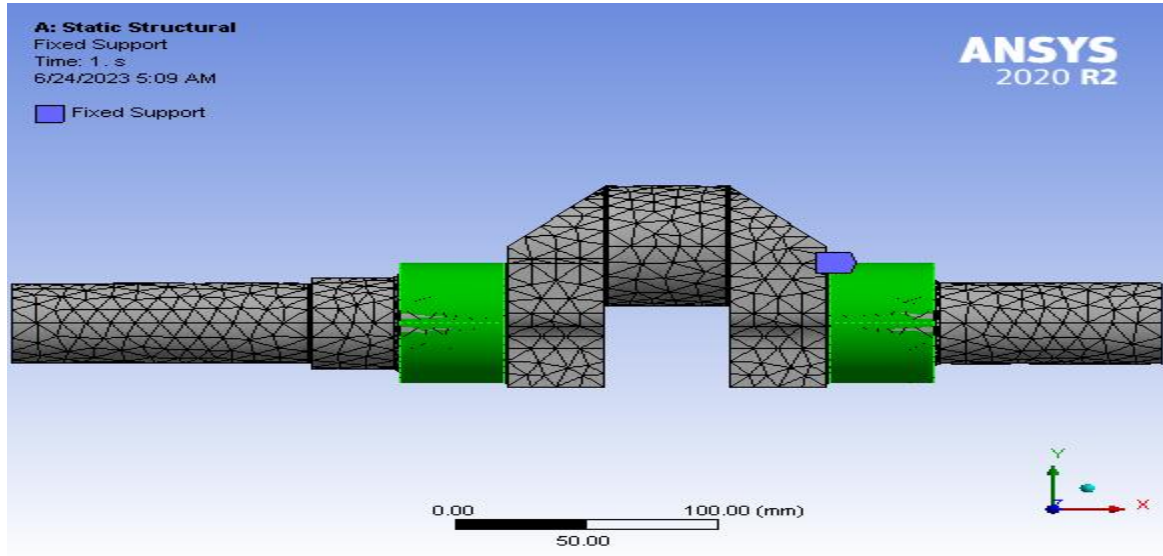


Figure 3.6 boundary condition cylindrical supports

3.6 Boundary Condition and Load Apply

The crankshaft is constrained with two journal bearings on both sides which are press fitted to the crankshaft that allows the crankshaft rotate about the main axis. In a combustion chamber the maximum load is applied to the crankshaft when a piston is at top dead center. The load is applied perpendicular to the surface of the crankpin to represent a piston load at top dead center. In this study a force of 104.9KN is applied on the crankpin at the top dead center position of the piston.

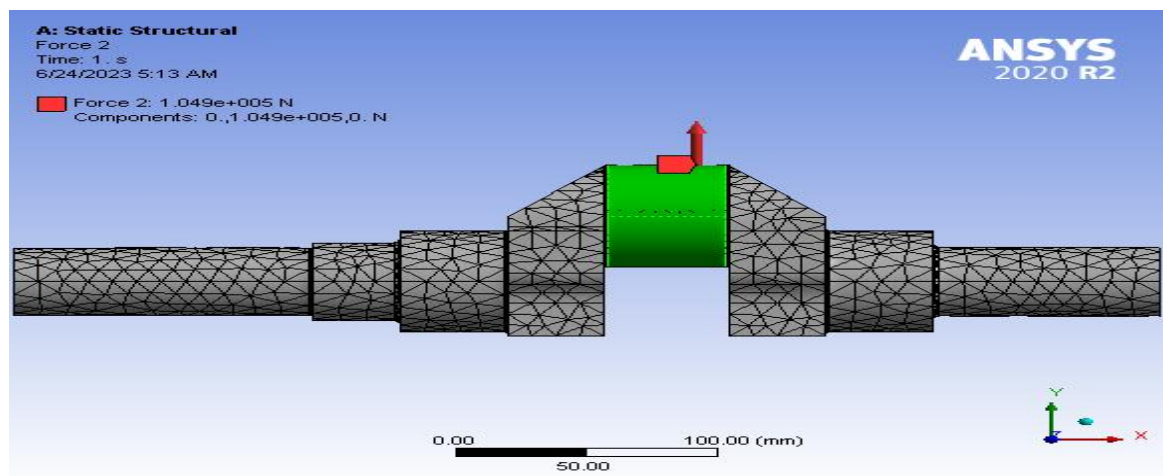


Figure 3.7 Load apply on crankshaft

3.7 Conditions

- Crankshaft is assumed as a beam with two supports.
- Geometry and material optimization will perform
- The Analytical calculation will study
- The crankpin is subjected to various forces and analyzed in two positions which is at the position of maximum bending (when the crank is at dead center and twisting
- Force is acting on the center of crankpin or at dead center (Rahul Kumar et al,)
- The maximum value of the tangential force lies when the crank is at an angle of 25-35 from the dead center for petrol engines and for 30-40 diesel engines (Jaimin Brahmbhatt, et al)
- Piston force is calculated using maximum cylinder pressure, and bore diameter of engine cylinder (Mr. Suraj K. Kolhe
- k_b Combine shock, fatigue factor for bending $=k_t$ and $=$ Combine shock, fatigue factor for torsion $= 1$ (Mr. Anant B. et al,)
- In order to find the thrust in the connecting rod (FQ), angle of inclination of the Connecting rod with the line of stroke (i.e. angle θ), Take $\theta = 4^\circ$ (Rinkle Garg, et al,)

3.7 Analytical methods

3.7.1 Methods

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 crankshafts develop two types of stress on the crankpin during the working cycles which are bending stress and shear stress due to torsional moment the crankshaft the crankshafts subjected to maximum bending when the crank is in dead center. The failure may occur at this position due to cylinder pressure reaches maximum. The crank pin is subjected to twisting moment when the cranks are at an angle. Due to maximum twisting load the shear stress may occur crank may fail. Therefore the design of crankpin must considering the two crank positions.

Analysis the crankshaft, the gas pressure is the input parameter to determine the force on the cylinder which causes the crankshaft for max bending and torsion load. In the cylinder

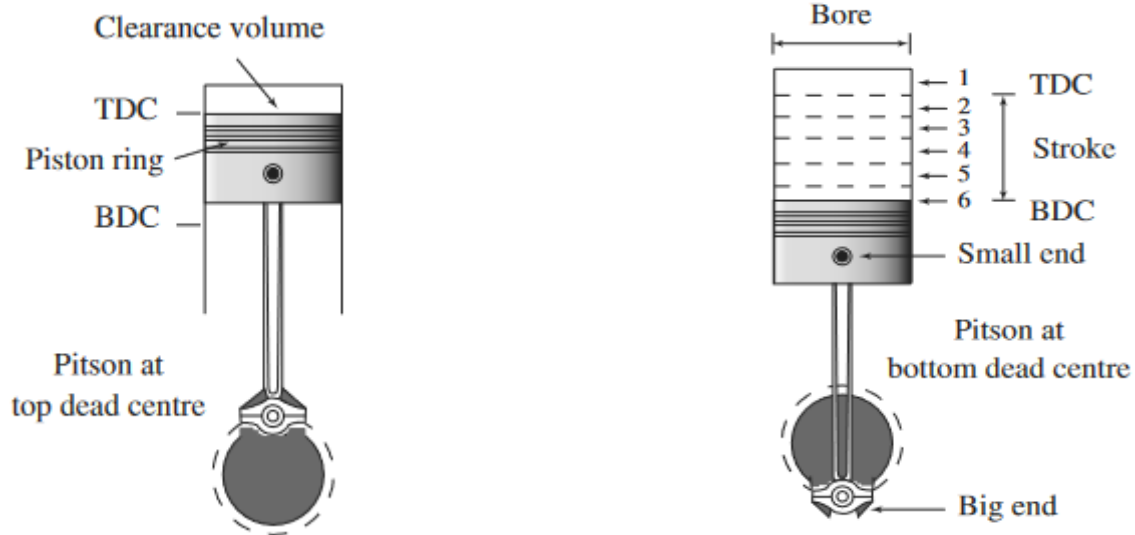


Figure 3.8 part name of piston

Top Dead Centre (TDC): It is the dead center when the piston is farthest from the crankshaft.

Bottom Dead Centre (BDC): It is the dead center when the piston is nearest to the crankshaft.

Displacement or Swept Volume (V_s): The nominal volume swept by the working piston when travelling from one dead center to the other is called the displacement volume. It is expressed in terms of cubic centimeter (cc) and given by

$$v_s = AL = \frac{\pi}{4} D^2 L \quad 3.8$$

3.7.2 Crank at Dead Center

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

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

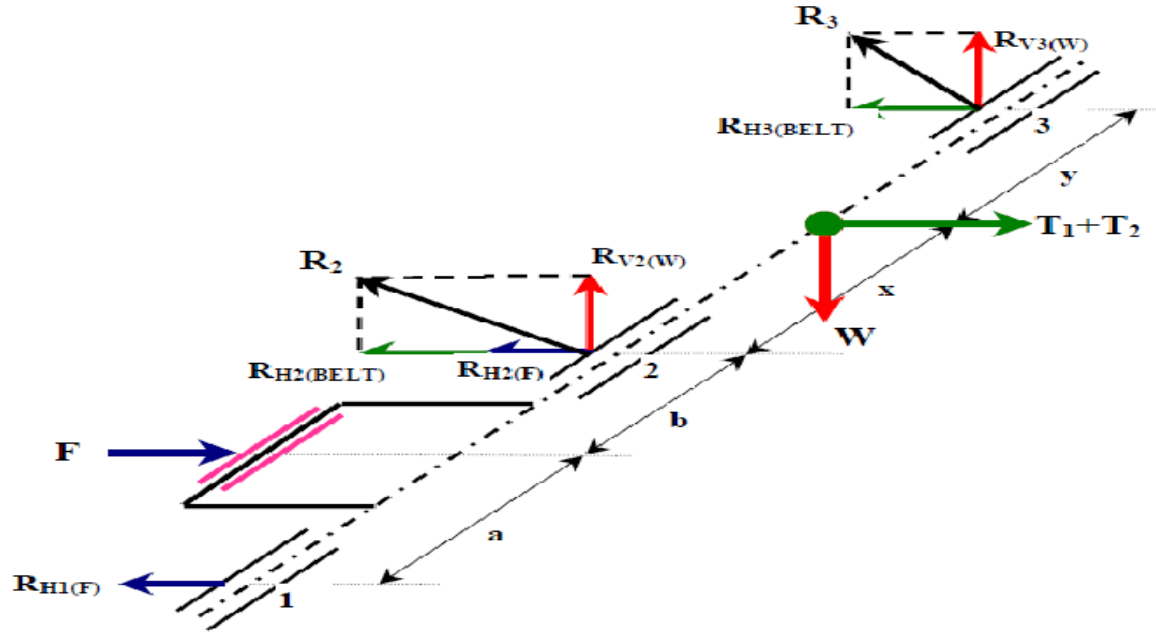


Figure 3.9 Force analysis of crank at dead center

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

$$F_s = \frac{\pi B^2}{2} P \quad 3.10$$

Where,

F_p = force acting on the cylinder due to gas pressure

B = diameter of the piston

P = gas pressure in the max bending angle

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 $R_{H2}(F)$

$$\sum M = 0 \quad 3.11$$

Assume that the clockwise direction as positive and anticlockwise direction as negative bending moment.

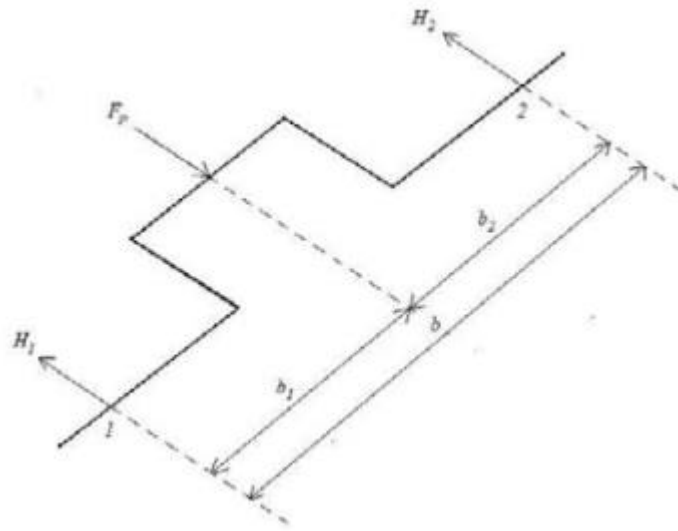


Figure 3.10 Crank at dead center

$$FP(b_1) - RH_2(b_1+b_2) = 0 \quad 3.12$$

$$R_{H2} = \frac{F_P * b_1}{b_1 + b_2} \quad 3.13$$

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

$$R_{H1(F)} + R_{H2(F)} - F_P = 0 \quad 3.14$$

By substituting equation 3.12 in equation 3.14 we get

$$R_{H1} = F_P - \frac{F_P b_1}{b_1 + b_2} \quad 3.15$$

When the shaft is subjected to the bending moment, the bending stress will be drive from the general formula as follows.

$$\frac{M}{I} = \frac{\sigma}{c} \quad 3.16$$

Where M =internal bending moment

C=perpendicular distance from natural axis to farthest point

I= momentum of inertia

σ =bending moment

$$M = \frac{I \cdot \sigma}{c} \quad 3.17$$

Substituting in 3.17 $c = \frac{d_c}{2}$ and $I = \frac{\pi(d_c)^4}{64}$

$$M = \frac{\pi}{32} \sigma_b d_c^3 \quad 3.18$$

$$\sigma_b = \frac{32M}{\pi d_c^3} \quad 3.19$$

Where M = Is Bending moment, and

d_c =is diameter of the crankpin

σ_b =is allowable stress in bending

3.7.3 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 $25^\circ = 30^\circ$ from the dead center for petrol engines and $30^\circ = 40^\circ$ for diesel engines

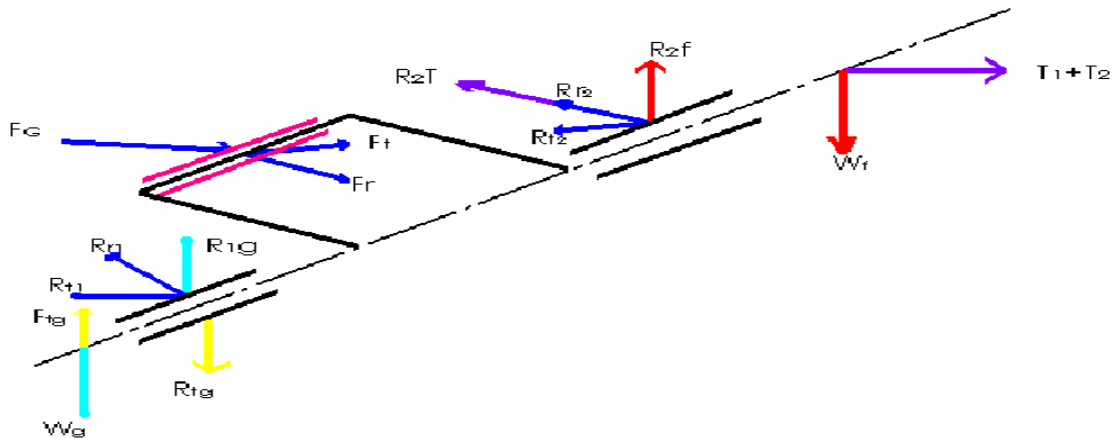


Figure3.11 Crank at maximum twisting moment

Tangential force will have two reactions RH1FT and RH2FT at bearing 1 and 2 respectively. Radial force will have two reactions RH1FR and RH2FR at bearing 1 and 2 respectively. The

reactions at the bearings 2 and 3 due to belt pull (T1+T2) 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 to Radial force

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

First calculate summation of bending movement

$$\Sigma M_R = 0 \quad 3.20$$

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

$$F_r(b) + R_{H1FR}(a + b) = 0 \quad 3.21$$

$$R_{H1FR} = \frac{F_r(b)}{(a+b)} \quad 3.22$$

$$\Sigma F_Y = 0 \quad 3.23$$

$$-F_R = R_{H1FR} + R_{H2FR} = 0 \quad 3.23$$

$$R_{H2FR} = \frac{F_R(a)}{(a+b)} \quad 3.24$$

The reactions due tangential force (F_T)

$$\Sigma M_T = 0 \quad 3.25$$

$$-F_T(b) + R_{H1FT}(a + b) = 0 \quad 3.26$$

$$R_{H1FT} = \frac{F_T(b)}{(a+b)} \quad 3.27$$

$$\Sigma F_Y = 0 \quad 3.28$$

$$-F_T + R_{H1FT} + R_{H2FT} = 0 \quad 3.29$$

$$R_{H2FT} = \frac{F_T(a)}{(a+b)} \quad 3.30$$

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 $\emptyset$), Take $=l/r = 4$, the maximum value of tangential force lies when the crank is at an angle of 35° from 30° to 40° for diesel engines.

$$\sin \emptyset = \sin \theta / (l/r) = \sin 35^\circ$$

Which implies $\emptyset = 8.24^\circ$ we know that thrust in the connecting rod or the force exerted on the crankpin by the piston

$$F_Q = \frac{F_P}{\cos \emptyset} \quad 3.31$$

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

$$F_r = F_Q \cos(\theta + \emptyset) \quad 3.32$$

$$F_t = F_Q \sin(\theta + \emptyset) \quad 3.33$$

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

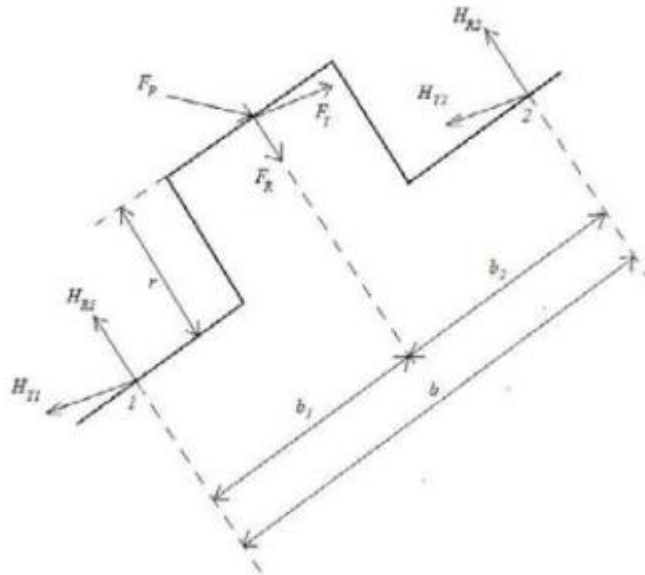


Figure3.12 Reaction at bearing tangential and radial force

$$R_{T1} = \frac{F_t b_1}{b} \quad 3.34$$

$$R_{T2} = \frac{F_t b_2}{b} \quad 3.35$$

Reaction at bearing 1&2 due to radial force

$$R_{r1} = \frac{F_r b_1}{b} \quad 3.36$$

$$R_{r2} = \frac{F_r b_2}{b} \quad 3.37$$

The bending moment at the center of the crankpin,

$$M_b = R_{r2} b_2 \quad 3.38$$

Twisting moment

$$T_M = R_{T2}(r) \quad 3.40$$

Equivalent twisting moment on the crankpin

$$T_e = \sqrt{(M_{CR})^2 + T_1^2} \quad 3.41$$

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

$$M_{EV} = \sqrt{(k_b M_C)^2 + \frac{3}{4}(K_t T_C)^2} \quad 3.42$$

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

K_T = combined shock and fatigue factor for torsion (Take $K_T = 1$)

$$M_{ev} = \frac{\pi}{32} (d_c)^3 \sigma_v \quad 3.43$$

$$\sigma_v = \frac{32}{\pi} * \frac{M_{EV}}{(d_c)^3} \quad 3.44$$

$$T_E = \frac{\pi}{16} * (d_c)^3 * \tau \quad 3.45$$

$$\tau = \frac{16T_E}{\pi d^3} \quad 3.46$$

3.7.4 Strain on crankshaft

Based on these materials, the strain of the shaft would analysis by analytical as well as numerical methods.

$$\varepsilon = \frac{\delta_v}{E} \quad 3.47$$

Where ε = strain

δ_v = Stress

E = Young's modulus

CHAPTER FOUR

4.1 Result and discussion

4.1.1 Analytical results

$$\text{Clarence volume } V_c = V_2 = \frac{\pi_s B^2}{4(r_c - 1)}$$
$$9.793 \times 10^{-5} m^3$$

$$\text{Displacement } v_{d=\frac{\pi B^2 2r}{4}}$$
$$1.86 \times 10^{-3} m^3$$

$$\text{Maximum volume } v_1 = v_{max} = v_d + v_c$$
$$1.959 \times 10^{-3} m^3$$

$$\text{Maximum pressure of gases } p_1 v_1^\gamma = p_2 v_2^\gamma$$
$$7.338 \times 10^6 mpa$$

