ADDIS ABABA UNIVERSITY
ADDIS ABABA INSTITUTE OF THECHNOLOGY
SCHOOL OF MECHANICAL AND INDUSTRIAL ENGINEERING

Static Structural and Life Prediction Analysis of Feeder Table Head Shaft for Wonji-Shoa Sugar Factory Using ANSYS

A Thesis Submitted to the School of Mechanical and Industrial Engineering in Partial Fulfillment of the Degree of Masters of Science in Mechanical Engineering (Mechanical Design)

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Declaration

This is to certify that, the thesis prepared by Tsegay Hagos, the title is optimization and fatigue analysis of feeder table head shaft using ANSYS for an application of wonji-shoa sugar factory. This thesis is original work and it has not been submitted for a degree in any university or institution.

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Abstract

This paper deals with the problem occurred in the feeder table head shaft in wonji-shoa sugar factory. The total length of the feeder table head shaft is 11.877 meter. And the total capacity of the cane to feed is 6250 tone cane per day. The feeder table head shaft is subjected to repetitive load condition (bending and twisting). From the result of static structural analysis, the existing and hollow feeder table shaft is failed due to fatigue. To avoid the failure of the feeder table head shaft, the stress induced on the shaft is studied and reduce the maximum stresses occurs on the shaft. By modifying a new feeder table head shaft the stress is reduced comparing with the existing and hollow feeder table head shaft. The composition of the material is known by using the Spectro test in the Akaki basic metals industry. The models of the feeder table head shafts are prepared by CATIA software and import to ANSYS workbench. The finite element analysis has been performed on the feeder table head shaft in order to model and analyses static structural and life prediction analysis of the feeder table head shaft models. The result of the modified feeder table head shaft is compared with the existing and hollow feeder table head shaft based on different criteria's which are maximum shear stress, maximum von-misses stress, fatigue life, fatigue factor of safety and deflection of the shafts. Finally, based on the comparison of the models result the modified feeder table head shaft is selected. The modified feeder table head shaft has better results of fatigue factor of safety, von-misses stress, maximum shear stress, deflection and fatigue life cycle with the same applied loads to the existing and hollow feeder table head shafts by increasing the diameter and changing its material.

Keywords: FEM, feeder table head shaft, shaft Failure Analysis, fatigue analyses, static structural analysis of shaft.

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Abbreviation

1. CATIA — Computer Aided Three Dimension Interactive Application
2. ANSYS — Analysis System
3. FEM — Finite Element Method
4. 3D — Three Dimension
5. TCD — Tone cane per day
6. IS — Indian standard
7. ASME — American Society of Mechanical Engineers
8. C — Carbon
9. Si — Silicon
10. Mn — Manganese
11. P — Phosphorous
12. S — Sulfur
13. Cr — Chromium
14. Mo — Molybdenum
15. Ni — Nickel
16. Al — Aluminum
17. Co — Cobalt
18. Cu — Copper
19. Nb — Neodymium
20. Ti — Titanium
21. V — Vanadium
22. W — Tungsten
23. Pb — Lead
24. Sn — Tin
25. As — Arsenic
26. Zr — Zirconium
27. Ca — Calcium
28. Ce — Cerium
29. B — Boron
30. Zn — Zinc
31. La — Lanthanum
32. Fe — Iron

CHAPTER 1

1. Introduction

1.1. Background

A shaft is a rotating device that usually circular cross-section solid or hollow, which is used to transmit power and rotational motion in machinery and mechanical equipment in different applications. It is supported by bearings and most of shafts are subjected to fluctuating loads of combined torsion and bending with different degrees of stress concentration. Generally, shafts are not of uniform diameter but are stepped, keyways, sharp corners etc. The stress on the shaft at a particular point varies with rotation of shaft there by introducing fatigue. for such shafts the problem is fundamentally fatigue loading. Failures of such components and structures have engaged scientists and engineers extensively in an attempt to find their main causes and thereby offer methods to prevent such failures. Rotating parts are very widely used as a part of mechanical system to perform large number of functions. Shaft is a rotating machine element used to transfer power from one point to another, shaft can be solid or hollow. The shafts are used in different machines, automobiles, aircrafts etc. [1, 2,3].

The total length of the feeder table head shaft is 11.877 meter. Chain and sprocket are mounted on it, and the total capacity of the cane to feed is 6250 TCD.

Figure 1.1: Existing feeder table head shaft in wonji-shoa sugar factory

The damaged existing feeder table head shaft as shown in figure 1.3: below, considering that the failure occurred at the intermediate shaft during operation and that the shaft broken down.

Figure 1.2: Broken existing feeder table head shaft in wonji-shoa sugar factory

There are two numbers of feeder tables which are cane feeding equipment which helps in feeding the cane into the cane carrier as and when required and also fills up gaps left over by other cane handling and feeding devices. The cane is unloaded over the table first by the cane unloader and then feed into the cane carrier [4].

The cane carrier is the moving apron which conveys the cane into the factory and which assures the feed to the mills by transporting the cane. Effective feeding requires an elevated hopper, and the cane must be raised to this high level from the level of the plot. The carrier always includes a sloping portion (Fig.1.1).

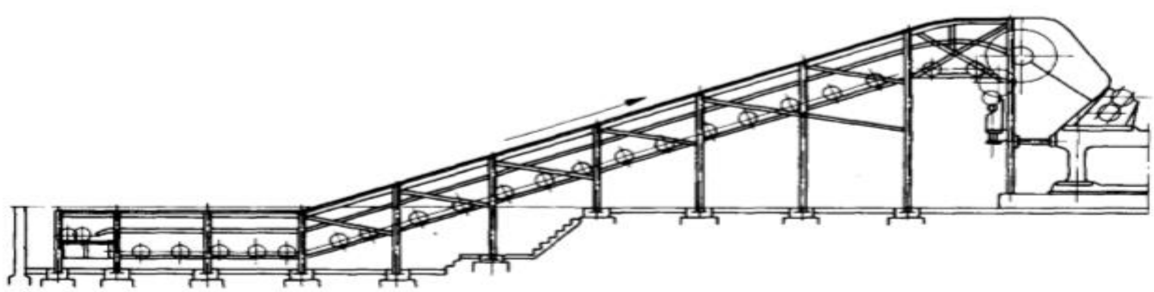

Figure 1.3: Cane carrier

The slope of the inclined portion varies from 15° to a maximum 22°. The inclination cane carrier in wonji-shoa sugar factory is 16.5^0 [5, 6].

This study has conducted on the feeder table head shaft in wonji-shoa sugar factory. The modeling of the feeder table head shaft done by using CATIA software and it imports to ANSYS workbench. The finite element analysis has been performed on the feeder table head shaft in order to modeling and analysis of the static structural and predict fatigue life analysis of the feeder table head shaft.

1.2. Statement of the problem

Wonji-shoa sugar factory have so many assembled components to perform their functions. Most of the parts are big and highly mechanical. Feeder table head shaft is highly responsible and important component being used for the industry works. Existing feeder table head shaft in wonji-shoa sugar factory had frequent failure history, according to the data found from the company. The feeder table head shaft is subjected to continuous loads (bending and twisting). Due to the failure of the parts a major problem brought a big loss in the industry. The company tried to solve the problem of the existing feeder table head shaft. They solved the problem by changing the material and by avoiding the intermediate solid shaft. But after the solution the shaft was failed and also, the material used for the shaft was unknown. This paper tries to modify a new feeder table head shaft to have a better life time.

