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Optimization and Fatigue Analysis of Crane Hook Using Finite Element Method

A Thesis Submitted for the Partial Fulfilment of Degree of Masters of Science in Mechanical Engineering (Design)

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Abstract

The failure of crane hook occurs because of the stress induced due to repetitive loading and unloading conditions. These are the causes of fatigue failure of the crane hook. To minimize the failure of crane hook, the stress induced in the crane hook is studied and reduced the maximum stress than existing (trapezoidal) crane hook. By modifying the geometry (cross section) the stress of the crane hook is reduced comparing with the standard crane hook. In this study, the cross section has been selected as a basic parameter to optimize the hook design which carries 4.5-tons load. Because of the stress reduction the fatigue life of the model-3 crane hook goes to increase (have better life comparing with standard and other models of crane hook). Here, there are four types of cross-section for crane hook namely model-1, model-2, Model-3 and trapezoidal (standard). These crane hooks are modelled and analysed using SOLIDWORK and ANSYS software respectively, and the results of each modified crane hooks are compared with trapezoidal hook results based on different criteria's which are maximum stress, maximum deformation, fatigue life and weight of the hooks. Finally based on the comparisons of each modified modelling crane hooks with the standard crane hook, model-3 is selected. The fatigue life cycle of model-3 crane hook is increased by 22.78% from the standard crane hook with the same applied load. The maximum Von-Misses stress and weight of model-3 crane hook are reduced from trapezoidal (standard) crane hook by 3.461 MPa (4.133%) and 0.354 Kg (2.49%) respectively, and model-3 crane hook has better results of safety factor & fatigue life than standard crane hook with the same applied load. Therefore, model -3 crane hook is considered as optimum results.

Key words: Crane Hook, Finite Element Method, Weight Optimization, Fatigue Analysis

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CHAPTER 1

1. INTRODUCTION

Material-handling equipment is equipment that relate to the movement, storage, control and protection of materials, goods and products throughout the process of manufacturing, distribution, consumption and disposal. Material handling equipment is the mechanical equipment involved in the complete system. Material handling equipment is generally separated into four main categories: storage and handling equipment, engineered systems, industrial trucks, and bulk material handling. [1]

Crane hook is a curved bar and is used for lifting loads in cranes. Crane hook is the component which is generally used to elevate the heavy load and transfer it from one place to another in industries, factories and constructional sites. Crane hook is one of the main and important components of crane basically a hoisting fixture designed to engage a ring or link of a lifting chain or the pin of a shackle or cable socket and must follow the health and safety guidelines. [2, 3]

Cranes are classified as weight handling equipment (WHE). They are primarily designed to perform weight lifting and, with the proper attachment, excavating operations under varied conditions. A crane is an item of plant intended for raising or lowering a load and moving it horizontally including the supporting structure of the crane and its foundations. Cranes have many configurations to accomplish various construction and industrial operations and are typically identified by their carrier and type of boom. [4] Overhead crane, mobile crane, tower crane, telescopic crane, gantry crane, deck crane, jib crane, loader crane are some of the commonly used cranes. Figure 1-1 shows the crane with all components.

A **gantry crane** has a hoist in a fixed machinery house or on a trolley that runs horizontally along rails, usually fitted on a single beam (mono-girder) or two beams (twin-girder). The crane frame is supported on a gantry system with equalized beams and wheels that run on the gantry rail, usually perpendicular to the trolley travel direction. **Overhead Crane** known as a 'suspended crane', this type of crane work very similar to a gantry crane but instead of the whole crane moving, only the hoist / trolley assembly moves in one direction along one or two fixed beams, often mounted along the side walls or on elevated columns in the assembly area of factory. The **tower crane** is a modern form of balance crane. Fixed to the ground (and sometimes attached to the sides of structures as well), tower cranes often give the best combination of height and lifting capacity and are used in the construction of tall buildings. A **mobile crane** is a crane capable of travelling over a supporting surface without the need for fixed runways and relying only on gravity for stability. A **loader crane** (also called a knuckle-boom crane or articulating crane) is a hydraulically-powered articulated arm fitted to a truck or trailer, and is used for loading/unloading the vehicle. A **jib crane** is a type of crane where a horizontal member (jib or boom), supporting a moveable hoist, is fixed to a wall or to a floor-mounted pillar. Jib cranes are used in industrial premises and on military vehicles. **Deck Crane** is Located on the ships and boats, these are used for cargo operations or boat unloading and retrieval where no shore unloading facilities are available. A **telescopic crane** has a boom that consists of a number of tubes fitted one inside the other. A hydraulic or other powered mechanism extends or retracts the tubes to increase or decrease the total length of the boom. [5]

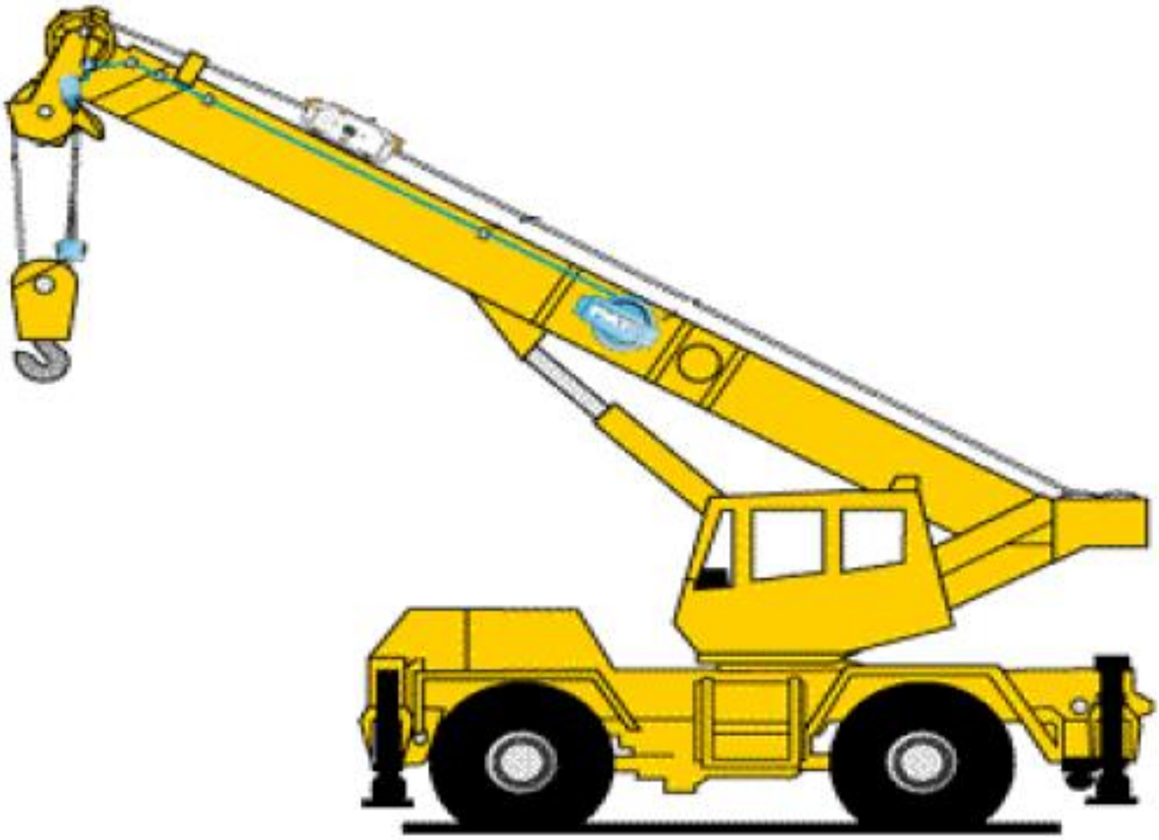


Figure 1-1: Crane with all components [4]

1.1. Types of Crane Hooks

Crane hooks are classified depending on their manufacturing materials as well as the intended usage and, (amongst other factors) according to this, some aspects are more important than others. Different types of crane hooks can be classified according to their shapes, method of manufacture, mode of operation or other unique characteristics. They are made in a variety of styles to meet specific needs and they are rated for loads of specific type and size.

1.1.1. Types of Crane Hooks based on their shape

There are two types of crane hook depending on their shape which are single crane hook and double crane hook. As the name suggests, the main difference between these two options is the number of hooks included, and there are different sub-types possible such as the C-hooks (which is essentially a single hook variant with a slightly different shape). Figure 1-2 and figure 1-3 shows the single and double crane hooks.

Single Crane Hooks are the right choice if the machinery deals with loads of up to 75 tons; this lifting hook is very simple and easy to use.

Figure 1-2**:** Single Crane Hooks []

Double (Ramshorn) Crane Hook is similar in conception, but their design provides superior bearing which is suitable for heavier loads of over 75 tons. A **Ramshorn hook** is a shank hook with two throat openings, sometimes called sister hooks or twin hooks. Commonly they are used in applications with shipyard cranes and container cranes. There are two types of Ramshorn hooks: The **Ramshorn Form A hook,** which has a solid lower hook design and the **Ramshorn Form B hook which have hole at the lower hook design.** The hole of hook is used to attach rigging.

Figure 1-3: Double (Ramshorn) Crane Hook []

1.1.2. Types of Crane Hooks based on their manufacturing methods

When it comes to the manufacturing methods used to create crane hooks, there are two main manufacturing types available which are forging crane hooks and laminated crane hooks.

Forging Crane Hooks are forged from a single piece of high quality steel with low carbon which is cooled slowly to ensure optimum stress resistance. These hooks typically feature very simple manufacturing and are also very simple to use, since they're made of a single piece.

Forging is defined as a metal working process in which the useful shape of work piece is obtained in solid state by compressive forces applied through the use of dies and tools.

Laminated Crane Hooks are a little more complex, and are comprised of several steel plates riveted together to ensure higher stress resistance and increased safety. These hooks are conceived in a way that allows the hook to keep working even if some of the individual parts are damaged. Laminated hooks are available as single hooks or double laminated hooks. Figure 1-4 indicates the single laminated crane hook.

Figure 1-4: Laminated crane hook [40]

Generally, crane hooks are made from wrought iron or carbon steel. For heavy duty crane hooks low alloy steels are used but material is not the only factor behind its enormous load bearing capacity. Steel grade, heat treatment and forging are equally important to make a durable crane hook. Proper forging is very important. Forging provides better structural integrity than any other metal working processes. It eliminates any kind of defect such as gas pockets or voids in the hook which can affect its long term performance; thus increasing its strength, toughness, load bearing capacity and fatigue resistance.

Carbon steel is a popular material for production of crane hook. It is available in numerous grades, and can be heat-treated to improve its strength, ductility, and machinability. There are four main grades of carbon steel: low carbon steel, medium carbon steel, high carbon steel and very high carbon steel. Depending on the amount of carbon present in the material, carbon steel forgings are hardenable by heat treatment to increase yield and impact strength as well as wear resistance. Material cost of carbon steel forging is relatively much lower than other steel forgings, especially compared with stainless steel forging.

The load is usually handlled by means of chain or rope slings attached to hook. Ther are two most popular design hooks which are standard (single) and ramshorn (double) hooks. Standard and ramshorn hooks may be flat-die or close die forged or else made of a series of sharped plates. One piece forged hook is used to lifting loads up to 100tons. Mostly hooks are forged from low carbon steel. In the process of production hooks are carrfully anealed after forging and machining. [6]

Due to continuous working of crane hook nanostructure of crane hook are changes and some problems like weakening of hook due to wear, tensile stresses and excessive thermal stresses these are some other reason of failure. Hence continuous working of crane hook may increase the magnitude of these stresses and eventually result in failure of crane hook. Due to some design modification all the above mentioned failures may be prevented. [7, 42]

Motivation to Work on Weight Optimization and Fatigue Analysis of Crane Hook

In Mesifin Industrial Engineering (MIE) branch of automotive assembly plant during lifting of the small vehicles accident is happened because of crane hook failure, and this failure caused to vehicle damage. This accident is happened by didn’t consider the lifting ability and life cycles of the crane hook. Therefore, here is interested to the fatigue and stress analysis of crane hook, and this analysis helps to identify lifting ability and estimate the available life of crane hook.

1.2. Statement of Problem

A crane is subjected to continuous loading and unloading condition. These will cause fatigue failure of the crane hook and lead to serious accidents. Due to continuous loading cycle the life (the ability to resist the applied load) of crane hook will reduce. Therefore, the fatigue life and fatigue damage the crane hook must be analysed (estimated).

To reduce the failure of crane hook, the stress induced in the hook must be analysed and reduced the stress as much as possible with having better other results. Actually, the analysis of optimized hook stress is reduced compared to the standard hook by means of increasing the contact area (high stress concentration area of cross section) and by design concept this means indirectly strength improving of crane hook. By modifying the cross-section stress and weight of the crane hook can reduce with comparing the standard crane hook, And the fatigue life also be better than the standard crane hook.

1.3. Objective

1.3.1. General Objective

The general objective of this study is to optimize and fatigue analysis of crane hook using finite element method

1.3.2. Specific Objective

The specific objective of this study is to:

- ✓ Reduce stress induced on the high stress concentration area of crane hook
- ✓ Get have better fatigue life than standard crane hook
- ✓ Compare the optimized and trapezoidal (standard) crane hooks with respect to deformation, stress and minimum available life

1.4. Methodology used for this thesis

The overall weight optimization and fatigue analysis of crane hook is done by using FEM. The first step is Literatures survey from different sources such as journals, books, websites and other sources. And then find the gap of the literatures (problem) is identified. The 2D and 3D models of the crane hooks are prepared by autocad and SolidWork softwere and the FEM simulation is done by Ansys Workbench 17.2. The numerical method results are compared with analytical methods. The new modified crane hook models are compared with the trapezoidal (existing) crane hook based on the FEM analysis and selected the best one crane hook.

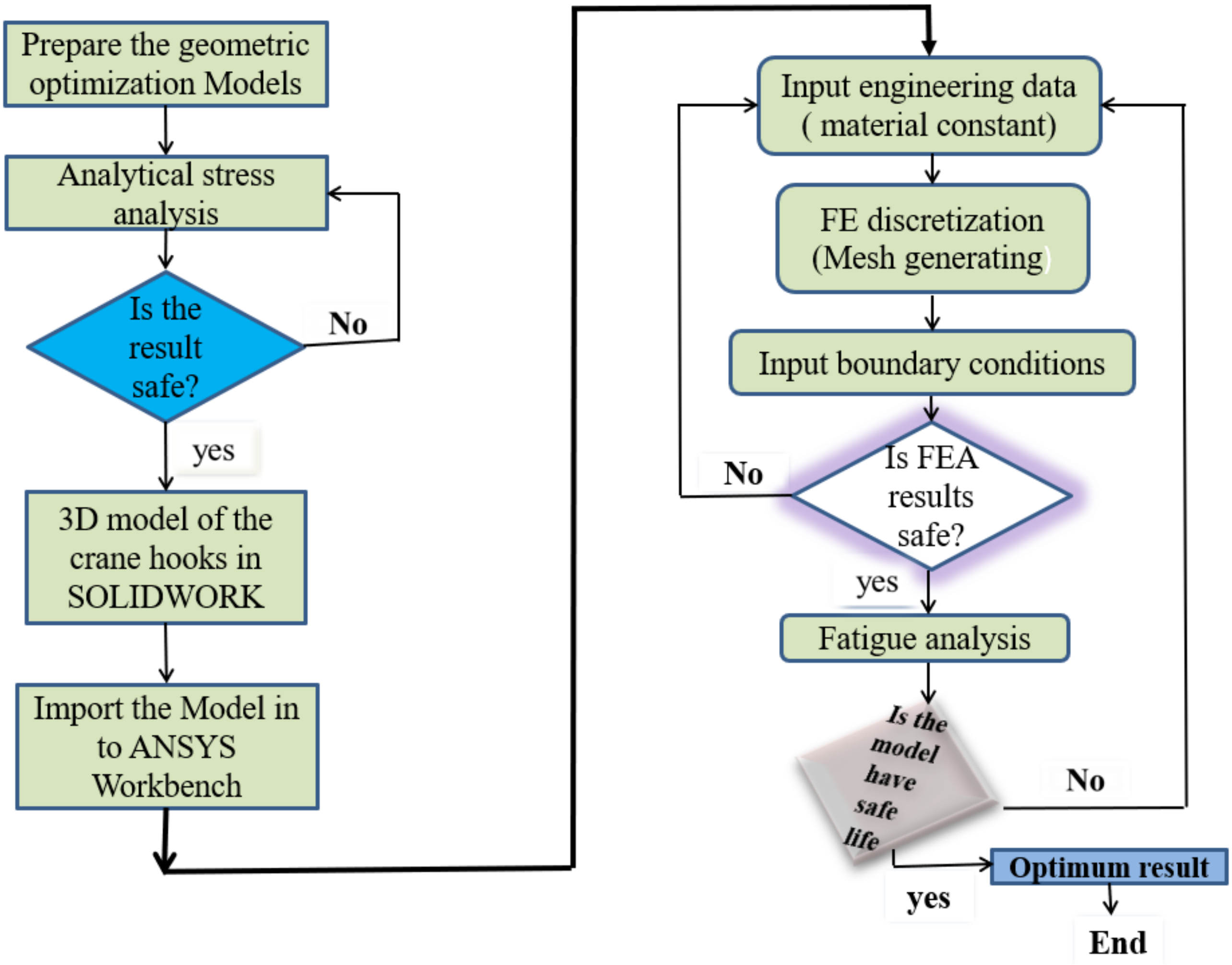


Figure 1-5: Methodology chart

1.5. Significance of the Study

The importance of this study is to reduce the weight of the crane hook which means to make light weight of crane hook as well as to save the waste of material for Manufacturer Company. The weight reduction of crane hook is indirectly making light to the overall weight of crane. This helps to save (reduce) fuel consumption of the crane. The manufacturer company produced the crane hooks by mass; therefore, it will save more materials and it is good saving in terms of cost. The other important of this study is to reduce the maximum stress induced on the crane hook (make having better fatigue life than the existing crane hook). The fatigue life is checked by analysis fatigue sensitivity of the crane hooks and this analysis helps to estimate the fatigue life with the applied over load. If the maximum stress is reduced the durability of the crane hook will improve.

1.6. Scope and Limitation of the study

This study is focus on stress and weight optimization of crane hook by modified its cross section. And also fatigue analysis of crane hook using FEM is concerned. Then compare the modified and standard crane hooks based on their results. Other optimization and analysis methods are not considered in this study.

1.7. Organization of the Study

This thesis is divided in to five chapters. **Chapter one** is an introduction to the report. It comprises the background, statement of the problem, objective of the research, organization, methodology, scope & limitation of the study and the significance of the study are introduced. **Chapter two** includes the devoted to theoretical and empirical literatures. Discussed historical background of crane hook, the methods of previous thesis, journals and books related to studies of crane hook. **Chapter three,** structural design part of the thesis, materials and methods are also developed of the study that fundamental to achieve of the specific objectives of the thesis. And also included the analysis part (i.e. the heart of the study). The descriptions and steps of how to finite element analysis of any object using ANSYS. The **fourth chapter** is about result and discussion of the study. Compare the results of each modified models of crane hooks with trapezoidal (standard) crane hook. The **last chapter** of this study is included the conclusions, recommendations and future works.