4.1.2 The crank at the dead center

The force acting on the cylinder due to gas pressure

$$F_s = \frac{\pi B^2}{4} P$$
$$104.920 \text{KN}$$

The reaction force RH1 and RH2 in bearing 1 and 2

$$R_{H1} = F_P - \frac{F_P b_1}{b_1 + b_2}$$
$$52.2980 \text{ KN}$$

$$R_{H2} = \frac{F_P * b_1}{b_1 + b_2}$$

52.2980 KN

The maximum bending moment at center of the crank pin

$$M_{maxb} = R_{r2}b_1$$

4249.21KN

The bending stress of crank pin

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

101N/mm²

4.1.3 Crank at an angle of maximum twisting moment

The thrust force exerted on the crankpin by the piston

$$F_Q = \frac{F_P}{\cos \phi}$$

106.01KN

Thrust on the crank shaft can be split into the radial component and tangential

$$F_r = F_Q \cos(\theta + \phi)$$

76.433KN

Tangential Component

$$F_t = F_Q \sin(\theta + \phi)$$

71.877KN

Reaction at bearing 1&2 due to tangential force

$$R_{T1} = \frac{F_t b_1}{b}$$

36.423 KN

$$R_{T2} = \frac{F_t b_2}{b}$$

$$36.199\text{KN}$$

Reaction at bearing 1&2 due to radial force

$$R_{r1} = \frac{F_t b_1}{b}$$

$$38.729\text{KN}$$

$$R_{r2} = \frac{F_t b_2}{b}$$

$$38,490\text{KN}$$

$$\text{Bending moment center of crank pin} = R_{r2} \times b_1$$

$$3146.731\text{KN-mm}$$

$$\text{Twisting moment } T_M = R_{T1}(r)$$

$$2367.495\text{KN-mm}$$

$$\text{Equivalent twisting moment on the crankpin } T_e = \sqrt{(M_{CR})^2 + T_1^2}$$

$$3937.886\text{KN}$$

According to distortion energy theory, the Von-Misses stress induced in the crank-pin is (Mr. Anant B. et al,)

$$M_{EV} = \sqrt{(k_b M_e)^2 + \frac{3}{4}(K_t T_m)^2}$$

$$4439.676\text{KN/mm}$$

$$\text{Von misses stress induces in crankpin } \sigma_v = \frac{32}{\pi} \frac{M_{EV}}{(d_c)^3}$$

$$107.193\text{N/mm}^2$$

Shear stress induce in crankpin

$$\tau = \frac{16T_E}{\pi d^3}$$

$$47.5\text{N/mm}^2$$

4.1.4 Strain on crankshaft

Based on these materials, the strain of the shaft would analysis by analytical as well as numerical methods.

$$\varepsilon = \frac{\delta_v}{E}$$

Where δ_v =Stress

E =Younges modules

ε =streaan

Table 4.1property of selected material

selected materials	young's modulus
Aluminum alloy	710000Mpa
Cast iron	178000
Alloy steel 4130	2060000
Cast steel	200000Mpa

Strain for aluminum alloy

$$\frac{107.2\text{Mpa}}{700000\text{Mpa}}=0.000153$$

Strain for cast iron

$$\frac{107.2\text{Mpa}}{178000\text{Mpa}}=0.000602$$

Strain for alloy steel

$$\frac{107.2\text{Mpa}}{206000\text{Mpa}}=0.000520$$

Strain for cast alloy steel

$$\frac{107.2\text{Mpa}}{120000\text{Mpa}}=0.000536$$

4.2 Static FEA result of crank shaft

Static analysis of crankshaft from ANSYS 2020R2. Finite element analysis was performed to obtain the variation of the stress magnitude at critical locations. Total deformation, von-missis stress, equivalent strain, Shear stress, factor of safety of alternative materials.

Total deformation of alternative materials

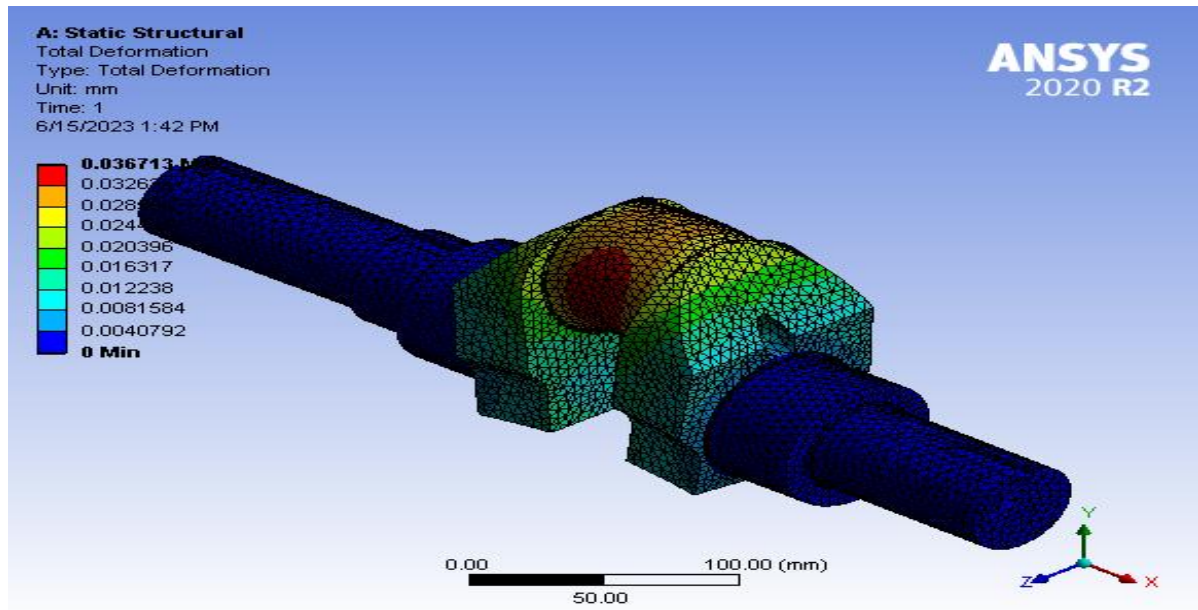


Figure 4.1 Total deformation of aluminum alloy

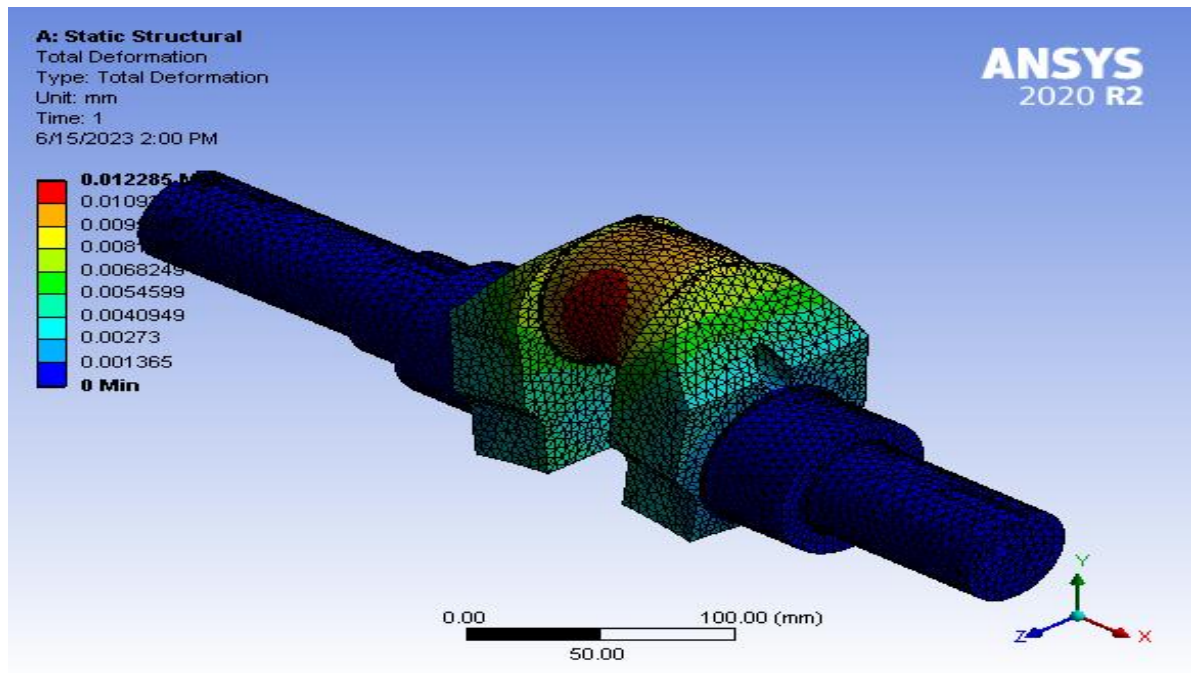


Figure 4.2 Total Deformation of alloy steel

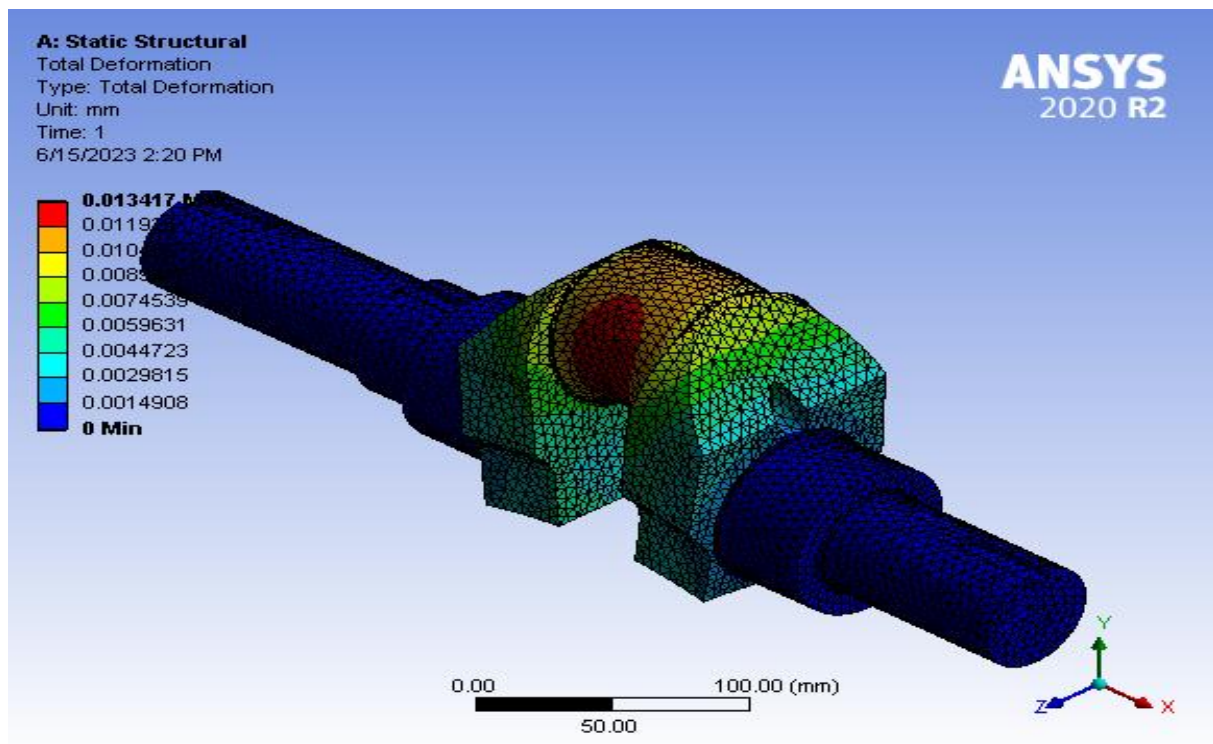


Figure 4.3 Total Deformation of cast iron

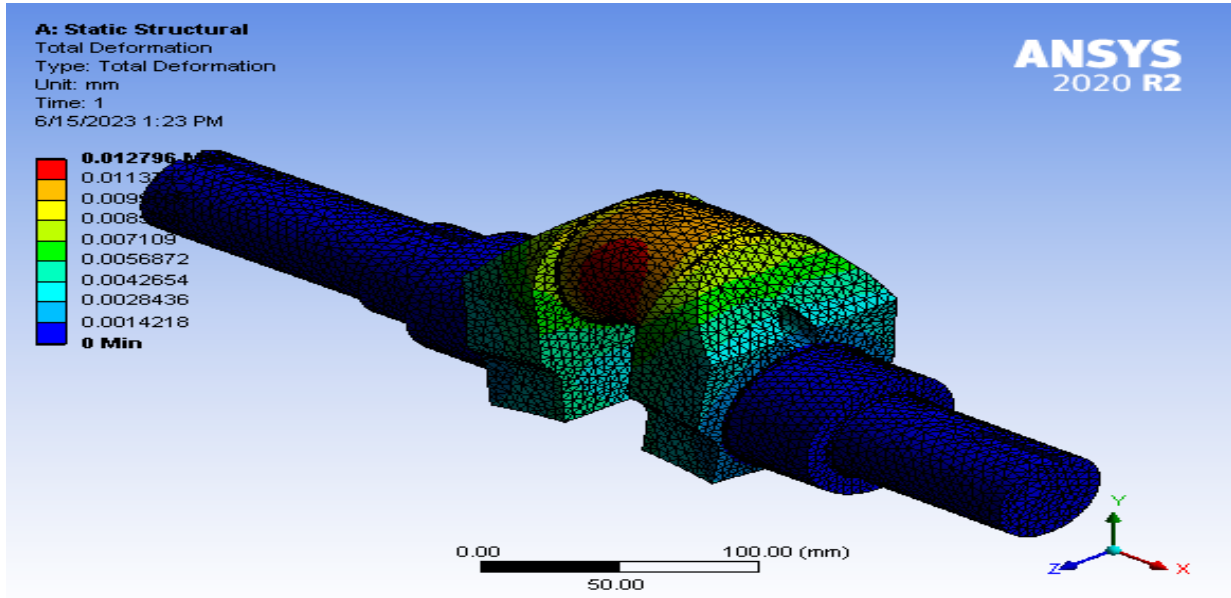


Figure 4.4 Total deformation of cast alloy steel

The total Deformation is the degree to which a structural element is displaced under a load of 104920N. The deformation analysis was performed in four different materials to evaluate the maximum deformation position on the crankshaft. The distribution of deformation under maximum stress is shown from figure 4.1-4.4. The maximum deformation value under maximum stress was in the position of the crankpin of the crank shaft. The maximum deformation values of the crankshaft in four different materials wear between 0.032-010 maximum deformations occurs in aluminum alloy and minimum deformation occurred in alloy steel