1.3. Objective

1.3.1. General objective

The general objective of this paper is static structural and life prediction analysis of the feeder table head shaft for an application of wonji-shoa sugar factory using ANSYS.

1.3.2. Specific objective

- ✓ Modeling of the feeder table head shaft with the existing dimension.
- ✓ Conduct appropriate mechanical analysis of the failures feeder table head shaft using ANSYS.
- ✓ Static structural and life prediction of the feeder table head shaft.
- ✓ Modify the feeder table head shaft to have a good life time.
- ✓ Reduce the stress of the existing feeder table head shaft.
- ✓ Get a better fatigue life of the feeder table head shaft.

1.4. Significance of the Study

The importance of this thesis study is to solve the problem of the feeder table head shaft by modifying a new feeder table head shaft to operate safely and to get a better fatigue life. The modification of the feeder table head shaft uses for the factory to save their cost of repairing for the damage, reduce wastage time due to breakage and to avoid failure of the feeder table head shaft. The other importance of this study is to reduce the maximum von-misses stress, maximum shear stress, deflection and to have high fatigue life cycles (infinite life) of the feeder table head shaft to operate safely without failure.

1.5. Scope and Limitation of the study

This paper is focused on static structural and life prediction analysis and modification of the feeder table head shaft in wonji-shoa sugar factory by changing its diameter of the intermediate solid shaft and by changing its material. And also fatigue life analysis of the feeder table head shaft using ANSYS is concerned. Then, the new modified feeder table head shaft is compared with the existing and hollow shaft based on their results. The vibration and welding analysis are not considered in this study.

1.6. Organization of the Study

This document contains five chapters. Chapter 1: is an introduction part that includes background, statement of the problem, objective of the research, organization, methodology, scope & limitation of the study, the significance of the study and motivation on fatigue analysis of the feeder table head shaft are introduced. Chapter 2: includes the relative literatures about different shafts based on their different methods, how they analyzed, get the problems and it includes identifying the gap. Chapter 3: includes modified of a new feeder table head shaft to achieve the specific objectives of the thesis. And also included a static structural analysis of the models shaft by finite element analysis using ANSYS and also Spectro test uses to know the composition of the material in Akaki basic metals. Chapter 4: is about the result and discussion of the study. Compare the results of each model of the shaft includes deformation, maximum shear stress, von-misses stress, fatigue life cycle and fatigue factor of safety of the existing and new modified feeder table head shaft. And finally, chapter 5: is included the conclusions, recommendations and future works on the study.

CHAPTER 2

2. Literature review

Many researches have been conducted on failure analysis of shafts using various methods for different applications by various authors. The journals and papers that relevant studies to the shaft are discussed on this subtitle are categorized under the methods (finite element method, test method) and materials.

2.1. Methods

2.1.1. Finite element method

Ganesh M. et.al [7]. They studied about failure analysis of drive shaft used in transfer coal conveyor. An investigation was performed to determine the failure root cause and factors affecting on transfer coal conveyor shaft. ANSYS analysis was performed to quantify the stress distribution in the shaft. The stresses caused the crack initiation. The mechanical design and material selection of the shaft was not appropriate for its intended service. And they avoided the failure by redesign of new shaft with higher diameter & with two sprocket arrangement on shaft for shearing a maximum load on the shaft. Finally, they concluded that the shaft failed due to stress concentration at the end of sprocket where the diameter of the shaft was small to carry load of coal.

Deepan et.al [8]. They studied about Optimization of shaft design under fatigue loading using Goodman method, Considering the drive system, forces and the torques acting on the shaft determined using which the stresses occurring in the failure section was calculated. Stress analysis was also carried out by using finite element method (FEM) and the results were compared with the calculated values. Stress concentration factors at the endurance limit using Goodman's method, fatigue factor of safety and the theoretical number of cycles sustained by the shaft before the failure was estimated. The failure occurred due to torsional-bending fatigue. Endurance limit, fatigue safety factor was calculated and fatigue life of the shaft was estimated.

Finally, they concluded the increase in fillet radii of the shaft and change in the position of the support decreases the stress concentration factor, increase the endurance limit and the fatigue factor of safety of the shaft.

Mahesh L. et.al [9]. They studied about life estimation of crankshaft using finite element method. Finite element analysis was performed to obtain the variation of stress magnitude at critical locations. As a result, critical location on the crankshaft was obtained. Due to the repeated bending and twisting the crankshaft failed. Accurate prediction of fatigue life was very important to insure safety of components and its reliability and the S-N approach used to predict the fatigue life of crankshaft.

D. K. Padhal et.al [10]. They studied about Analysis and Failure Improvement of Shaft of Gear Motor by considering the drive system forces and torque acting on the output shaft were determined using which the stresses occurring at the failure section were calculated. Stresses analysis was also carried out with analytical method and comparing these results with the software results that was done using the ANSYS software after getting all the different parameter redesign of shaft was done and again analyzing the shaft using ANSYS.

Finally, they concluded, the stresses find out by analytical method and by using software. There was some parameter to improve the shaft life; By changing the material of good shear strength, the shaft may sustain in the maximum loading condition. Re-design the shaft with modified diameter. Select the Proper no. of motor having the least diameter of shaft. All the above parameter that improves quality of shaft of gear motor leads to uneconomical for company.

Shinde V. V [11]. in his paper weight reduction and analysis of sugar mill roller has investigated by using FEA techniques. From the result he concluded that two sugar roller mill with hollow shaft is preferable and it offers much greater safety factors against failures.

Surekha S. et.al [12]. They studied about the Static analysis of a crankshaft of a single cylinder 4-stroke petrol Engine. The modeling of the crankshaft was created using CATIA-V5 Software. Finite element analysis (FEA) was performed to achieve the variation of stress at critical areas of the crank shaft using the ANSYS software and apply the boundary conditions. The results were drawn Von-misses stress and shear stress induced in the crankshaft.

And finally, they concluded;

- ✓ The maximum deformation occurs at the center of the crankpin neck surface.
- ✓ The maximum stresses appear at the fillet areas between the crankshaft journal and crank cheeks and near the central point journal.
- ✓ The value of von-misses stresses that comes out from the analysis was less than the material yield stress so their design was safe.

Gurkan Irsel, [13]. Studied about an axle shaft subjected to static deformation and fatigue damage was considered by using computer-aided engineering. The shaft system was modeled with CATIA and subjected to fatigue analyses with Ansys workbench. The damaged fracture surfaces of the shaft have been examined. Thus, the fatigue and damage were revealed in greater detail. The axle shaft was exposed to bending. The fatigue resistance of shafts decreases with bending deformation in the axle shaft. and the maximum von-Mises stress value, total deformation; fatigue life and safety factor values which were performed with ANSYS for the damaged shaft. When the shaft is exposed to deformation damage, the increase in shaft rotation increases the fatigue life instead of reducing.