CHAPTER 2

2. LITERATURE REVIEW

Gopichand A.et al. (2013) has worked optimization of design parameters is carried out using Taguchi method. Considered total three parameters and made mixed levels a L16 orthogonal array. The design parameters for crane hook are area of cross section, material and radius of crane hook. Under parameter of cross sectional area included rectangular, triangular, circular and trapezoidal. And the materials are structural steel and cast iron with two different radiuses of 150 mm & 200 mm. the parameters have 4 levels and 16 models of crane hooks by mixed the parameters. Depending results of SN ration plots for each parameters the optimum combination is determined. The optimum combinations of input parameters for minimum Von-Misses stresses are triangular cross section, cast iron material and 200 mm radius of curvature. Analytical method for stress calculation curved beam flexure formula is used when the curvature of the member is pronounced as in cases of hooks and rings for different cross sections mathematical analysis of stress. [8]

Nishant soni .et al. (2013) has worked with objective of the optimization of the mass for cane hook-under the effect of static load comprising the peak pressure load. He used finite element analysis for the shape optimization of crane hook as well as for validation of final geometry. Also considered geometry and manufacturing constrain during optimization process and results show that optimized cane hook is 14% lighter then original crane hook. [9]

Rashmi Uddanwadiker.et al. (2011) has calculated the stress pattern produced due to the load on hook by using curved beam theory. He compared the analytical result of stress and the stress estimated from the FEM analysis and found that there was 8.26% error between them. His whole study is an initiative to establish a FEA procedure, by validating the results with the help of photo elasticity. From the analysis he found the area at which high stress concentration occurs. For the design improvement if the inner side of hook at the portion of maximum stress is widened then the stress will get reduced. [10]

Pradyumnakeshri maharana .et al. (2012) has estimated hook dimensions for various cross section and topology by keeping the depth and cross section area. It has been concluded from his work that the trapezoidal section was the best cross sectional with minimum von mises stresses and deformations under the given load. [11]

Shaban M.et al. has studied the stress pattern of crane hook in its loaded condition. A solid model of crane hook is prepared with the help of ABAQUS software to obtain the real time pattern of stress concentration in 3D model of crane hook. By estimating the stress concentration area and shape modification is possible to increase its working life and reduce the failure rates. The whole work is a step to form a FEA procedure by validating the results for the calculation of stresses. They investigated if the inner curvature of hook is widened the stress will be reduced. The value and location is very much important factor in reducing the failure. [12]

Patel A.et al. (2013) has worked on reduction of weight of girder which has reduced the cost and increased life of girder. They made a mathematical design for crane component by using ANSYS workbench. They also design hook by using Trapezoidal cross sectional area. [13]

Takuma Nishimura.et al. (2010) has studied damage factor estimation of crane hooks to recognise the tendency of the load condition. It has been estimated that loading conditions were crucial to the crane-hook damages. They used FEM to estimate the relation between the load condition and its deformation. They get the result from the work that the load condition lies between the most downward point and the tip-end point, the direction is toward the gravity direction and the tip-end stress will be reduced. Load-deformation database that has the relation between the load condition of crane hook and its deformation using numerical calculation is constructed. After the completion of study, they found that load acts in downward position and tip-end position and load direction is not downward normal in damaged hook. [14]

Santosh Sahu.et al. (2012) made a model of crane hook of trapezoidal using CATIA V5R20.Then estimated the location of stress after Appling the 2-ton load using FEM. They also analysed the effect of variation in length of two parallel sides of trapezoidal hook on stress. [15]

Chetan N. Benkar.et al. (2014) worked on crane hook for estimated the stress pattern in its loaded condition by preparing a 3-D solid model with the help of ANSYS 14 workbench. By considering various cross sectional area, he obtained real time pattern of stress concentration on the models of crane hook. He calculated stress pattern for various cross section topology such as rectangular, triangular, trapezoidal, and circular by keeping the cross sectional areas are constant and compare the change in result for different topology of cross section. And he found that rectangular cross sectional area gives minimum stress and deformation level. [16]

E. Narvydas.et al. (2012) calculated the stress concentration at shallow notches and smooth lifting hook. And he considered for the durability evaluation and machine element the stress concentration factor is very important. Result is obtained and used with selected generic equation for the stress concentration factor without the usage of FEM. By comparison the shallow notches and smooth trapezoidal cross section hooks investigated the stress concentration factors at the shallow notches of the lifting hooks of trapezoidal cross-section were established by fitting the selected generic equations with comparing to the FEA results. And the difference of the results of the fitted equations and the FEA results were in a range of 3%. In his design rule of the lifting hooks requires ductile material to avoid brittle failure. [17]

Tripathi Yogesh .et al. (2013) has conducted FEM analysis to study the stress pattern of crane hook in its loaded condition, for that a solid model is made with help of CATIA and analysis by using ANSYS14.0. For the correctness of result, the stress in hook compared with the winker- Bach theory. The induced stresses as obtained from Winkler-Bach theory for curved beams are compared with results obtained by ANSYS software. The results are in close harmony with a percentage error of 10.36%. And in his study concluded the complete study is an initiative to establish an ANSYS based Finite Element procedure, by validating the results, for the measurement of stress with Winkler-Bach theory for curved beams. [18]

Y. Torres.et al. (2010) has studied in identifying the cause that led to failure of the crane hook in service. In these studies, includes the comparison of standard governing UNE 58-509-79, UNE-EN 1677-1 & UNE-EN 1677-5 and their experimental results. And also includes visual & microscopic observation, chemical composition analysis, metallographic analysis and simulation of thermal history of hook. Finally, they are decided the following to decrease or eliminate the failure of hook. The steel must be manufactured using electrical melting furnaces & oxygen converters. And the other recommendations or results are the content of aluminium should be higher than 0.025%, but nitrogen cannot exceed 0.0075% and maximum allowable content of sulphur is 0.03%. [19]

Manee-ngam A.et al. (2017) has studies aims for 2 tons lifting hook designed to be the appropriate size for using. By considering of the cost, the strength, the security and the aesthetics by using the optimal designed method together with the highest weight method. The methodology to design the hook has three-step processes. The first step is to create a lifting hook by ISO 7597: 2013 standard (Forged Steel Lifting Hooks with Latch, Grade 8) to be an initial lifting hook then analysed the axial load of 2 tons by the finite element method. The second step is to create and analyse the new design hook by the optimal design. The third step is the results of standard hook sand the new design hook with the same materials depend on the cost, the strength and the safety. And decided the new design hook is better than the standard hook. [20]

Mahesh Solanki.et al. (2014) has studied by selecting having different cross sectional areas hooks which are circular, rectangular, triangular & trapezoidal cross sections and analysis the stress induced in various cross sections. And compare each to other with the results of design calculation. In their studies also observed the parameters that affect the weight reduction. Material type, cross section area & radius of curvature parameters that affect the weight of crane hook. [21]

J.D. Costa.et al. (2001) has concerned with the surface treatment (ion-nitriding) of fretting fatigue and fatigue resistance of 34CrNiMo6 steel. Tests are made on a servo-hydraulic machine under tension for both treated and non-treated specimens. And the test parameters are involving the applied displacement, fretting pressure, fatigue stress amplitude and stress ratio. They proofed by helping of experiments and also putted the chemical and mechanical properties. They conclude the following points depending on SEM and X-ray results of the tested specimens. The first conclusion is the process of ion nitrided improves fatigue resistance of the 34CrNiMo6 steel in the range of lives analysed. The treatment increases the hardness of the surface layer and introduces compressive residual stress preventing crack initiation in the surface layer. As a consequence, crack initiation occurs from internal discontinuities for applied stresses higher than those of the untreated specimens. And finally they conclude fretting fatigue lives of untreated specimens were significantly lower than those specimens treated with ion nitride. Crack initiation in the fretting fatigue of specimens treated with ion nitride also occurs from internal discontinuities. Hence, fretting damage does not seem to influence the life of the specimen. [22]

Sampath S. et al. (2014) has studied on the design and validation of hook used in double spring balancer using finite element method hook is modeled in Creo and analysed in ANSYS software. The ansys analysed result of stress is compared with the theoretical result which is obtained using curved beam design equations. The cross section of hook is T-shape (T- section) and applied different loads from 20 to 100 Kg and compare with stress results. And conclude the design is safe to lift a weight up to 1000 N. [31]

Prashant R. et al. (2015) has Studied about Structural Analysis and improving performance of Crane Hook. And also compare the Manufacturing process of crane hook. Forging is preferred to casting as the crane hooks produced from forging are much stronger than that produced by casting. The reason been in casting the molten metal when solidifies, it has some residual stresses due to non-uniform solidification. Thus casted crane hooks cannot bear high tensile loads. And decide increases the stress slightly though reducing the cost of material. Finally, discus from the stress analysis it is observed that the cross section of max stress area. If the area on the inner side of the hook at the portion of max stress is widened, then the stresses will get reduced. Analytically if the thickness is increased by 3 mm, stresses are reduced by 17%. Thus the design can be modified by increasing the thickness on the inner curvature so that the chances of failure are reduced considerably. [32]

Amandeep Singh. Et al. (2015) has studied about weight optimization by using the lengths of two parallel sides of the cross-section of crane hook are varied and different candidates are obtained for loading capacity of 30 ton on the basis of Mass, total Displacement and Von-Mises stress. It is done to reduce weight of the hook as well as the material cost. And prepared 24 candidates to lift 30 tons by changing the dimentions of the hook cross section and compared this basis on weight, maximum stress and total deformations. Based on this, best three candidates are selected out of the 24 candidates. These best candidates are considered and fatigue analysis is performed on these candidates. The material selected is AISI 4340 150. Along with the mechanical properties of AISI 4340 150, some fatigue parameters are also listed. On the basis of fatigue analysis results obtained, Candidate No.3 can withstand the maximum number of fatigue cycles before failure i.e., the minimum fatigue life obtained is 8.805E7repeats which is better than the other two candidates. Then Candidate No.3 is selected on the basis of fatigue analysis having maximum fatigue life and it is compared with the actual crane hook model. [33]

Jayesh Rajendra Chopda.et al. (2015) in this work considered 50 kN as the working load lifting capacity. And Design, analysis and optimization of EOT Crane hook is carried out. Pro-Engineer software used for modeling and ANSYS software used for obtaining analysis results. For performing optimization, shape optimization is used as a tool. And based on stress concentration the cross-sectional dimensions are changed. Geometry is modified of the selected cross-section shape of crane hook. Here sections are created at twelve locations which are used for modifying dimensions as per stress concentration obtained. Six iterations are performed and the iterations are in deferent area of the hook. Then compared with the basic case (standard hook) and the best optimized iteration is selected which is iteration 6. The iteration 6 is on the top part (straight part) of the hook. Recommended best optimized cross-section is given for manufacturing and testing. [34]

K. S. Raghu Ram.et al. (2017) has studied about Stress Analysis of Traverse Beam Crane Hook (laminated) Used in Steel Melting Shops. The crane hook is fabricated with eight plates of 25 mm thickness and cut to the required size and dimensions and the plates are riveted. The design is done to carries 125 tons load. For this load the stress is found to be high and so high carbon steel with higher yield strength is suggested. Four materials are selected to design and compare the results each to other. The materials are Carbon steel, Cast Steel, SAE 1025 Water quenched, SAE 1096 Oil Quenched steel. The solid model which is created using CATIA v5 is imported into ANSYS. The IGES extension file is imported from catiav5 in ANSYS. Then compare the stress results of each material of hooks. Finally, Carbon steel gives the optimized results and the material is recommended for the fabrication purpose. [35]

Ejaj R. Khan. Et al. (2017) studied about Design and Analysis of Crane Hook with Different Materials. The crane hook is designed by analytical method and design is done for the different materials which are ASTM grade 60(grey cast iron), high strength low alloy steel, structural steel, SAE 1040 and wrought iron. By using curved beam design, the analytical method is done. After the analytical method, by using modeling software (CATIA) the design and modeling of crane hook is done. Then import thr IGS file of modelling in to ANSYS Workbench to finite element amalysis. The FEA gives different results depending on the material type. The high strength low alloy steel material gives minimum stress then taken as optimum result. [36]

Mamta R. Zade.et al. (2017) studied in stresses and Fatigue Analysis of Crane Hook with different materials. Having trapezoidal cross section hook is selected by comparing with other cross sections like rectangular, circular and triangular. Trapozoidal cross section is selected for further static structural analysis with different materials. The design of the hook is done by analytical method. After the analytical method design and modeling of hook is done in modeling soft-ware (CATIA). And the FEA is done by using ANSYS Work bench. By using different materials which are aluminium alloy, structural steel, wrough iron and compare the results of FEA. Depending on the structural analysis results Material for crane hook wrough iron is selected to fatiue analysis. By considering the actual hook material is structural steel and then comparing with wrough iron based on fatigue analysis. And decided wrough iron is best based on the results. [37]

Kunjan B. Vanpariya.et al. (2016) design analysis and weight optimization of crane hook. By selected the as most preferable based on the revised literature survey. The used method for Weight optimization of hook is decrease the cross section of area. To minimize cross section area used variables and constraints parameters. The variable parameters are height of cross section (h) and outer width of cross section (bo). And constraints parameters are inner width of cross section (bi), Hook radius (C) and the Stress inner side (σi) must be less than or equal the Allowable stress of the material. The used material for this analysis is high tensile steel (AISI4140). For Weight Optimization of Lifting Hook, MATLAB software is used for Genetic Algorithm and Prepare a new Model According to Optimization. The 3D Model is made in Proe Wildfire 5.0 and step file import in ANSYS Workbench, then analysis done with 1.5 tonne load applied. After the analysis compare the new results of FEA with the existing crane hook results. [38]

Mahesh Solanki.et al. (2015) studied I weight optimization of crane hooks for different cross sectional. Model different cross sectional hooks which are rectangular, circular, triangular and trapezoidal crane hooks. These hooks are analysis by analytical method and finite element analysis method. Depending on the analysis results of all cross sectional select the trapezoidal cross section. And also compare the analytical stress and von misses stress of the all cross sectional hooks for 20 tons applied load. For the FEA analysis cad model of crane hook is prepared in Creo 2.0. Anysis workbench 14.5 is used for static structural analysis for crane Hook. The used material for this study is SAE 1040 steel. By changing the cross sectional area, and removing material from low stress concentration area in crane hook and then comparing design stress. Compare of results between old model and modified model based on weight, deformation and stress. And decided the new model of crane hook is optimizing result. [39]

Sunil J. Tiwari.et al. (2015) Review the various papers to design the laminated ladle hook and optimization of the same using Finite Element Analysis tool without compromising on the strength & keeping the safety within the specified limits as per the standard. Then conclude laminated crane hooks are widely used handling industry for lifting liquid metal kept much ladles. For view point of safety, the factor of safety kept much higher side hence hook becomes bulky. In ductile fracture, the crack propagates continuously and is more easily detectible and hence preferred over brittle fracture. In brittle fracture, there is sudden propagation of the crack and hook fails suddenly. This type of fracture is very dangerous as it is difficult to detect. The finite element method has been evolved as important tool for design an optimization of the structure. Finally recommended very few article have been published so far regarding stress analysis and design optimization of laminated crane hook. [40]

Niranjan Desai.et al. (2016) Studied optimize the performance of the crane hook based on stress, geometry, and weight. A single load is considered and multiple cross sections including square, circular, and trapezoidal are analyzed. The analysis takes the form of theoretical calculations and finite element analysis through the use of SOLIDWORKS Simulation. The results show that for a given cross sectional area, a trapezoid cross section of a hook has better performance in terms of maximum stress than a circular or square cross section. The parameters of the trapezoidal cross section are varied to determine the values that provide the optimal performance. And decided the highest value for *h* will give the lowest weight. But it is important to keep the proportions of the hook. A very large value of *h* will increase the overall extents of the hook profile and create inefficiencies in packaging. Depending on the results trapezoidal cross section is selected to check with other materials. The materials considered in this optimization are A-36 steel, 6061-T6 Aluminium, and Ti-6AL-4V Titanium. The result in percent decrease of the weight between steel and aluminium or titanium is as large as 80%. This indicates that in terms of performance, aluminium or titanium a clear choice over steel. [41]

Omkar P. et al. (2015) have Studies about trapezoidal crane hook design the hook by using curved beam design theory. The CAD model of a lifting crane hook is initially prepared from the Analytical Design. The designed trapezoidal section crane hook is modeled using UG-NX 8.0. Then import the CAD model into ANSYS Workbench. By applied the load until 20 tons estimated the results of FEM analysed. The model of hook is checked by applied loads of 15, 18 and 20 tons. And select three different still materials which are structural steel, AISI 4140 steel and AISI 4340 steel. The von-misses stress developed in the model by FEM method these results Von-Misses stresses are being compared with Taguchi L9 orthogonal array for specific results. Suggest about fatigue failure of brittle and ductile materials. And also about other failures which are Bending stresses combined with tensile stresses, weakening of hook due to wear, plastic deformation due to overloading, and excessive thermal stresses. And conclude AISI-4340 for 20-ton load is best suitable material for the manufacturing of the crane hook by considering the Taguchi results of stress and deflection. [42]

Studied on the stress pattern of crane hook in its loaded condition, the stresses are obtained by using finite element analysis and Winkler-Batch formula method in different cross- sections are compared. A solid model of crane hook is prepared by using Creo Parametric 2.0 (Computer Aided Three Dimensional Interactive Application) software. The FEM analysis of stress distribution in 3D model of crane hook is obtained using ANSYS software. By starting as the initial point, the material type, cross section area and radius are design parameters that affect the weight of crane hook. To reduce the stress in the cross section of the hook is done by changing its design parameters and increased its mass material. In his study prepared three trapezoidal cross sections having different inner width of the cross section and analysis using analytical and FEM analysis. Compare the results of each modelled crane hooks with respect to analytical & FEM stresses, weight and deformation. Finally put in percentage the value of increased weight and reduced stresses induced on the crane hooks. Then concluded trapezoidal crane hook cross section 3 is better based on the stress value. But in his study less work has been done on the optimization of crane hook because the weight is increased from the standard hook. [43]

This work is to design a crane hook using different materials which are Forged Steel AISI 4140, Grey cast iron, Carbon Steel ASTM A148, Stainless Steel 316(marine). A 3-D model of the crane hook was designed based on the sketch using NX-CADD modeling software and analysis by ANSYS workbench to calculate the total deformation and von mises stress distribution. And compared with the results of each material modelled crane hooks. The concluded Forged Steel AISI 4140 and Stainless Steel 316(marine) are more suitable for making crane hooks. [44]

Improving the durability of E.O.T. crane stracture and optimize the ramshorn hook material for improving its solidity by using FEM. The analysis consists of three major components of E.O.T. crane which are main longitudinal girder, cross girder and ramshorn crane hook. The analysis of ramshorn crane hook is based on the three different materials are En-22, steel-20 and forged steel. And also the trapezoidal cross sectional ramshorn crane hook is checked with another two cross sections which are round and rectangular cross sections. After analysis compared according to the results of Von-Misses stress and total deformation of the trapezoidal cross section ramshorn crane hooks with different materials. Then decided En-22 material is most excellent from another two materials and trapezoidal cross section is better than the round and rectangular cross sections. [45]

Studies the Stress Analysis of Ramshorn crane Hook with Different Cross Sections. By only changing cross sectional shape of the ramshorn crane hook and keep constant material properties, applied load and area of critical cross section for all selected cross sections. The selected cross sections are circular, T section and I section. And the effect of cross sectional shape on stress induced is analysed. The model of Ramshorn hook has been prepared in SolidWorks. And the FEM stress and total deformation analysis of hook is done using ANSYS 14 workbench. The deflection and von Mises stresses have been compared for all selected cross sectional shapes of hook. Then concluded Ramshorn hook with I-cross section is best as it gives minimum stress and deformation value under given load in comparison with other counterparts. [46]

Studied at experimentally investigating the effects of normalizing, and conventional hardening on the hardness, microstructure and toughness of AISI 4340 steel. The specimen was machined to ASTM standards and then different tests like microstructure analysis, hardness test, impact test, and were carried out after the heat treatment processes. It was found the impact resistance value of specimen in normalizing is too high compared to conventional hardening specimen. [52]

the study was focused on investigate the effect of chain installation condition on stress distribution that could eventually cause disastrous failure from sudden deformation and geometric rupture. To estimate the effect of different load cases on structural rigidity of the chain which is used in the crane utilized 3D modelling and finite element analysis. And used CATIA for modelling and ANSYS for the analysis of the chain. Based on the results of finite element analysis decided the stress is strongly affected by the condition of the chain installed in the hook. [53]

2.1. Gap of the Literature

Generally, the literatures stated above studied about the stress on the hooks by using different methods and about causes to fracture of crane hooks. Compare the trapezoidal cross section hook with other regular cross sections which are circular, triangular and rectangular cross sections depending on stress then they decided trapezoidal is best. Some studies are available about weight reduction of hook by changing the material type and cross sections; but in their work when the weight is reducing the stress is increase with comparing the trapezoidal (standard) hook. But there is no work about optimization of weight and stress parallelly by modified the cross section of the crane hook. Which means reducing the weight and stress by comparing the trapezoidal (standard) crane hook with modified one.

By taking this gap, here the weight and maximum stress modified crane hook are reduced parallelly with comparing the trapezoidal (standard) crane hook. And also estimated the fatigue life of the modified crane hook is greater than the trapezoidal (standard) crane hook.

CHAPTER 3

3. MATERIALS AND METHODS

3.1. Materials

3.1.1. Dimensions of standard crane hook

Acording to [47] the proportional dimensions of single shank hook are indicated below. In figure 3-1 and table 3-1 shows all parameters of the shank crane hook are described and the standard (proportional) dimensions are listed. The high and low stress concentration area of cross section are indicated; which are modification area of crane hook to get better results of weight and maximum stress of the crane hook.

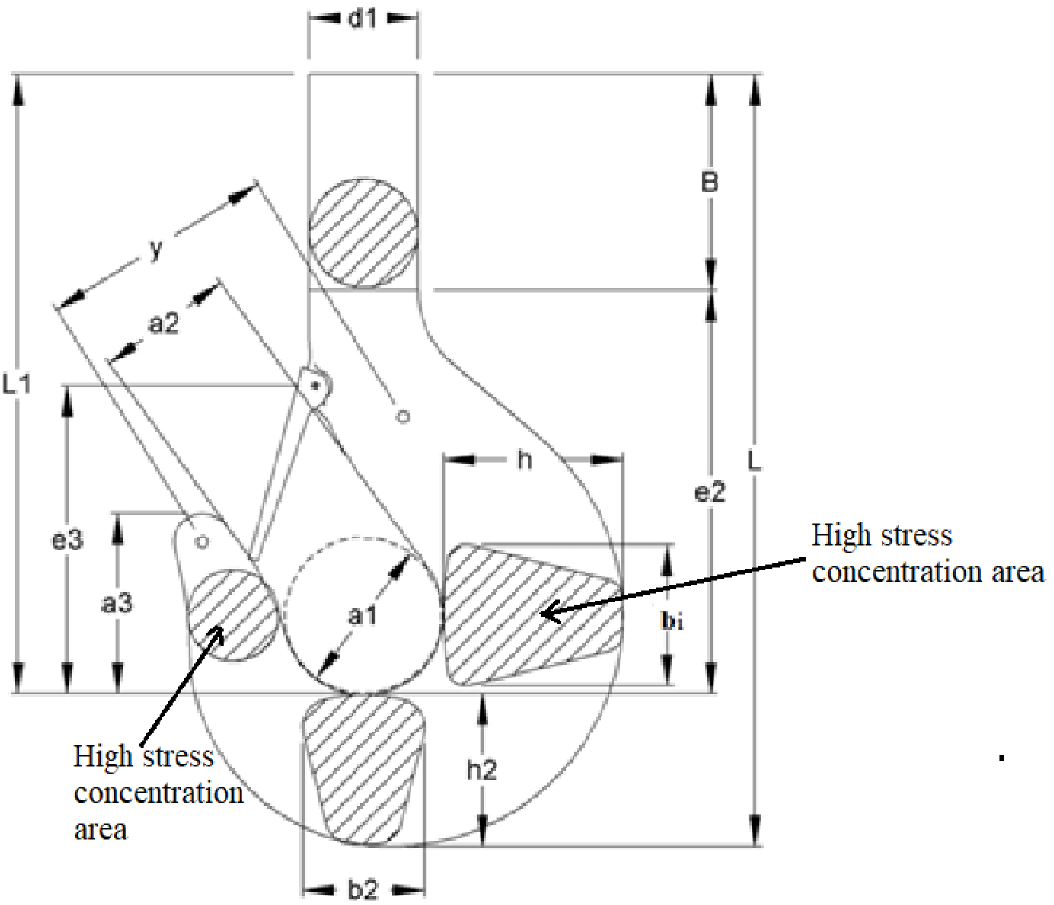


Figure 3-1: dimensions cross sectional view of the standard (trapezoidal cross section crane hook) [47]

Table 3-1: The overall dimensions of the trapezoidal (standard) crane hook [47]

Distance from top to the load applied portion (L1) = 318	Length of the shank (B) = 103
Inner width of cross section at load applied portion b2 = 60	Total height of the crane hook (L) = 393
Distance from lock pin to applied load surface (e3) = 165	Distance from bottom of the shank to the load applied portion (e2) = 215
The height of nose part of the crane hook (a3) = 90	Inner width of high stress concentration area of cross section (bi) = 71

Gap of the curvature (a2) = 63	Height of high stress concentration area of cross section (h) = 90
The diameter of inner curvature of the crane hook (a1) = 80	Height of cross section at load applied portion (h2) =75

For analytical stress analysis method used only the high concentration stress area dimensions which are height of the cross section (h), inner and outer width of the cross section (b_i) and (b_o). Because the analytical stress analysis method is done using curved beam flexure formula (Winkler-Bach formula for curved beam).