equivalent von-miss stress of selected materil

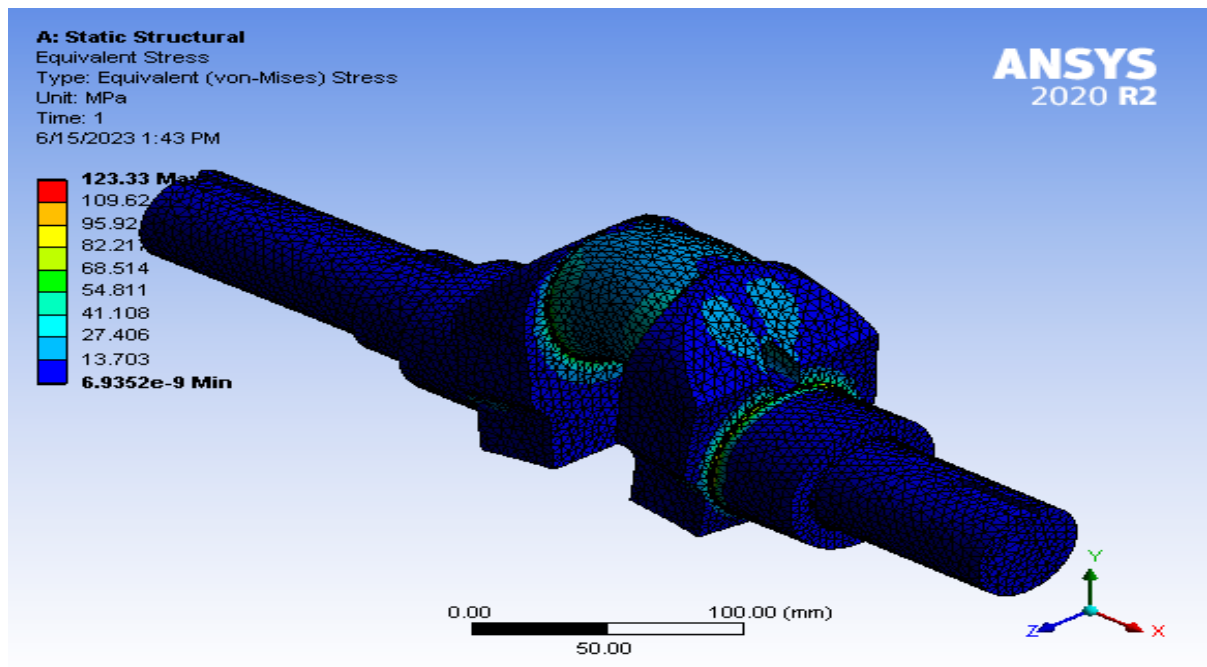


Figure 4.5 equivalent von-miss stress aluminum alloys

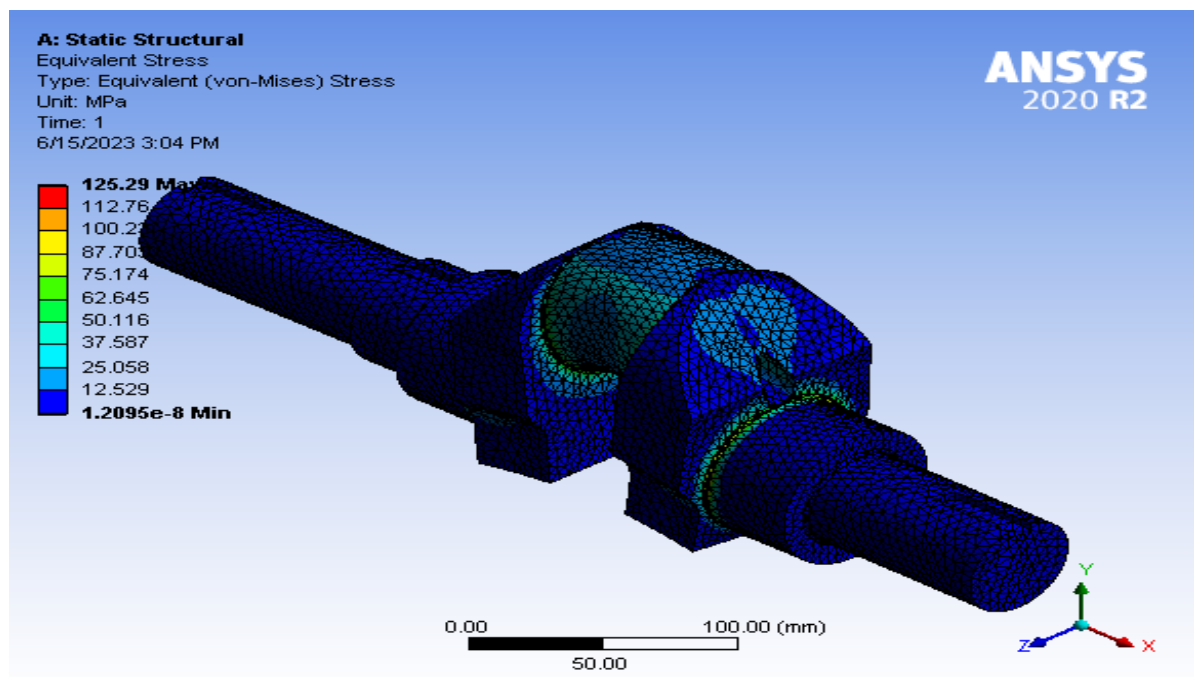


Figure 4.6 equivalent von-misses stress of alloy steel

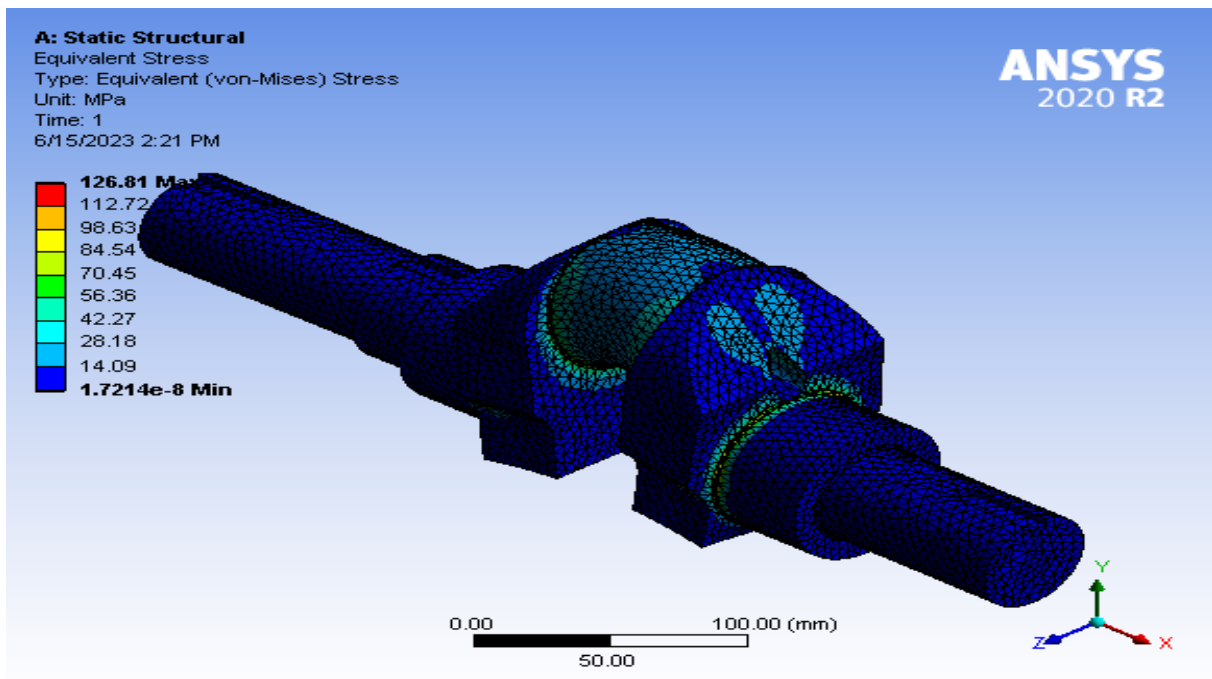


Figure 4.7 Equivalent Von- miss stress of cat iron

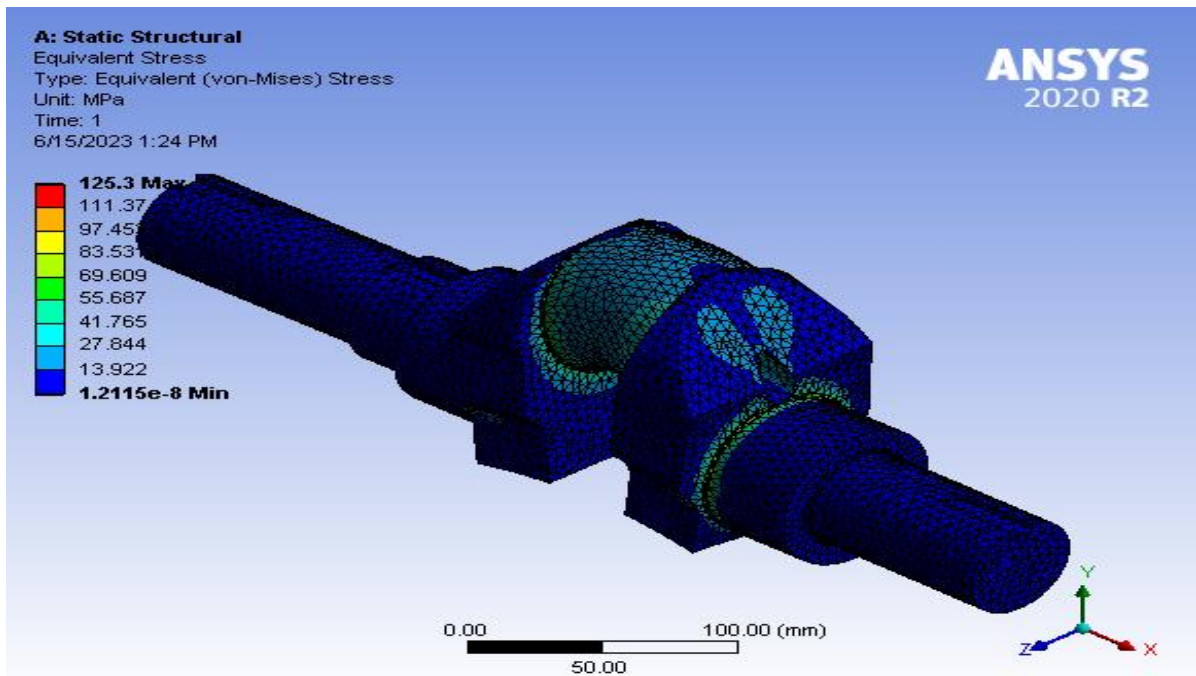


Figure 4.8 Equivalent von-miss of cast alloy steel

The von mises stress analysis results was shown between figures 4.5-4.8. In the analysis results, the maximum von mises stress concentration occurred in the journals fillets. This implies, when the stress values was beyond the yield stress of the materials the crankshaft was failed during load applied of 104920N. However, the values of these von-misses stress were far less than the yield stress of each material. The von misses stresses of the crankshaft in four materials were between 109.62Mpa-112.76Mpa. The von misses stress of 109.62 Mpa is the minimum stress which was occurred in a material of aluminum alloy and the maximum von misses stress was occurring in a material of alloy steel which implies that the stress concentration of alloy steel was greater than all three materials

Static FEA result of shear stress crank shaft

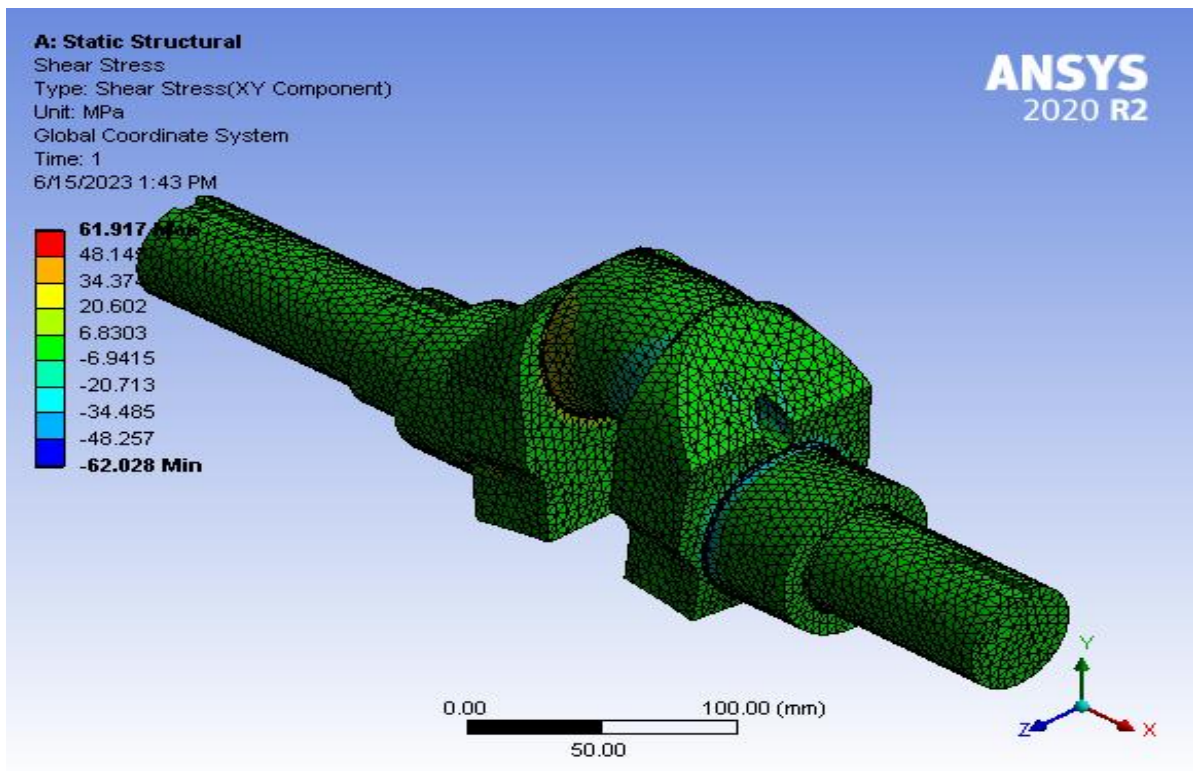


Figure 4.9 shear stress aluminum alloy

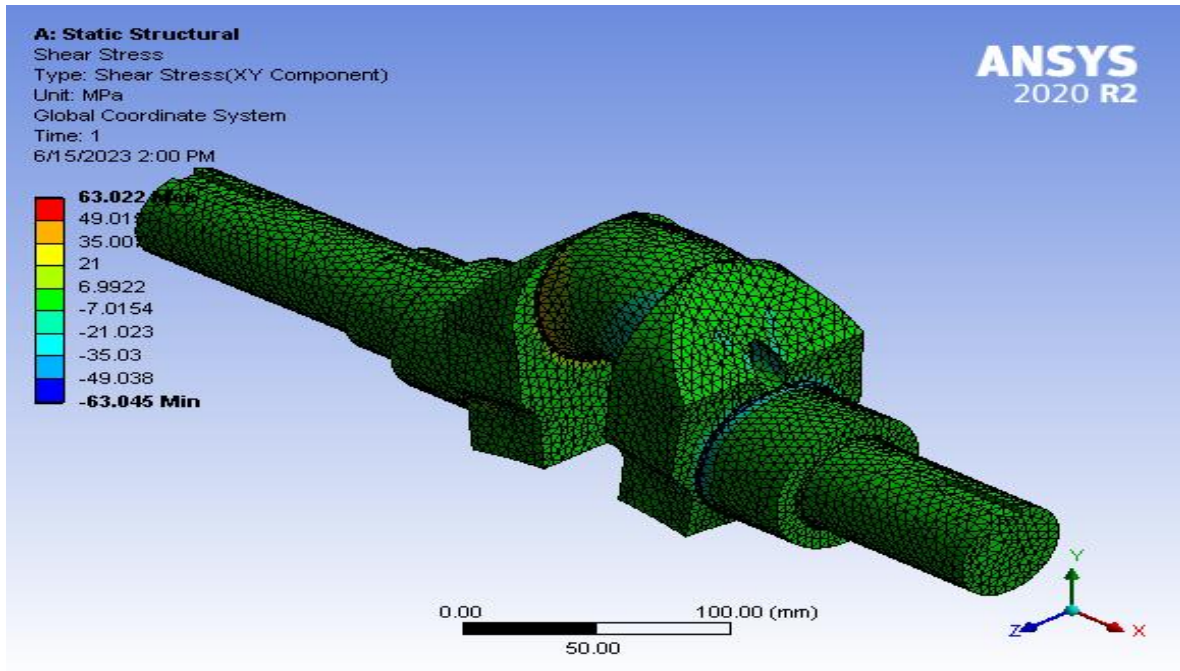


Figure 4.10 Shear stress of alloy steel

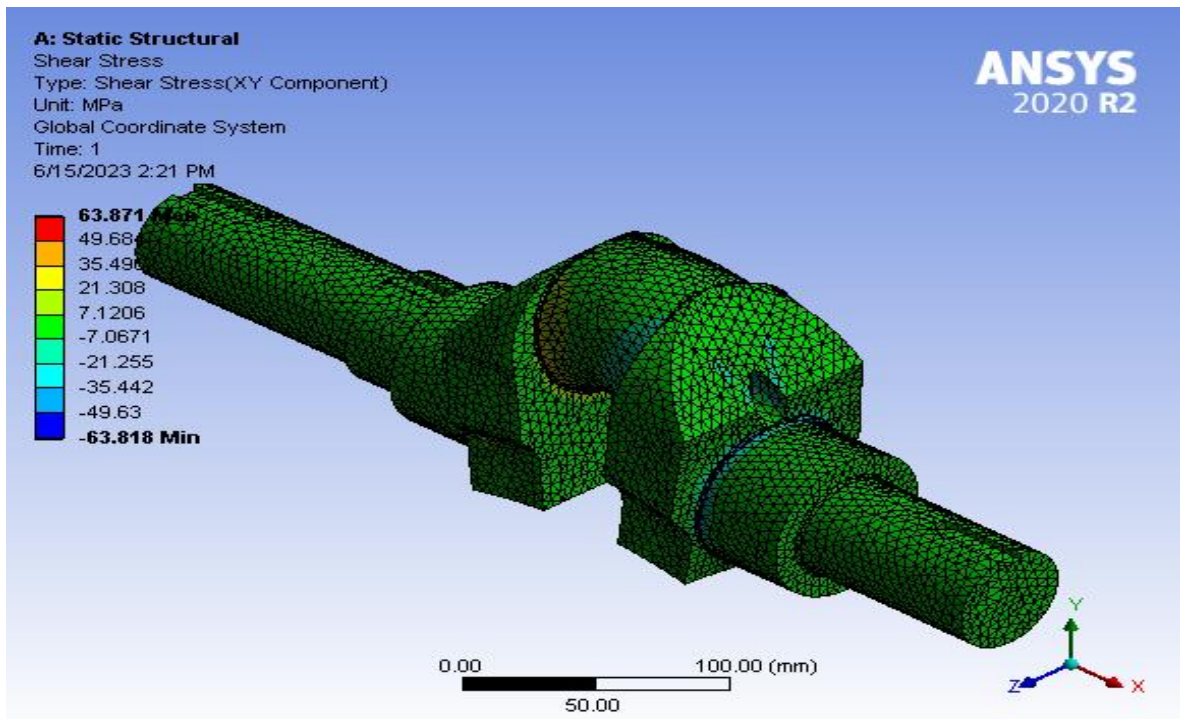


Figure 4.11 Shear stress of cast iron

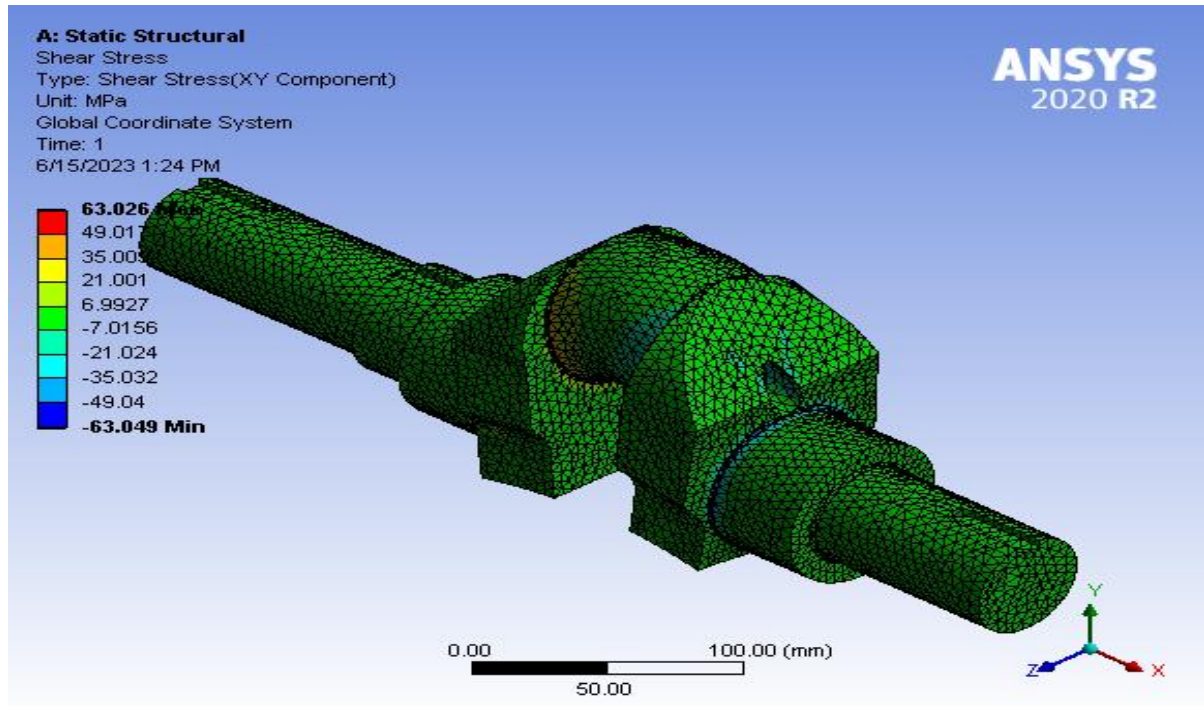


Figure 4.12 Shear Stress of cast alloy steel

The shear stress was analyzing point out the maximum shear stress position. The shear stress analysis results were shown between figures 4.9-4.12. In the analysis results, the shear stress concentration occurred in the journals fillets; the values of this shear stress were far less than the material shear stress of each material. The shear stresses of the crankshaft in four materials were between 48.15Mpa-49.49Mpa. The shear stress of 48.15Mpa is the minimum shear stress which was occurred in a material of aluminum alloy and the maximum shear stress was occurring in a material of alloy steel which implies that the shear stress concentration of alloy steel was greater than all three materials