Sihong Yan [14]. Studied about fatigue life prediction and structural optimization on torsional shaft. the parametric modelling of the torsion shaft was carried out with Solid works, and the model was transferred to the ANSYS Workbench through the data interface. A parameterized model of torsion shaft was developed based on Solid works and imported into ANSYS Workbench by data interface. At the same time, based on the ANSYS Workbench, the transient stress analysis and fatigue analysis of the parameterized model are carried out.

On this basis, the influence of different structures on the fatigue life of the torsion shaft was analyzed, and the structure of the torsion shaft was improved. Based on the fatigue life prediction method of torsion shaft, the influence of different inner diameter values on fatigue life and structure quality of hollow torsion shafts was studied. And based on the influence rule, the inner diameter of the torsion shaft under the condition of the lightest weight and enough fatigue life was calculated.

In Conclusion of their study the fatigue life analysis method of torsion shaft based on ANSYS Workbench was presented. The modelling and simulation of the torsion shaft were completed, and the results were compared with the actual situation, indicating the applicability of the method.

Pramod J. et.al [15]. They studied about shaft analyzed by using finite element analysis technique for stresses and deflections. The total work carried out in stages first stage was static analysis. In this stage pump shaft was analyzed for stresses and deflection and same results were verified. Result values obtained for deflection and stresses compared in both cases. Finally, they concluded whatever design made for is safe and shaft will not fail for current working condition.

Samta Jain et.al [16]. Reviewed of modeling, static structural analysis and modal analysis of engine camshaft. Camshafts were also subjected to varying contact fatigue loads due to the contact of the plunger on the cam. Camshafts are rotating components with critical load. These exact values were needed to be determined to avoid failure in camshaft. The studies were generally undertaken to predict the structural behavior of camshaft using three-dimensional finite element stresses. ANSYS stress values were conventionally then compared with the theoretical stress values. In the work presented the FEM (Finite Element Method) was used to determine the stress distribution of the Cam Shaft under the usual loading. Also, the critical frequencies of the cam shaft under the loading were identified and the failure of the Cam Shaft was avoided.

Ravi Kumar Goel et.al [17]. In their study the modeled of crankshaft was designed in the ANSYS Workbench and then Finite Element Analysis was carried out under the same loading conditions. static analysis was done to find the maximum deformation and maximum stress point. Apart from this strain, shear stress and total deformation contours were also plotted. The results obtained were compared with the existing results and the crankshaft with optimum design was selected.

The following conclusions can be drawn from the analysis conducted in the study: On the basis of the current studies performed, it can be concluded that the design parameter of connecting rod can be modified in such a way so that sufficient improvement in the existing results can be obtained. During the design optimization, weight of the crankshaft was also reduced. It was found that the maximum stress point region was at the projection of the center main journal shaft and crank arm.

P.Parveen Sulthana1 et.al [18]. In their paper they discussed about the drive shafts in commercial heavy-duty vehicle that subjected to cyclic loads due to variation in the torque demanded by the varying road loads. In their project an attempt had been made on prediction of fatigue life of the drive shaft using FEA technique. Structural analysis was carried out on the FE model of drive shaft, and potential areas of high stress concentration were obtained. Von Mises stress, fatigue damage, fatigue life, factor of safety and total deformation were the results of fatigue analysis. It was observed that from the static analysis that the roots of the splines were the areas of high stress concentration. The fatigue analysis revels that the drive shaft fails in the region of high stress concentration as expected. The fatigue life of the component was found to be infinite for design pay load.

Ashwani Kumar Singh et.al [19]. Studied about statics analysis conducted on a nickel chrome steel and structural steel crank shafts from a single cylinder four stroke engine. Finite elements analysis was performed to obtain the variation of stress magnitude at critical locations. The load was then applied to the FE model and boundary condition where applied as per the mounting conditions of the engine in the ANSYS Workbench.

Finally, they concluded that, that FEA was a good tool to reduce time consuming Ansys workbench. The maximum deformation appears at the center of crankpin neck surface. The maximum stress appears at the fillets between the crankshaft journal and crank cheeks and near the central point journal. The edge of main journal was high stresses area. The value of the von-Misses stresses that comes outs from the analysis for less than material yield stress so the design was safe and it optimized to reduce the material and cost.

1.1.2. Test method

Gys Van Zyl et.al [20]. Studied about failure analysis conveyor pully shaft and the shaft conveyor belt drive failed in service and investigation was performed in order to determine the failure root cause and conditional factors. The investigation methods included visual examination, optical and scanning electroscope analysis, chemical analysis of the material and mechanical tests. a finite element was also performed to quantify the stress distribution in the shaft.

It was concluded that the shaft failed due to fatigue and that the failure was caused by improper reconditioning of the shaft during repetitive repair.

S. sitthipong, P. et.al [21]. Studied about the failure analysis of metal alloy propeller shafts. The shaft fractured at shoulder with evidence of torsional- bending fatigue. Chemical composition, micro-structural characterization, fractography, hardness measurements, and finite element simulation were used for the analysis. Fatigue crack has initiated at the shoulder. The geometry of the fillet also promoted the initiation crack because the fillet was erroneously designed. And they concluded that all these factors produced fatigue failure.

G.A. Nassef, A. et.al [22]. In their paper the discussed about analysis of a failed rocker arm shaft of a passenger car engine. Microscopic investigations of the failed shaft revealed the presence of microcracks close to the supporting holes. The failure had occurred by the rapid crack propagation because of the lower fracture toughness of the martensite. And finally, the concluded the failure might occurred by consequential failure, improper lubrication and faulty manufacturing (including assembly).

Sh. Zangeneh, M. et.al [23]. In this paper they studied about the experimental and numerical simulation of the AISI 304L stainless steel shaft. The analysis methodology focused on fracture surface examination and finite element method (FEM) simulation using Abaqus software for stress analysis and the experimental procedure was, the failed shaft was inspected macroscopically by means of optical and scanning electron microscopy (SEM) while a great care was taken to avoid damage of fracture surfaces. Atomic absorption spectrometry was employed to determine chemical composition of the alloy to determine the hardness of the alloy, Brinell hardness (HRB) measurements were carried out on several points from the outer surface to the central zone of the polished surface of the shaft. The results showed that mechanical properties and chemical composition of the failed shaft were in acceptable range.

Finally, they conclude that inadequate fillet radius size and more importantly machining grooves on the shaft surface was the main cause of the shaft failure.

R.W.Fuller,J.Q. et.al [24]. Studied about Failure analysis of AISI 304 stainless steel shaft. This failure analysis methodology focused on observation gathering, preliminary visual examination and record keeping, nondestructive testing, mechanical testing, selecting/preservation of fractures surfaces, macroscopic examinations, microscopic examinations, metallography, failure mechanism determination, chemical analysis mechanical failure analysis, testing under simulated service conditions, and final analysis and report.

The application of the methodology was demonstrated in the failure analysis approach, they pinpointed the primary mode of failure and developed a means of circumventing this type of failure in the future. The results show that the steel shaft failed due to intergranular stress cracking (sensitization during welding) at the heat affected zones (weld plugs).