3.1.2. Material Selection

3.1.2.1. Material comparison depending on mechanical properties

In many applications, the behaviour of a component in service is influence by several other factors besides the properties of the material used in its manufacture. This is particularly true for the cases where the component or structure is subjected to fatigue loading, the fatigue resistance can be greatly influenced by the service environment, surface condition of the part, method of fabrication and design details. In some cases, the role of the material in achieving satisfactory fatigue life is secondary to the above parameters, as long as the material is free from major flaws. Commonly used material type for design against fatigue is steel. Steels are widely used as structural materials for fatigue application as they offer high fatigue strength and good process ability at relatively low cost. The optimum steel structure for fatigue is tempered marten site, since it provides maximum homogeneity. Steel with high hardenability gives high strength with relatively mild quenching and hence, low residual stresses, which is desired in fatigue applications. Normalized structure, with their finer structure give better fatigue resistance than coarse pearlite structure obtained by annealing. [24]

Table 3-2: Comparison of the materials based on their mechanical properties [50, 51]

Properties	**Stractural Steel**	**AISI 4340 Steel, Normalized**	**AISI 4340 Steel, Annealed**	**AISI 4130 Steel, Normalized**	**Units**

Elastic modulus	200000	205000	205000	205000	N/mm^2
Poisson's ratio	0.3	0.32	0.285	0.285	N/A
Shear modulus	76923	80000	80000	80000	N/mm^2
Mass density	7850	7850	7850	7850	Kg/m^3
Ultimate Tensile strength	460	1110	745	670	N/mm^2
Yield strength	250	710	470	436	N/mm^2

The above table is compared four different materials which are structural steel, AISI 4340 Steel normalized, AISI4340 Steel annealed and AISI 4130 steel normalized based on their mechanical properties. The structural steel is not heat treated and also the strength the material is very small with comparing the others. As mentioned in the above the normalized material is better than the annealed material due to the fatigue resistance. Therefore, depend on the above comparison criteria's AISI 4340 Steel is better than the others. Therefore, for this study selected AISI 4340 steel normalised by consideration of the above steel criteria's.

3.1.2.2. Material comparison based on Analytical analysis of Fatigue S-N Curve

There are five fatigue strength factors to reduce the strength or life of the steel material. The temperature factor, C_T, accounts for the fact that the strength of a material decreases with increased temperature. The reliability factor, C_R, acknowledges that a more reliable (above 50%) estimate of endurance limit requires using a lower value of endurance limit.

Surface factor C_S, is the effect of surface finish. For this study the surface factor for forged is 0.6. This means there are no effect of surface scratches and geometric irregularity at the stress concentration. Parts that are more than 50 mm in diameter and that are subjected to reverse bending should carry a gradient factor C_G and the load factor C_L for bending are0.8 and 1 respectively.

$$S_e = C_L C_G C_S C_T C_R S_e' \tag{1}$$

Where: $S_e' = 0.5S_u$

But the value of correction factors is 0.48 depending on the assumption and the standards of the material type. Therefore, endurance limit and the correction endurance limit are 555 MPa and 266.4 MPa.

The Strength of 1000 cycles for bending load type

$S_m = 0.9S_u = 999$ MPa

Determine method of Fatigue S-N Curve diagram values

On the ANSYS workbench by default displays the S-N curve of structural steel, but the material of this study is AISI 4340 Steel normalized. The S-N curve values of this material can be solved using the generalized S-N formula.

$$\log_{10}(S) = A \log_{10}(N) + B \tag{2}$$

The two constants A and B are unknown, and can be determined as follows. Let the strength at $N = 10^6$ be $S = S_e$ (endurance limit), let the strength at $N = 10^3$ be $S = S_m$. Substituting these values in Equation (2), can obtain:

$\log_{10}(S_e) = A \log_{10}(10^6) + B$

$\log_{10}(S_m) = A \log_{10}(10^3) + B$

Solving for A and B:

$$A = \frac{1}{3} \log_{10}\left(\frac{S_e}{S_m}\right) \quad \text{and} \quad B = \log_{10}\left(\frac{S_m^2}{S_e}\right) \tag{3}$$

Thus the general S-N formula is given by:

$$\log_{10}(S) = \frac{1}{3}\log_{10}(\frac{S_e}{S_m})\log_{10}(N) + \log_{10}(\frac{S_m^2}{S_e}) \quad (4)$$

Now by substituting any value number of cycles (N) in equation (4) can find the corresponding strength (alternating stress).

Similarly, can determine the S_m, S_e and fatigue S-N diagram values for structural steel, AISI 4340 steel annealed and 4130 steel normalized by using general S-N formula. The table is compares three different materials based on the S-N curve values. In the previous section These materials are compared with mechanical properties and fatigue resistance ability. Here also compared with available fatigue life (S-N curve results). Table 3-3 shows comparison of the values of S-N curve for structural steel, AISI 4340 steel normalized, AISI 4340 steel annealed and 4130 steel normalized. Table 3-3 shows comparison of alternating stress and corresponding number of cycles for each material. The values of alternating stress are determine using equation (4) by substituting numbers of cycles.

Table 3-3: comparison of the values of S-N curve for structural steel, AISI 4340 steel normalized, AISI 4340 steel annealed and 4130 steels normalized

Number of cycles (N)	Alternating stress of structural steel in MPa	Alternating stress of AISI 4340 steel normalized in MPa	Alternating stress of AISI 4340 steel annealed in MPa	Alternating stress of AISI 4130 steel normalized in MPa
10	999.28	2411.31	1618.4	1455.48
20	875.16	2111.8	1417.38	1274.69
50	734.42	1772.19	1189.44	1069.7
100	643.2	1552.06	1041.7	936.83
200	563.3	1359.28	912.31	820.47
2000	362.58	874.91	587.22	528.1
10000	266.47	643.02	431.57	388.13

20000	233.38	563.15	377.97	339.92
10^5	171.52	413.88	277.79	249.82
$2*10^5$	150.21	362.47	243.28	218.79
10^6	110.4	266.4	178.8	160.8
10^7	71.06	171.47	115.09	103.5

Most of literatures presented in this paper are structural steel material used to make (manufacture) the crane hooks. But for this study AISI 4340 steel normalized selected with compared other two materials as well as structural steel described in section 3.1.2.1 and its better. Now will also compare based on the S-N curve or fatigue life of the materials.

Figure 3-2 is compared the materials which are structural steel, AISI 4130steel normalized, AISI 4340 steel normalized and AISI 4340 steel annealed depending on the fatigue life (S-N curve). As shown in the figure 3-2 the graph is plotted by using results of table 3-3. The results of AISI 4340 steel normalized shows at stress reaches 2411.31 MPa can survive 10 cycles and at 266.4 MPa the fatigue life cycle is 10^6 cycles. And the AISI 4340 steel annealed counter plot shows 10 and 10^6 cycles can survive at 1618.4 MPa and 178.8 MPa respectively. The maximum number of cycle which is 10^6 of the structural steel is at 110.4 MPa and the minimum cycles are at 999.28MPa. The results of AISI 4130 steel normalized shows at stress reaches 1455.48 MPa can survive 10 cycles and at 160.8 MPa the fatigue life cycle is 10^6 cycles. Therefore, AISI 4340 steel normalized have better life than the other two materials.

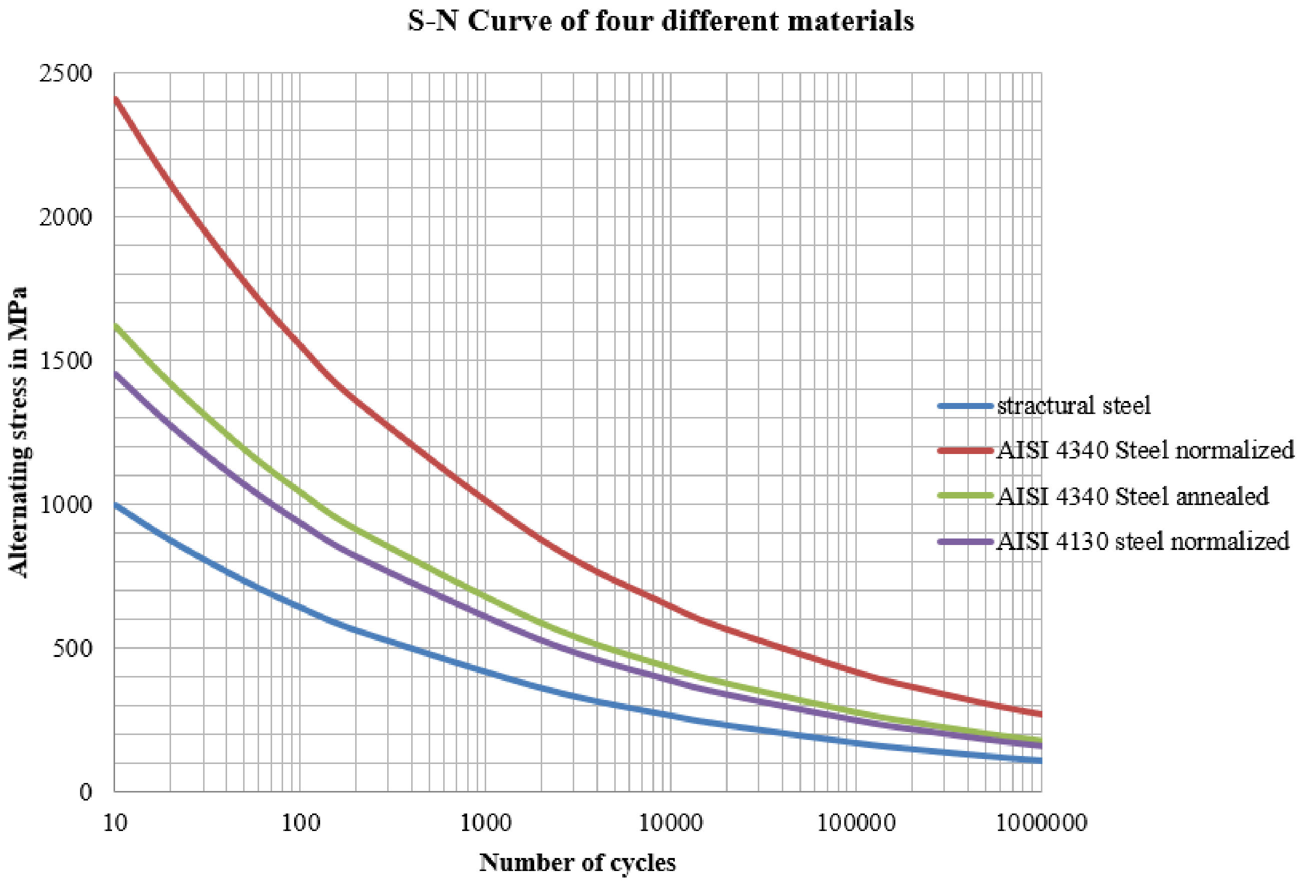


Figure 3-2: Comparison of S-N curve for structural steel, AISI 4340 normalized and annealed steel

3.2. Analysis and optimization Methods of crane hook

Design optimization is an important engineering design activity. The process of determining the best design is called optimization. In general, design optimization determines values for design variables such that an objective function is optimized while performance and other constraints are satisfied. The use of design optimization in engineering design continues to increase, driven by more powerful software packages and the formulation of new design optimization problems motivated by the decision-based design (DBD) framework. Design optimization model is a subjective process that requires engineering judgment and technical skills. In a given design situation, there are likely to be many variables, parameters, constraints, and criteria related to different performance attributes. Thus, there are a variety of relevant optimization models from which to choose. [49]

The design and analysis of crane hook is done by considered as curved beam. Crane hooks with trapezoidal, circular, rectangular and triangular cross section are commonly used. From literatures presented in this paper trapezoidal cross section hook is better than the other cross sections. This study deals with the study of crane hook consists the trapezoidal cross sections as well as the modification of standard hook in which the weight optimization is done by changing cross section. In this study there are three modified cross sectional hooks. Sometimes an accident may occur due to stress concentration factor in it so stress analysis is necessary before make it applicable. And stress analysis is studied in ANSYS workbench 17.2. Now the model and analysis both are carried out by software's. So; it must be designed to deliver maximum performance without failure.

In this study there are four type of cross-section for crane hook namely model-1, model-2, Model-3 and trapezoidal (standard). The theoretical stress calculation is done depend on the cross-section type. All four cross-sections are modelled in SOLIDWORK 17 software. Then, saving. igs file is used further for analysis. After modelling the four types of crane hook cross-sections its. igs file which is saved in SOLIDWORK is imported in ANSYS workbench software and by applying load and proper boundary constraints its analysis is obtained. The results of stress obtained in ANSYS are compared with theoretical calculations and based on the result section having least stress concentration is taken for further optimization.

For this method of optimization, geometry optimization is used as a tool. And based on high and less stress concentration the cross-sectional dimensions are changed. Here there are three new modelled (modified) cross sections which are model 1, model 2 and model 3. In the method of model 1 by reduced and changed the high stress concentration area of cross section. Which are by varying the inner and outer width of the cross sections and keep the height of cross section is constant. In the method of model 2 the geometry is modified by reducing the both sides of the curved member of hook cross section. In the case of model 3 by increase the cross section of high stress concentration area. And reduce and change the cross section of low stress concentration area. Finally, the new modified crane hook models are compared with the trapezoidal (existing) crane hook based on the FEM analysis and selected the best one crane hook.

For this study used three types of geometric modifications to optimize the weight and maximum stress of the hook. Which are model-1, model-2 and model-3. And the results of each modified hooks are compared with trapezoidal hook results based on different criteria's which are max stress, maximum equivalent strain, maximum deformation, weight and fatigue life of the hooks. The applied load for all cross section is the same which is 4.5 tons. And compared the modified hooks each to other then selected the best (better) model.

3.2.1. Methods of Modified Geometry 1 (Model -1)

For this modified hook (Model-1) the independent parameters of high stressed area of cross section are width of inner side (b_i) and width of outer side (b_o). There are three Iterations having different width of inner side (b_i) and width of outer side (b_o) but the cross sectional areas are the same for all Iterations. By adjust (varying) these parameters analysed the stress and selected cross section having better maximum equivalent stress which is Iteration 2 to compare with other model hooks (model-2 and model-3). And the dependent parameters are material properties, the radius of curvature, height of cross section and applied load. By analysed of selected cross section the result of analysed functions are in safe condition.

The two parallel sides of crane hook cross-section namely 'b_i' & 'b_0' are taken as design variables where 'b_i' is width of inner side and 'b_o' is width of outer side of the cross section of crane hook.

Table 3-4: Analysed variables and analysed functions for this model-1 optimization case

Analysed variables	Analysed functions
✓ width of outer side (bo)	✓ Weight (reduced the weight)
✓ width of inner side (bi)	✓ Maximum (von-mises) stress (less than the yield stress of the material)
	✓ Total deformation
	✓ Equivalent elastic strain

Table 3-5: comparison of three different Iterations to select one of these

cross sections	Area of the cross section (A)	width of outer side of the cross section (b_0)	width of inner side of the cross section (b_i)	Maximum Equivalent (von-mises) stress
Iteration 1	4702.5 mm^2	36.5mm	68mm	86.83 MPa
Iteration 2	4702.5 mm^2	33.5mm	71mm	85.30 MPa
Iteration 3	4702.5 mm^2	35mm	69.5mm	86.04 MPa

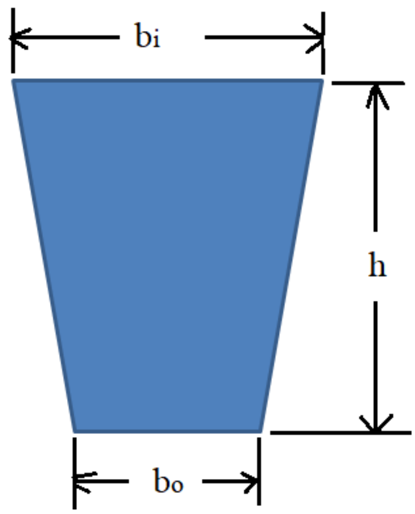


Figure 3-3: cross section of selected iteration (iteration 2)

The dimensions of 'b_i' & 'b_0' for this crane hook 71mm & 33.5mm for the big cross section hook (maximum stress concentration area of cross section). The height and cross sectional area of all iterations are the same the only variables are 'bi' & 'b_0'. Therefore, they have the same weight the comparison is only based on the stress. Based on this, best one iteration is selected out of the 3 iterations to compare with other model hooks (model-2 and model-3) shown in table3-5. By changing (varying) the analysed variables obtained a new optimized design. The above comparison results are obtained by analytical analysis. And Based on the stress comparison of the above 3 iterations, iteration 2 is having better max stress concentration (having less than the other two trials).

3.2.2. Methods of Modified Geometry 2 (Model -2)

In this optimization method the geometry is modified by reducing the both sides of the curved member of hook cross section. The load is applied on the curved inner surface of the hook. Actually the modified cross section is thin at the middle. But there is no any reaction force on the opposite surface (outer surface) of the applied force. Therefore, it is not subjected to buckling.

Figure 3-4: cross section of model 2 crane hook

3.2.3. Methods of Modified Geometry 3 (Model -3)

As shown in the figure3.3 below the minimum stress is produced in the nose portion of crane hook. So by reducing and changing the low stress concentration (nose part of the crane hook) can minimize the weight of the crane hook. The shape of this nose Part of crane hook is changed from circular to rectangular and reduce the cross sectional area. There is no that much effect in stress pattern produced in the modified portion of the crane hook. And by increase the dimension of high stress concentration area of cross section can reduce stress induced on the crane hook. To keep the proportion of shape and size of the crane hook height of high stress concentration area (h) is the same with trapezoidal (standard) crane hook and the only changed cross section parameters are width of inner and outer fibers which are b_i and b_o.

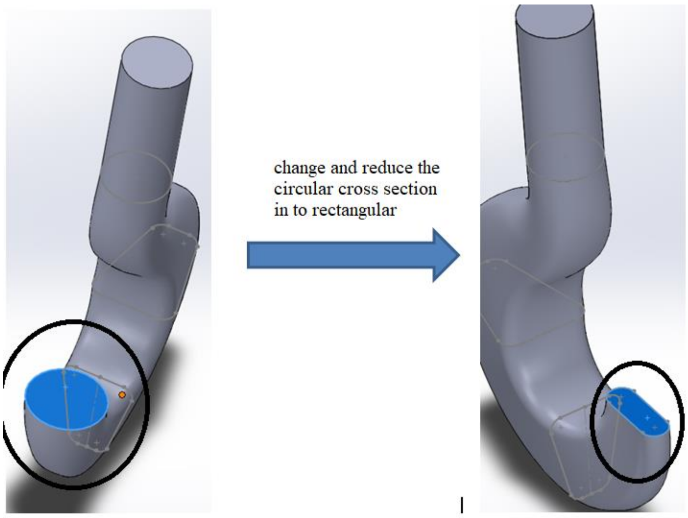


Figure 3-5: Change and reduce the circular cross section at the nose part of the crane hook and increase inner width of the high stress area (b_i)

3.3. Analytical Methods in Structural Design of Crane Hook

According to the curved beam design theory components having curved beam portions are frequently subjected to axial or bending loads or to a combination of bending and axial loads. Structural design of crane hook is designed by considering as curved beam. The distribution of stress in a curved flexural member is determined by using the following assumptions.

- The cross section has an axis of symmetry in a plane along the length of the beam.
- Plane cross sections remain plane after bending.
- The modulus of elasticity is the same in tension as in compression. [23]

The load condition of hook is the combination of bending moment and direct tensile load. The inner surface of the hook is subjected to tensile stress and the outer surface subjected to compression stress due to bending moment and direct tensile load.

According to the literatures presented in this paper concluded having trapezoidal cross section crane hook is better than the other cross sectional hooks which are rectangular, circular and triangular.

3.3.1. Stress analysis for trapezoidal crane hook

To design the trapezoidal cross section hook first draw should have to draw the curved beam with its cross section to shows that the neutral and centroidal axes are not coincident. And also show the parameters and the applied load on the curved beam (hook).