Static FEA result of equivalent static strain of crank shaft

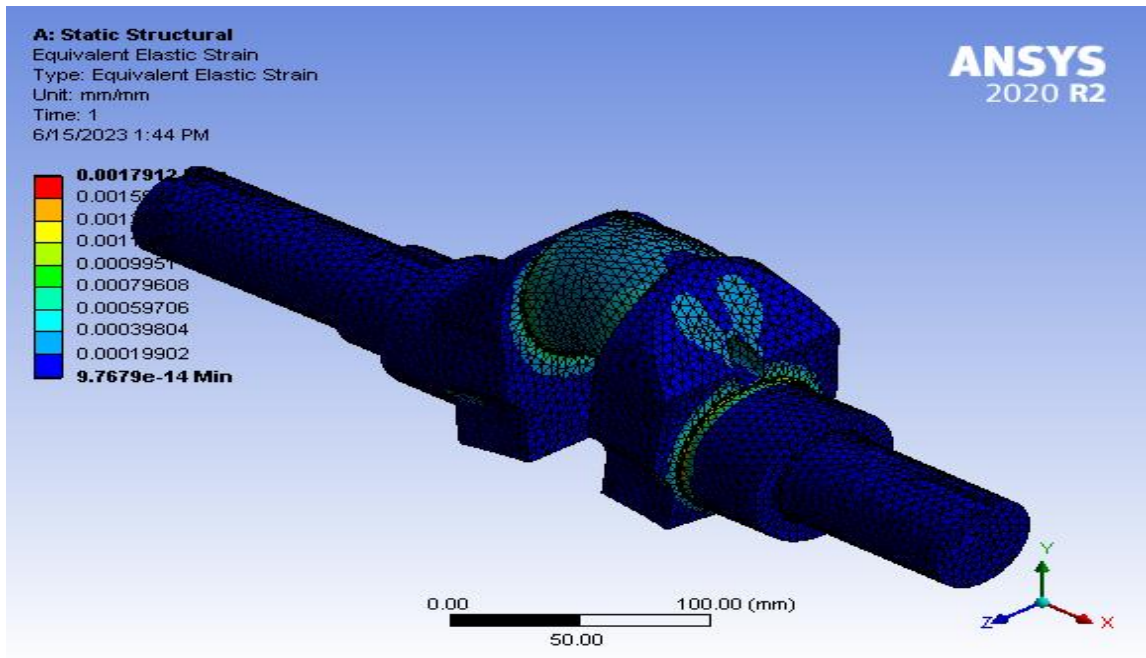


Figure 4.13 Equivalent elastic Strain of aluminum alloy

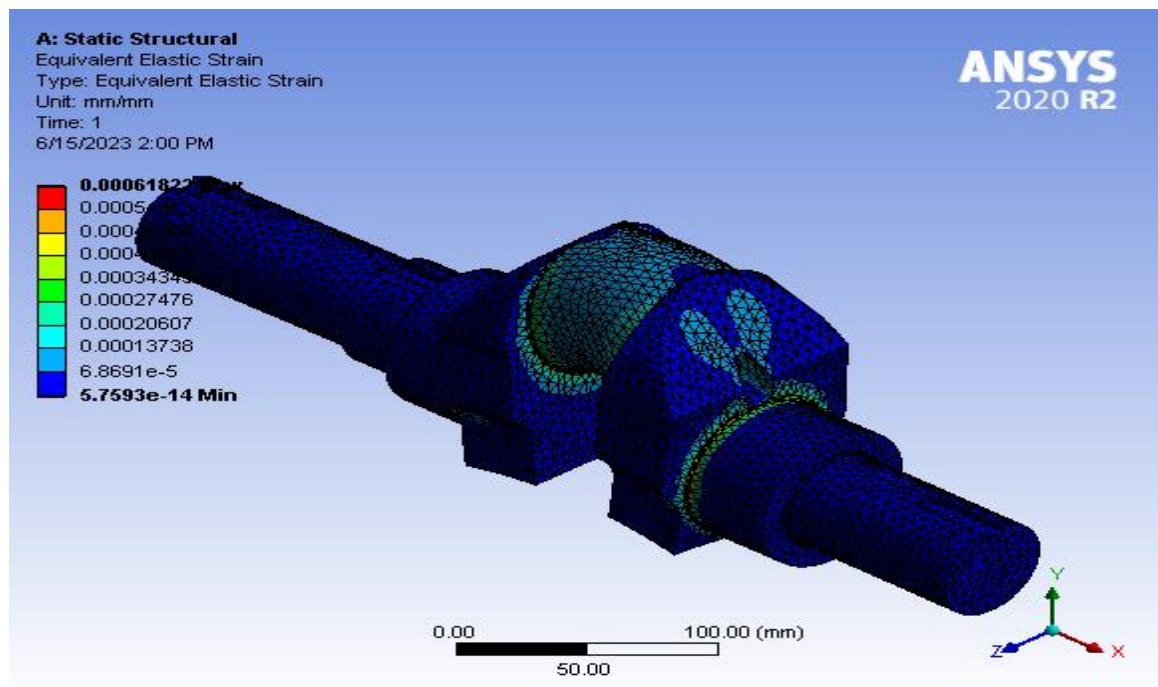


Figure 4.14 Equivalent elastic Strain of alloy steel

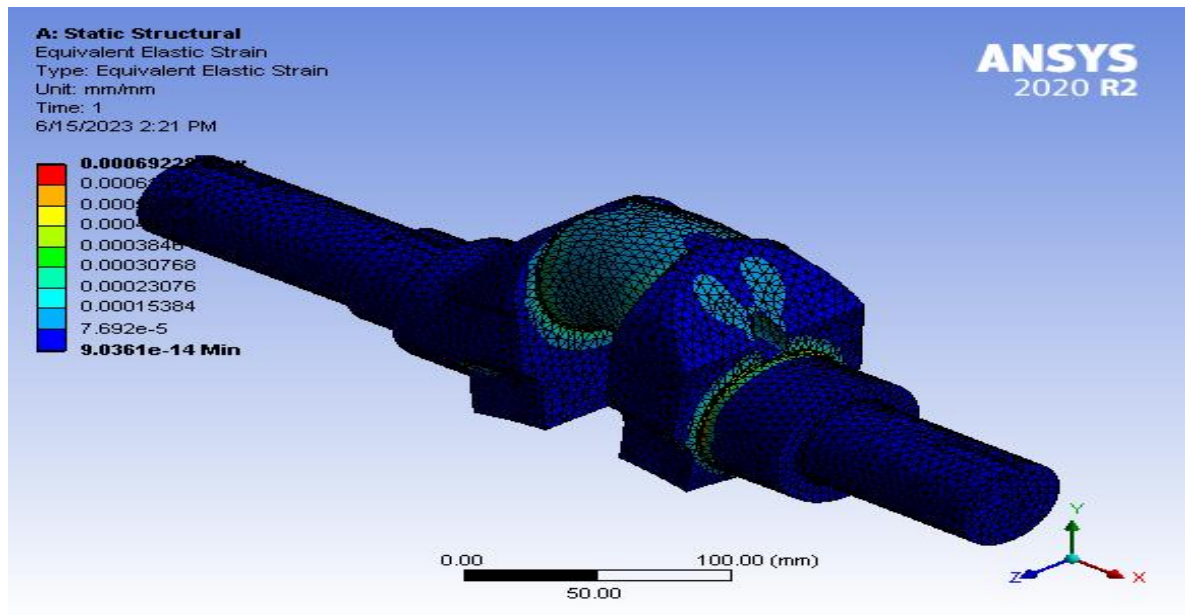


Figure 4.15 Equivalent Elastic strain of cat iron

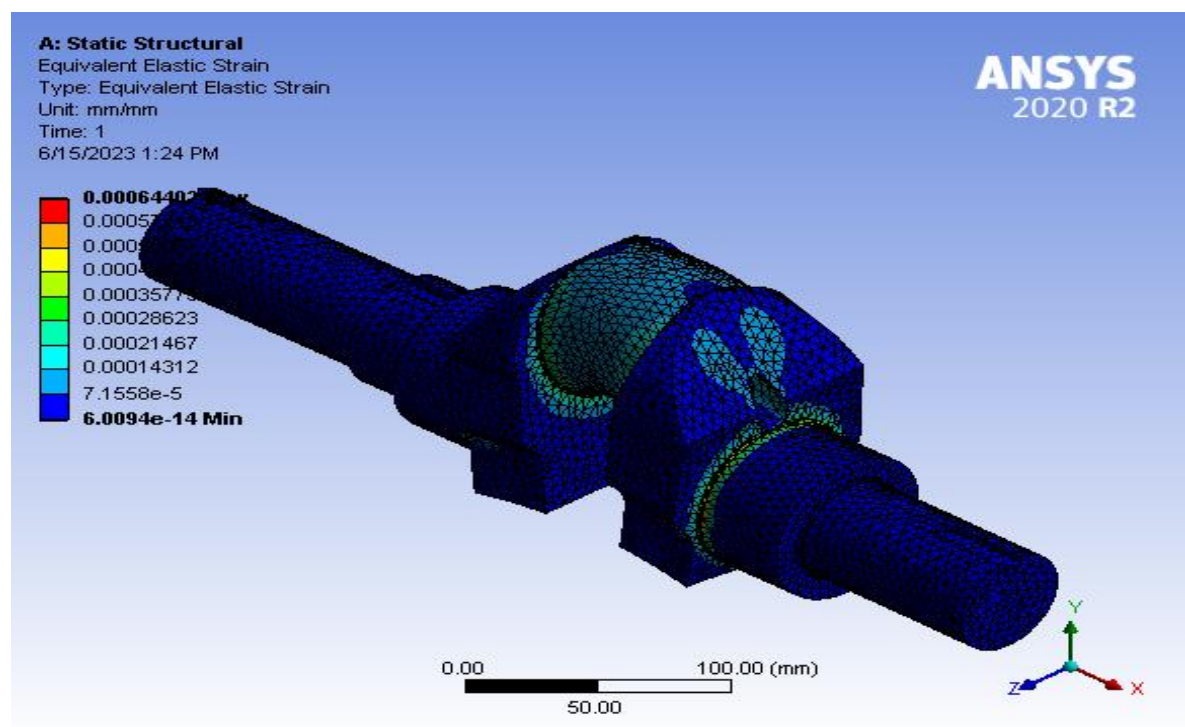


Figure 4.16 Equivalent Strain of cast alloy steel

The crankshaft had the maximum elastic strain in the journal fillet area. The maximum strains were analysis from 0.000602-0.0015 in four materials. From these the minimum stain was appeared in alloy steel and the maximum strain was appeared in aluminum alloy

Static FEA result of equivalent factor safety crank shaft

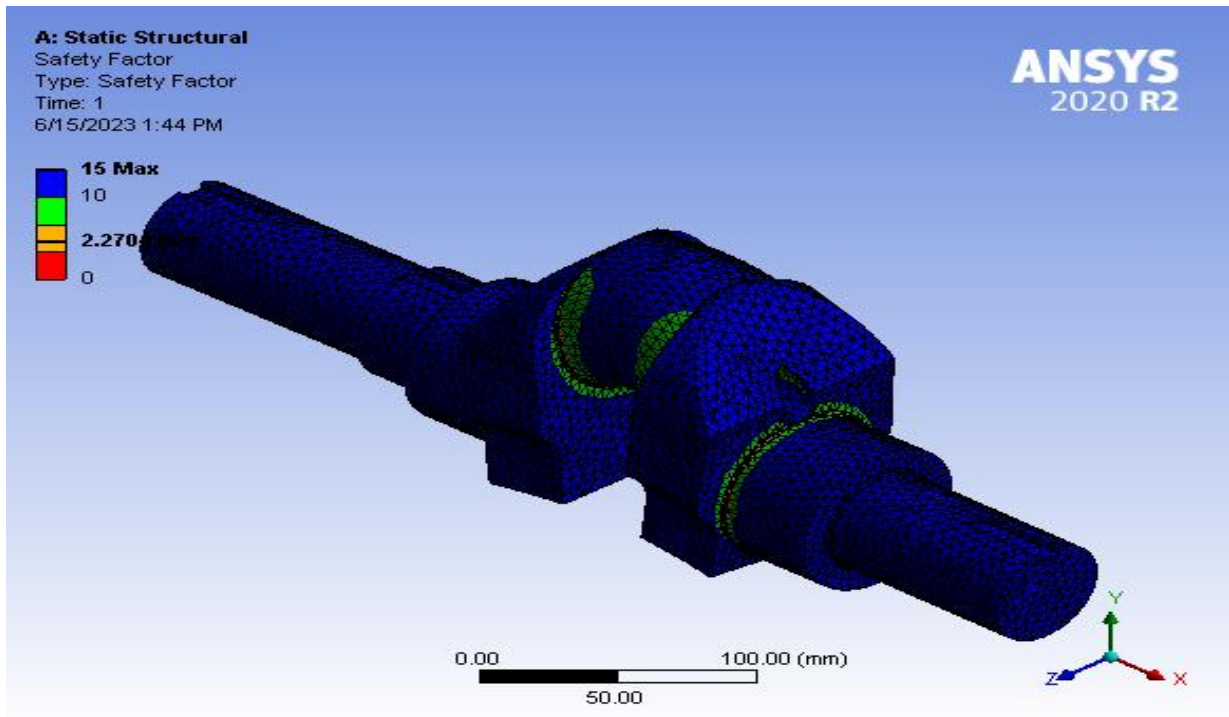


Figure 4.17 Safety factor of aluminum alloy

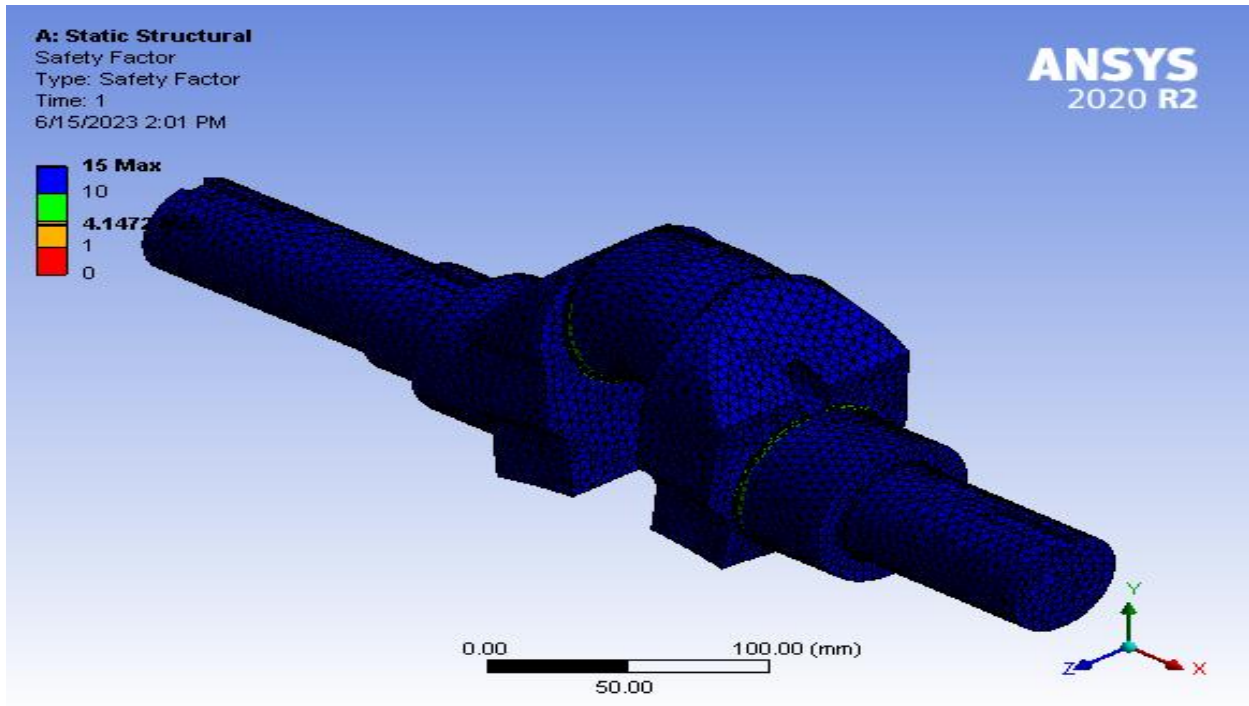


Figure 4.18 Safety factor of alloy steel

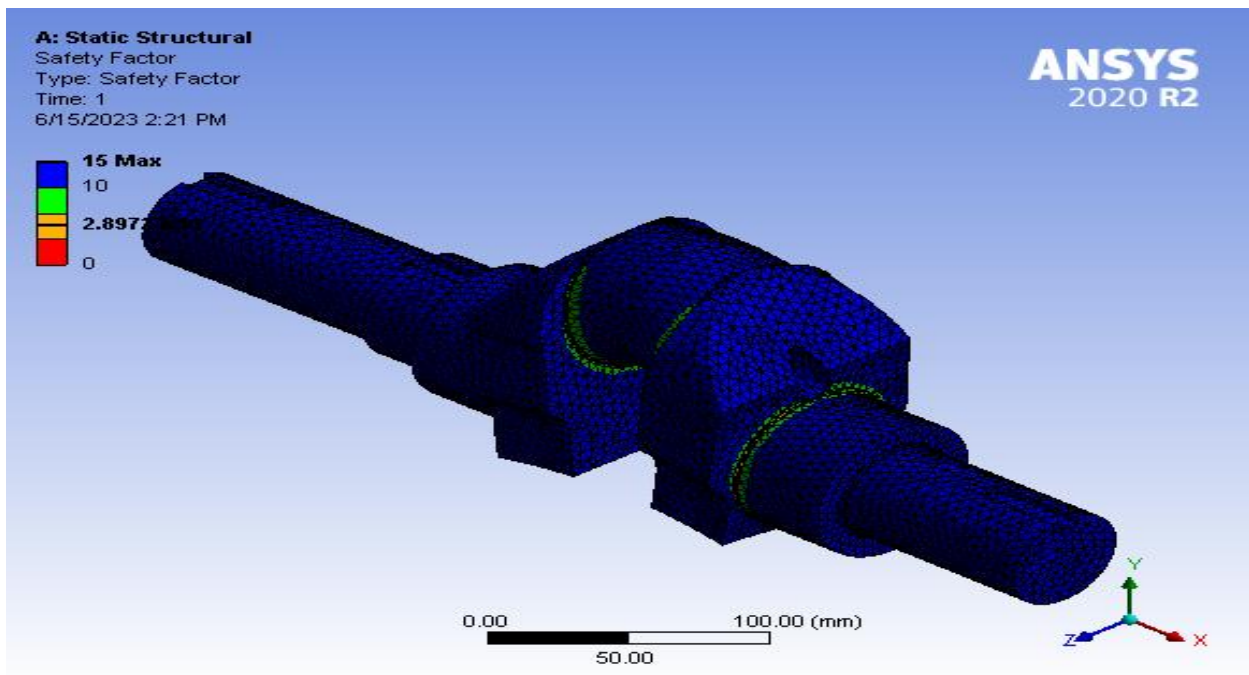


Figure 4.19 Safety factor of cast iron

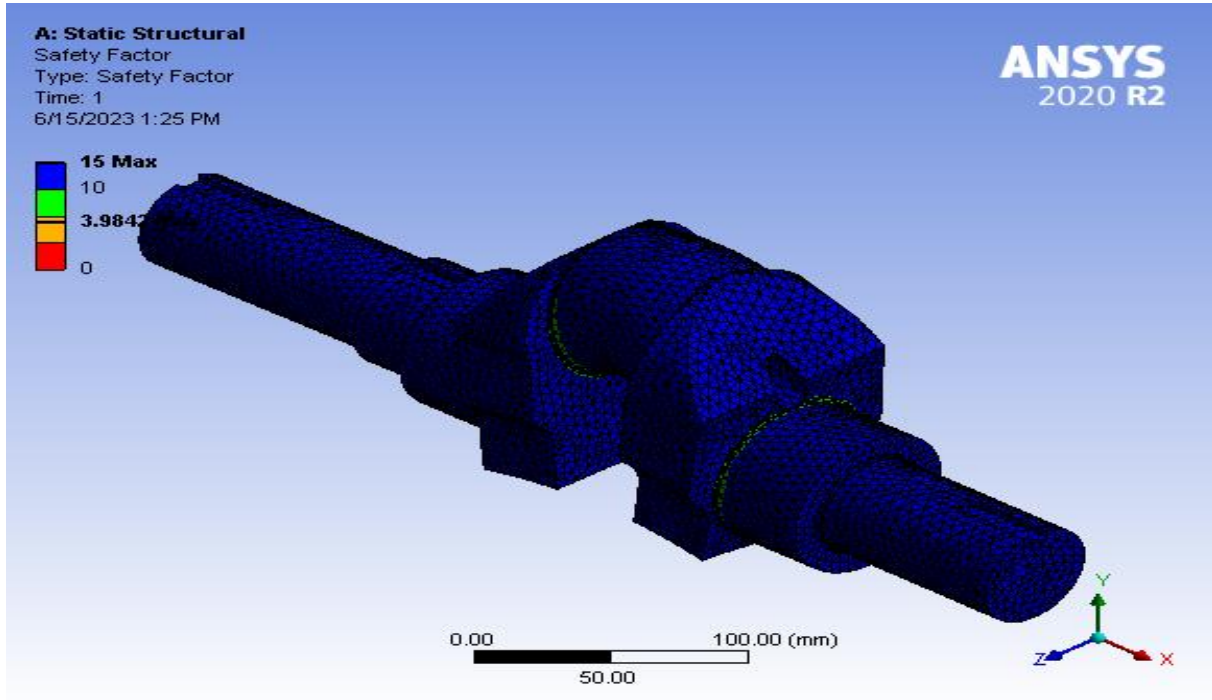


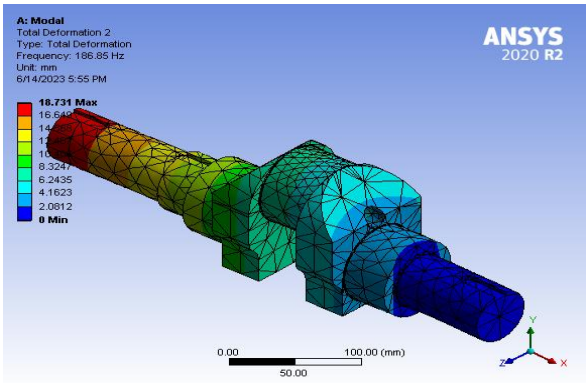
Figure 4.20 Factor of safety of cast alloy steel

Figure 4.17-4.20 shows the factor of safety (FOS) obtained in the crankshaft under the applied load of 104920N on the crankpin. As shown in the figure above the minimum factors of safety were varied from 2.2 - 10 depending on the four materials. These factors of safety in each material mean that the materials have safety to withstand the load without failure up to the load times the safety factor. So the material alloy steel had more chance withstand more loads than the other. However, this material is weighted than the other materials. Next to alloy steel, the cast iron had the chance to carry more than the current applied loads without fracture. So cast iron is the optimum material to use the for crankshaft which is cheap in cost and light in weight.

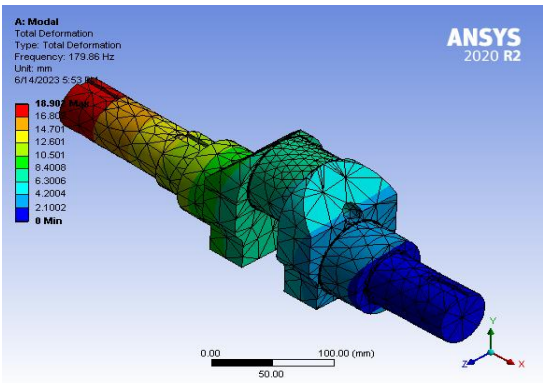
4.2.1 Modal Analysis Results of Crankshaft

Modal analysis is the process of determining the inherent dynamic characteristics of a system in forms of natural frequencies and damping factors In this paper, we obtain the first 7 order natural frequency and main vibration mode of the crankshaft, and try to check these frequencies in practice, so as to prevent the production resonance.