A. Lanzutti, A. et.al [25]. Studied about failure analysis of gears, shafts and keys of centrifugal washers failed during life test. The aim of the test apparatus was to simulate the service conditions of the components in order to evaluate their mechanical resistance and to verify that the life of the appliance was consistent with the project specifications

Premature failures occurred during the life test were considered in their work. In particular, the components interested by failure were the gear’s teeth or the keys used to transfer force from the shaft to the gear. The failure of these components was observed after about 3000 cycles. The microstructure of the gears and shafts, that exhibited premature failure together with that of intact components, were investigated by optical microscopy. The chemical composition of these components was characterized By Glow Discharge Optical Emission Spectroscopy. The mechanical properties of the components were investigated by Vickers microhardness tests. The experimental results obtained in this work indicate that the failure of the components could be related to the quality of the materials of the components. In particular, the life test used in this work proved to be an important tool to evaluate the risk of a possible reduction of the products life due to non-conformity with materials specifications.

B. Engel et.al [26]. They studied the failure modes of the shaft of the rotary draw bending machine were inspected. Furthermore, stress and deflection analysis of the shaft subjected to combined torsion and bending loads were carried out by an analytical method and compared with a finite element analysis method.

The theoretical fatigue strength, correction factors, and fatigue life sustained by the shaft before damaged was estimated by creating a stress-cycle (S-N) diagram. Shaft failure analysis and fatigue life estimation consisted with some steps that, Material analysis and hardness tests were performed to identify the kind and strength of the material that the shaft made of Brinell hardness tests were done on the shaft surface by using Wolpert tester. Material analysis test was carried out by using Spectro technology to identify the chemical composition of the shaft material. An optical examination and surface roughness measurements on the shaft surface were conducted.

Fatigue life estimation by creating a calculated stress-cycle diagram was made. In their conclusion, the shaft needs some surface treatments to increase the reliability and fatigue life.

2.2. Material

2.2.1. Shaft materials

Santosh Y. et.al [27]. studied about static structural analysis of conventional sugar mill roller by using FEA method to carry out the stress analysis. The results for maximum shear stress on the Top, Feed, and Discharge roller are calculated analytically and compared with the results from software. Static structural analysis of all three rollers was done using forged steel materials 45C8 (C - 0.35-0.45 %, Mn - 0.60 to 0.90%), as per the standard IS: 1570-1979 for analyzing the results. Due to results finally, it concluded that the maximum shear stress value of top, feed & discharge rollers was less than yield strength, therefore all three rollers was safe.

Subrata Kr Mandal et.al [28]. In their paper they described about the failure analysis of intermediate shaft. The shaft has been made from a stainless steel, 40C8. The shaft was found to be bending failure. The investigation was carried out in order to establish whether the failure was the cause or a consequence of the accident. The shaft assembly was prone to bending failure after relatively a short operational time; modifications to the detail design were changes to the detail design would substantially improve the resulting safe life of shafts in service.

The design modifications as the current shaft assembly was prone to bending failure after relatively a short operational time, modifications to the detail design was essential to longer life obtained. Material fatigue strength Grade 40C8 IS: 1570-88 stainless steel grade is used for the shaft.

Some of the most effective and simple alterations modification was as follows: Shortening the total length of the shaft, increasing the diameter of the shaft and Increase the fillet radius.

Bhumesh J. et.al [29]. They studied about static structural and fatigue analysis of single cylinder engine crank shaft by identify and solve the problem using modeling and simulation techniques. The structural and fatigue analysis of the crank shaft used by ANSYS software. The finite element analysis has been performed on crank shaft in order to optimize the weight and manufacturing cost. Their objective was to identify the effect of the stress on crank shaft by compare various materials and to provide possible solution. The material for crank shaft was EN 9 and other alternate materials they use for comparison, the materials were SAE 1045,1137,3140 and Nickel cast iron.

Their result obtained the total deformation, fatigue life and the von-misses stress of all materials and they conclude by comparing the materials SAE 1137 was better than the others.

K Prasanth et.al [30]. They designed an automotive manual transmission shaft for its better performance by using four materials to show the comparable their results. The materials were Aluminum 7075-T6, Titanium alloy Ti6Al4V, AISI 18Ni steel annealed and, Magnesium alloy AZ80A-T6. They use finite element analysis software. And also, they discussed that among the four materials the titanium alloy (Ti6Al4V) provides the better torque transmission. Finally, the weight and stress optimized for different materials were compared with the titanium alloy. Hence results revealed that the Ti6Al4V had the better result than other which were considered.

Fernando Casanova [31]. Studied about failure analysis and redesign shaft for sugar cane transport. According the manufactures, AISI 1045 steel was the material of the shaft; however, it was found that the material of some shafts did not have the specifications of AISI 1045 steel. Moreover, even if the shafts were made with good quality AISI 1045 steel, the stresses are so high that the element will fail due to fatigue. Finally, some recommendations were given to increase the reliability of the shafts, and a new design and manufacturing process were proposed.

CHAPTER 3

3. Methods and conditions

3.1. Methodology

The overall static structural and life prediction analysis of the feeder table head shaft is done by using ANSYS software. The first step is Literatures survey from different sources such as manual from wonji-shoa sugar factory, journals, books and websites. The 3D models of the feeder table head shaft are prepared by CATIA and it import to Ansys Workbench 17.2. Then the new modified feeder table head shaft models are compared with the existing and hollow feeder table head shaft based on the ANSYS analysis result.

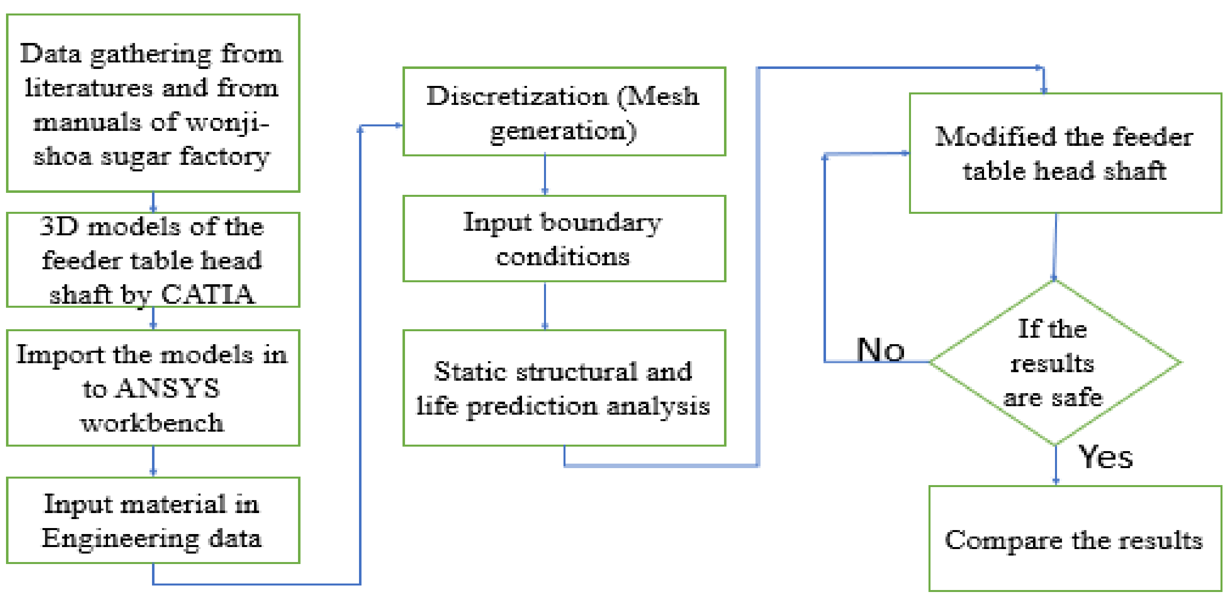

Figure 3.1: Methodology chart

3.2. Spectro test method

3.2.1. AKAKI basic metals Industry Spectro test measurement for the material C45 carbon steel.

Figure 3.2: measuring composition material of the spacemen using Spectro test

Figure 3.2: Indicates the measuring of the composition material of the feeder table head shaft by taking sample from wonji-shoa sugar factory in AKAKI basic metals by using Spectro. It was tested 6 times and it takes the average.