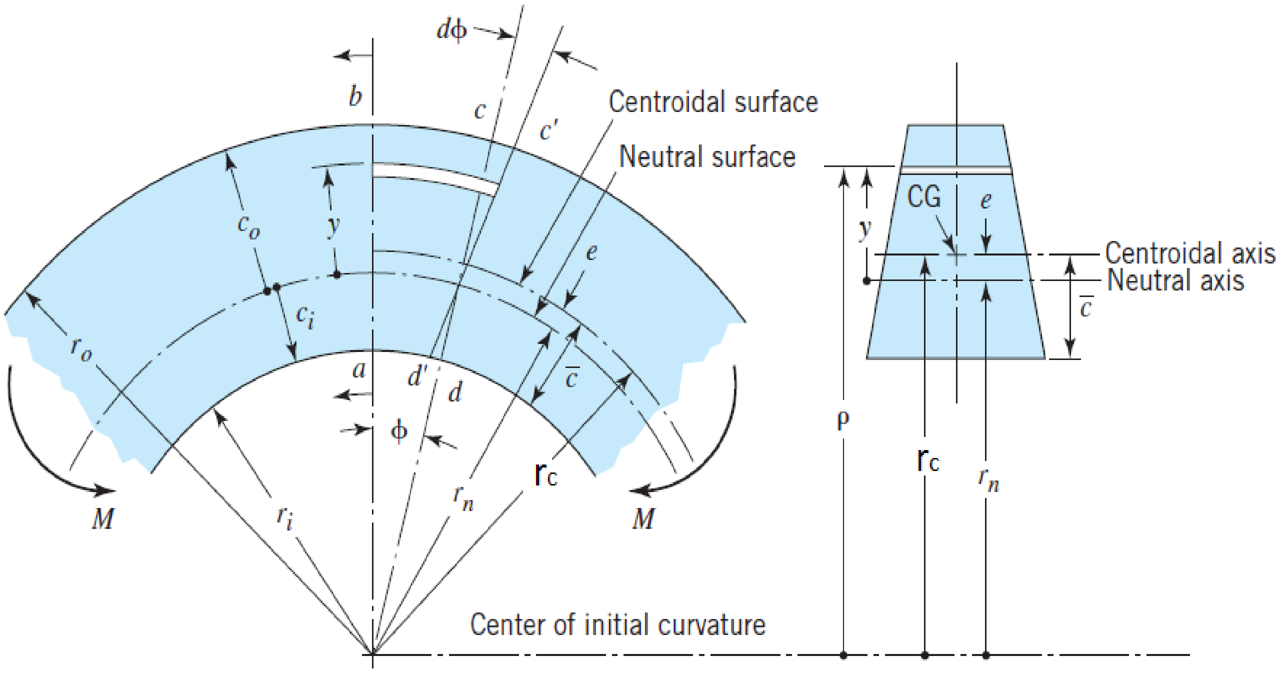


Figure 3-6: curved beam with its cross section to shows that the neutral and centroidal axes are not coincident. [48]

The purpose of this design is to get safe condition of the hook due to maximum stress by using appropriate design dimensions of the hook to lifting the applied load.

Radius of outer fiber (r_o) = 130 mm

Radius of inner fiber (r_i) = 40 mm

Width of the outer fiber (side) of the cross section (bo) = 35.5 mm

Width of the inner fiber (side) of the cross section (bi) = 71 mm

Depth of section (h) = 90 mm [47]

The cross sectional area of trapezoidal cross section is;

$$A = (bi + bo)\frac{h}{2} \tag{5}$$

$$A = (71 + 35.5)\frac{90}{2} = 4792.5 \text{ mm}^2$$

Compute the radius of centroidal axis (rc) based on the cross section type. This is the distance from origin of the curvature to the centroidal axis of the curved beam (hook). For trapezoidal cross section to calculate the radius of centroidal axis is used the following equation.

$$rc = ri + \frac{h}{3} + \frac{bi + 2bo}{bi + bo} \tag{6}$$

$$rc = 40 + \frac{90}{3}\left(\frac{71+2*35.5}{71+35.5}\right) = 80 \text{ mm}$$

Compute the radius of neutral axis (rn) based on the cross section type. This is the distance from origin of the curvature to the neutral axis of the curved beam (hook). For trapezoidal cross section to calculate the radius of neutral axis is used the following equation.

$$\text{rn} = \frac{A}{bo - bi + [\frac{biro - bori}{h}] \ln(ro/ri)} \tag{7}$$

$$\text{rn} = \frac{4792.5}{35.5 - 71 + [\frac{(71*130)-(35.5*40)}{90}] \ln(130/40)} = 71.76 \text{ mm}$$

Neutral axis and centroidal axis are not coincident. Therefore, there is a gap (distance) between the axes and the distance is called eccentricity (e).

$$e = r_c - r_n \tag{8}$$

$$e = 80 - 71.76 = 8.24 \text{ mm}$$

Compute the bending moment about the radius centroidal axis (rc). The force is through the center of curvature. The maximum applied load (P) of this design is 4.5 tons.

$$M = P.rc \tag{9}$$

$$M = 4500Kg \text{ x } 9.81m/s^2 \text{ x } 80 \text{ x } 10^{-3} \text{ m} = 3531.6 \text{ Nm}$$

Calculate the distance from the neutral axis to the inner (Ci) and distance from the neutral axis to outer surface (Co).

$$C_i = r_n - r_i \quad \text{and} \quad C_o = r_o - r_n \tag{10}$$

$$C_i = 31.76mm \quad \text{and} \quad C_o = 58.24 \text{ mm}$$

Calculate the critical maximum stresses at the inside and outside surfaces due to the bending moment. The stress in the inner surface (fiber) of the hook is tension and in the outer surface is compression.

The stress in the inner fiber

$$\sigma_i = \frac{M\,\text{Ci}}{A\,e\,r\text{i}} \tag{11}$$

$$\sigma_i = \frac{3531.6 \text{ Nm x } 0.03176 \text{ m}}{0.0047925\text{m2 x } 0.00824\text{m x } 0.04\text{m}}$$

$$\sigma_i = 71 \text{ MPa}$$

The stress in the outer fiber

$$\sigma_o = \frac{-M\,\text{Co}}{A\,e\,r\text{o}} \tag{12}$$

$$\sigma_o = \frac{3531.6 \text{ Nm x } (-0.05824\text{m})}{0.0047925\text{m2 x } 0.00824\text{m x } 0.13\text{m}}$$

$$\sigma_o = 40.06 \text{ MPa (compression)}$$

The hook is stretched to downward by the applied load the dueto the load the hook is subjected to direct tensile stress.

$$\sigma_t = \frac{P}{A} \tag{13}$$

$$\sigma_t = \frac{4500\ x\ 9.81}{0.0047925}$$

$$\sigma_t = 9.21\ \text{MPa}$$

Calculate the resultant stresses at the inner and outer surface add or subtract maximum stresses at inner and outer surfaces from direct stresses (using superposition) because the hook is subjected to bending and direct tensile stress.

The resultant stress in the inner surface (fiber)

$$\sigma_{Ri} = \sigma_t + \sigma_i = 80.21\ \text{MPa} \tag{14}$$

The resultant stress in the outer surface (fiber)

$$\sigma_{Ro} = \sigma_t - \sigma_o = 30.85\ \text{MPa (compression)} \tag{15}$$

3.3.2. Stress analysis for model -1 crane hook

All cross sectional dimension except width of outer side of the cross section of model 1 crane hook are the same with trapezoidal crane hook cross sections. The changed dimension is only width of the outer side of the cross section. Therefore, Model 1 crane hook can design with the same procedure of the trapezoidal crane hook. Figure 3-7 shows the cross sectional dimensions of high stress concentration area of model-1 crane hook.

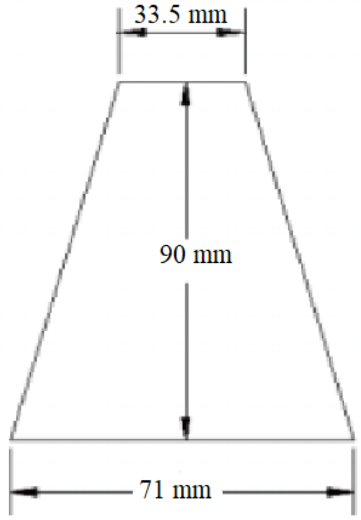


Figure 3-7: Cross section of the High stress concentration area of model-1 crane hook

Radius of outer fiber (r_o) = 130 mm

Radius of inner fiber (r_i) = 40 mm

Width of the outer fiber (side) of the cross section (bo) = 33.5 mm

Width of the inner fiber (side) of the cross section (bi) = 71 mm

Depth of section (h) = 90 mm

The cross sectional area of model 1 cross section is;

$$\mathrm{A} = (bi + bo)\frac{h}{2} = 4702.5 \text{ mm}^2$$

Radius of centroidal axis (rc) for model 1 crane hook

$$rc = ri + \frac{h}{3}\frac{bi+2bo}{bi+bo} = 79.62 \text{ mm}$$

Compute the radius of neutral axis (rn) for model 1 crane hook

$$\mathrm{rn} = \frac{A}{bo-bi+[\frac{biro-bori}{h}]\ln(ro/ri)} = 71.435 \text{ mm}$$

There is a gap (distance) between the neutral and centroid axes and the distance is called eccentricity (e).

$$e = r_c - r_n = 79.62 – 71.435 = 8.185 \text{ mm}$$

Compute the bending moment about the radius centroidal axis (rc).

$$M = P.rc = 4500Kg \times 9.81 m/s^2 \times 79.62 \times 10^{-3} \text{ m} = 3514.825 \text{ Nm}$$

Calculate the distance from the neutral axis to the inner (Ci) and distance from the neutral axis to outer surface (Co).

$$C_i = r_n - r_i \quad \text{and} \quad C_o = r_o - r_n$$

$$C_i = 31.435\text{mm} \quad \text{and} \quad C_o = 58.565 \text{ mm}$$

Calculate the maximum stresses at the inside and outside surfaces due to the bending moment.

The stress in the inner fiber

$$\sigma_i = \frac{M \text{ Ci}}{A\, e\, r\text{i}} = 71.748 \text{ MPa}$$

The stress in the outer fiber

$$\sigma_o = \frac{-M \text{ Co}}{A\, e\, r\text{o}} = 41.14 \text{ MPa (compression)}$$

The direct tensile stress of model 1 crane hook

$$\sigma_t = \frac{P}{A} = 9.39 \text{ MPa}$$

Calculate the resultant stresses at the inner and outer surface add or subtract maximum stresses at inner and outer surfaces from direct stresses (using superposition) because the hook is subjected to bending and direct tensile stress.

The resultant stress in the inner surface (fiber)

$$\sigma_{Ri} = \sigma_t + \sigma_i = 81.14 \text{ MPa}$$

The resultant stress in the outer surface (fiber)

$$\sigma_{Ro} = \sigma_t - \sigma_o = 31.75 \text{ MPa (compression)}$$

Therefore, the stress concentrated cross section of model 1 crane hook is subjected to 81.14 MPa tensile stresses on the inner fiber and 31.75 MPa compression stress on the outer fiber.

3.3.3. Stress analysis for model -2 crane hook

For the model -2 cross section have some different method to calculate the radius of neutral axis (r_n) and radius of centroidal axis (r_c) because the cross section is complex. Determining r_c for complex cross section is can be done easily by most CAD programs. For this cross section used SolidWork to determine the r_c. According to solidwork the value of r_c is 75.96 mm and the moment of area (I) is 4068382.6 mm^4. The methods for determining the didtance from neutral to centroid axes (e) are needed to use simple approximation of e, it can be shown that. Figure3-8 shows the high stress concentration area of cross section for model-2 crane hook.

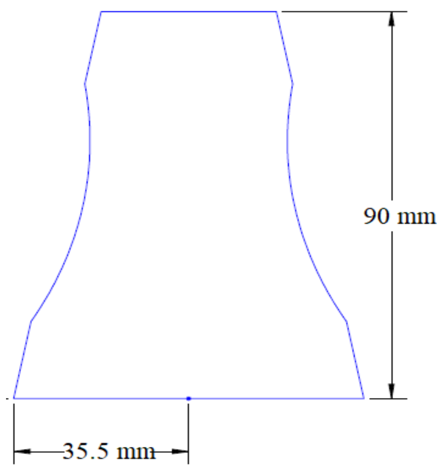


Figure 3-8: high stress concentration area of cross section for model-2 crane hook

$$e \approx \frac{I}{rc\,A} \tag{16}$$

$$e \approx 6.08 \text{ mm}$$

To determine the radius of neutral axis r_n

$$r_n = r_c - e$$

$$r_n = 69.88 \text{ mm}$$

The bending moment about the centroid of the cross section

$$M = P.rc = 4500Kg \times 9.81 m/s^2 \times 75.96 \times 10^{-3}\ m = 3353.2542\ Nm$$

Calculate the distance from the neutral axis to the inner (Ci) and distance from the neutral axis to outer surface (Co).

$$C_i = rn - ri \quad \text{and} \quad C_o = r_o - r_n$$

$$C_i = 29.88\ mm \quad \text{and} \quad C_o = 60.12\ mm$$

Calculate the resultant stresses at the inner and outer surface add or subtract maximum stresses at inner and outer surfaces from direct stresses (using superposition) because the hook is subjected to bending and direct tensile stress.

The resultant stress in the inner surface (fiber)

$$\sigma_{Ri} = \sigma_t + \sigma_i$$

$$\sigma_{Ri} = \frac{P}{A} + \frac{M\,Ci}{A\,e\,ri} = 103.54\ MPa$$

The resultant stress in the outer surface (fiber)

$$\sigma_{Ro} = \sigma_t - \sigma_o$$

$$\sigma_{Ro} = \frac{P}{A} - \frac{M\,Ci}{A\,e\,ri} = -47.87\ MPa$$

3.3.4. Stress analysis for model -3 crane hook

To keep the proportion of the crane hook size andshape the parameters (dimensions) are the same with trapezoidal (standard) crane hook. The only modifies are width of inner and outer high stress concentration area of cross section which are b_i and b_o.

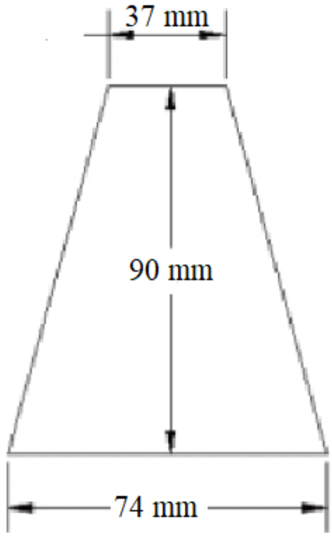


Figure 3-9: cross section of high concentration area of cross section model 3 crane hook

Radius of outer fiber (r_o) = 130 mm

Radius of inner fiber (r_i) = 40 mm

Width of the outer fiber (side) of the cross section (bo) = 37 mm

Width of the inner fiber (side) of the cross section (bi) = 74 mm

Depth of section (h) = 90 mm

The cross-sectional area is;

$$A = (bi + bo)\frac{h}{2} = 4995 \text{ mm}$$

The radius of centroidal axis (r_c) and radius of neutral axis (r_n)

$$rc = ri + \frac{h}{3} + \frac{bi+2bo}{bi+bo} = 80 \text{ mm}$$

$$rn = \frac{A}{bo-bi+[\frac{biro-bori}{h}]\ln(ro/ri)} = 71.764 \text{ mm}$$

The bending moment about the radius centroidal axis

$$M = P.rc = 3531.6 \text{ Nm}$$

The hook is stretched to downward by the applied load the dueto the load the hook is subjected to direct tensile stress.

$$\sigma_t = \frac{P}{A} = 8.838 \text{ MPa}$$

The stress in the inner fiber (σ_i) and the stress in the outer fiber (σ_o)

$$\sigma_i = \frac{M\ \mathrm{Ci}}{A\ e\ r\mathrm{i}} = 68.17 \text{ MPa}$$

$$\sigma_o = \frac{-M\ \mathrm{Co}}{A\ e\ r\mathrm{o}} = 38.456 \text{ MPa (compression)}$$

The resultant stress in the inner surface (fiber)

$$\sigma_{Ri} = \sigma_t + \sigma_i = 77.01 \text{ MPa}$$

The resultant stress in the outer surface (fiber)

$$\sigma_{Ro} = \sigma_t - \sigma_o = -29.618 \text{ MPa (compression)}$$

Table 3-6: analytical analysis results of crane hooks

Crane hook	Bending moment about centroidal axis (M) in Nm	Resultant stress in the inner surface (σ_{iR}) in MPa	Resultant stress in the outer surface (σ_{oR}) in MPa
Trapezoidal crane hook	3531.6	80.21	-30.85
Model-1	3514.8	81.14	-31.75
Model-2	3353.25	103.54	-47.87
Model-3	3531.6	77.01	-29.618

The above table 3-6 shows the analytical results of bending moment and maximum stress of the crane hooks. Model-3 crane hook have smaller maximum stress than others. and the maximum stress model-2 is large with comparing the other crane hooks.

3.4. Geometrical Modelling and Numerical Analysis Using Ansys

ANSYS is general-purpose finite element analysis (FEA) software package. Finite Element Analysis is a numerical method of constructing a complex system into very small pieces called elements. The software implements equations that govern the behaviour of these elements and solves them all; creating a comprehensive explanation of how the system acts as a whole. These results then can be presented in tabulated or graphical forms. This type of analysis is typically used for the design and optimization of a system for too complex and difficult to analyse by hand. Systems that may fit into this category are too complex due to their geometry, scale, or governing equations. The product offers a complete set of elements behaviour, material models and equation solvers for a wide range of engineering problems. In addition, ANSYS mechanical offers thermal analysis and coupled-physics capabilities involving acoustic, piezoelectric, thermal structural and thermal electric analysis. [28]

For the analysis of this study is using ANSYS workbench 17.2 under static structural. Static structural have the following input requirements to analysis the object. 3D model, material properties generate mesh and applied load. Then it solves the problem and presents the results. The main difference of numerical methods from analytical methods is an extra step called mesh generation. This step divides the complex model into small elements that become solvable in an otherwise too complex situation. The procedures that have to be carried out are:

- Building the geometry; Constructing the three-dimensional representation of the object (hook) using the SOLIDWORK. It can be modelled by using ANSYS mechanical APDL but for this study planned to model by using SOLIDWORK.
- Defining of Material Properties or engineering data; now the ANSYS workbench17.2 is opened and from the menus of static structure must define the engineering data (material properties). And then import the SOLIDWORK model in to the geometry of ANSYS workbench.
- Generating Mesh; At this point ANSYS understands the makeup of the parts which defines how the modelled system should be broken down into finite pieces. Input the mesh sizing depends on the object to analyse.

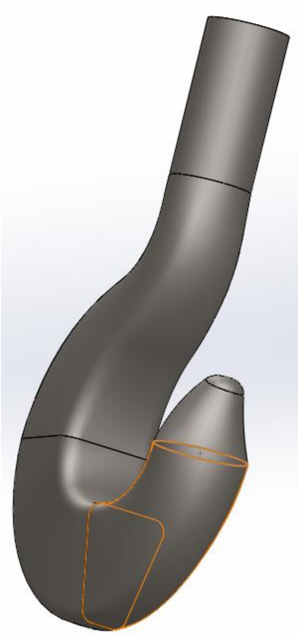
- Applying Loads; once the system is fully designed, the last task is to burden the system with constraints, such as physical loadings or boundary conditions.
- Reading the Solution: This is the difficult step, because ANSYS needs understanding within, what state (steady state, transient), the problem must be solved.
- Presentation of the Results; after the solution has been obtained, there are many ways to present ANSYS' results such as tables, graphs, and contour plots.

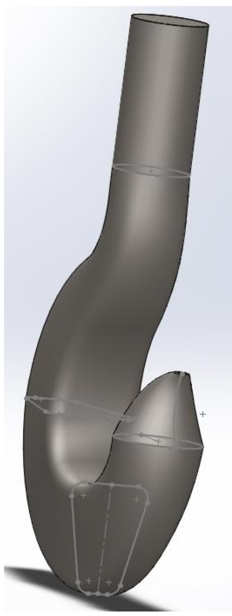
3.4.1. Geometrical Modelling of the Hooks

SolidWorks is design automation software. SolidWorks is used by students, designers, engineers, and other professionals to produce simple and complex parts, assemblies, and drawings. Today this software is used around the world to design products, develop machinery, and create production systems. Mechanical engineering, industrial design, and transport technologies are just a few of the functions in which SolidWorks software is successfully used as an advanced tool by designers and engineers.

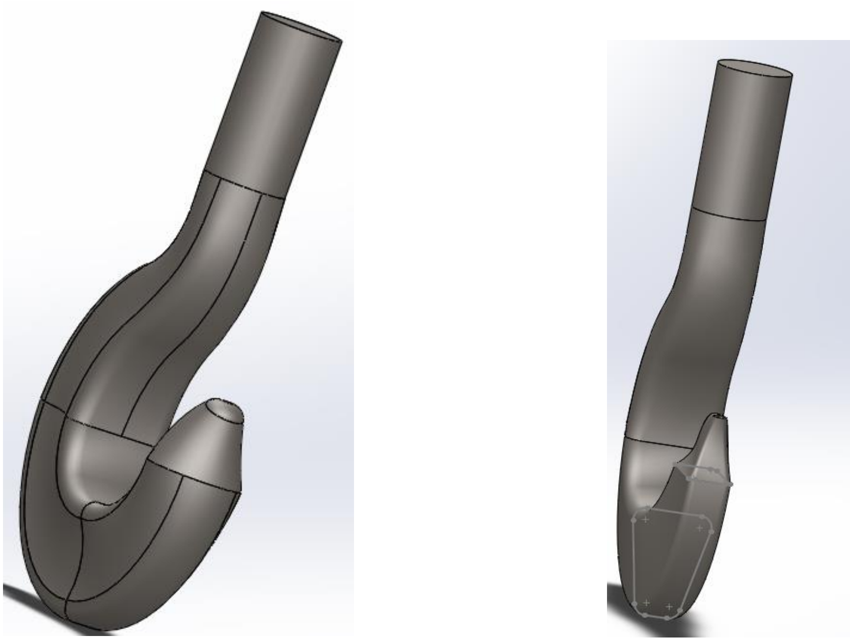
Therefore, for this study SolidWork is selected to design or 3D modelling of the hooks because the hooks have complex shape and cross section. And solid work is easy to design these kinds of parts.

As mentioned above for this study the geometrical modelling of the crane hooks are done using SolidWork 17. Then the 3D modelling is imported to the ANSYS workbench 17.2 to finite element analysis.





(a) Trapezoidal (b) Model 1



(c). Model 2 (d). Model 3

Figure 3-10: The 3D models of different cross sectional hooks which are trapezoidal, Model-1, Model-2 and Model-3.

The models of different cross-sectional crane hooks which are trapezoidal, Model-1, Model-2 and Model-3 cross sections are shown in the figure above3-10. The models are done by using SolidWork 17 and the purpose of the models is to visualise (describe) the shape of the hooks.