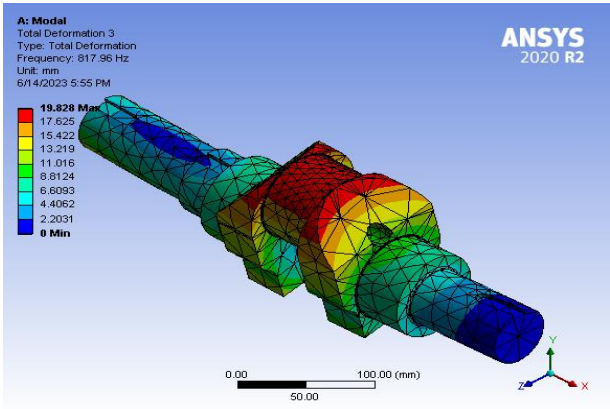
Modal analysis of aluminum alloy



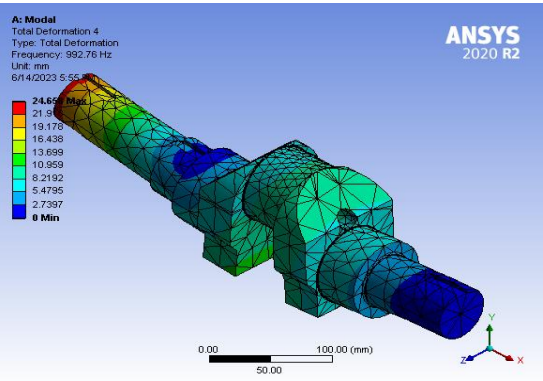
First order modal



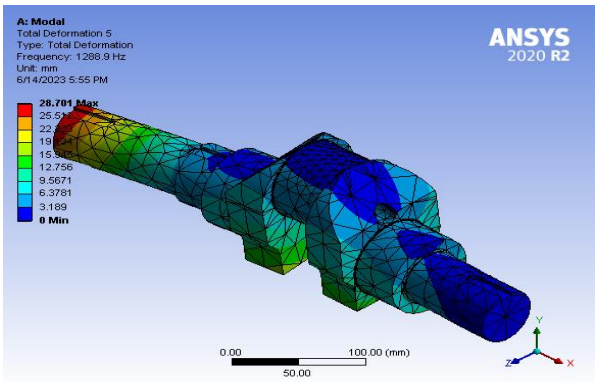
second order modal



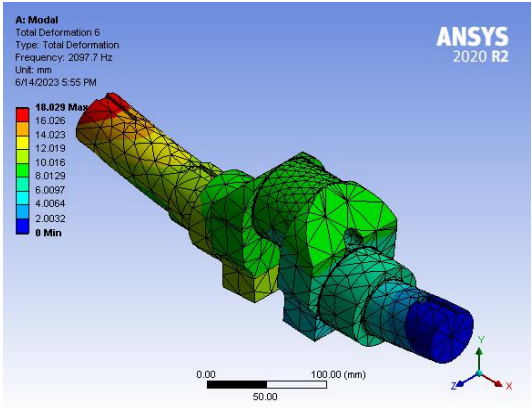
Thired ordar modal



forthe ordar modal mod



Fifth order modal

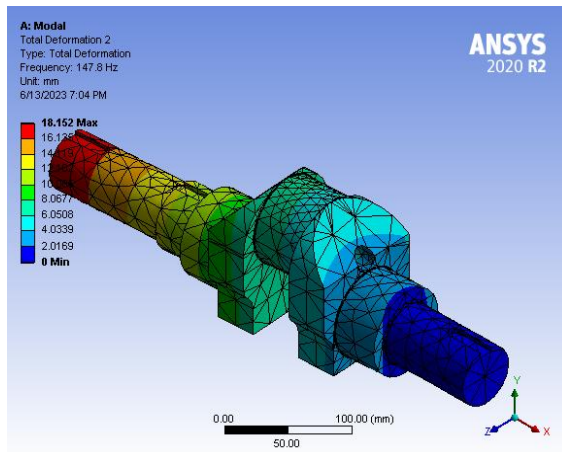


six order model

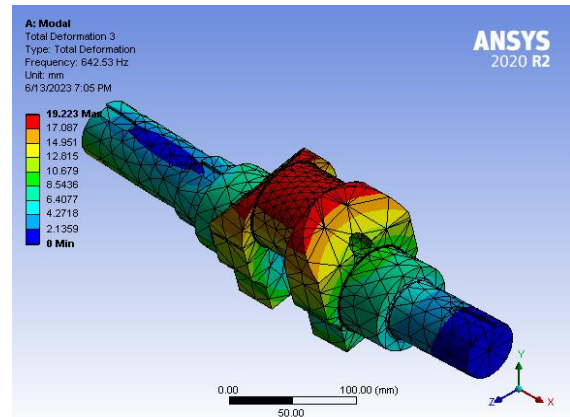
Figure 4.21 modal analysis aluminum alloys

The vibration characteristics (the mode shape and the natural frequency) of these aluminum alloy materials were found from the modal analysis of crankshaft. The first modal of 176.65 hz and the six modal 2058.7 hz

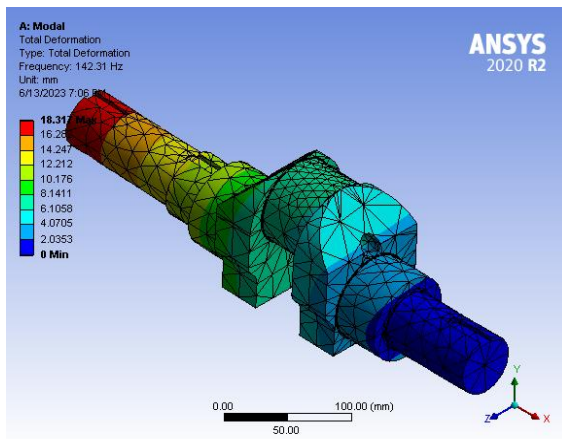
Modal analysis of alloy steel



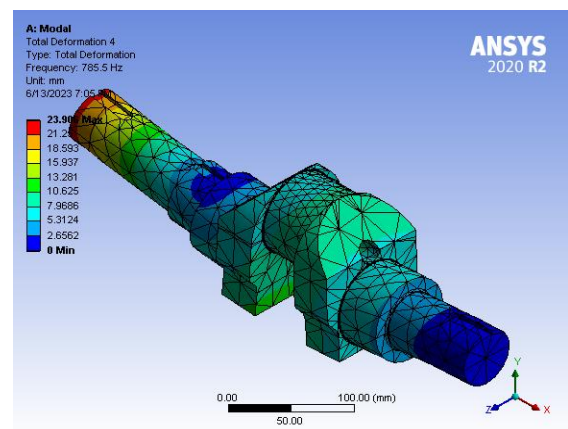
First order modal



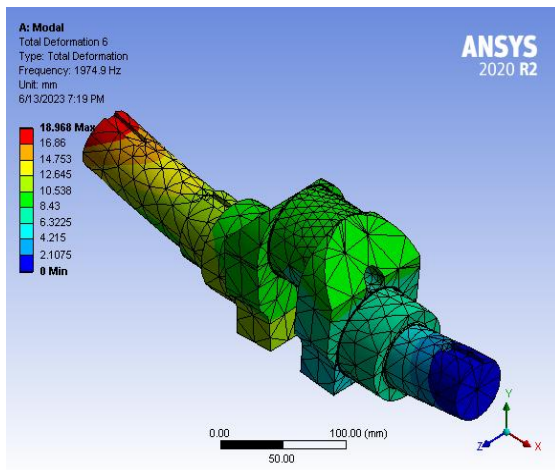
second order modal



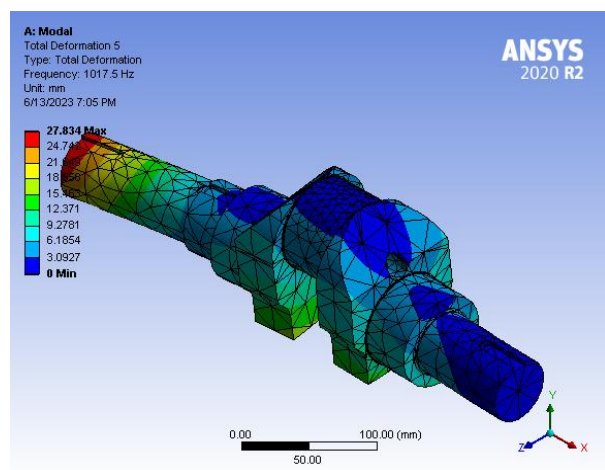
Third order modal



forth order modal



Fifth order mode

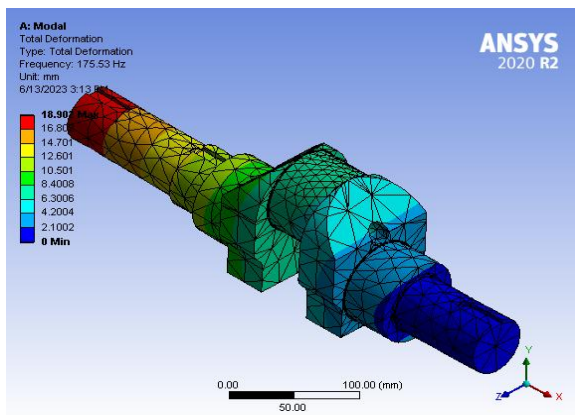


sixe order mode

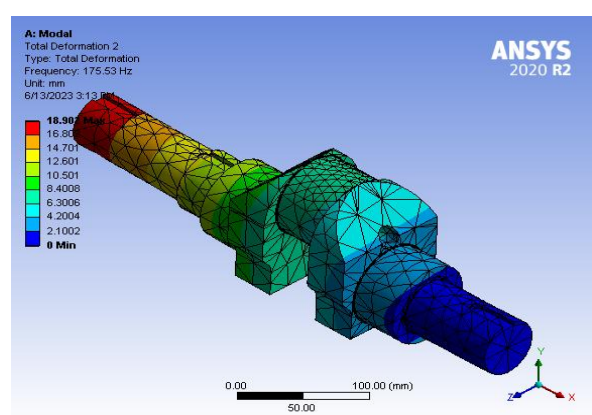
Figure 4.22 model analyses of alloy steel

The vibration characteristics (the mode shape and the natural frequency) of these alloy steel materials were found from the modal analysis of crankshaft. The first modal of 179.86 Hz and the six modal 2097.7 Hz

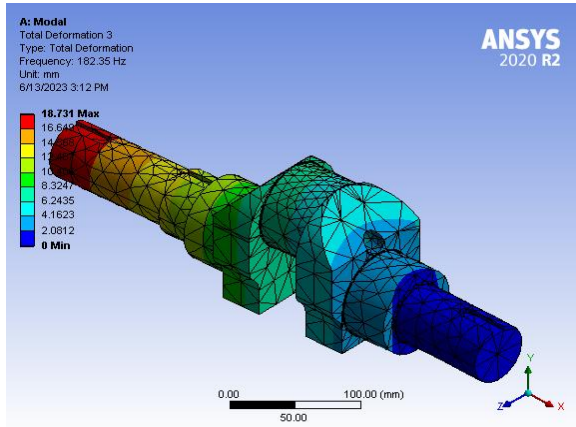
Modal analysis of cast iron



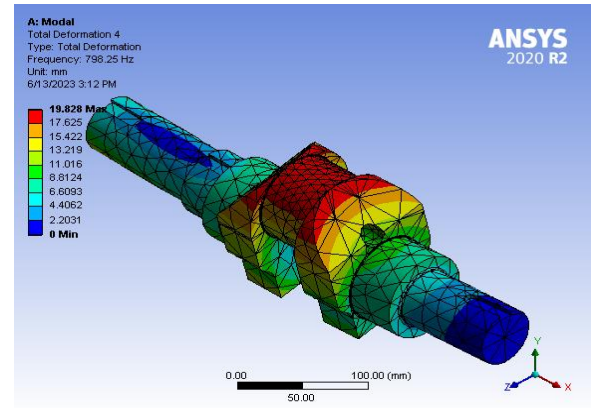
Frist order modal



second order modal



Third order mod



forth order mod

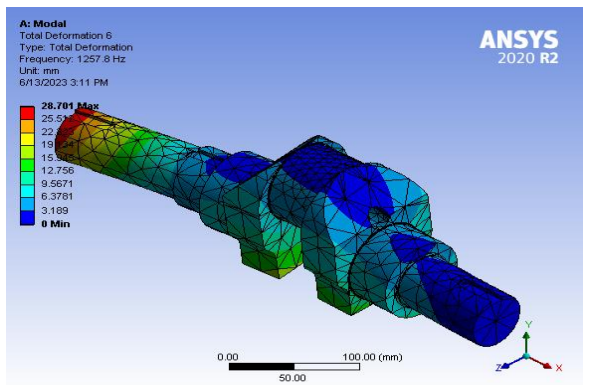
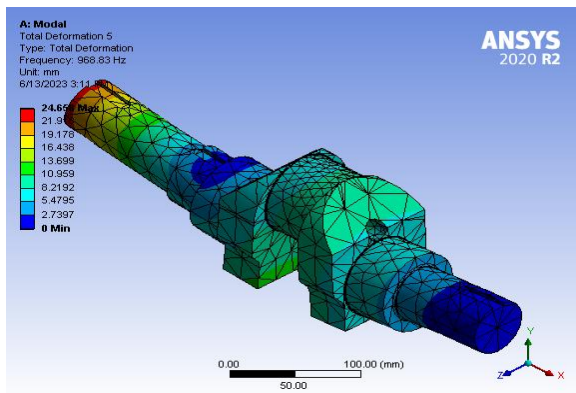
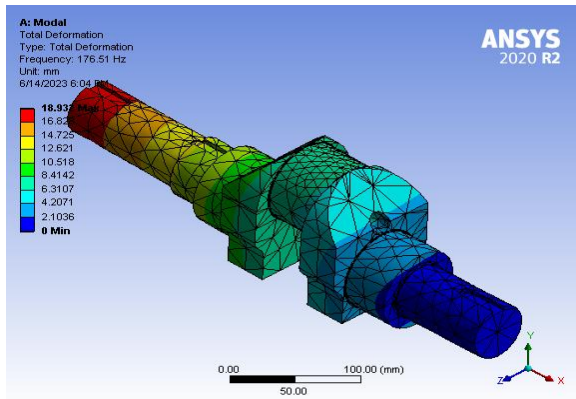


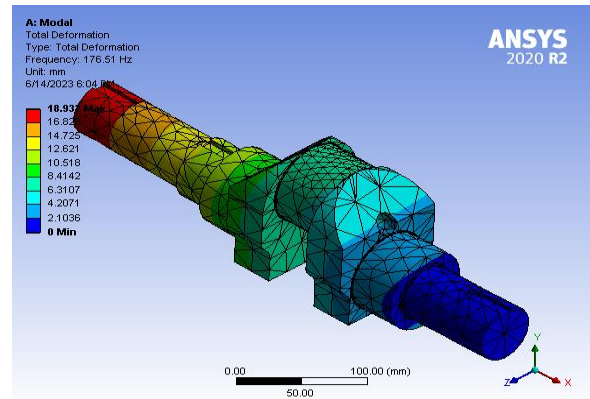
Figure 4.22 modal analysis of cast iron

The vibration characteristics (the mode shape and the natural frequency) of these cast iron materials were found from the modal analysis of crankshaft. The first modal of 173 Hz and the six modal 2002

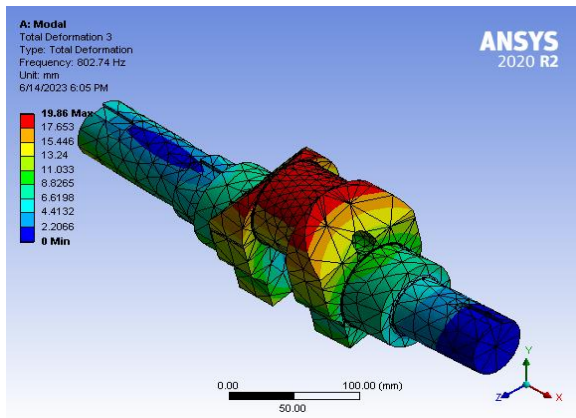
Modal analysis of cast alloy steel



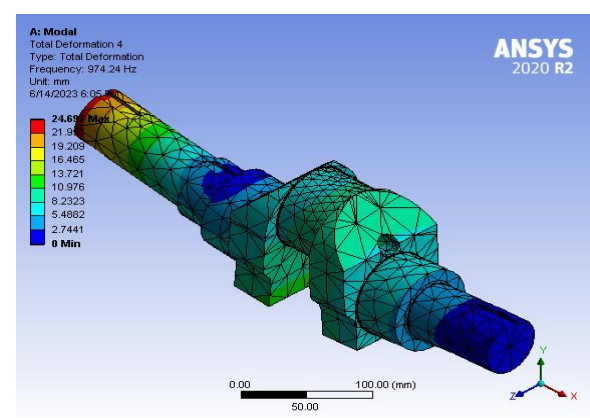
First order modal



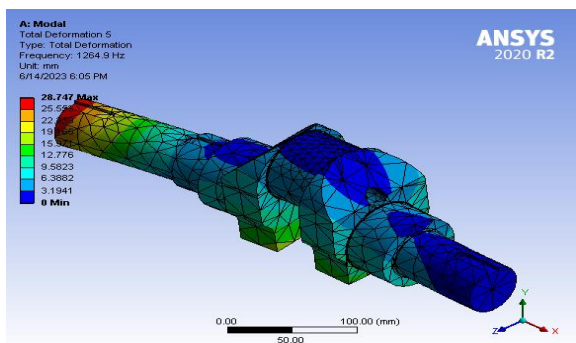
second order modal



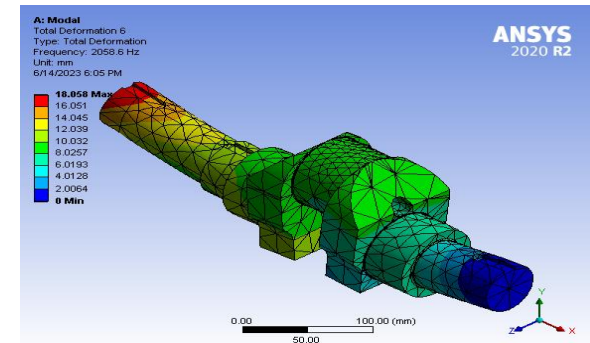
Third order modal



fourth order modal



Fifth order modal



six order modal

Figure 4.24 cast alloy steel

The vibration characteristics (the mode shape and the natural frequency) of these cast alloy steel materials were found from the modal analysis of crankshaft. The first modal of 176.51 Hz and the six modal 2090 Hz

Table4.2: The modal analysis results in four different materials

Modal	Frequency in Hz			
	Aluminum alloy	Cast iron	Alloy steel	Cast alloy
1	176.65	173.42	179.86	176.51
2	183.37	180.42	186.85	183.37
3	793	798.39	817.96	802.74
4	975.1	967.57	992.76	950.23
5	1260.8	1305.7	1288	974.24
6	2058.7	2039.2	2097.7	1264.9

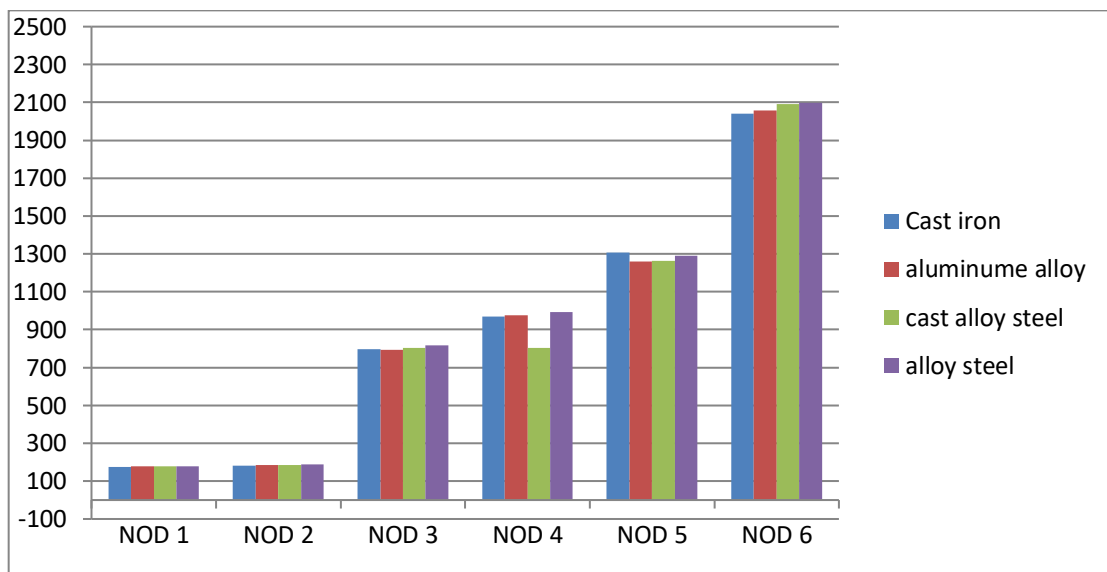


Figure4. 40 Variations of modal frequencies of the four crankshafts.

The vibration characteristics (the mode shape and the natural frequency) of these materials were found from the modal analysis of crankshaft. The only lower frequency will be the greatest source of the engine which vibrates. The natural frequencies of the Materials were presented in table 4.2. The different mode shapes of the crankshaft were as shown in figure 4.21-4.24. The lowest frequency was found to be 173 Hz from four materials and in the

different vibration mode shapes. The speed of the engine is 1600 rpm at which the base frequency of the engine 27.2Hz. The base frequency of the engine was far less than the first natural frequency which was found from the results. So, the possible action of resonance of the crankshaft is very low and crankshaft is not running in critical speed.