Table 3.1: Spectro test results in AKAKI basic metals industry for the material C45 carbon steel.

Measurement	C	Si	Mn	P	S	Cr	Mo	Ni	Al
1	0.46	0.27	0.42	0.023	0.011	0.097	0.061	0.060	0.020
2	0.45	0.26	0.43	0.020	0.012	0.097	0.058	0.056	0.021
3	0.48	0.24	0.45	0.025	0.012	0.10	0.058	0.045	0.016
4	0.47	0.26	0.48	0.021	0.030	0.11	0.061	0.048	0.021
5	0.5	0.23	0.48	0.023	0.013	0.11	0.055	0.035	0.019
6	0.46	0.22	0.51	0.24	0.014	0.11	0.050	0.030	0.014
Average	0.47	0.25	0.46	0.023	0.015	0.10	0.057	0.046	0.019

Measurement	Co	Cu	Nb	Ti	V	W	Pb	Sn	As
1	0.004	0.12	0.003	0.018	0.002	0.030	0.20	0.023	0.026
2	0.004	0.12	0.003	0.027	0.002	0.030	0.20	0.021	0.026
3	0.004	0.13	0.003	0.025	0.002	0.030	0.20	0.023	0.017
4	0.005	0.13	0.003	0.013	0.002	0.030	0.19	0.025	0.028
5	0.005	0.13	0.003	0.017	0.002	0.030	0.19	0.023	0.019
6	0.005	0.13	0.003	0.027	0.002	0.030	0.17	0.023	0.018
Average	0.005	0.13	0.003	0.022	0.002	0.030	0.19	0.024	0.023

Measurement	Zr	Ca	Ce	B	Zn	La	Fe
1	0.006	0.0005	0.006	0.0007	0.006	0.004	98.1
2	0.006	0.0006	0.007	0.001	0.005	0.004	98.1
3	0.005	0.0010	0.006	0.0005	0.004	0.003	98.1
4	0.006	0.001	0.008	0.001	0.005	0.003	98.1
5	0.005	0.001	0.006	0.0006	0.006	0.004	98.1
6	0.004	0.0005	0.006	0.0006	0.003	0.003	98.1
Average	0.005	0.0009	0.006	0.0008	0.005	0.004	98.1

Table 3.2: The average result of Spectro test results in AKAKI basic metals industry for the material C45 carbon steel.

Measurement	C	Si	Mn	P	S	Cr	Mo	Ni	Al
Average	0.47	0.25	0.46	0.023	0.015	0.10	0.057	0.046	0.019

Measurement	Co	Cu	Nb	Ti	V	W	Pb	Sn	As
Average	0.005	0.13	0.003	0.022	0.002	0.030	0.19	0.024	0.023

Measurement	Zr	Ca	Ce	B	Zn	La	Fe
Average	0.005	0.0009	0.066	0.0008	0.005	0.004	98.1

After the feeder table head shaft with the original material IS 2062 carbon steel failed, the shaft replaced with unknown material. This unknown material is identified by using Spectro test in AKAKI basic metals by its composition. The material is C45 carbon steel.

3.3. Material selection method

There are many types of material use to make a drive shaft to improve from the failure occur. So, the manufacturer chooses high hardness and strength material to make sure the drive shaft can hold on a long time [33].

Shafts are made of various materials according to their applications and requirements. Medium carbon/low alloy steel is one of the common shaft materials. It is used where high strength and toughness are required. It combines deep hardenability with ductility/ toughness, strength and also it has good fatigue resistance [34].

3.3.1. Material selection based on Fatigue S-N Curve

3.3.1.1. Fatigue analysis by analytical approach

S-N curve approach and equivalent alternating stresses are used to obtain the fatigue life of feeder table head shaft. S-N Method Used to Predict Fatigue Life Due to the long-life and elastic loading conditions on the shaft, the stress-life (S-N) approach to life estimation is commonly used. S-N diagram plots stress S versus cycles to failure N. S-N test data are usually displayed on a log-log plot. There are two basic regimes, low-cycle fatigue (generally below 1000 stress cycles) and high-cycle fatigue (more than around 1000 but less than one million stress cycles). The slope of the line is much lower in low-cycle fatigue than in high-cycle fatigue. Low-cycle fatigue is any loading that causes failure below 1000 cycles.

The stress endurance limits of steel for three types of loading can be approximated as;

bending: $S'e = 0.5 * S_{ut}$,

axial: $S'e = 0.45 * S_{ut}$, and

torsion: $S'e = 0.29\ S_{ut}$

The fatigue strength for steel at which high-cycle fatigue begins can be approximated as;

bending: $S'l = 0.9 * S_{ut}$,

axial: $S'l = 0.75 * S_{ut}$, and

torsion: $S'l = 0.72 * S_{ut}$ [35]

Strength needs to be expressed as a function of loading cycles. The fatigue strength at any location between S'l and S'e can generally be expressed as;

$\log S'f = bs \log N't + C$ …….…………………………………..…………………….……… (1)

where bs is the slope and C the intercept of the finite-life portion of the S-N diagram. At the end points ($S = S'l$ for $N= 1000$ and $S=S'e$ for $N = 10^6$)

$\log S'l = bs \log (103) + C=3bs+C$ …………………………………………………….……….. (2)

$\log S'e = bs \log (10^6) + C=6bs+ C$………………………………………………………………. (3)

then by subtracting Eq. (3) from Eq. (2) it gives

$bs = - 1/3 \log (S'l / S'e)$ ……………………………………………………………………….. (4)

by Substituting Eq. (4) into Eq. (3) it gives

$C=2\log (S'l\ S'e) + \log S'e = \log [(S'l)^2 / S'e]$ ……………...…………………………………. (5)

Then the fatigue strength

$S'f = 10^{C} (N't)^{bs}$ for $10^3 \leq N\,t \leq 10^6$ ………………………………………… (6) [35].

Endurance limit and fatigue strength for IS2062 steel is calculated as follows;

$S'e = 0.5 * S_{ut} = 0.5 * 410 = 205$ Mpa for bending load

$S'e = 0.29 * S_{ut} = 0.29 * 410 = 118.9$ Mpa for torsional load.