3.4.2. Mesh Generating of the crane hook

ANSYS Meshing is a component of ANSYS Workbench. In the finite element method, the structure of interest is sub-divided in to discrete shapes called elements. Finite element methods have proved indispensable for physical simulation. These methods discretize the simulated domain for example, for this study the different cross sections crane hooks dividing in to many small elements. The most common element types include one dimensional beam, two dimensional elements, or three-dimensional bricks, typically triangles or quadrilaterals in two dimensions and tetrahedra or hexahedra in three. The complex of elements is the mesh. The Purpose of the Mesh generating is Domain is required to be divided into discrete cells (meshed) and Equations are solved at cell/nodal locations. Refine (smaller cells) used for high solution gradients and fine geometric detail. Solution accuracy & stability deteriorates as mesh cells deviate from ideal shape. [29, 30] In this study have four types of cross sections which are trapezoidal, mode-1, model-2 and model-3 cross sections. The mash size for all cross sections is the same which is 4 mm but to get accurate answer of FEA results the mesh size at high stress concentration area is 1 mm. depending on the volume of the models of crane hooks the number of elements and nodes for each model are different. The numbers of elements and nodes for each model of crane hook are listed in table 3-7.

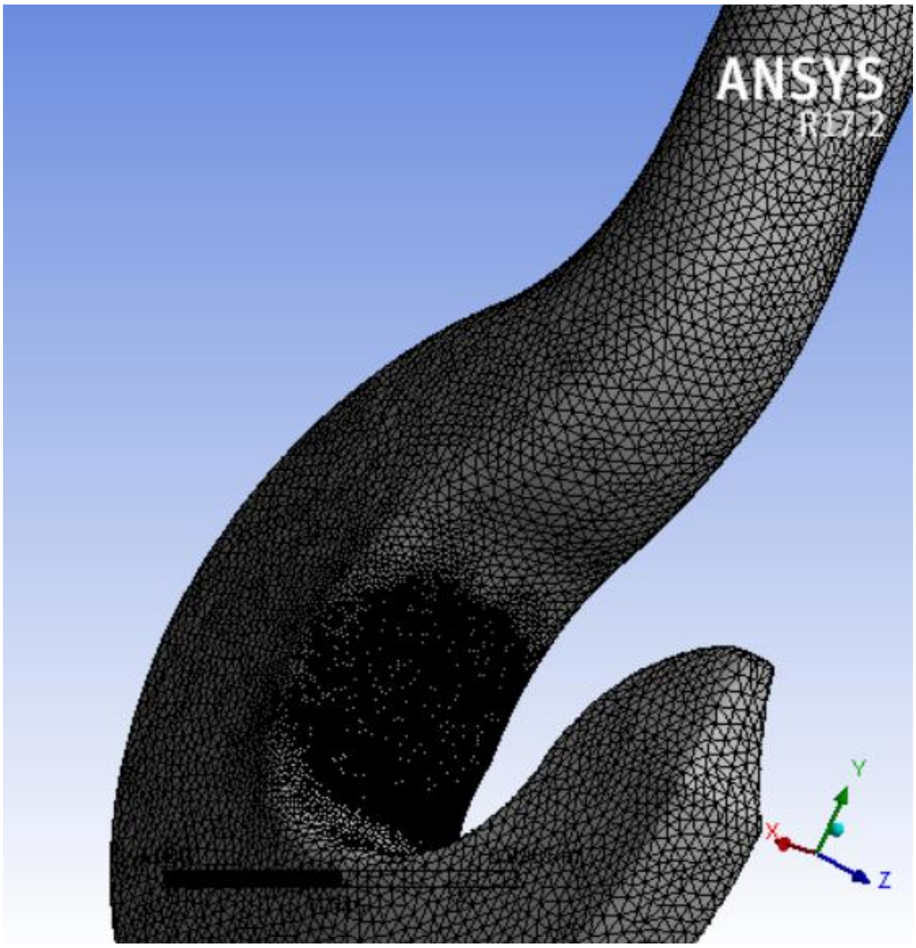


Figure 3-11: The mesh generating for the different cross section hooks

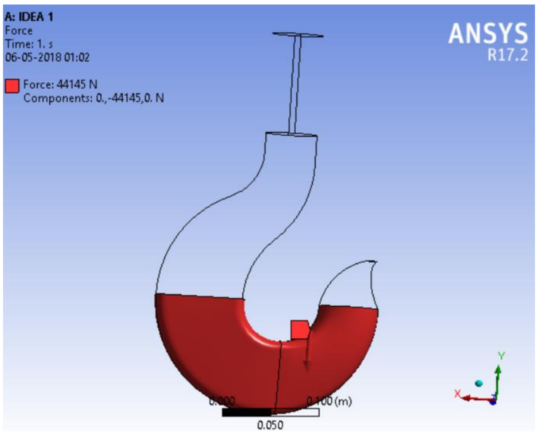
Table 3-7: indicated the mesh generating properties (number of elements and nodes)

Model of crane hook	Number of elements	Number of nodes
Trapezoidal	272408	391074
Model-1	270811	387911

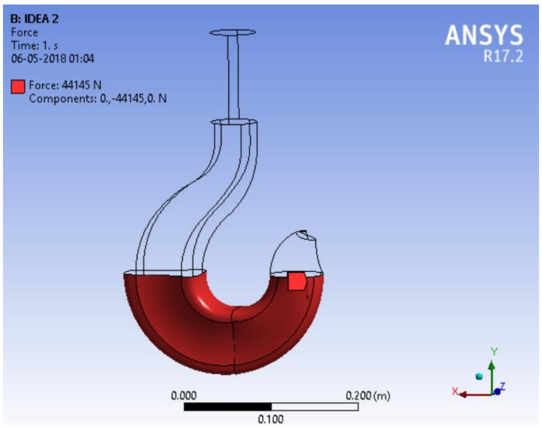
Model-2	256997	369467
Model-3	266904	383795

3.4.3. Load condition of the crane hook

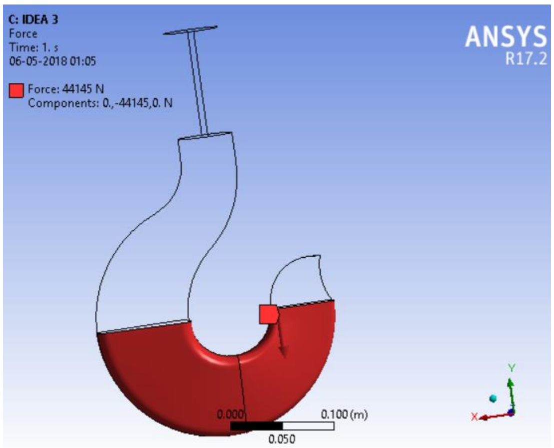
The load condition for all cross section is the same in magnitude and direction because the direction of applied load is the same for all cross sections. It is applied vertically to downward direction as shown in the figure 3-12. For these hooks the applied load is 4.5 tons (44145 N) and the result of total deflection, equivalent strain and Von-Mises stress is deferent depend on the cross section. The force is distributed on the red colours of the hooks and it is the same for all cross-sectional hooks.



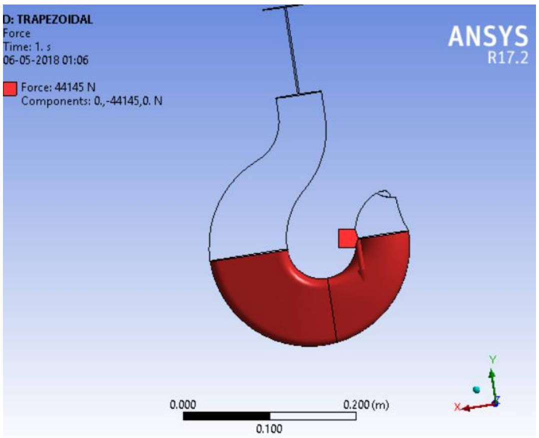
(a) Model-1 cross section



(b) Model-2 cross section



(c) Model-3 cross section



(d).Trapezoidal (standard) cross section

Figure 3-12: Applied force on the trapezoidal, Model-1, Model-2, and Model-3 cross section hooks.

3.5. Fatigue Analysis of Crane Hooks using Finite Element Method

Fatigue is the weakening of a material caused by repeatedly applied loads. It is the localized structural damage that occurs when a material is subjected to cyclic loading. Fatigue occurs when a material is subjected to repeat loading and unloading. If the loads are above a certain threshold, microscopic cracks will begin to form at the stress concentrators area or surface. Eventually a crack will reach a critical size, the crack will propagate suddenly, and the structure will fracture. The shape of the structure will significantly affect the fatigue life.

The objective of this analysis is to evaluate the fatigue failure or to evaluate the number of cycle the hook going to experience before it fails. Before proceed with the complete steps involved in the fatigue failure analysis.

While many parts may work well initially, they often fail in service due to fatigue failure caused by repeated cyclic loading. Characterizing the capability of a material to survive the many cycles a component may experience during its lifetime is the aim of fatigue analysis. In a general sense, Fatigue Analysis has two main methods which are Strain Life and Stress Life. There are available within the ANSYS Fatigue Module. Fatigue analysis can be classified in to high cycle fatigue (HCF) which is greater than 10^5 cycles and low cycle fatigue (LCF) which means less than 10^5. For this study used stress life. Stress life of the hook which means the total life of the hook. Total life hook is the summation of crack initiation and crack life of the hook. The hook is experienced the number of cycle greater than 10^5cycles which means it is HCF and it is inclusive of infinite life.

The analysis type is stress life because for this study is interested to estimate total life (crack initiation and crack life). The fatigue strength factor and scale factor are equal to one which means there is no surface imperfection and cracks on this model (surface of geometry). The mean stress theory is chosen Goodman theory and the stress component is equivalent von-mises stress.

3.5.1. Fatigue Life and Damage estimation of hooks by FEM

Fatigue life is a mechanical and scientific term that relates to how long an object or material will last before completely failing because of concentrated stresses. There are a number of different factors that can influence fatigue life including the type of material being used, structure, shape and temperature changes. In most cases, fatigue life is calculated as the number of stress cycles that an object or material can handle before the failure. There are several different types of stress values that are considered when computing fatigue life, including the maximum stress value, which is usually less than ultimate tensile stress limits. When a material is put into use, the design can increase the stress that is put on the object. For instance, certain sharper angles, such as the corners in a square object, can be a significantly higher stress area than a rounded area, which disperses the weight and stress of a load more evenly over a larger area. Smooth transitions or fillets will increase the fatigue strength of the structure. Fatigue damage is defined as the design life divided by the available life. For Fatigue Damage, values greater than 1 indicate failure before the design life is reached.

3.5.2. Safety factor of crane hooks

Safety factor (SF) is a term describing the load carrying capacity of a system beyond the expected or actual loads. Essentially, the factor of safety is how much stronger the system is than it needs to be for an intended load. For most metal (ductile) materials, it is often required that the factor of safety be checked against both yield and ultimate strengths. The yield calculation will determine the safety factor until the part starts to plastically deform. The ultimate calculation will determine the safety factor until failure. To choose the appropriate design factors are based on several considerations, such as the accuracy of predictions on the applied loads, strength, wear estimates, and the environmental effects.

CHAPTER 4

4. RESULT AND DISCUSSION

In this portion is discussed about the whole results of the crane hooks. To achieve the objective of this paper stress and fatigue analysis are done on the different modelled crane hooks. Then should have discuss about these results and compare the results each to other.

4.1. Static Structural Analysis of Crane Hooks

Here the results of total deformation, maximum Von-Misses stress, weight and equivalent elastic strain of the crane hooks are used to compare the crane hooks each to others.

4.1.1. Static structural analysis of trapezoidal crane hook

This section discusses the static structural analysis of trapezoidal crane hook. Figure 4-1. (d) shows the von-misses stress, total deformation and equivalent elastic strain of trapezoidal (existing) crane hook by applied load of 4.5 tons. The stress analysis has been done by ANSYS software, where the results have been presented by contours and numerical values. For the trapezoidal crane hook the maximum von-misses stress result from ANSYS is 84.149 MPa. And also, the maximum total deformation and equivalent elastic strain results are 0.2755 mm and 0.0004105. The weight of trapezoidal (standard) hook indicated in the ANSYA workbench properties of geometry is 14.233 Kg.

4.1.2. Static structural analysis of Model 1crane hook

The ANSYS analysis results of the **model 1** crane hook as follow presented by contours and numerical values. For this crane hook the maximum von-misses stress, total deformation and equivalent elastic strain is 85.563 MPa, 0.2785 mm and 0.0004174 respectively. And the weight of model-1 hook indicated in the ANSYA workbench is 13.984 Kg.

Table 4-1: Comparison of static structural analysis of model 1 crane hook with trapezoidal crane hook

Cross section of hook	**Load (tons)**	**Maximum Total deformation (mm)**	**Max Equivalent elastic strain (m/m)**	**Max Equivalent (von-mises) stress (N/m^2)**	**Weight (kg)**
Model-1	4.5	0.2785	$0.4174 x 10^{-3}$	$85.563 x 10^{6}$	13.984
Trapizoidal	4.5	0.2755	$0.4105 x 10^{-3}$	$84.149 x 10^{6}$	14.233

In the above table 4-1 the model-1 is compared with trapezoidal depending on criteria of maximum stress, strain, deformation and weight of the hooks. The total deformation and maximum equivalent elastic strain almost the same. Slightly model-1 is increased than trapezoidal. The total deformation and equivalent elastic strain of model-1 are increased by 0.003mm and 0.000007.

The basic comparison must be based on the stress and weight of the hooks. Because the safety of the hook is depending on the stress induced on the hook. The increased stress of model-1 from trapezoidal is 1.414 MPa (1.68%) but it is under safe condition. And the reduced weight model-1 from trapezoidal is 0.249 kg (1.75%).

4.1.3. Static structural analysis of Model 2 crane hook

Results are obtained from analysis of Crane hook using ANSYS for 4.5tons load acting on Crane hook. The maximum Von-Mises stress is 87.682 MPa, total deformation is 0.28443 mm and the maximum equivalent elastic strain is 4.277×10^{-4}. And the weight of model 2 crane hook is 13.426 Kg.

Table 4-2: Comparison of static structural analysis of **model 2** crane hook with trapezoidal crane hook

Cross section of hook	**Load (tons)**	**Maximum Total deformation (mm)**	**Maximum Equivalent elastic strain (m/m)**	**Maximum Equivalent (von-mises) stress (N/m^2)**	**Weight (kg)**
Model-2	4.5	0.2844	$0.4277x10^{-3}$	$87.682x10^6$	13.426
Trapizoidal	4.5	0.2755	$0.4105x10^{-3}$	$84.149\ x10^6$	14.233

The above table 4-2 is compared trapizoidal hook and model-2 with different results which are total deformation, equivalen elastic strain and von-mises stress. The weight of model-2 is reduced by 0.807 kg(5.716%) compared to the trapizoidal hook which are shows the weight on the above table 4-2. Actually the stress on the model-2 is greater than the trapizoidal by 3.533 MPa(4.2%) with the same applied load. But its with in the limit stress of the material, then it is safe as concept of design.

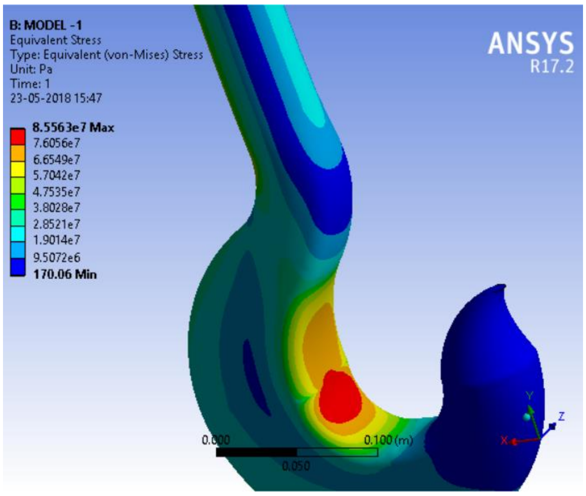
4.1.4. Static structural analysis of Model 3 crane hook

From the result of ANSYS the maximum von-misses stress acting on the inner fiber of the Model 3 crane hook is 80.93 MPa. The maximum total deformation and the maximum equivalent elastic strain are 0.30225 mm and 0.3989.

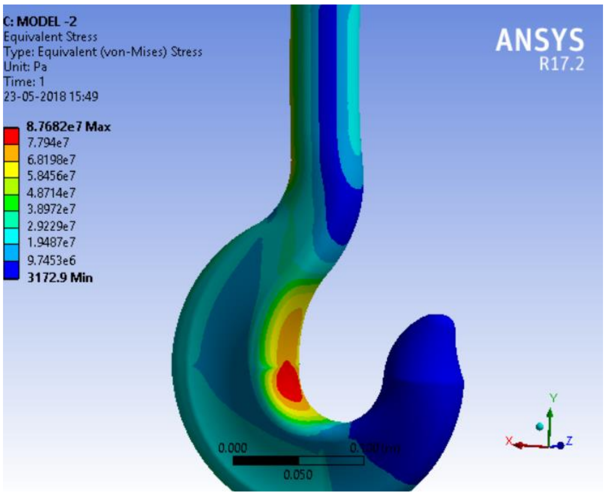
Table 4-3**:** comparison of trapezoidal and model-3 hooks

Cross section of hook	**Load (tons)**	**Maximum Total deformation (mm)**	**Maximum Equivalent elastic strain (m/m)**	**Maximum Equivalent (von-mises) stress (N/m^2)**	**Weight (kg)**
Model-3	4.5	0.3022	$0.3989x10^{-3}$	$80.93\ x10^6$	13.879
Trapizoidal	4.5	0.2755	$0.4105x10^{-3}$	$84.149\ x10^6$	14.233

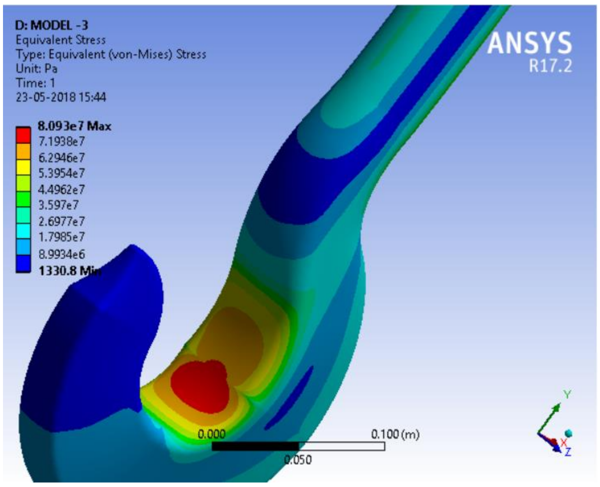
In the above table 4-3 the model-3 is compared with trapezoidal with criteria of maximum stress, maximum equivalent elastic strain, total deformation and weight of the hooks. The maximum equivalent elastic strain of model-3 crane hook is less than the trapezoidal. But total deformation of model-3 is greater than trapezoidal. The total deformation of model-3 increased by 0.0267mm and the maximum equivalent elastic strain reduced by 0.0000116 from trapezoidal. The stress of model 3 is better than the trapezoidal stress. The reduced stress and weight of model-3 from trapezoidal is 3.22 MPa (3.825%) and 0.354 kg (2.487%).



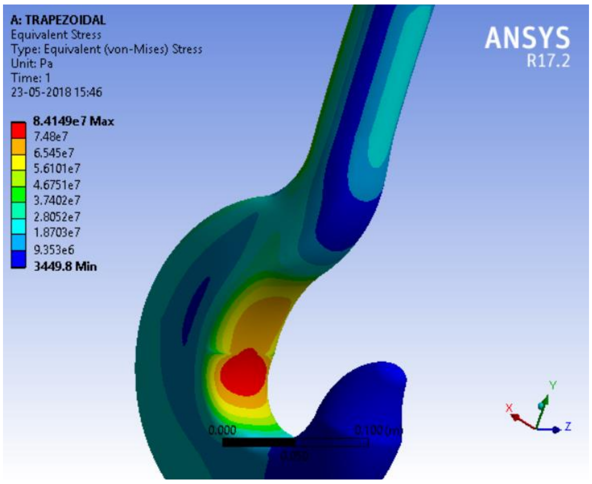
(a). Equivalent (Von-Misses) stress model -1



(b). equivalent (Von-Misses) stress model -2



(c). Equivalent (Von-Misses) stress model -3



(d). Equivalent (Von-Misses) stress Trapezoidal

Figure 4-1: Maximum equivalent (Von-Misses) stress of modelled crane hooks by the maximum applied load

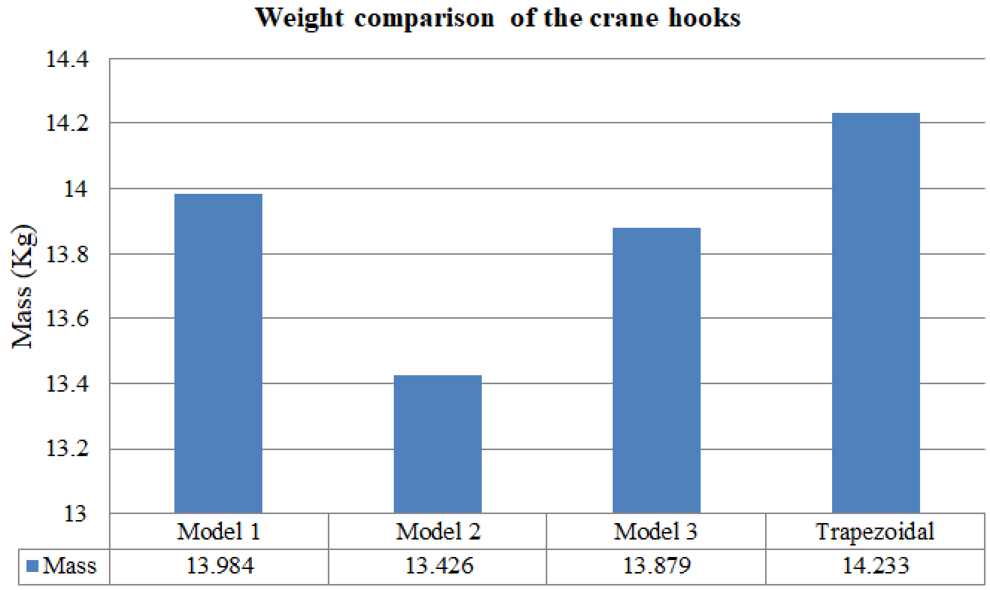
Figure 4-1 shows the results of maximum stresses of the all modelled crane hooks. Model-3 has better result of stress with comparing the others. And the result of model-2 is greater than the other modelled crane hooks.