4.3 Topology optimization

4.3.1 Mesh generation

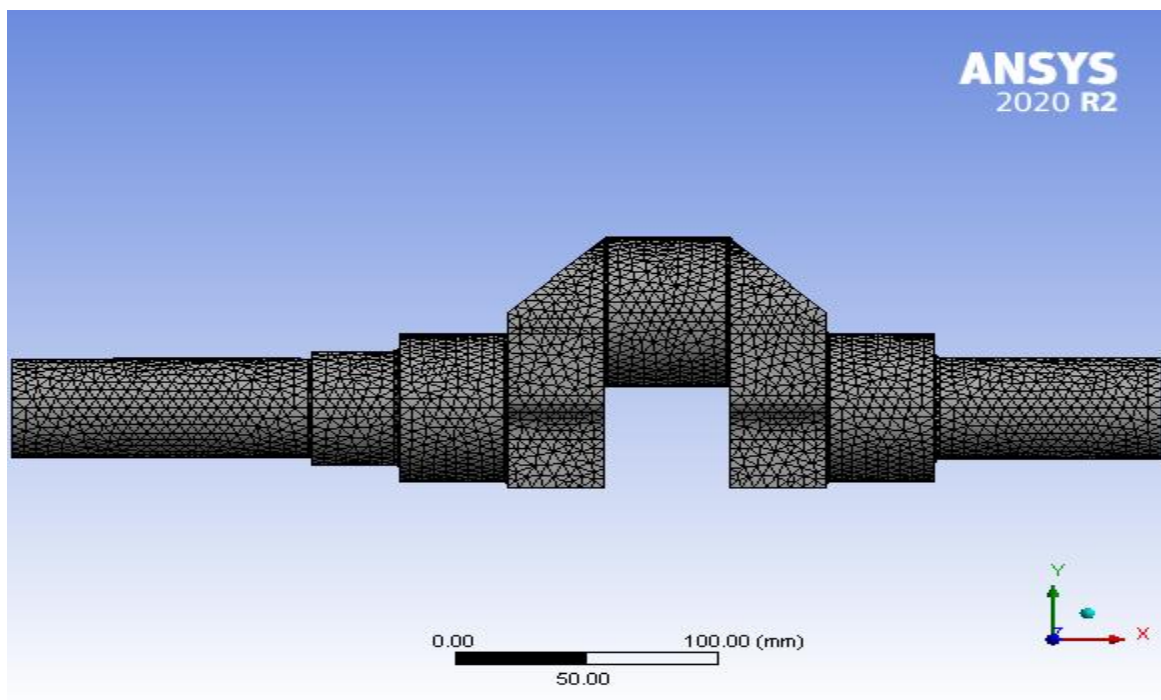


Figure 4.25 Meshes generation of original crank shaft

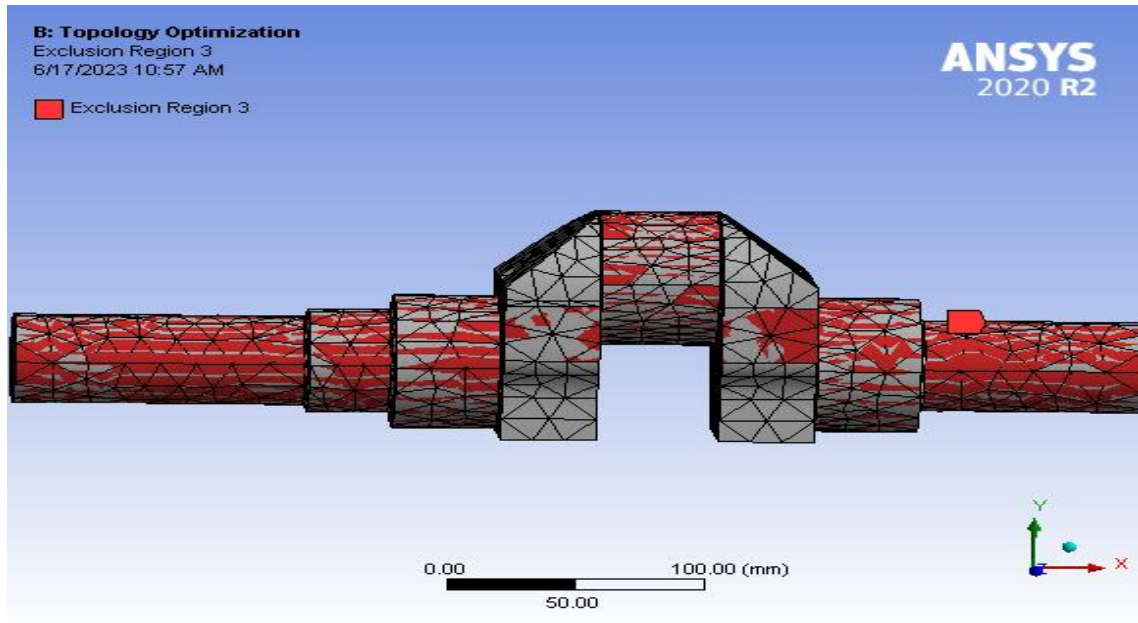


Figure 4.26 Selecting optimization regions

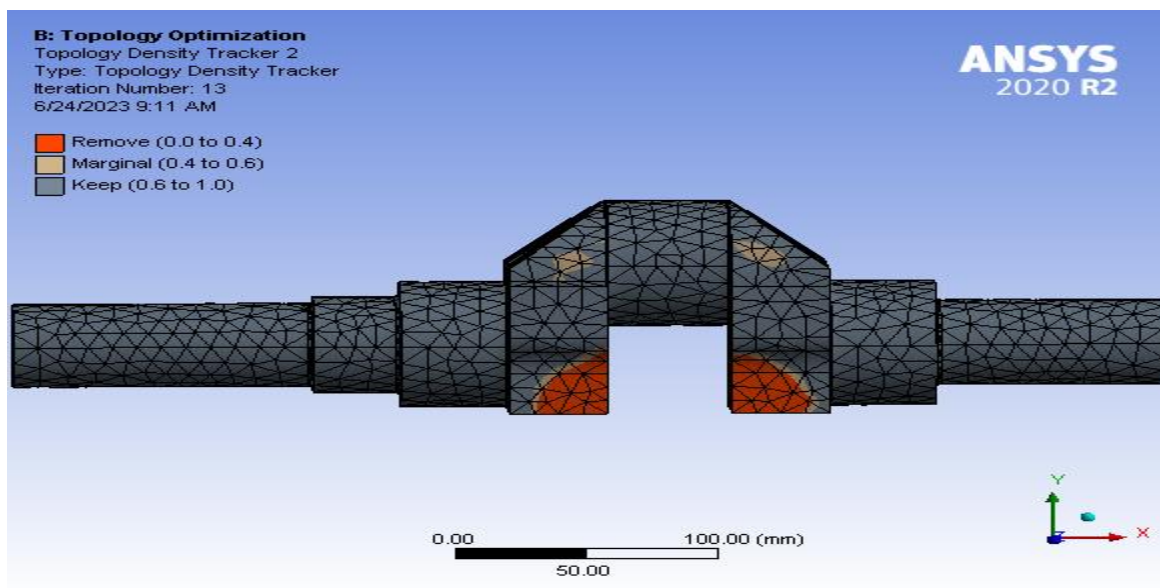


Figure 4.27 Topology density tracer removed part of crankshaft

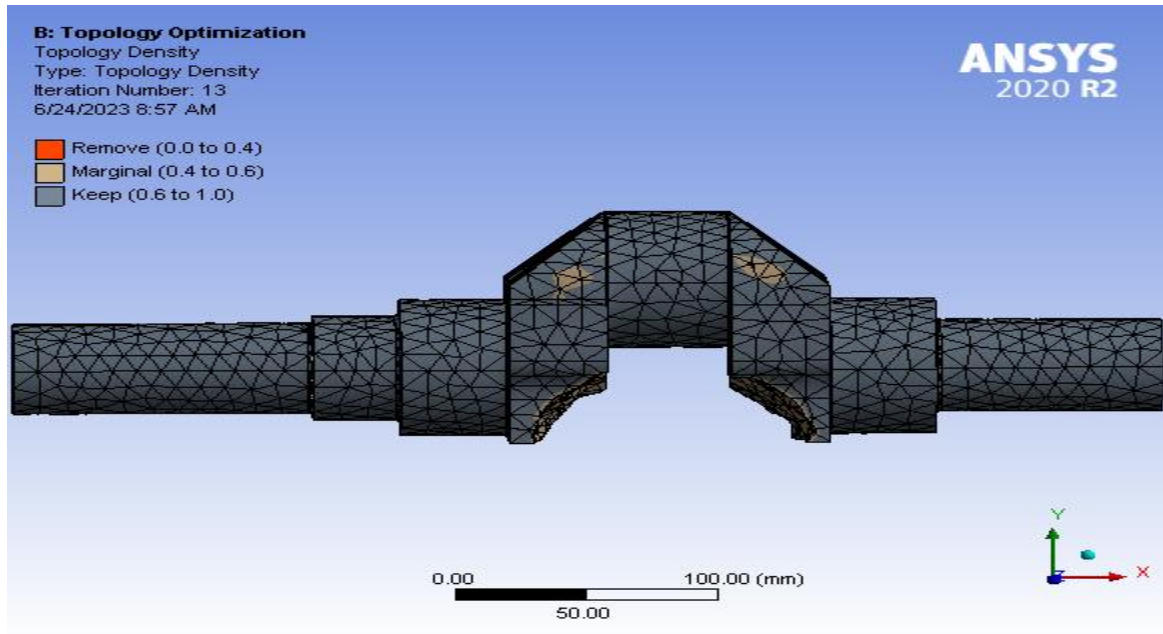


Figure 4.28 Optimized part of crank shaft

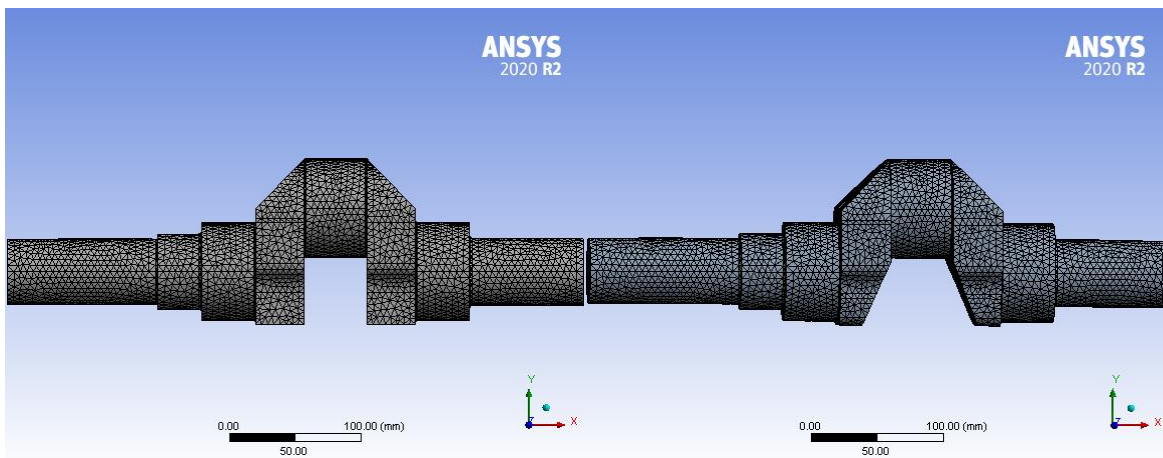


Figure 4.29 Original material and optimized crank shaft

4.4 Static FEA analysis of optimized crank shaft

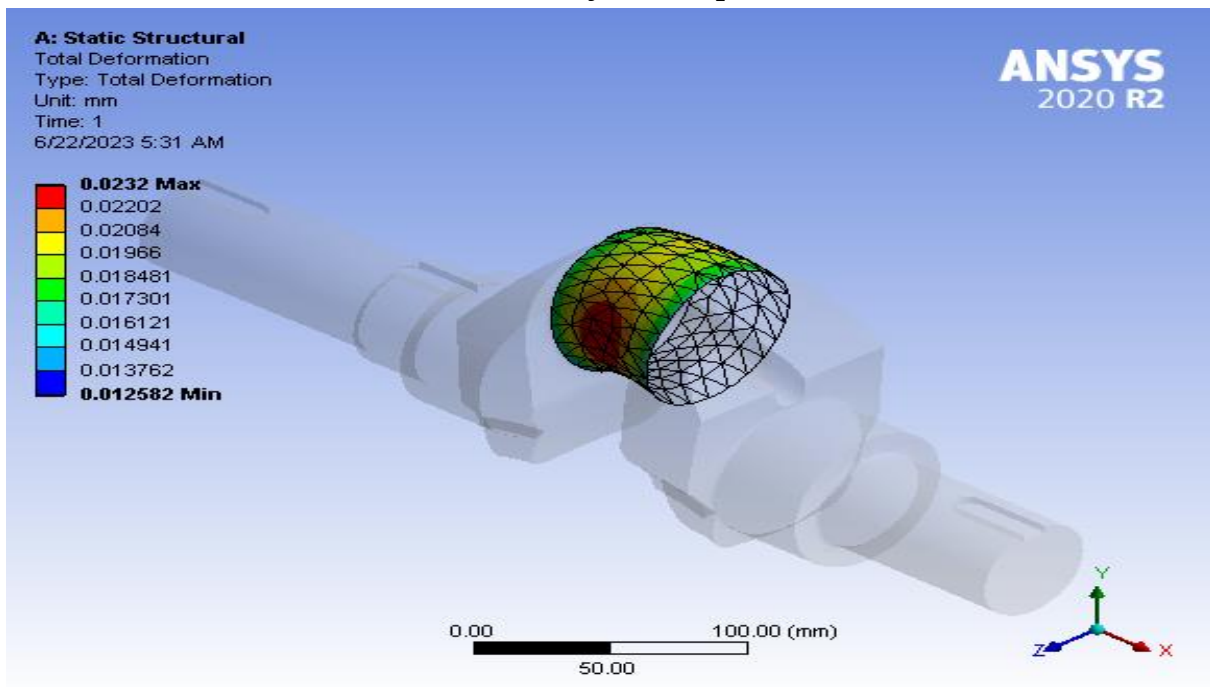


Figure 4.30 Total Deformation of cast iron

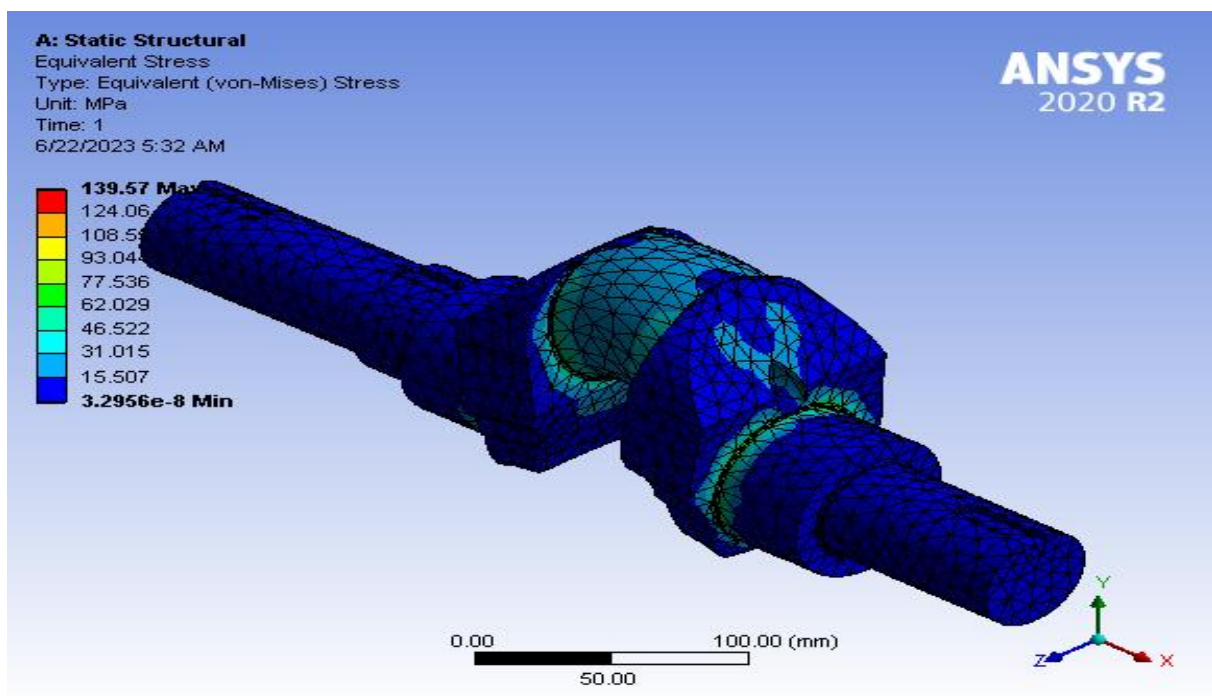


Figure 4.31 Equivalent von-misses stress of optimized crank shaft

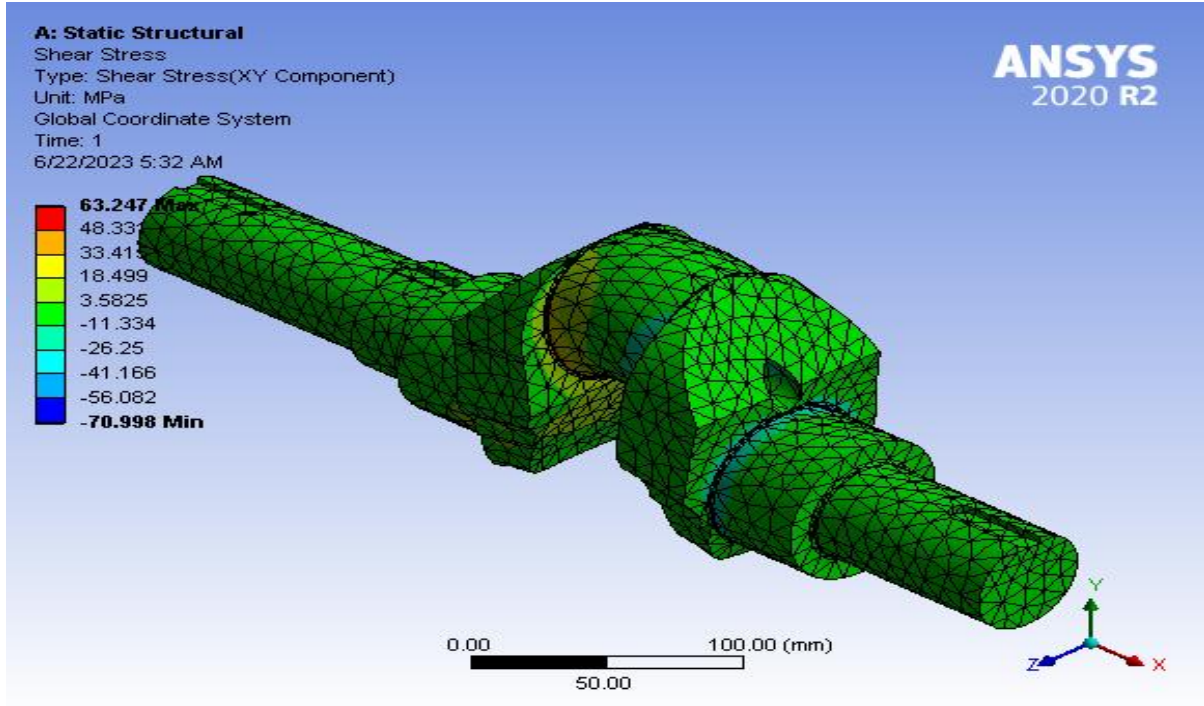


Figure 4.32 Shear Stress of optimized crank shaft

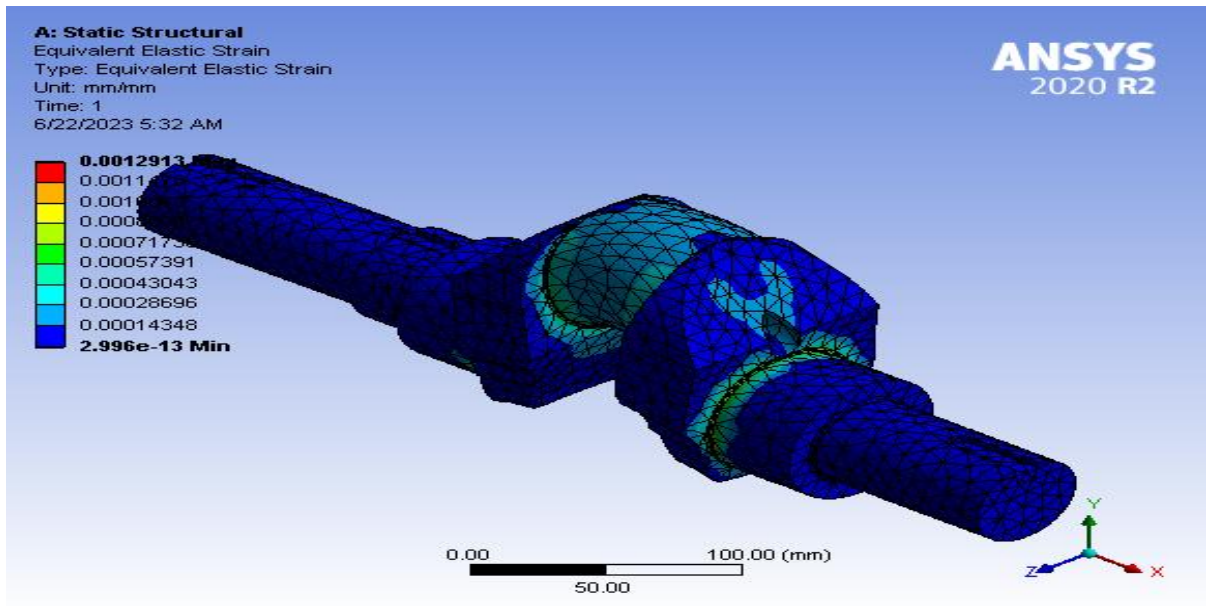


Figure4.33 Equivalent Elastic Strain of optimized crank shaft

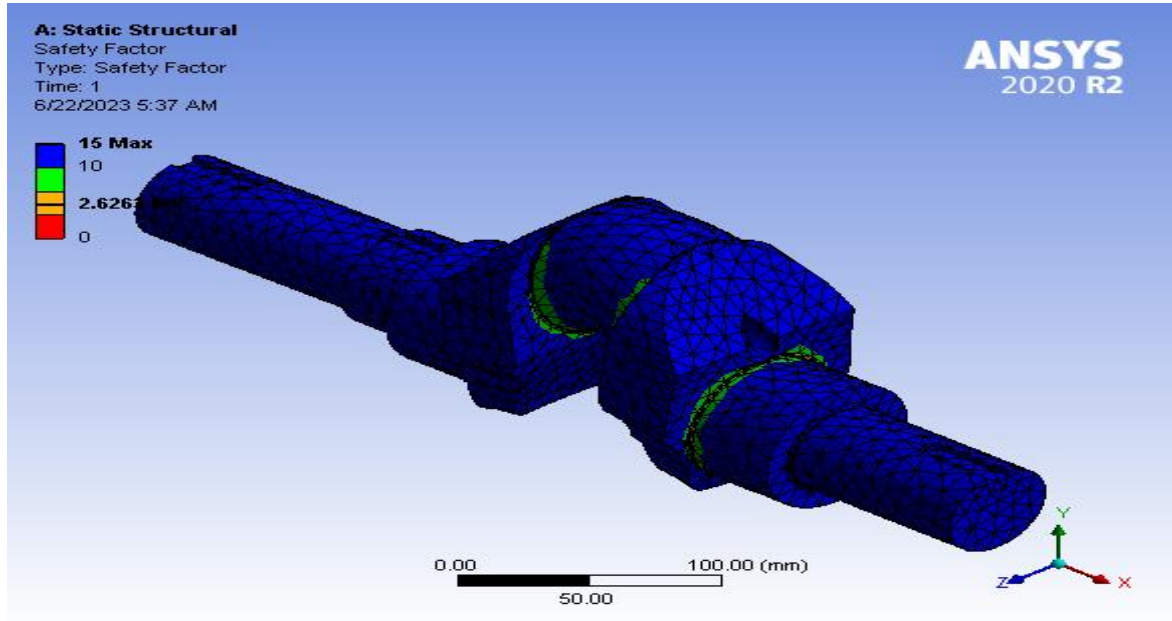


Figure4.34 Safety Factor of optimized crank shaft

4.5.2 Geometry optimization

The topology optimization is done in ANSYS 2020 R2 software with the constraint of the crankpin diameter and length, journal diameter and length. As a result the counter weight is optimized in 0.66 kg by weight, which 5.28 % from the original crankshaft. After this optimization, static and modal FEA of crank shaft is done in ANSYS software to check crankshaft's the static and dynamic conditions