Endurance limit and fatigue strength for C45 steel is calculated as follows;

$S'e = 0.5 * S_{ut} = 0.5 * 710 = 355$ Mpa for bending load

$S'e = 0.29 * S_{ut} = 0.29 * 710 = 205.9$ Mpa for torsional load.

Endurance limit and fatigue strength for AISI 4340 steel is calculated as follows;

$S'e = 0.5 * S_{ut} = 0.5 * 1110 = 555$ Mpa for bending load

$S'e = 0.29 * S_{ut} = 0.29 * 1110 = 321.9$ Mpa for torsional load.

And similarly, for the material AISI 18Ni 350 steel

$S'e = 0.5 * S_{ut} = 0.5 * 2415 = 1207.5$ Mpa for bending load

$S'e = 0.29 * S_{ut} = 0.29 * 2415 = 700.35$ Mpa for torsional load.

There are some fatigue strength factors to reduce the strength or the life of the steel material. Then by considering the correction factors for the endurance limit it finds the modified endurance limit (Se).

The fatigue strength factors to reduce the strength or life of the steel material are temperature factor, accounts that the strength of a material decreases with increased temperature. The reliability factor, acknowledges that a more reliable (above 50%) estimate of endurance limit, Surface factor is the effect of surface finish, load factor, size factor and other miscellaneous factors. By considering the surface factor (0.915), the size factor (0.616), reliability factor (0.702), temperature factor (1), load factor (0.59) and other miscellaneous factors.

Therefore, the values of the correction factors are 0.233448 then by considering the factors the modified endurance limit calculated as follow;

$Se = S'e * C_{load} * C_{size} * C_{surf} * C_{temp} * C_{reliab}$

For the material IS2062 steel based on bending and torsional loads.

Se = 205 * 0.233448 =47.857 Mpa

Se = 118.9 * 0.233448 = 27.756 Mpa

And for the material 45C8 steel based on bending and torsional loads.

Se = 355 * 0.233448 =82.87 Mpa

Se = 205.9 *0.233448 = 48.06 Mpa

The S-N approach is used to predict the fatigue life of the feeder table head shaft. Now by substituting any value number of cycles (N) in equation (6) it can find the corresponding strength (alternating stress). Table 3.3: shows comparison of alternating stress and corresponding number of cycles for IS2062, C45, AISI 4340 and AISI 18Ni 350 steel material. The values of alternating stress are determined using equation (4) by substituting numbers of cycles from 10-10^7 cycles.

Table 3.3: Comparison of S-N curve of the materials IS2062, C45, AISI 4340 and AISI 18Ni 350 steel based on bending load

Number of cycles (N)	Alternating stress of IS2062 (MPa)	Alternating stress of C45 steel (MPa)	Alternating stress of AISI 4340 steel (MPa)	Alternating stress of AISI 18Ni 350 steel (MPa)
10	1440.15	2490.577	3898.54	8480.12
50	894.81	1547.44	2422.244	5267.66
100	728.987	1260.6658	1973.349	4291.449
500	452.94	783.27	1226.08	2666.36
10^3	369	638.116	998.86	2398.31
10^4	186.76	322.998	505.61	1099.53
10^5	94.547	163.49	255.92	614.486
10^6	47.858	82.756	129.54	281.71
10^7	24.22	41.889	65.57	142.82

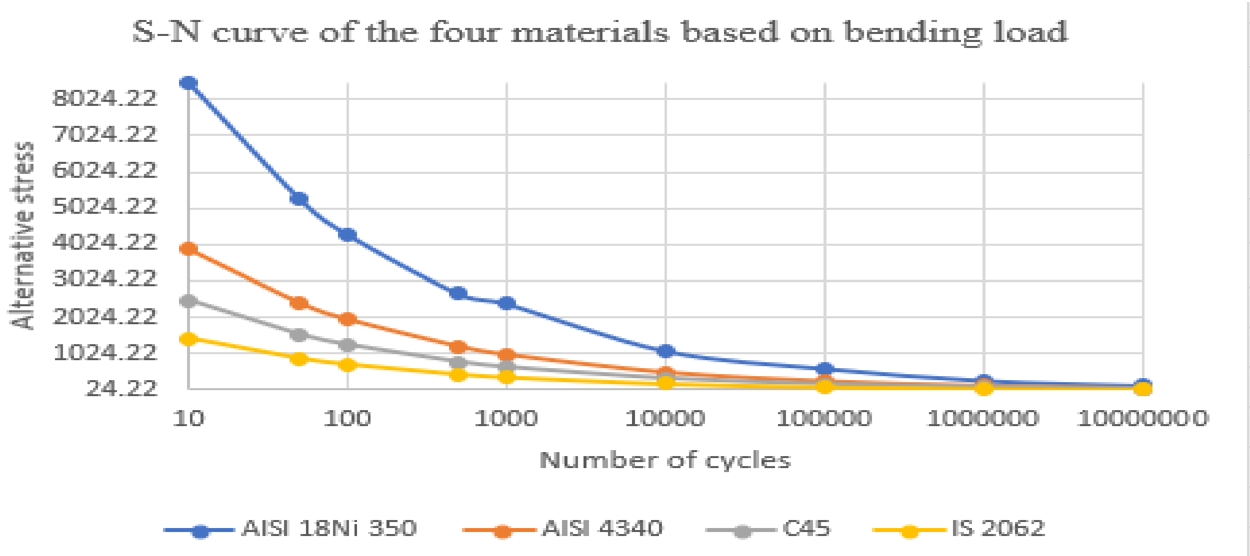

Figure 3.3: Comparison of S-N curve of the materials IS2062, C45, AISI 4340 and AISI 18Ni 350 steel based on bending load

Accurate prediction of fatigue life is very important to insure safety of the shaft and its reliability. Then Table 3.3: and Figure 3.3: compared the materials (IS2062, C45, AISI 4340 and AISI 18Ni 350) depending on the fatigue life (S-N curve) based on bending load. As shown in the figure the results of IS2062 steel shows the fatigue life at 10 cycles the stress is 1440.15 Mpa, the fatigue life cycle of 10^7 cycles the stress is 24.22 Mpa.C45 steel shows the fatigue life at 10 cycles the stress is 2490.577 and the fatigue life at 10^7 cycles the stress is 41.889 Mpa. AISI 4340 steel shows the fatigue life at 10 cycles the stress is 3898.54 Mpa and at 10^7 cycles the stress is 65.57 Mpa And AISI 18Ni 350 steel fatigue life at 10 cycles the stress is 8480.12 Mpa and at 10^7 cycles the stress is 142.82 Mpa based on bending load. Therefore, AISI 18Ni 350steel has better life than the other materials, so it is selected for the feeder table head shaft.