4.2. Stress and Weight Comparison of the modelled crane hooks

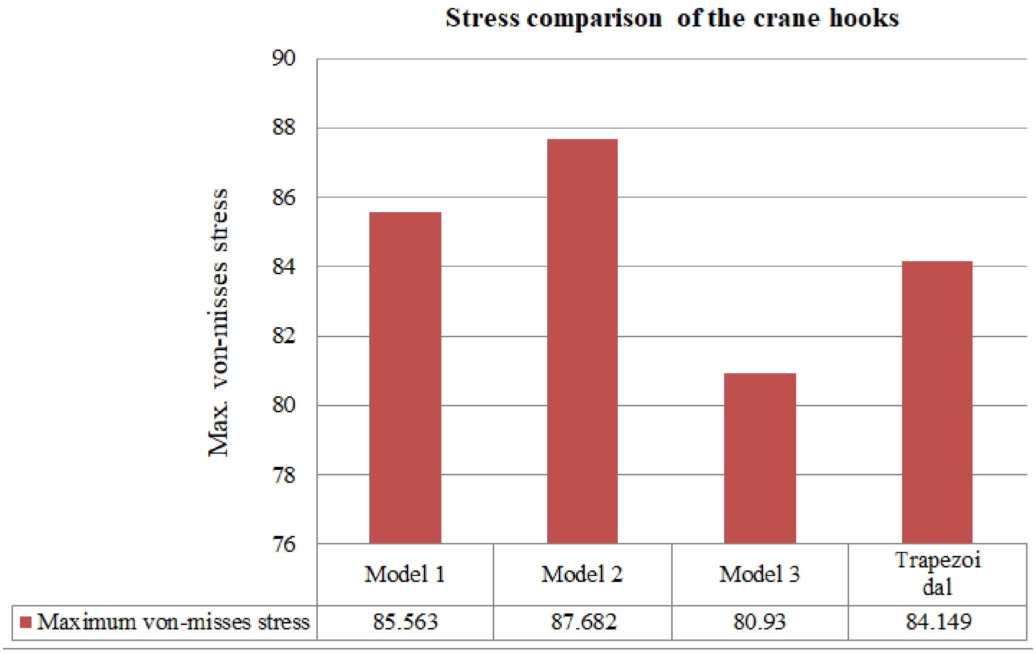
This portion discusses about the comparison of weight and maximum stress of new modelled crane hooks (model 1, model 2 and model 3) and the standard (trapezoidal). As we know the reduction of weight of any mechanical part has more advantages and increasing of stress is reversely because it will decrease the life time of the material (hook). The safety of the hook is depending on the stress induced on the hook and the main objective of this study is weight optimization with safe life condition. Therefore, the comparison must be based on stress and weight of the hooks. Then here compare the hooks based on the above two things.

Table 4-4: Comparison of hooks depending on maximum stress and weight of the crane hooks

Type of crane hook	Maximum von-Misses stress in (MPa)	Weight of the crane hook In (Kg)
Model 1	85.563	13.984
Model 2	87.682	13.426
Model 3	80.93	13.879
Trapezoidal	84.149	14.233



(a) Weight comparison of crane hooks



(b) Comparison of Maximum Von-Misses stress of the hooks

Figure 4-2: Maximum Von-Misses stress and weight comparison of the modelled crane hooks.

The table 4-4 and figure 4-2 shows the comparison of stress and weight of the modelled crane hooks. When each optimization models are comparing with the trapezoidal (standard) crane hook;

The weight of model-1 crane hook is reduced by 0.249 Kg and the maximum Von-Misses stress induced on the model-1 crane hook is increased by 1.414 MPa with comparing the trapezoidal (standard) crane hook.

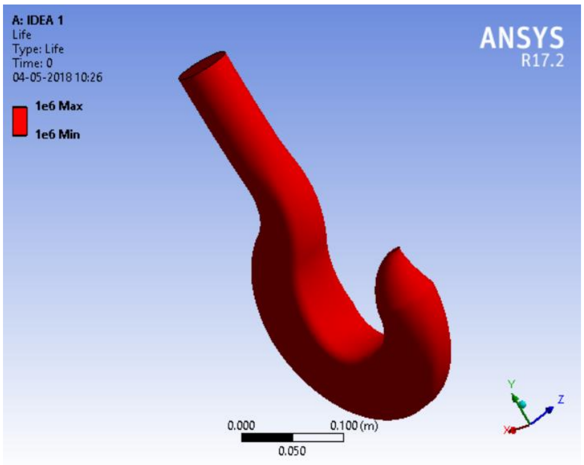
Based on the weight comparison the weight of Model-2 is smaller than the trapezoidal (standard) and other models of crane hooks but its maximum Von-Misses stress is greater than the all models of crane hooks. The weight is reduced by 0.807 Kg and the maximum Von-Misses stress is increased by 3.533 MPa from trapezoidal (standard) crane hook.

Model-3 crane hook is having better results of weight and stress than trapezoidal (standard). The weight and stress of model-3 crane hook reduced by 0.354 Kg (2.49%) and 3.461 MPa (4.133%). Therefore, based on the FEM analysis of weight and maximum Von-Misses stress of model-3 is better than the other crane hooks.

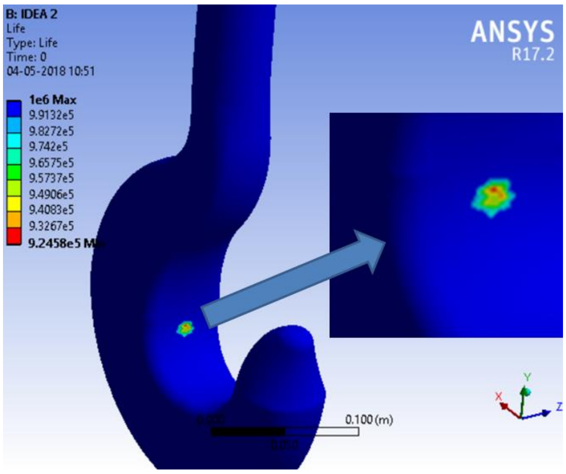
4.3. Fatigue Analysis of Crane Hooks

4.3.1. Fatigue Life and Damage estimation of hooks by FEM

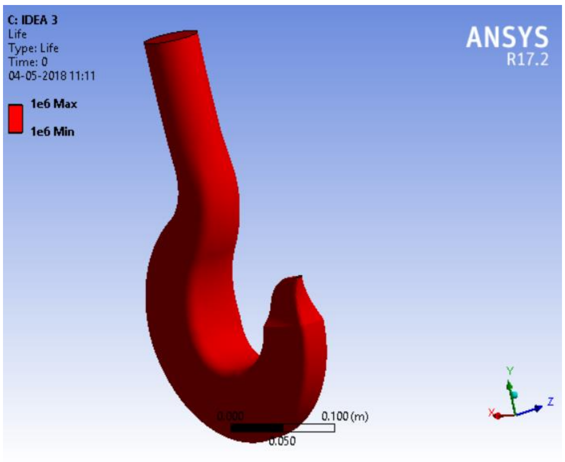
Fatigue life is the available life for the given fatigue analysis. Fatigue life can be over the whole model or parts, surfaces, edges and vertices. This result contour plot shows the available life for the given fatigue analysis.



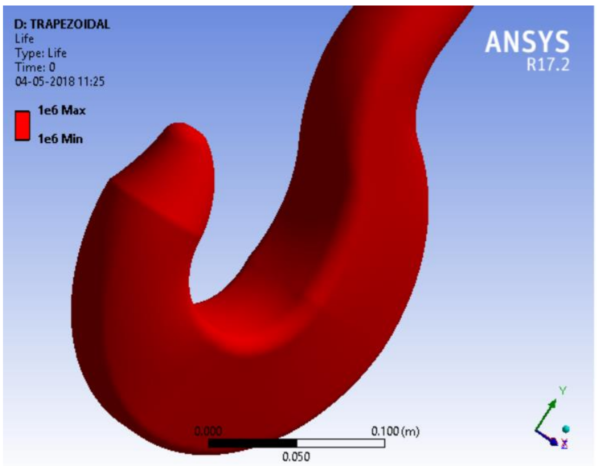
(a). fatigue life of model 1



(b).fatigue life of model 2



(c). fatigue life of model 3



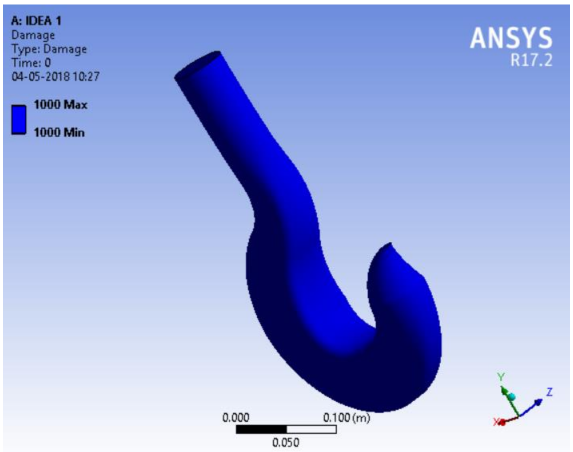
(d) fatigue life of trapezoidal

Figure 4-3 The fatigue life estimatted of the hooks by FEM

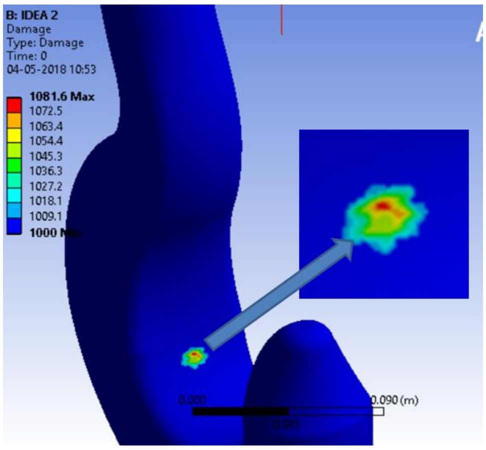
From the details of figure 4-4 clearly indicated the maximum and minimum life of model-1, model-3 and trapezoidal hooks are the same which is 10^6 cycles. Therefore, these can survive infinite life of cycles. But the minimum number of cycles for model-2 is 9.2458 x 10^5. The localized area is going to fail before it reaches infinite life of cycles. This means a localized stress concentration is happening because of fatigue loading.

Hook is highly responsible component then it must be safe at all conditions. The working process of crane is over head of the human beings and other fixtures (materials). If it is fail in the working time can happened serious accident. Therefore, should have design for infinite life of cycles (more than 10^6 cycles).

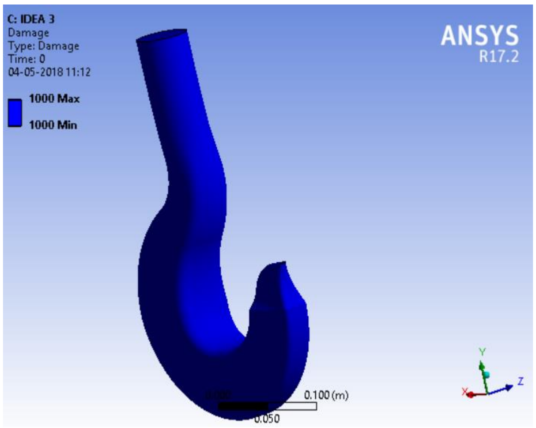
Fatigue damage is defined as the design life divided by the available life. For Fatigue Damage, values greater than 1 indicate failure before the design life is reached. This result contour plot shows the fatigue damage for the given fatigue analysis hooks. Therefore, model-2 crane hook result indicates failure can happen before its design life.



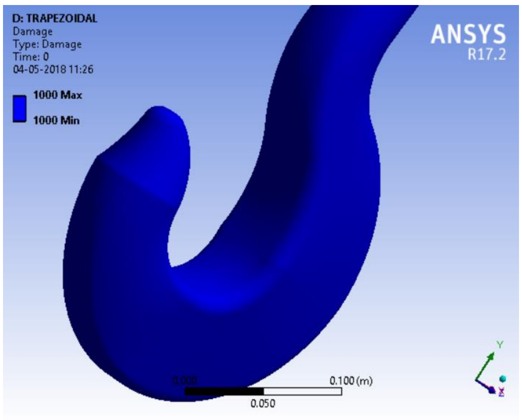
(a). model-1 hook



(b) model-2 hook



(c). model-3 hook



(d) trapezoidal hook

Figure 4-4: Estimated Fatigue damage of hooks

4.3.2. Comparison of the crane hooks with total deformation verses fatigue life

The total deformation of all models of crane hooks are greater than the trapezoidal (standard) crane hook. Should have check the effect of the increased total deformation on the life of crane hook by varying the applied load. Table 4-5 shows comparing of fatigue life versus total deformation by varying the applied load. The applied load is varied from 90% up to 150% of the design load (maximum load) to get different values of total deformation and fatigue life (numbers of cycle) depending on the applied load. Then compared the all crane hooks in figure 4-6 by using results of table 4-5.

Table 4-5: comparison of fatigue life and total deformation

	Trapezoidal		**Model-3**		**Model-1**		**Model-2**	
Load (N)	Total deformation (mm)	Numbers of Cycles	Total deformation (mm)	Numbers of Cycles	Total deformation (mm)	Numbers of Cycles	Total deformation (mm)	Numbers of Cycles
39730.5	0.24796	1000000	0.27202	1000000	0.25066	1000000	0.25599	1000000
44145	0.27551	1000000	0.30225	1000000	0.27851	1000000	0.28443	917780
48559.5	0.30306	691800	0.33247	849400	0.30636	632240	0.31287	554300
52974	0.33061	425020	0.36269	531560	0.33421	385260	0.34132	332520
57388.5	0.35816	261110	0.39292	332710	0.36206	234760	0.36976	200150
61803	0.38571	179160	0.42314	208250	0.38991	169210	0.3982	155320
66217.5	0.41326	140510	0.45337	161540	0.41776	132160	0.42665	120570

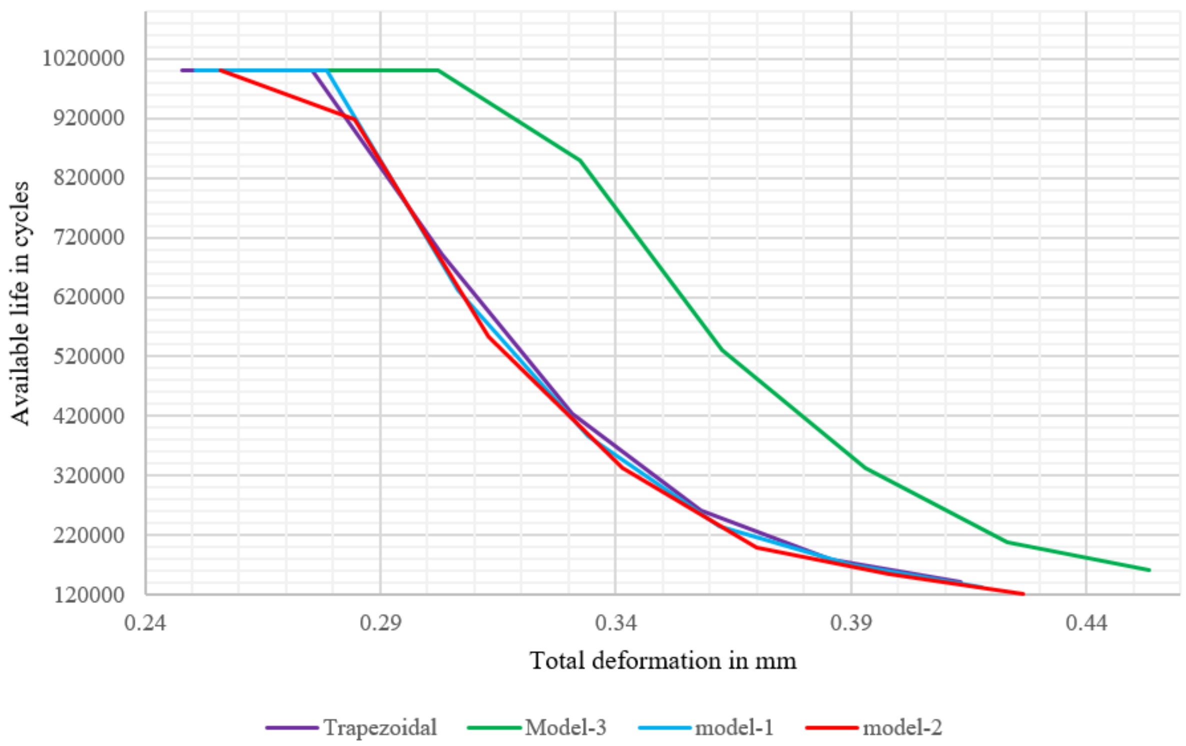


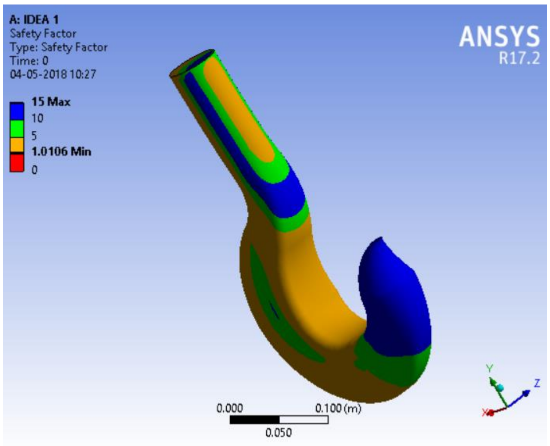
Figure 4-5: The graph of total deformation and available life of the crane hooks

Figure 4-6 indicates the graph of total deformation versus available life of each crane hooks. Actually, the total deformation of Model-3 is greater than the trapezoidal (standard) crane hook and it is occurring on the nose part of the crane hook (low stress concentration area of the crane hook). But it has not any effect on the life of the crane hook because the life of model-3 crane hook is better than the others at any condition. The green plot line on the figure 4-6 shows the graph of total deformation and the available life of model-3 crane hook.

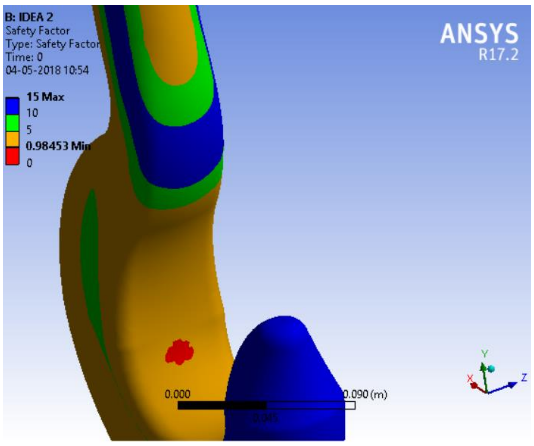
The results of model-1 and model-2 crane hooks is shows having less life than the trapezoidal (standard) crane hook with the same total deformation.

4.3.3. Fatigue Safety factor of the crane hooks

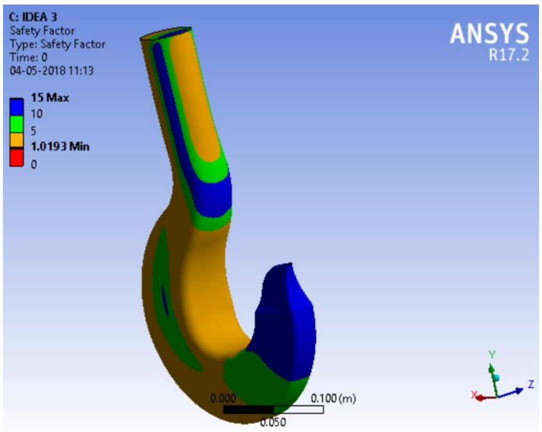
The results contour plot of figure 4-7 and figure 4-8 shows the maximum and minimum safety factor of the all modelled crane hooks. For fatigue safety factor values less than one indicates failure before the design life is reached. The maximum value of safety factor for all models of crane hook is 15.



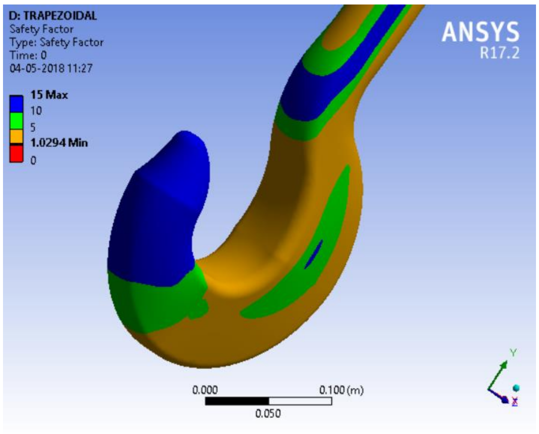
(a). model-1 hook



(b) model-2 hook



(c). model-3 hook



(d) trapezoidal hook

Figure 4-6: Safety factor contour plot of the crane hooks

The minimum safety factor results of all crane hooks except model -2 is greater than one at 4.5 tons applied load. This means it cannot fail before the design life. But the minimum value safety factor of model -2 is 0.98453 which means it can fail before the design life of the crane hook.

Table 4-6: Comparison of minimum fatigue safety factor with applied different (variable) loads.