Table 4.3 The Comparison of FEA result before and after topology optimization

Material	Before topology optimization					After topology optimization				
	Von-mises	Shear	Deformation	Strain	Mass	Von-mises	Shear	deformation	Strain	Mass(KG)
Cast iron	112.72 Mpa	49.6 Mpa	0.0119	0.0006	12.4	112.2 Mpa	48.33 Mpa	0.022	0.009	11.7

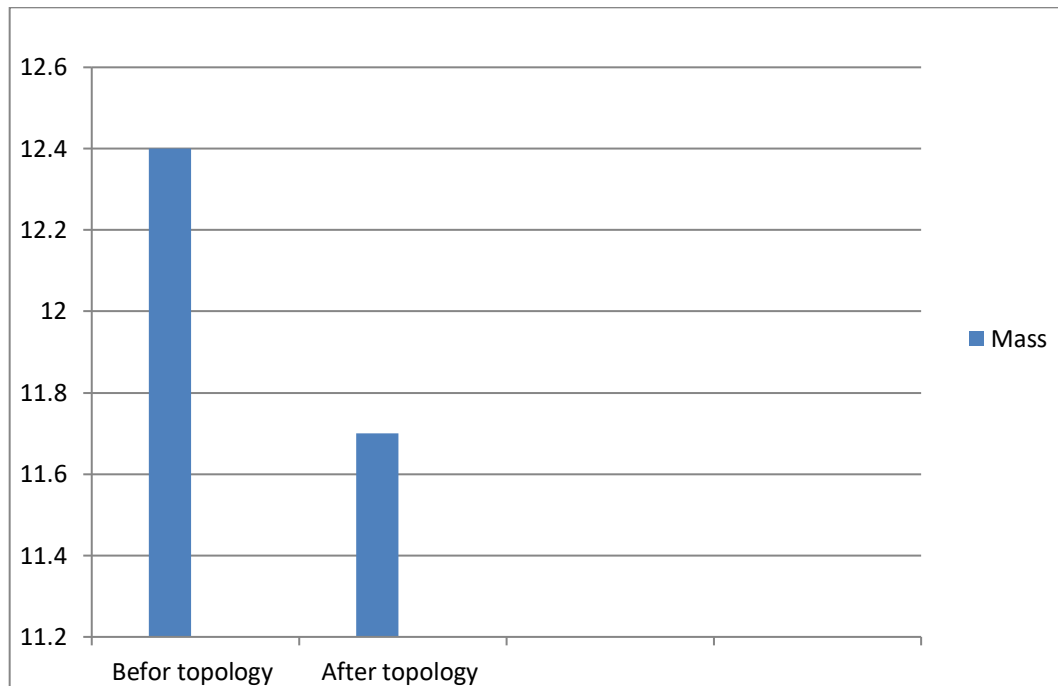
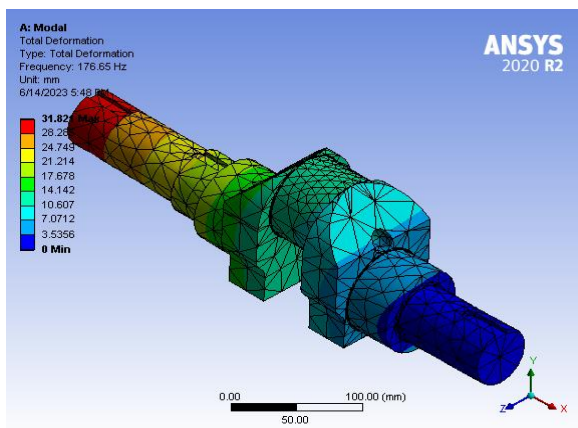
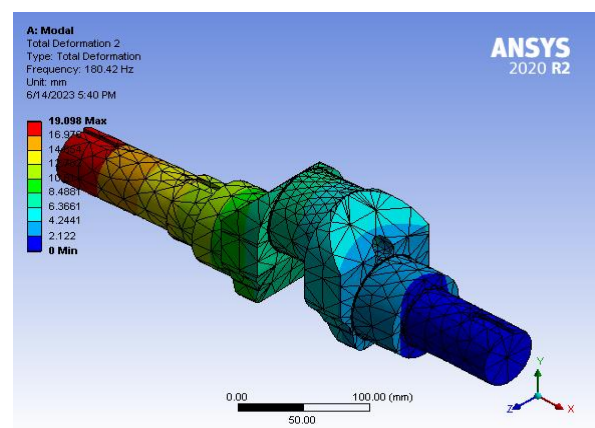


Figure4.41 Graphic Representation of weight before and after topology optimization

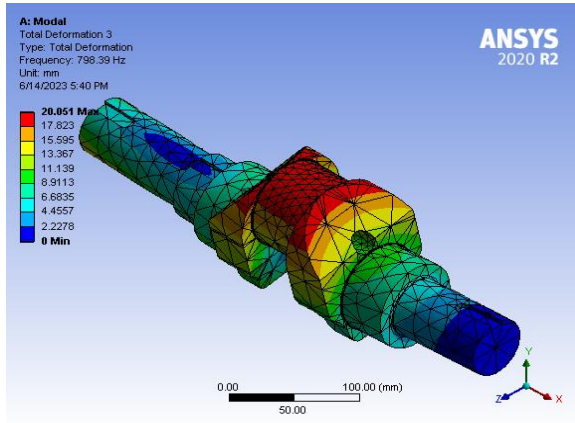
4.4.1 Modal analysis of optimized crank shaft



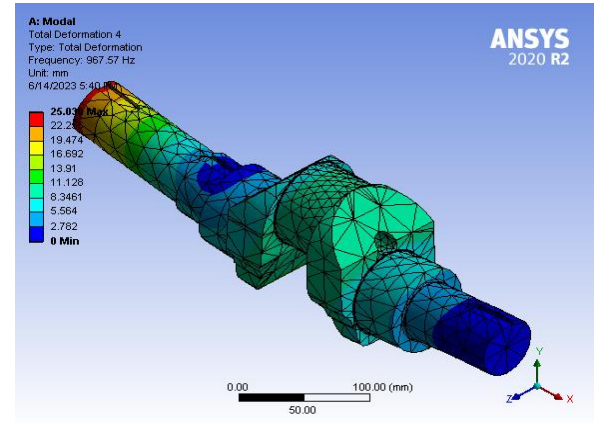
First order modal



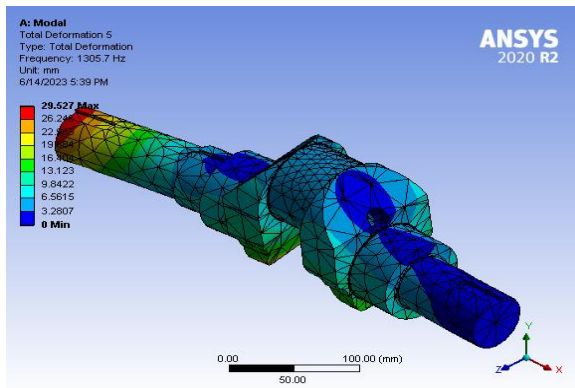
second order modal



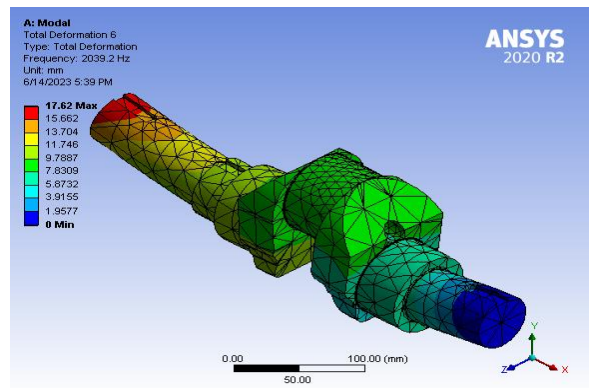
Third order modal



forth order modal



Fifth order modal



six order modal

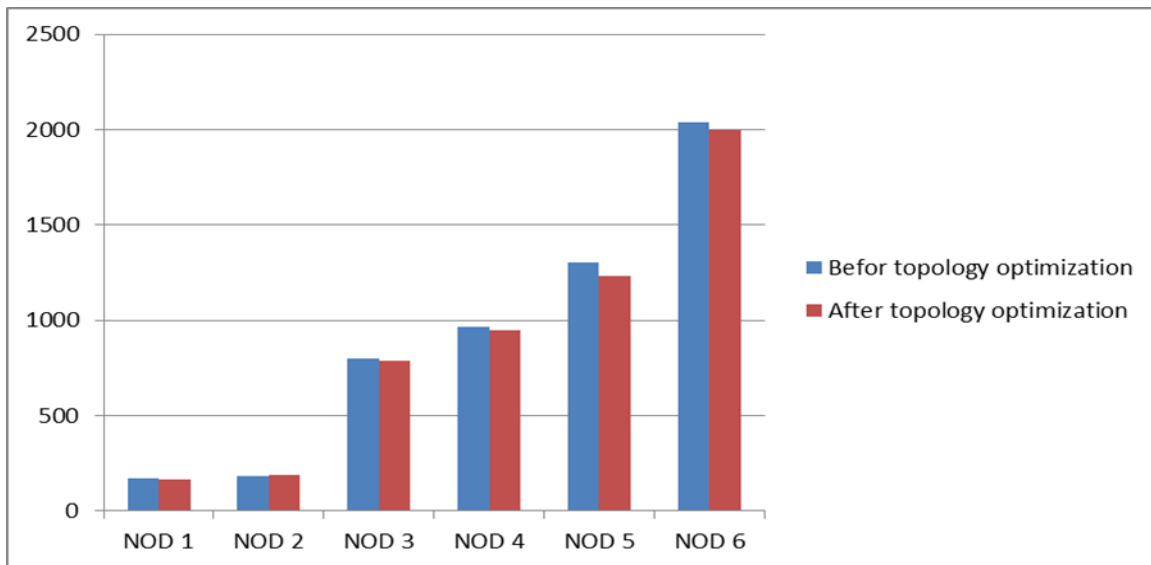
Figure 4.35 modal analysis of optimized crank shaft

The modal analysis of the crankshaft is done by using ANSYS workbench 2020R2 which is analyzed in four different materials. The first six modal shapes of each material are shown from figure 4.1-4.18 and they were compared each other as shown in table 4.1 It can be that the mode of vibration of the cast alloy steel crankshaft was slightly larger than the other three material crankshafts which imply that this crankshaft was not resonance before the other three material crankshafts was resonate. In mode shape order increase, the natural frequency also increase, implies that bring down the order, the natural frequencies which have the larger influence on the crankshaft to vibrate and resonate

Table 4.4 Modal analyses of cast iron before and after topology optimization

Nods	Cast iron Frequency in HZ	
	Before topology	After topology
Nod 1	173.42	162.91
Nod 2	180.42	187.42
Nod 3	198.39	787.88
Nod 4	967.57	945.82
Nod 5	1305.7	1238.8
Nod 6	2039.2	1998.2

Figure4.42 Variations of modal frequencies of crankshaft after topology optimization



4.5 Discussion

4.5.1 Material Optimization

The static FEA of crankshaft were in ANSYS 2020 R2 software. These analyses were conducted in four different materials. The alloy steel material which was originated from the original crankshaft and the others three were selected in material indexes depending on the strength and the density of materials.

Those were aluminum alloy, cast iron, and cast alloy steel. The static FEA of crankshaft were done which the total deformation, Von misses stress, shear stress and equivalent strain factor of safety analyze.

The total Deformation is the degree to which a structural element is displaced under a load. The deformation analysis performed in four different materials to evaluate the maximum deformation position on the crankshaft. The maximum deformation values of the crankshaft in four different materials wear between 0.011-0.032mm.in the figure 4.1-4.4 shows the maximum deformation value under maximum stress was in the position of the crankpin of the crank shaft the maximum deformation values occurs in aluminum alloy and minimum deformation occurs in cast alloy steel.

Table 4.5 Total deformation of materials

No	Material	Deformation (mm)	Load apply(104920n)
1	Cast alloy steel	0.0113	104920N
2	Alloy steel	0.0109	104920N
3	Cast iron	0.0119	104920N
4	Aluminum alloy	0.032	104920N

Von misses stress is a value used to determine if a given material will fracture. The von misses stresses of the crankshaft in four materials were between 109.62Mpa-112.76Mpa.The von misses stress analysis results was shown between figures 4.5-4.8 In the analysis results The von misses stress of 109.62Mpa is the minimum stress which occurred in a material of aluminum alloy and the maximum von misses stress was occurring in a material of alloy steel which that the stress concentration of alloy steel was greater than all three materials.

Table 4.6 von misses stress of materials

No	Material	Von misses (MPA)	Load apply(104920n)
1	Aluminum alloy	109.6	104920N
2	Cast alloy steel	111.37	104920N
3	Cast iron	112.72	104920N
4	Alloy steel	112.76	104920N

The shear force is an internal force caused by an applied force. The shear stress analysis results were shown between figures 4.9-4.12 in the analysis results, the shear stress concentration occurred in the journals fillets; the values of this shear stress were far less than the material shear stress of each material. The shear stresses of the crankshaft in four materials between 48.15Mpa-49.49.65Mpa. The shear stress of 48.115Mpa is the minimum shear stress which occurred in a material of aluminum alloy and the maximum shear stress was occurring in a material of cast iron which implies that the shear stress concentration of cast iron greater than all three materials.

Table 4.7 shear stress of material

No	Material	Shear stress (MPA)	Load apply(104920n)
1	Aluminum alloy	48.15	104920N
2	Alloy steel	49.015	104920N
3	Cast alloy steel	49.17	104920N
4	Cast iron	49.68	104920N

Elastic strain the ability of body to resist a distorting and return to its original size and shape when the force is removed. The crankshaft had the maximum elastic strain in the journal fillet area. The maximum strains were analysis from 0.000602-0.0015 in four materials. From these the minimum stain was appeared in alloy steel and the maximum strain was appeared in aluminum alloy

These factors of safety in each material mean that the materials have safety to withstand the load without failure up to the load times the safety factor The factor of safety (FOS) obtained in the crankshaft under the applied load of 104920N on the crankpin. As shown in the figure 4.17-4.20 the minimum factors of safety were varied from 2.2 - 10 depending on the four materials. So the material alloy steel had more chance withstand more loads than the other. However, this material is weighted than the other materials. Next to alloy steel, the cast iron had the chance to carry more than the current applied loads without fracture

The weight of the alloy steel (original) material is 13.6 and the other three materials were in between 4.79 kg-13.58 kg. Among those materials the list weigh aluminum alloy but the cost may not be preferable

Table 4.8 material weight

No	Material	Weight kg
1	Aluminum alloy	4.7
2	Cast iron	12.4
3	Cast alloy steel	13.48
4	Alloy steel (original))	13.6

The cast iron crankshaft is 12.4kg weight which is 8.8 % weight reduction from the original material by only changing the material alloy steel by cast iron we get 8.8 % reduced weigh

4.6 comparison result for material optimization

Table4.9 Comparison of analytical result and software result

NO	Material	Analytical result			Software result			Deformation	Mass Kg
	Crank shaft	Von-misses	Strain	Shear stress	Von-misses	Strain	Shear stress		
1	Aluminum alloy	107.9 mpa	0.00055	47.5 Mpa	109.6 Mpa	0.0015	48.15 Mpa	0.032	4.8
2	Cast iron	107.9 mpa	0.00061	47.5 Mpa	112.72 Mpa	0.00061	49.68 Mpa	0.0119	12.2
3	Alloy steel	107.9 mpa	0.00015	47.5 mpa	112.76 Mpa	0.00054	49.015 Mpa	0.0109	13.6
4	Cast alloy steel	107.9 mpa	0.0015	47.5 mpa	111.37 Mpa	0.00053	49.17 Mpa	0.0113	13.48

Figure4.36 Graphic Representation of Analytical and Software Von Misses Stress

The analytical analysis of the von misses were 107.9 Mpa which is slightly less than in the software analysis value. The error rate of the analytical and software results were vary from 1.5%-4.4%. The software analysis is more accurate than analytical analysis the alloy steel crankshaft has more stress than the other three material crankshafts and the aluminum alloy has less stress.

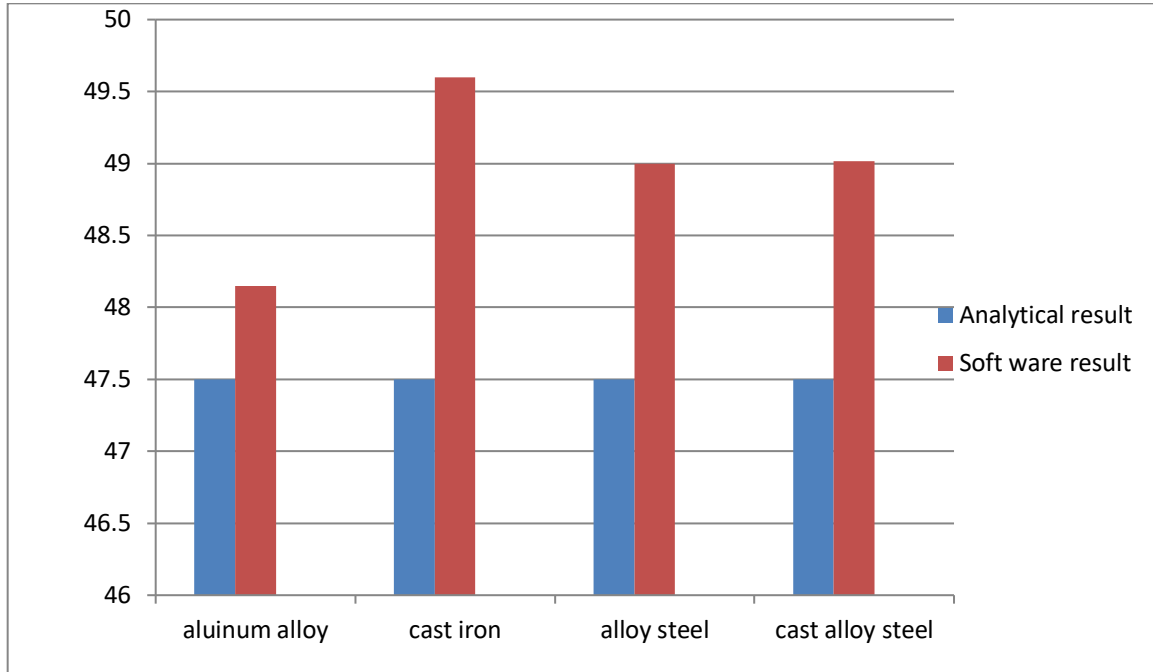


Figure4.37 graphical representation analytical and software result of shear stress

The analytical analysis of the shear stress 47.9 Mpa which is slightly less than in the software result. The error rate of the analytical and software results shear were vary from 0.5%-3.5%. The software analysis is more accurate than analytical analysis. The cast iron crankshaft has more stress than the other three material crankshafts and the aluminum alloy has less stress.