Table 3.4: Comparison of S-N curve of the materials IS2062, C45, AISI 4340 and AISI 18Ni 350 steel based on torsional load

Number of cycles (N)	Alternating stress of IS2062 (MPa)	Alternating stress of C45 steel (MPa)	Alternating stress of AISI 4340 steel (MPa)	Alternating stress of AISI 18Ni 350 steel (MPa)
10	1424.688	2470.58	3865.01	8410.07
50	821.252	1424.33	2228.07	4848.55
100	647.798	1123.569	1757.52	3824.72
500	373.419	647.75	1013.165	2205.01
10^3	294.55	510.975	799.19	1739.40
10^4	133.93	232.38	363.41	791.04
10^5	60.897	105.68	165.255	359.74
10^6	27.689	48.06	75.146	163.60
10^7	12.59	21.825	34.17	74.40

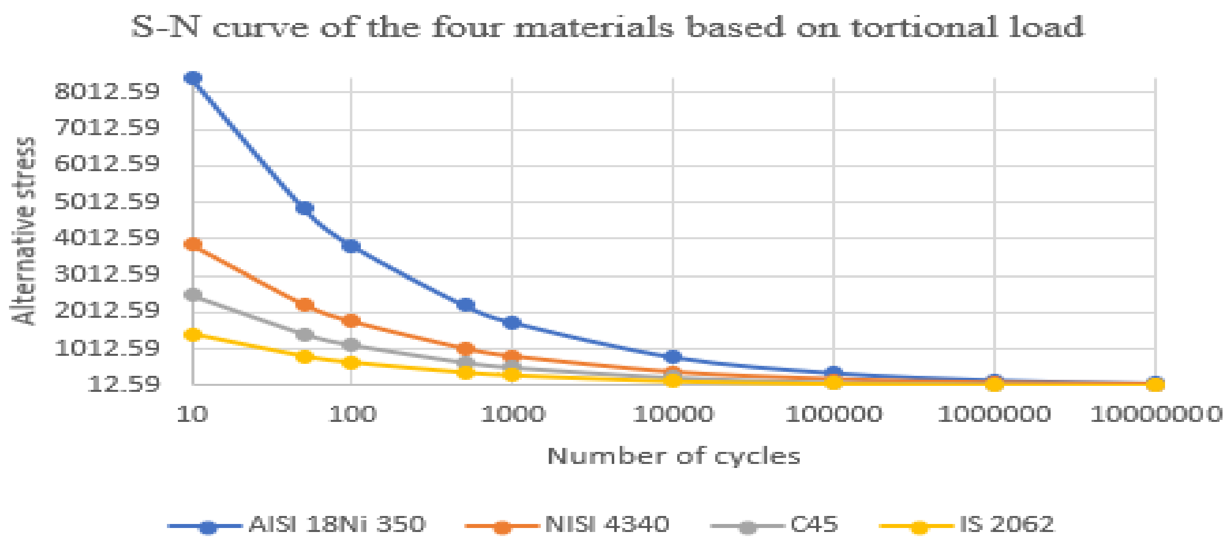

Figure 3.4: Comparison of S-N curve of the materials IS2062, C45, AISI 4340 and AISI 18Ni 350 steel based on torsional load

The materials IS2062, C45, AISI 4340 and AISI 18Ni 350 steel are compared depending on the fatigue life (S-N curve) in Table 3.4: and Figure 3.4. As shown in the figure the results of IS2062 steel shows the fatigue life at 10 cycles the stress is 1424.688Mpa, the fatigue life cycle of 10^7 cycles the stress is 12.59 Mpa. And the C45 steel shows the fatigue life at 10 cycles the stress is 2470.58 and the fatigue life at 10^7 cycles the stress is 21.825 Mpa. AISI 4340 steel shows the fatigue life at 10 cycles the is 3865.01 Mpa and at 10^7 the stress is 34.17 Mpa. And AISI 18Ni 350 steel the fatigue life at 10 cycles the is 8410.07 Mpa and at 10^7 the stress is 774.40 Mpa based on torsional load. Therefore, AISI 18Ni 350 steel has better life than the other materials, so it is selected for the feeder table head shaft based on torsional load.

Table 3.5: Fatigue strength as a function of ultimate strength for the materials of IS2062, C45, AISI 4340 and AISI 18Ni 350 steel (endurance limit Vs tensile strength)

Materials	Endurance limit		Tensile strength
	Based on bending load.	Based on torsional load.	
IS2062	47.856 Mpa	27.75 Mpa	410 Mpa
C45	82.87 Mpa	48.067 Mpa	710 Mpa
AISI 4340	129.56 Mpa	75.146 Mpa	1110 Mpa
AISI 18Ni 350	281.88	163.49	2415 Mpa

From the table 3.5: The endurance limit of AISI 18Ni 350 steel based on bending load 281.88 Mpa is better than the endurance limit of the other materials listed above. Similarly, the endurance limit based on the torsional load of AISI 18Ni 350 steel (163.49) is better than the other materials. So AISI 18Ni 350 steel is the best material for the shaft.

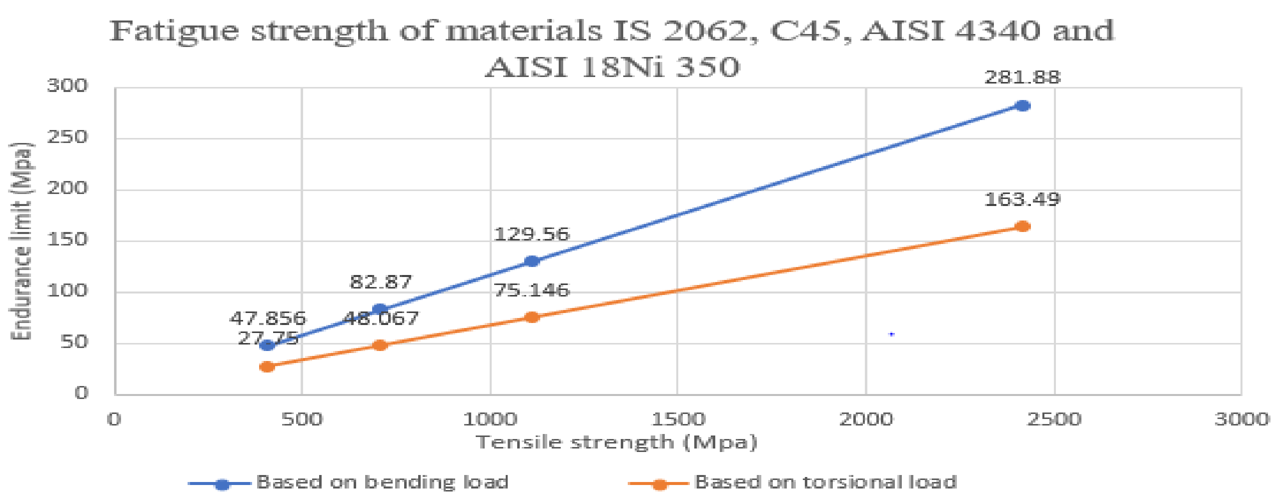

Figure 3.5: Comparison of fatigue strength of the materials of IS2062, C45, AISI 4340 and AISI 18Ni 350 steel based on endurance limit Vs tensile strength.

This figure 3.5: Describes the fatigue strength of the materials, IS2062 steel with 410 Mpa tensile strength, its endurance limit is 47.856 Mpa, for C45 steel with the tensile strength 710 Mpa the endurance limit is 82.87 Mpa, for AISI 4340 steel with the tensile strength 1110 Mpa the endurance limit is 129.56 Mpa. And for AISI 18Ni 350the tensile strength 2415 Mpa the endurance limit is 281.88 Mpa based on bending load. There for the endurance limit of AISI 18Ni 350 steel material is the best for the feeder table head shaft.