Applied load	Safety factor of Trapezoidal crane hook	Safety factor of Model-1 crane hook	Safety factor of Model-2 crane hook	Safety factor of Model-3 crane hook
22072.5	2.0487	2.0149	1.9662	2.1302
26487	1.7073	1.6791	1.6385	1.7752
30901.5	1.4634	1.4392	1.4044	1.533
35316	1.2805	1.2593	1.2289	1.3314
39730.5	1.1382	1.1194	1.0923	1.1835
44145	1.0244	1.0074	0.9831	1.0651
48559.5	0.93124	0.91586	0.89372	0.96829
52974	0.85364	0.83954	0.81925	0.8876

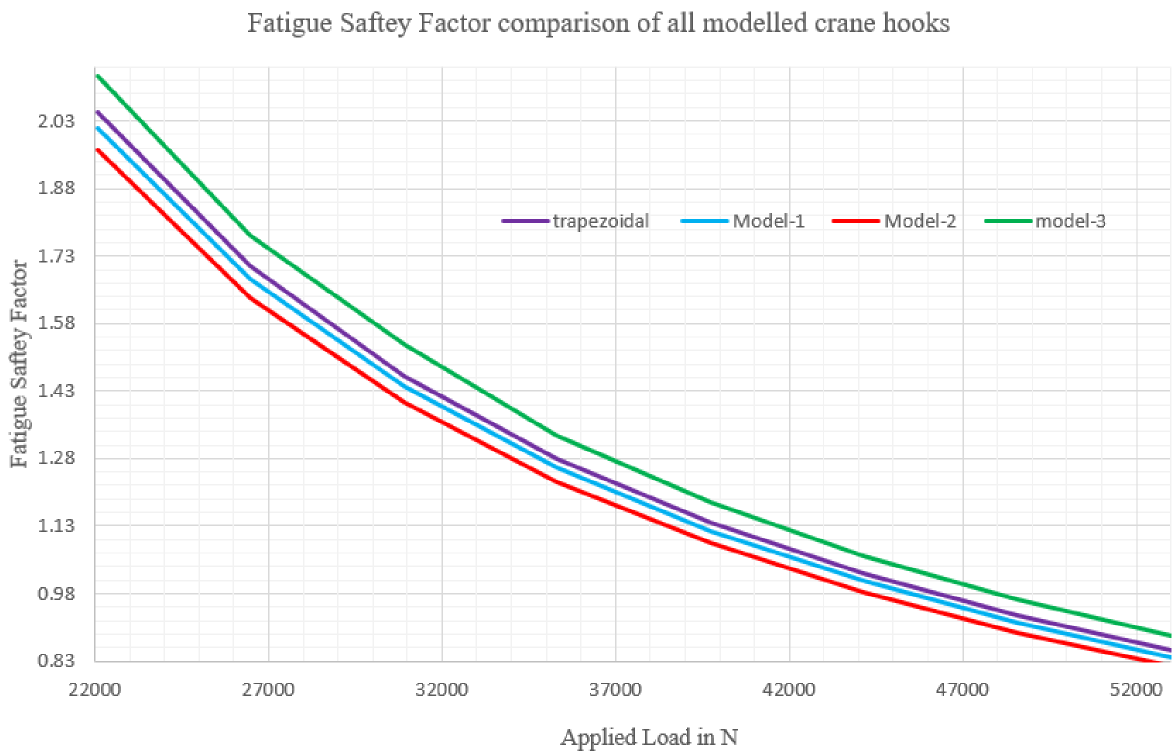


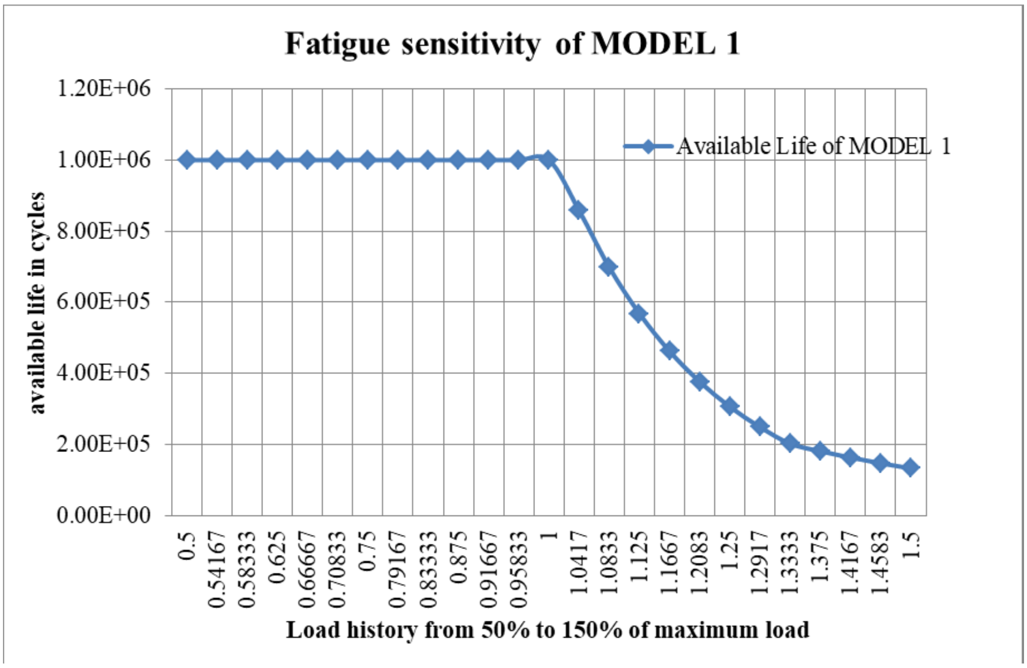
Figure 4-7: the fatigue safety factor of the modelled crane hooks by applied different loads

Table4-6 and figure 4-8 shows Comparison of minimum fatigue safety factor with applied variable loads. Naturally crane hook is subjected to loading and unloading condition with different loads. The loads are varied from 50% to 120% of the design load (maximum load) of crane hook. When the applied load is increase the safety factor is decrease for all models of crane hook. The maximum fatigue safety factor for all models crane hooks is 15 but the minimum value of safety factor is different. Then the comparison is based on the minimum value of fatigue safety factor.

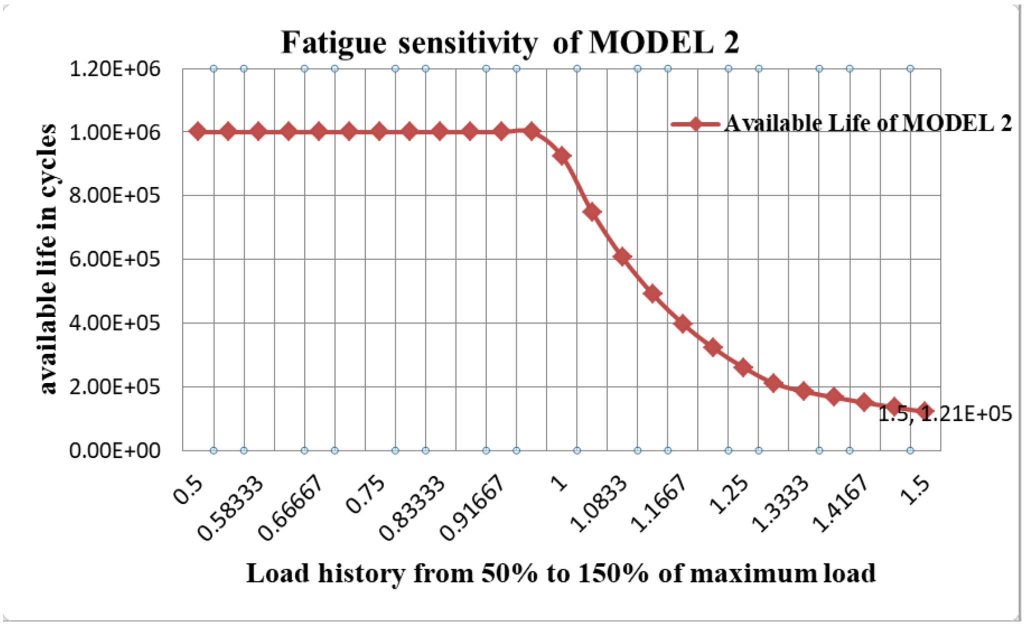
Depending on the comparison results of all crane hook models model-3 is better than the others as well as trapezoidal (standard) crane hook. Which means model-3 have better life than the others.

4.3.4. Fatigue sensitivity of hooks

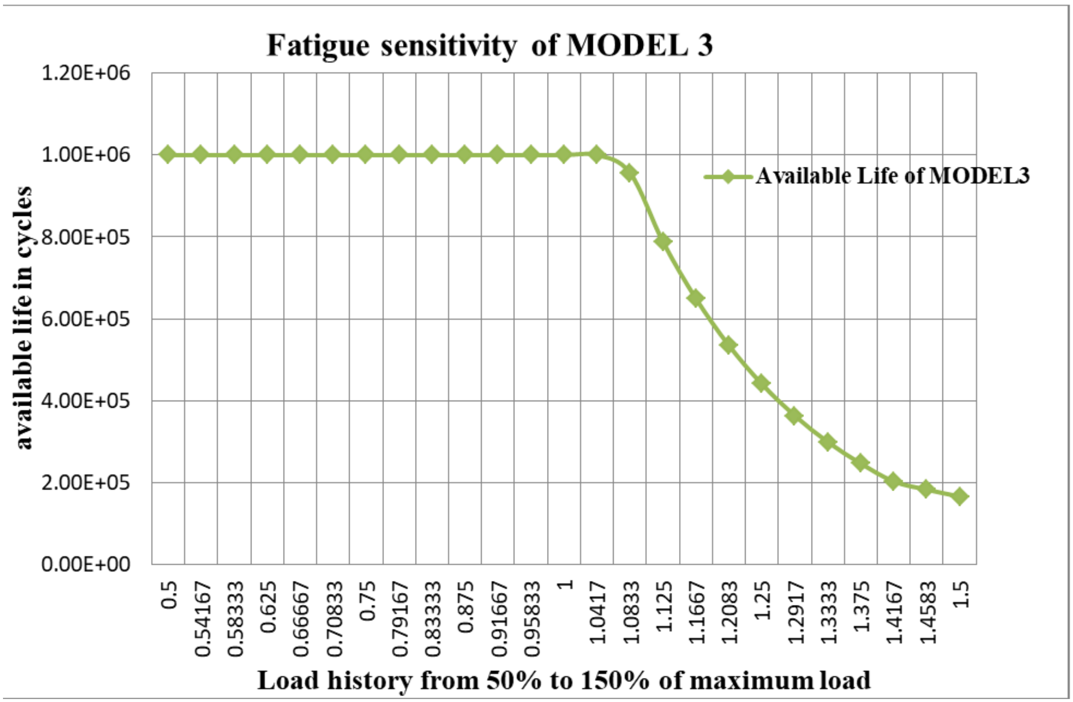
Fatigue sensitivity is shows how the fatigue results change as a function of the loading at the critical location on the model (hook). Sensitivity can found for life, damage or factor of safety. It can estimate in ANSYS by setting the number of fill point as well as the load variation limits. To estimate the fatigue sensitivity of this study used the load variation from 50% to 150% of the current load is applied. Which means checked the available life of the model by loading 0.5 up to 1.5 times of the current load (design load).



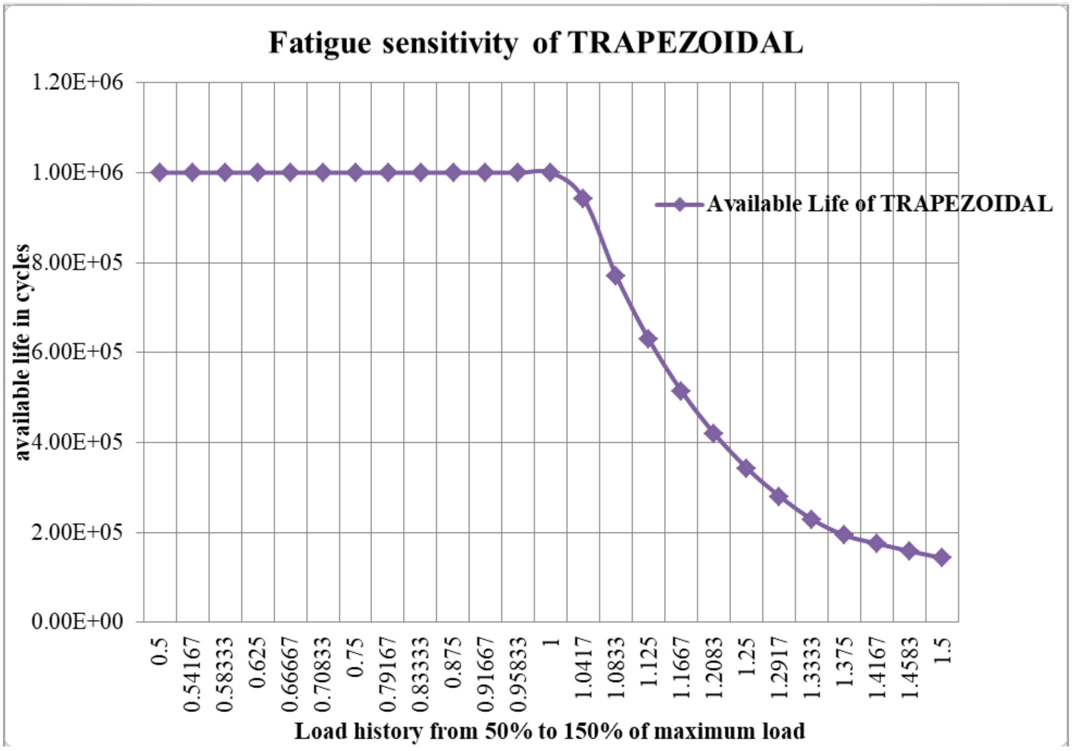
(a). Fatigue sensitivity of model-1 crane hook



(b). Fatigue sensitivity of model-2 crane hook



(c). Fatigue sensitivity of model-3 crane hook



(d). Fatigue sensitivity of trapezoidal crane hook

Figure 4-8: fatigue sensitivity chart of hooks for estimating available life in cycle depending on the load variation

From figure 4-9 (a, c, d) The fatigue sensitivity of the hooks (models) except model-2 at 1 or 100% of design load (44145N) are survived for 10^6 cycles (finite life). Which means if the applied load on the hooks is less than or equal of the design load the hooks are not fail before 10^6 cycles. But from figure4-9 (b) Fatigue sensitivity of model-2 at 1 (100% of design load) the available life is only $9.25x10^5$ cycles. This hook can fail before the design life.

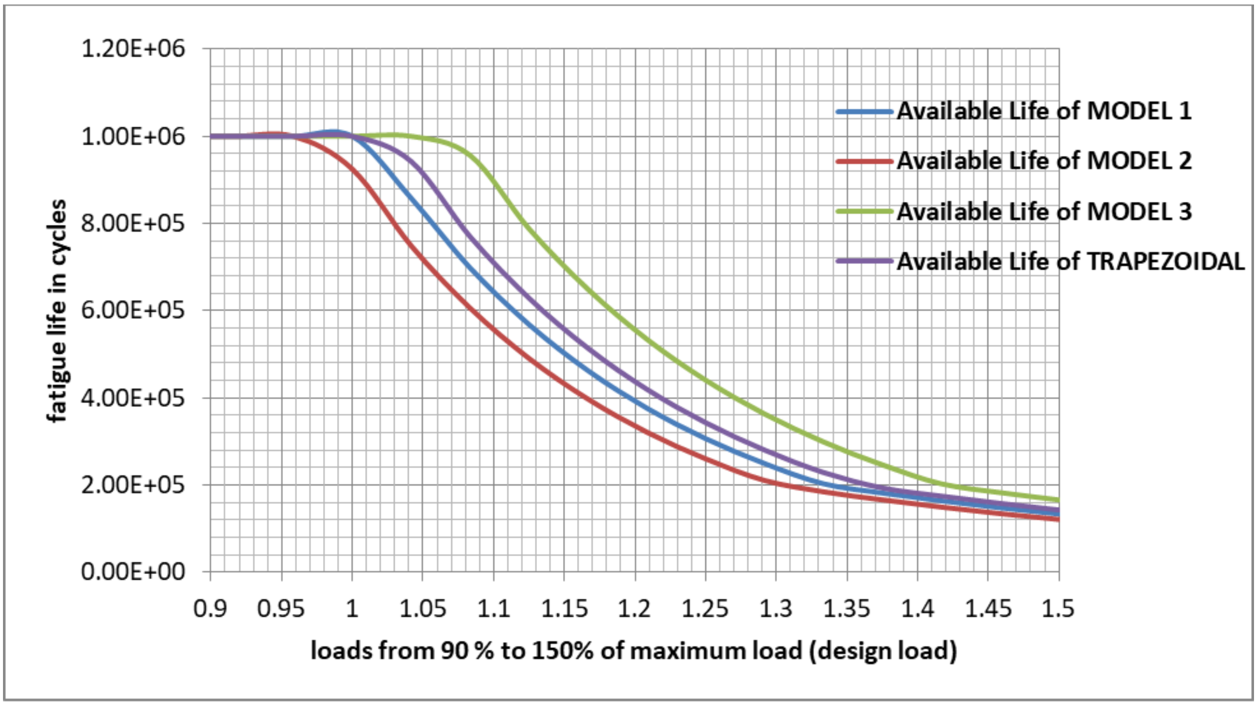


Figure 4-9: the difference of fatigue sensitivity for all modelled hooks

The above figure 4-10 is shows the comparison of the fatigue sensitivity of model-1, model-2, model-3 and trapezoidal crane hooks. The comparison of fatigue sensitivity of the crane hooks is done by applied load from 90% up to 150% of the maximum load (design load).

The trapezoidal and model-1 crane hooks are only surviving 10^6 cycles at 100% and less than of the maximum load (design load). When increasing the design load (4.5 tons) the fatigue life going to decreasing. And at 150% maximum load (1.5 times 4.5 tons) the fatigue life of trapezoidal (standard) and model-1 crane hooks are 1.41×10^5 and 1.32×10^5 respectively.

The infinite cycle (greater than 10^6 cycles) of model-2 crane hook is at less than 95.8% of the maximum load (0.958 times 4.5 tons). This means it can fail before the design life of the crane hook. because the design load is 4.5 tons and at this load the fatigue life of this crane hook is 9.18×10^5 cycles which means cannot survive for infinite life with applied load of 4.5 tons (design load). At 150% of maximum load (1.5 times 4.5 tons) model-2 crane hook can survive only 1.21 x 10^5 cycles.

Model-3 crane hook can survive the infinite life (above 10^6 cycles) up to 106% of maximum load (design load). This means the fatigue life of this crane hook is 10^6 and above cycles until the applied load greater than 1.06 times 4.5 tons (4.77 tons). At 150% of the maximum load (1.5 times 4.5 tons) the fatigue life of this crane hook is 1.62 x 10^5 cycles. The design of all models of crane hooks is considered to survive infinite life up to the applied load is 4.5 tons and less than, but model-3 crane hook is surviving infinite life up to applied load is reaches 4.77 tons.

The fatigue life model-3 crane hook is better than the other models and trapezoidal (standard) crane hooks at all conditions. Therefore, depend on the fatigue sensitivity model 3 crane hook is better than the other crane hook.

CHAPTER 5

5. CONCLUSION, RECOMMENDATION AND FUTURE WORKS

5.1. Conclusion

To conclude the results of each modified modelling crane hooks with the standard crane hook, the maximum Von-Misses stress and total deformation of model -1 and model -2 are increased. The fatigue life of both model-1 and model-2 crane hook is less than the standard crane hook. The maximum Von-Misses stress of model -3 is reduced and total deformation is increased. The fatigue life of model-3 crane hook is greater than the standard crane hook. Then model-3 crane hook has been selected as a best crane hook model.

From model-3 crane hook analysis get the following result which is achieved the objective of the thesis.

- ❖ The fatigue life of Model-3 crane hook is better than standard crane hook, and it is increased by 22.78% from the standard crane hook
- ❖ The maximum stress of model-3 crane hook is reduced by 3.461 MPa (4.133%) from the standard crane hook.
- ❖ The total deformation induced on model-3 crane hook is greater than the standard crane hook by 0.035 mm. But the increased value is very small with comparing the overall dimension of the crane hook. And it has no effect on the life of the crane hook, because still life of model-3 is better than standard crane hook at any applied load.
- ❖ The weight of model-3 crane hook is reduced by 0.354 Kg (2.487%) from the standard crane hook. This means the mass and volume of the crane hook is optimized.

Therefore model-3 crane hook is considered as optimized result.

5.2. Recommendation and Future works

In this thesis work stress and weight optimization is studied for different cross sectional crane hooks using finite element method (FEM). Now the Finite element method is widely used for mechanical analysis. Using FEM for stress analysis, weight optimization and fatigue life estimation have the many advantages. Some of these advantages are requiring a short period of time for analysis and less cost as compared to the experimental method.

The literature review allows Related to this title are very few articles have been published so need more extensive investigation on optimization of crane hook using geometrical modification. However, this paper can be extended to weight and stress optimization by varying radius of curvature of the crane hook. And also can be study about thermal analysis of crane hook.

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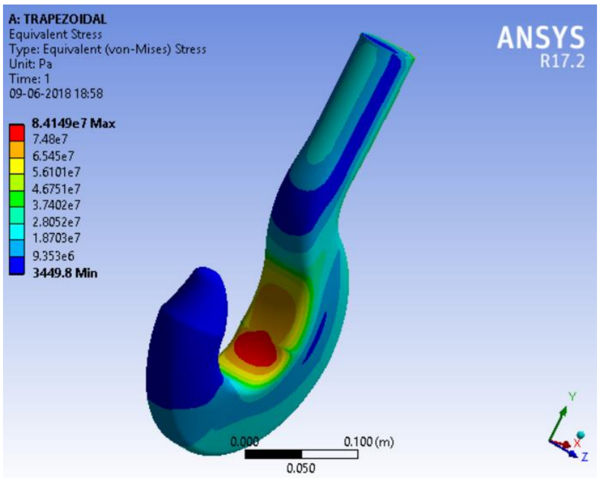
Appendix

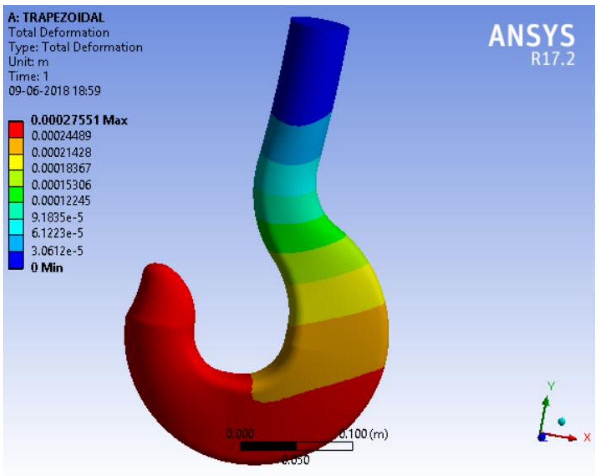
The analysis of maximum Von-Misses stress, total deformation, equivalent elastic strain and the fatigue analysis are done by using ANSYS WORKBENCH 17 for all different cross-sectional models of crane hooks.