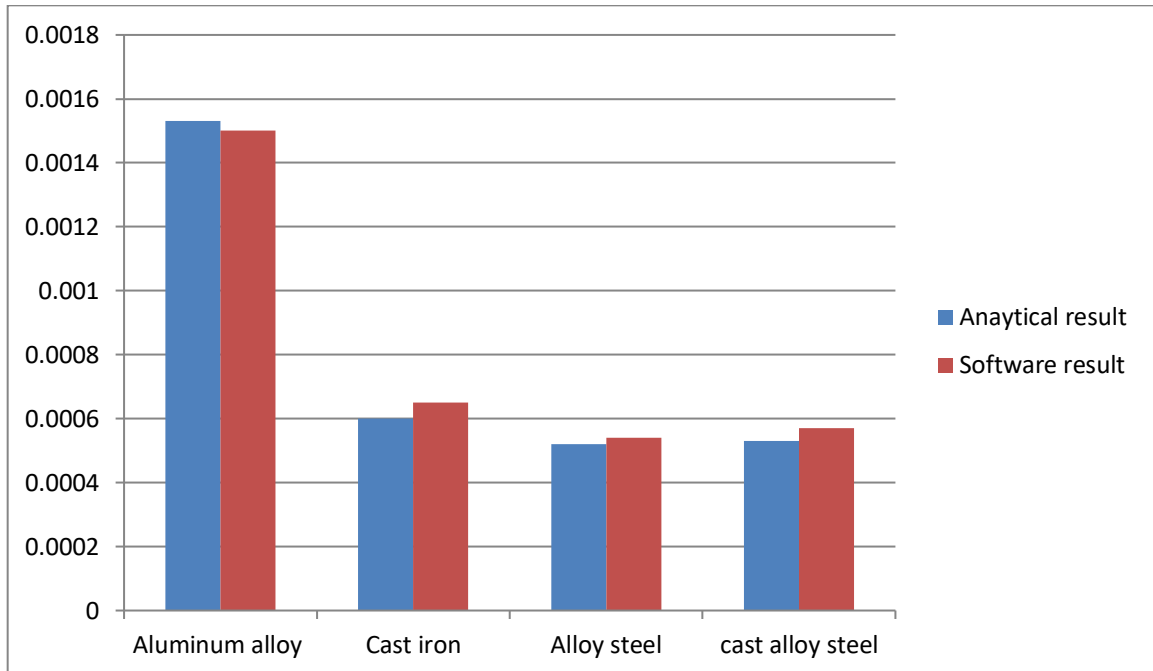
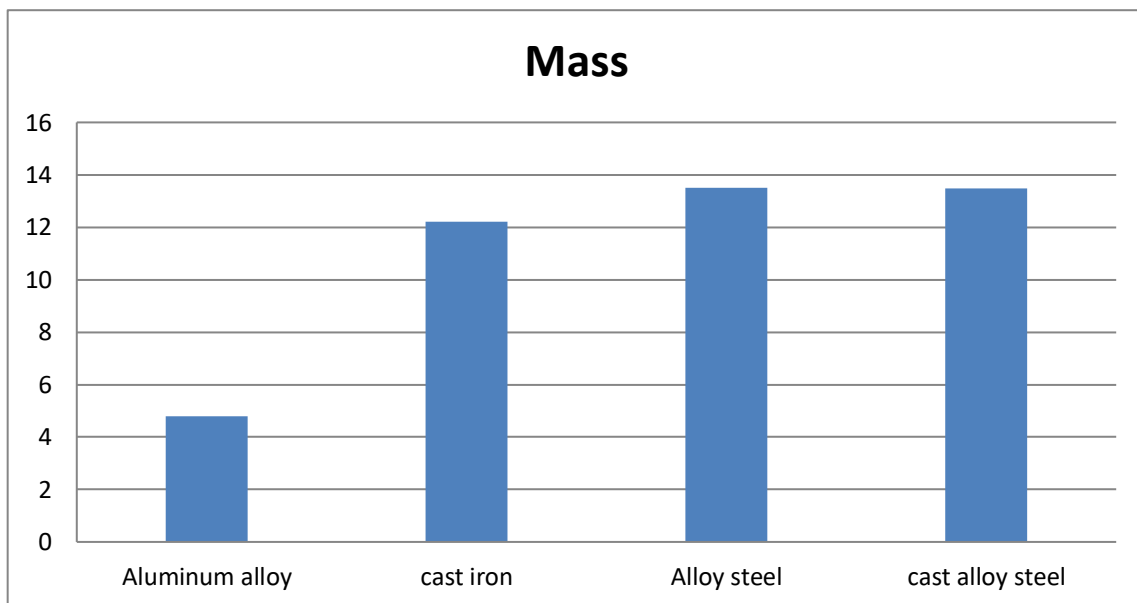


Figure4.38 graphical representation of analytical and software result of equivalent strain



. Figure4.39 Graphic Representation of Material Weight

The cast iron has less weight than the other except aluminum alloy. The alloy steel has more weight than the other

CHAPTER FIVE

5. CONCLUSION, RECOMMENDATION AND FUTURE WORK

5.1 Conclusion

The crankshaft model was created by SOLIDWORK for modeling the crank shaft. Then, the model imported into ANSYS software for static structural and modal analysis. The crankshaft is subjected to both torsional and axial loads As FEA Results show equivalent (Von-Misses) stress for all selected materials are below the limiting stress value of a crankshaft. The value of Von-misses stresses that comes out from the analysis is far less than material yield stress so our design is safe. Also the results of modal FEA give the information about mode shape vibration and frequency of the modes. The lowest free mode vibration of the crankshaft was 173 Hz, which is much greater than in forcing natural frequency of JSD 1800 engine which was 27.3 Hz. So, the possible action of resonance of the crankshaft is very low and crankshaft is not running in critical speed. The analysis of the crankshaft done using four different materials. These materials are Alloy Steel, cast alloy steel, Al Alloy, and cast iron. So we could select the optimum material depends on von misses stress, weight as well as cost. Therefore, on the static and modal analysis results the cast iron crankshaft gives the optimum results and more suggestible from the other due to less von miss stress among those materials , economically cheaper and minimum weight.

The weight of the alloy steel (original) material is 13.6kg and the other three materials where between 4.79 kg-13.58 kg. those materials the cast iron is the best crankshaft among the listed and 12.4kg which is 8.8 % weight reduce from the original material (alloy steel crank shaft), after topology optimization. The counter weight is optimized in 0.66 kg by weight, which 5.28 % from the cast iron crankshaft the studies have been achieved as a significant amount of weight reduced changing the material type and geometry optimization total of 14.08% from original alloy steel crankshaft.

5.2 Recommendation

From the result found analytical and ANSYS, the best way of analyzing is in ANSYS result. In this paper better lightweight crankshaft is done due to material and geometry size optimization. In this manner there are some recommendations given for the crankshaft. The heat treatment of the crankshaft must consider in the manufacturing of the shaft.

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,

- The static and modal analysis will be validated using experimental methods
- The thermal characteristics will be analyzing
- According to FE analysis, blue locations shown in Figure have low stresses the potential for material removal and weight reduction
- The further optimization of the crankshaft is possible in terms of stress and weight reduction by using various parametric modifications in the crankshaft.

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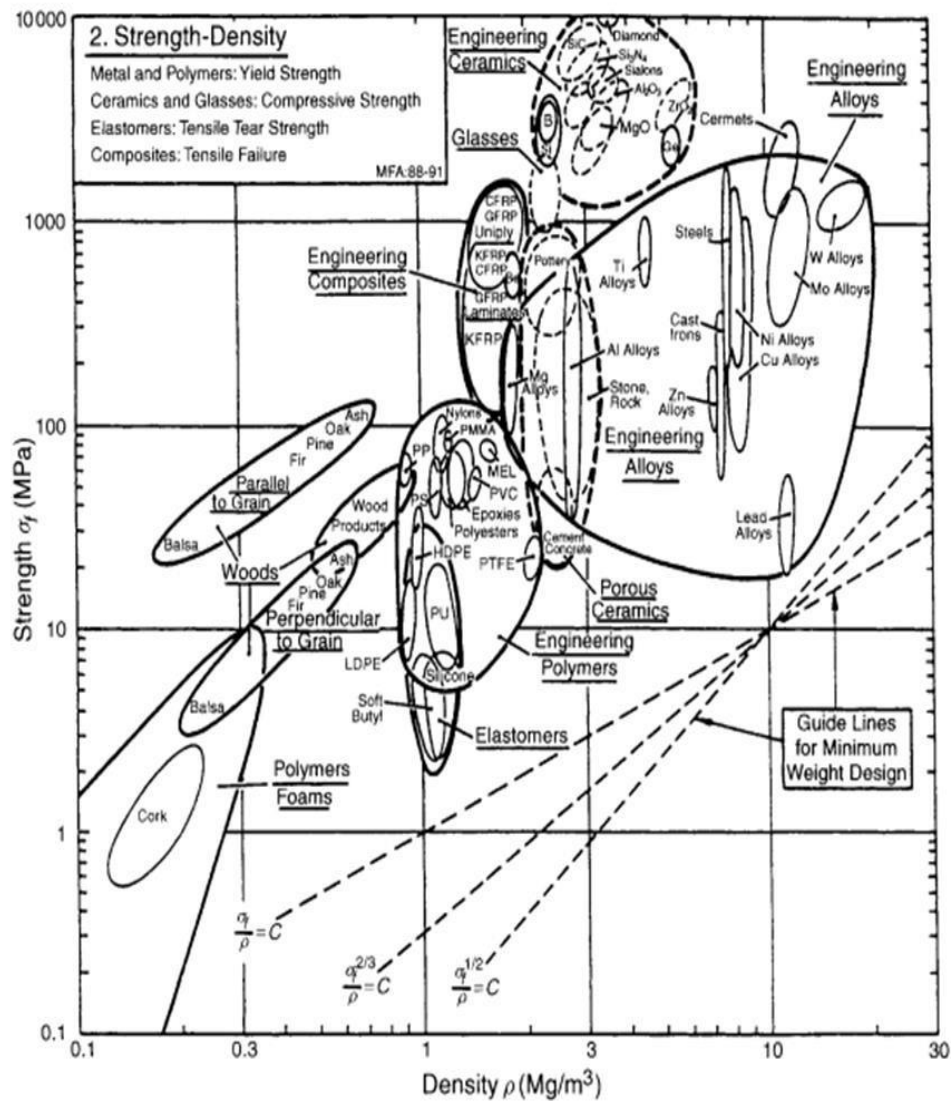
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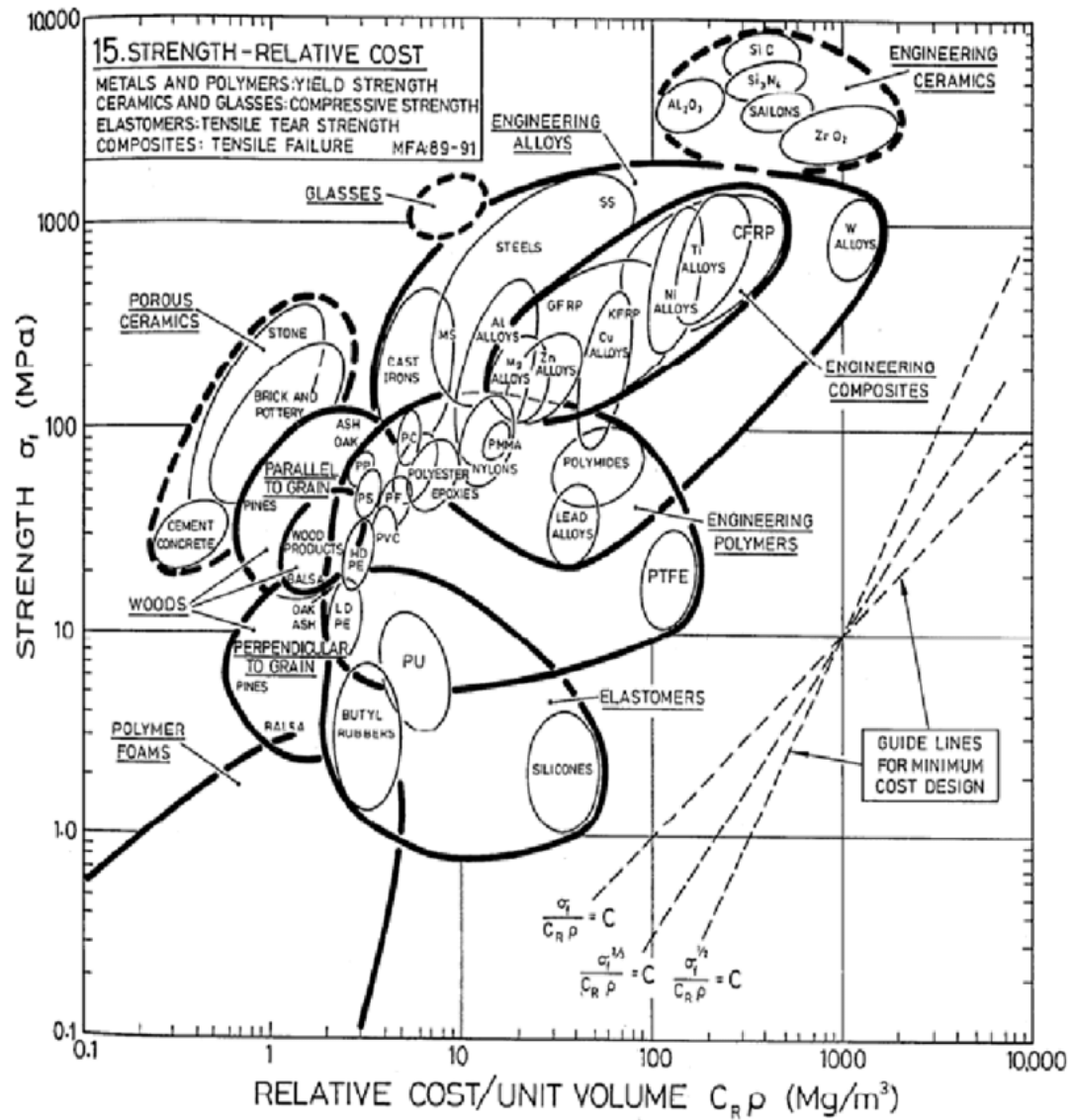
Appendices

A. The strength plotted against density, (yield strength for metals and polymers,

Compressive strength for ceramics, tear strength for elastomers and tensile strength for composites).



Ashby Diagram of strength versus relative cost/unit volume



Clarence volume $V_c = V_2 = \frac{\pi S B^2}{4(r_c - 1)}$

$$= \frac{\pi 0.13 (0.135)^2}{4(20-1)}$$

$$= 9.793 \times 10^{-5} m^3$$

Displacement $v_{d=\frac{\pi B^2 2r}{4}}$

$$= \frac{\pi (0.35^2) 13}{4}$$

$$= 1.86 \times 10^{-3} m^3$$

Maximum volume $v_1 = v_{max} = v_d + v_c$

$$= 9.793 \times 10^{-5} + 1.86 \times 10^{-3}$$

$$= 1.959 \times 10^{-3} m^3$$

Maximum pressure of gases

$$p_1 v_1^\gamma = p_2 v_2^\gamma$$

$$P_2 = P_{MAX} = P_1 \left[\frac{V_1^{1.4}}{V_c} \right]$$

$$= 1.013 \times 10^5 \left[\frac{(1.959 \times 10^{-3})^{1.4}}{9.193 \times 10^{-5}} \right]$$

$$= 7.338 \times 10^6 mpa$$

4.1.2 The crank at the dead center

The force acting on the cylinder due to gas pressure

$$F_S = \frac{\pi B^2}{4} P$$

$$= \frac{\pi (135)^2}{4} 7.33 \times 10^6 mp$$

$$=104.920\text{KN}$$

The reaction force R_{H1} and R_{H2} in bearing 1 and 2

$$R_{H1} = F_P - \frac{F_P b_1}{b_1 + b_2}$$

$$R_{H1} = 104.920\text{KN} - \frac{104.92\text{KN} \times 81.25}{81.25 + 80.75} =$$

$$= 52.2980 \text{ KN}$$

$$R_{H2} = \frac{F_P \cdot b_1}{b_1 + b_2}$$

$$= \frac{104.920\text{KN} \cdot 80.75\text{mm}}{162\text{mm}}$$

$$= 52.2980 \text{ KN}$$

The maximum bending moment at center of the crank pin

$$M_{maxb} = R_{r2} b_1$$

$$= 52.2980 \text{ KN} \times 81.25\text{mm}$$

$$= 4249.21\text{KN}$$

The bending stress of crank pin

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

$$= \frac{32 \times 4249.21\text{KN}}{\pi (75)^3\text{mm}}$$

$$= 101\text{N/mm}^2$$

4.1.3 Crank at an angle of maximum twisting moment

The thrust force exerted on the crankpin by the piston

$$\begin{aligned}
 F_Q &= \frac{F_P}{\cos \phi} \\
 &= \frac{104.920 \text{KN}}{\cos 8.24} \\
 &= 106.01 \text{KN}
 \end{aligned}$$

Thrust on the crank shaft can be split into the radial component and tangential

$$\begin{aligned}
 F_r &= F_Q \cos(\theta + \phi) \\
 &= 104.920 \text{KN} \cos(35 + 8.24) \\
 &= 76.433 \text{KN}
 \end{aligned}$$

Tangential Component

$$\begin{aligned}
 F_t &= F_Q \sin(\theta + \phi) \\
 &= 104.92 \sin(35 + 8.24) \\
 &= 71.877 \text{KN}
 \end{aligned}$$

Reaction at bearing 1&2 due to tangential force

$$\begin{aligned}
 R_{T1} &= \frac{F_t b_1}{b} \\
 &= \frac{76 \text{KN} \times 81}{81 + 62} \\
 &= 36.423 \text{KN} \\
 R_{T2} &= \frac{F_t b_2}{b} \\
 &= \frac{72.622 \text{KN} \times 80.75}{81.25 + 80.75} \\
 &= 36.199 \text{KN}
 \end{aligned}$$

Reaction at bearing 1&2 due to radial force

$$\begin{aligned}
 R_{r1} &= \frac{F_t b_1}{b} \\
 &= \frac{77.22KN \times 81.25mm}{162mm} \\
 &= 38.729KN
 \end{aligned}$$

$$\begin{aligned}
 R_{r2} &= \frac{F_t b_2}{b} \\
 &= \frac{77.22KN \times 80.75mm}{162mm} \\
 &= 38,490KN
 \end{aligned}$$

$$\begin{aligned}
 \text{Bending moment center of crank pin} &= R_{r2} \times b_1 \\
 &= 38.729KN \times 81.25mm \\
 &= 3146.731KN- mm
 \end{aligned}$$

Twisting moment

$$\begin{aligned}
 T_M &= R_{T1}(r) \\
 &= 36.423KN \times 65mm \\
 &= 2367.495KN-mm
 \end{aligned}$$

Equivalent twisting moment on the crankpin

$$\begin{aligned}
 T_e &= \sqrt{(M_{CR})^2 + T_1^2} \\
 &= \sqrt{(3146.73)^2 + (2367.495)^2} \\
 &= 3937.886KN
 \end{aligned}$$

According to distortion energy theory, the Von-Misses stress induced in the crank-pin is (Mr. Anant B. et al,)

$$\begin{aligned}
 M_{EV} &= \sqrt{(k_b M_e)^2 + \frac{3}{4}(K_t T_m)^2} \\
 &= \sqrt{(1 \times (3937.886))^2 + \frac{3}{4}(1 \times (2367.495))^2} \\
 &= 4439.676 \text{KN/mm}
 \end{aligned}$$

Von misses stress induce in crankpin

$$\begin{aligned}
 \sigma_v &= \frac{32 * M_{EV}}{\pi (d_c)^3} \\
 &= \frac{32 * 4439.676 \text{KNmm}}{\pi (75 \text{mm})^3} \\
 &= 107.193 \text{N/mm}^2
 \end{aligned}$$

Shear stress induce in crankpin

$$\begin{aligned}
 \tau &= \frac{16 T_E}{\pi d^3} \\
 &= \frac{16 \times 3937.886 \text{KN}}{\pi (75)^3} \\
 &= 47.5 \text{N/mm}^2
 \end{aligned}$$