Appendix A:

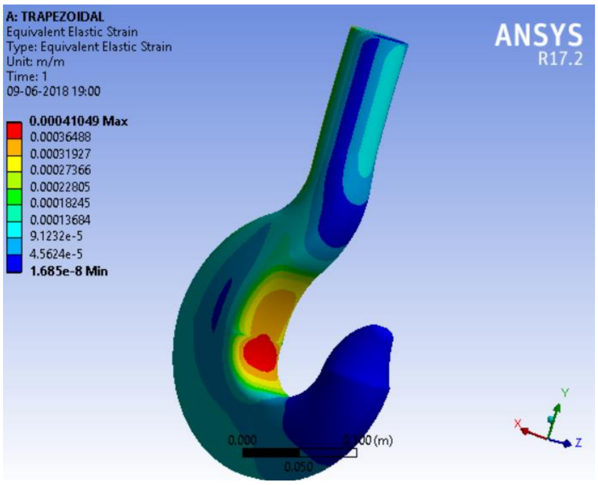
Trapezoidal Crane Hook



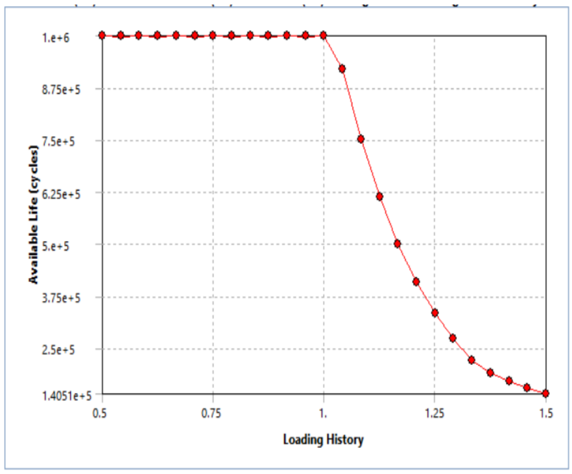
a. Von-Misses stress



b. Total deformation



c. Equivalent elastic strain



d. fatigue sensitivity

Figure A1: Results of stress, deformation, elastic strain and fatigue sensitivity of trapezoidal crane hook

Solution

Table A1: Model (A4) > Static Structural (A5) > Solution (A6) > Results

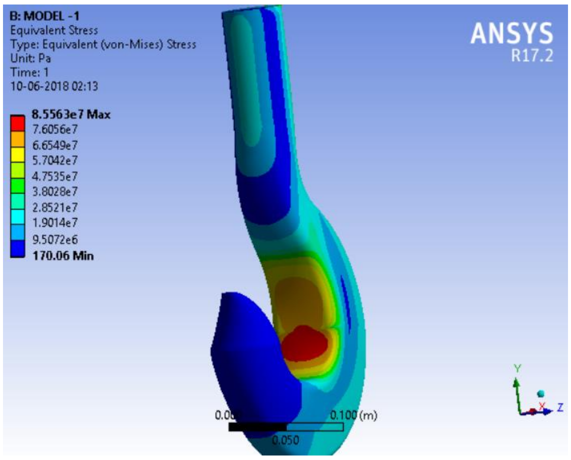
Object Name	*Equivalent Stress*	*Total Deformation*	*Equivalent Elastic Strain*
State	Solved		
Scope			
Scoping Method	Geometry Selection		
Geometry	All Bodies		
Definition			
Type	Equivalent (von-Mises) Stress	Total Deformation	Equivalent Elastic Strain
Integration Point Results			
Display Option	Averaged		Averaged
Results			
Minimum	3449.8 Pa	0. m	1.685e-008 m/m
Maximum	8.4149e+007 Pa	2.7551e-004 m	4.1049e-004 m/m

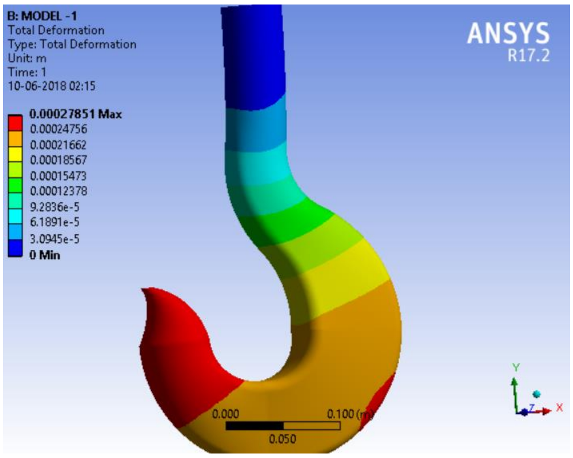
Table A2: Model (A4) > Static Structural (A5) > Solution (A6) > Fatigue Tool > Results

Object Name	*Life*	*Damage*	*Safety Factor*
State	Solved		
Scope			
Scoping Method	Geometry Selection		
Geometry	All Bodies		
Definition			
Type	Life	Damage	Safety Factor
Results			
Minimum	1.e+006 cycles		1.0244
Maximum		1000.	

Appendix B:

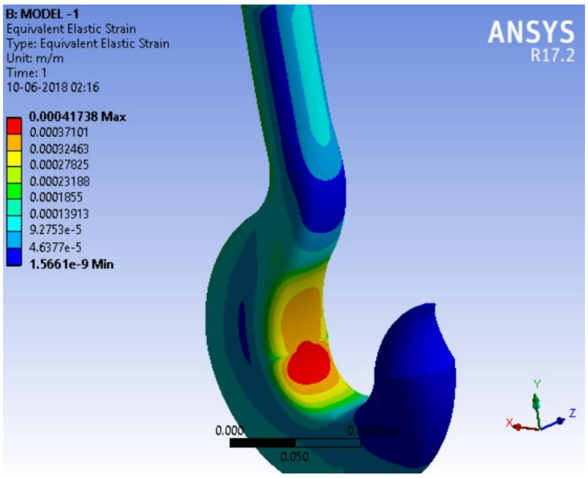
Model-1 Crane Hook

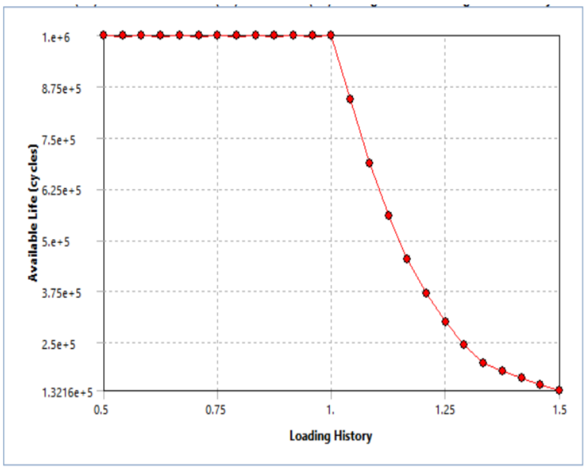




a. Von-Misses stress

b. Total deformation





c. Equivalent elastic strain

d. fatigue sensitivity

Figure B1: Results of stress, deformation, elastic strain and fatigue sensitivity of model-1 crane hook

Solution

Table B1: Model (B4) > Static Structural (B5) > Solution (B6) > Results

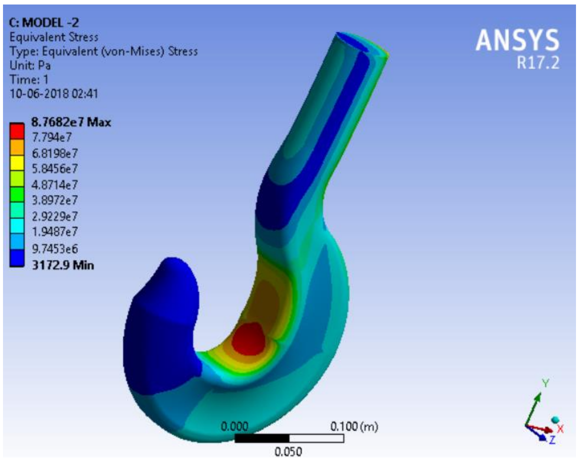
Object Name	*Equivalent Stress*	*Total Deformation*	*Equivalent Elastic Strain*
State	Solved		
Scope			
Scoping Method	Geometry Selection		
Geometry	All Bodies		
Definition			
Type	Equivalent (von-Mises) Stress	Total Deformation	Equivalent Elastic Strain
Integration Point Results			
Display Option	Averaged		Averaged
Results			
Minimum	170.06 Pa	0. m	1.5661e-009 m/m
Maximum	8.5563e+007 Pa	2.7851e-004 m	4.1738e-004 m/m

Table B2: Model (B4) > Static Structural (B5) > Solution (B6) > Fatigue Tool > Results

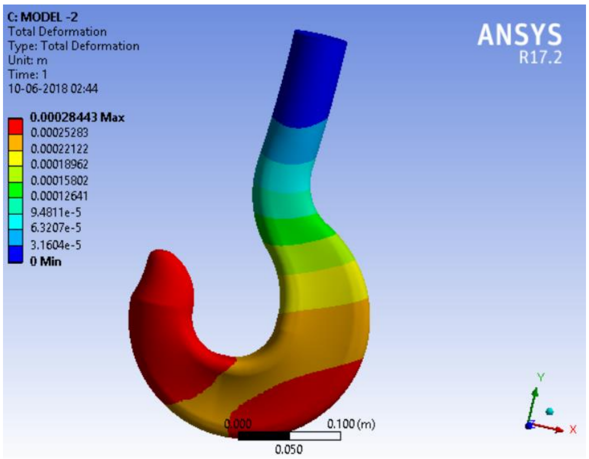
Object Name	*Life*	*Damage*	*Safety Factor*
State	Solved		
Scope			
Scoping Method	Geometry Selection		
Geometry	All Bodies		
Definition			
Type	Life	Damage	Safety Factor
Results			
Minimum	1.e+006 cycles		1.0074
Maximum		1000.	

Appendix C:

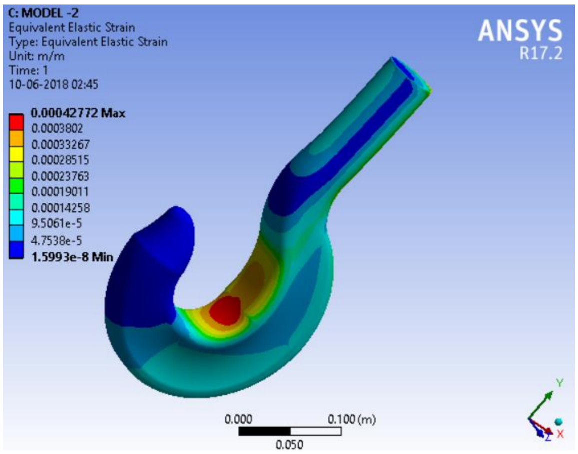
Model-2 Crane Hook



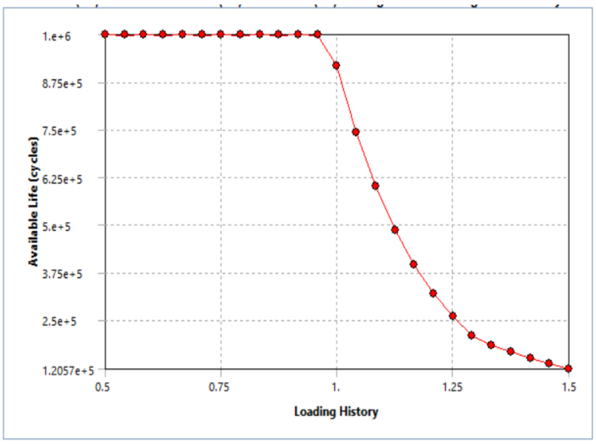
a. Von-Misses stress



b. Total deformation



c. Equivalent elastic strain



d. fatigue sensitivity

Figure C1: Results of stress, deformation, elastic strain and fatigue sensitivity of model-2 crane hook

Solution

Table C1: Model (C4) > Static Structural (C5) > Solution (C6) > Results

Object Name	*Equivalent Stress*	*Total Deformation*	*Equivalent Elastic Strain*
State	Solved		
Scope			
Scoping Method	Geometry Selection		
Geometry	All Bodies		
Definition			
Type	Equivalent (von-Mises) Stress	Total Deformation	Equivalent Elastic Strain
Integration Point Results			
Display Option	Averaged		Averaged
Results			
Minimum	3172.9 Pa	0. m	1.5993e-008 m/m
Maximum	8.7682e+007 Pa	2.8443e-004 m	4.2772e-004 m/m

Table C2: Model (C4) > Static Structural (C5) > Solution (C6) > Fatigue Tool > Results

Object Name	*Life*	*Damage*	*Safety Factor*
State	Solved		
Scope			
Scoping Method	Geometry Selection		
Geometry	All Bodies		
Definition			
Type	Life	Damage	Safety Factor
Results			
Minimum	9.1778e+005 cycles		0.9831
Maximum		1089.6	

Mass properties of mode-2 crane hook

To analytical stress analysis used solidwork soft ware to identify the mass properties of the high stress concentration area of cross section. Because its irregular shape and difficult to calculate.

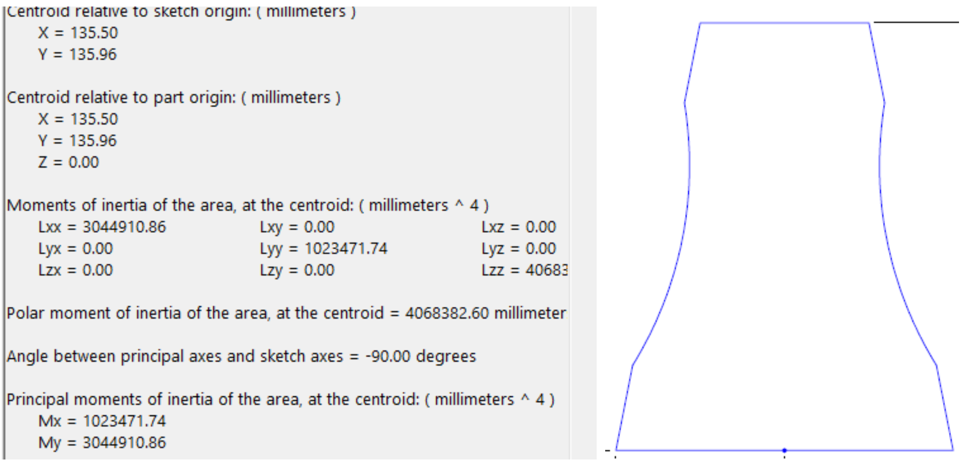
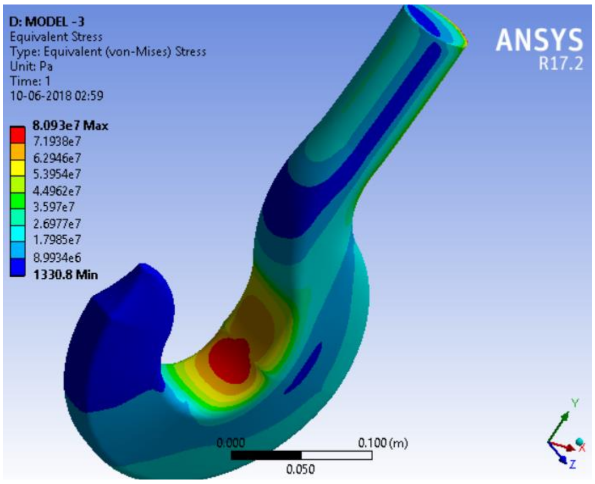


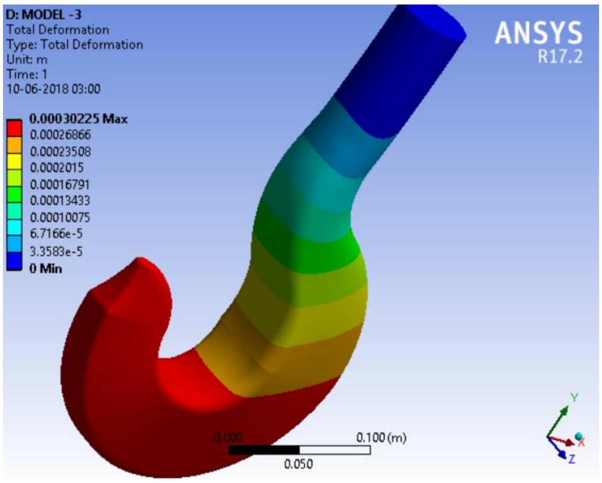
Figure C2: mass properties for cross section of model-2 crane hook

Appendix D:

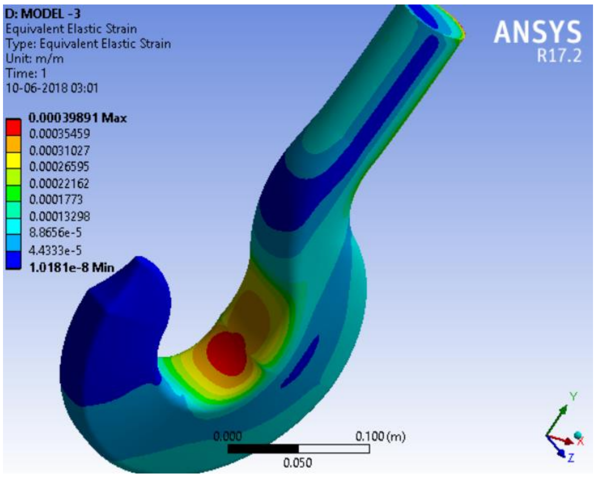
Mode-3 Crane Hook



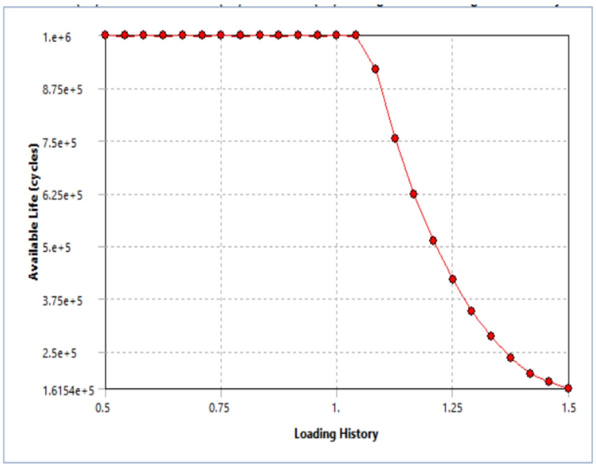
a. Von-Misses stress



b. Total deformation



c. Equivalent elastic strain



d. fatigue sensitivity

Figure D1: Results of stress, deformation, elastic strain and fatigue sensitivity of model-3 crane hook

Solution

Table D1: Model (D4) > Static Structural (D5) > Solution (D6) > Results

Object Name	*Equivalent Stress*	*Total Deformation*	*Equivalent Elastic Strain*
State	Solved		
Scope			
Scoping Method	Geometry Selection		
Geometry	All Bodies		
Definition			
Type	Equivalent (von-Mises) Stress	Total Deformation	Equivalent Elastic Strain
Integration Point Results			
Display Option	Averaged		Averaged
Results			
Minimum	1330.8 Pa	0. m	1.0181e-008 m/m
Maximum	8.093e+007 Pa	3.0225e-004 m	3.9891e-004 m/m

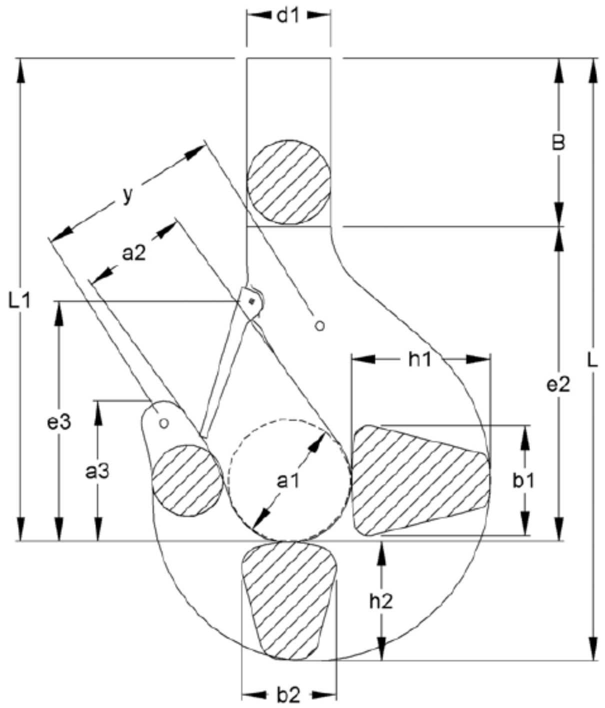
Table D2: Model (D4) > Static Structural (D5) > Solution (D6) > Fatigue Tool > Results

Object Name	*Life*	*Damage*	*Safety Factor*
State	Solved		
Scope			
Scoping Method	Geometry Selection		
Geometry	All Bodies		
Definition			
Type	Life	Damage	Safety Factor
Design Life		1.e+009 cycles	
Results			
Minimum	1.e+006 cycles		1.0651
Maximum		1000.	

Appendix E:

Standard Dimensions of trapezoidal crane hook

These dimensions are taken from Columbus McKinnon Corporation (CM) Shank Crane Hooks and Rigging Products. From this source only taking the size and shape proportionality of the crane hook. For this study used the fourth one from figure E1: indicated by arrow.



a1	a2	a3	B	b1	b2	d1	e2	e3	h1	h2	L	L1	y
56	45	64	78	45	38	36	146	118	56	48	272	224	
63	50	72	86	53	45	42	167	132	67	58	311	253	
71	56	80	95	63	53	48	190	148	80	67	352	285	
80	63	90	103	71	60	53	215	165	90	75	393	318	