3.3.2. Material selection based on their mechanical properties

Different materials are being used in shafts for various applications. The existing feeder table head shaft is made from IS2062 carbon steel. This IS 2062 steel is the previous material they use before the material changed to C45 steel that got from manual. After the feeder table head shaft failed the material IS 2062 steel changed to C45 carbon steel in wonji-shoa sugar factory. There are many different options to choose. Each type of steel has different benefits. The main differentiating factors is the amount of carbon that is mixed with iron during production. Other materials, mainly metals, can be added to change the physical properties. The additive can change tensile strength, ductility, or toughness. Carbon steel can be segregated into three main categories, low carbon steel sometimes known as mild steel, medium carbon steel and high carbon steel. Low carbon steel (mild steel) contains less than 0.30% carbon.

Medium carbon steel has a carbon contain up to 0.60%. This product is stronger than low carbon steel and are quite often hardened and tempered using heat treatment. Medium carbon steel is ductile and strong, with long-wearing properties. And medium carbon steel is typically employed in conjunction with alloys such as chromium, nickel and molybdenum produce high strength, wear resistance and toughness. So, it uses for machine components that require optimal combinations of strength and toughness. And high carbon steel commonly knowns as carbon tool steel. It typically has a carbon content up to 1.5%. once heat treated it becomes extremely hard and brittle [38,39].

The feeder table head shaft modified based on the strength and stiffness design method. Design based on Strength is carried out so that stress at any location of the shaft should not exceed the material yield stress. Design based on Stiffness in such case depends on the allowable deflection and twist of the shaft. The AISI 18Ni 350 steel is strong material so, it is selected to the feeder table head shaft.

The comparison based on their mechanical properties [40,27,45, 46].

Table 3.6: Material comparison by their mechanical properties of the materials

Properties	IS 2062	C45 steel	AISI 4340 steel	AISI 18Ni 350 steel
Yield strength	240 Mpa	380 Mpa	710 Mpa	2363 Mpa
Ultimate Tensile strength	410 Mpa	710 Mpa	1110 Mpa	2415 Mpa
Elastic modulus	$2*10^5$ Mpa	$2.1*10^5$ Mpa	$2.05 * 10^5$ Mpa	$2.0 * 10^5$ Mpa
Poisson's ratio	0.3	0.31	0.29	0.3
Mass density	7850 Kg/m^3	7850 Kg/m^3	7850 Kg/m^3	7850 Kg/m^3
Shear modulus	76.9 Gpa	80 Gpa	80 Gpa	77 Gpa

In this work mainly focus on AISI 18Ni 350 steel is compared with the currently used in the industries. And AISI 18Ni 350 steel has more advantages when compared to the other materials listed above.

Mild steel cannot be hardened by heat treatment. Because heat treatment affects carbon. If there is not enough carbon in a steel, there won't be any significant improvement. And the other disadvantage is mild steel cannot be used in highly stressful applications as the infrastructure of constrictions. And as shown in the above table 3.6: comparison of the two materials IS2062, C45, AISI 4340 and AISI 18Ni 350 steel is based on their mechanical properties. The IS2062 is an Indian standard material that is mild steel and its strength the material is very small with comparing to the other steels. Therefore, depend on the above comparison criteria AISI 18Ni 350 steel is better and it is selected for this study.

3.4. Finite element method

FEM analysis is very efficient and simple method for achieving stresses at different loading condition, according to forces applied to the feeder table head shaft. The finite element analysis is a numerical procedure for analysis of complicated shapes and structures. In this method the geometrical model is divided into small elements and each element are connected by nodes. Each node is having some degrees of freedom based on the number of nodes, degrees of freedom, material properties and element stiffness matrix is generated for each element. Selection of the type of element affects directly on the accuracy of results. Accuracy of result is increased either by increasing the number of elements or by selecting higher order element. Analysis of feeder table head shaft is done for observing maximum stresses, deformation and von-Mises stresses and fatigue life cycle of the shaft when different forces such as a load of the canes and torque due to power transmission are applied on it.

Finite element method is a numerical method for solving a differential or integral Equation. It has been applied to a number of physical problems, where the governing differential equations are available. The method essentially consists of assuming the piecewise continuous function for the solution and obtaining the parameters of the function in a manner that reduces the error in the solutions [41].

Finite Element Method reduces degree of freedom from infinite to finite with the help of discretization i.e. (nodes & elements) [42].

3.4.1. Modeling of the feeder table head shaft

A three-dimensional model of the feeder table head shaft is made by using modeling software CATIA by using the appropriate dimensions from the existing shaft and it imports to ANSYS workbench software. ANSYS is the most powerful and widely used software of its kind in the world.

3.4.1.1. Modeling of the existing feeder table head shaft

The model of this existing feeder table head shaft is modeled by CATIA software with the existing dimensions. Its length is 11.877 m, the external and internal diameter of the hollow is 790 mm and 750 mm and the diameter of the solid intermediate shaft is 300mm. The material used for this shaft is IS20762 carbon steel.

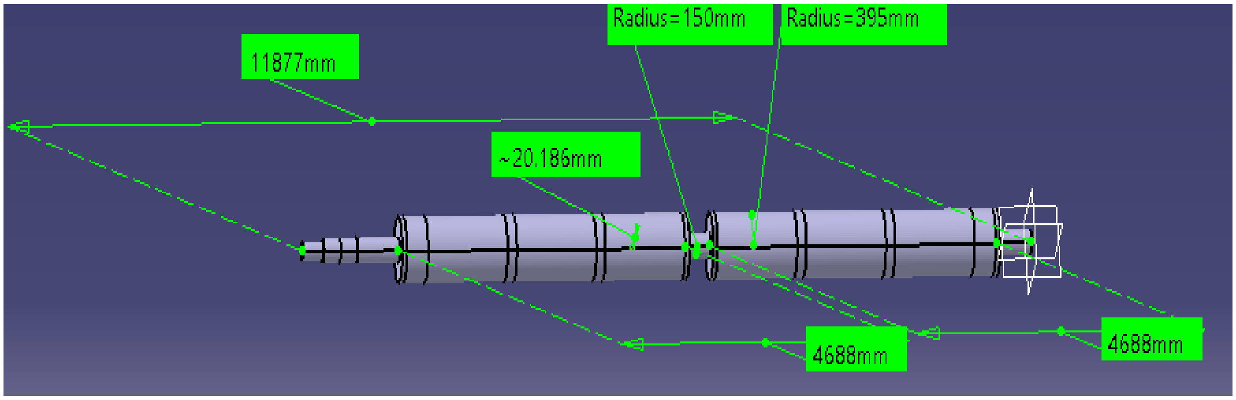

Figure 3.6: 3D model of the existing feeder table head shaft

3.4.1.2. Modeling of the hollow feeder table head shaft

This hollow feeder table head shaft is modelled by CATYA software. Its length is 11.877 mm with the external, internal diameter 750mm and 712mm. In this model the intermediate solid shaft is avoided and its thickness is increased from 20 mm to 39mm with the material IS2062 Indian standard mild steel.

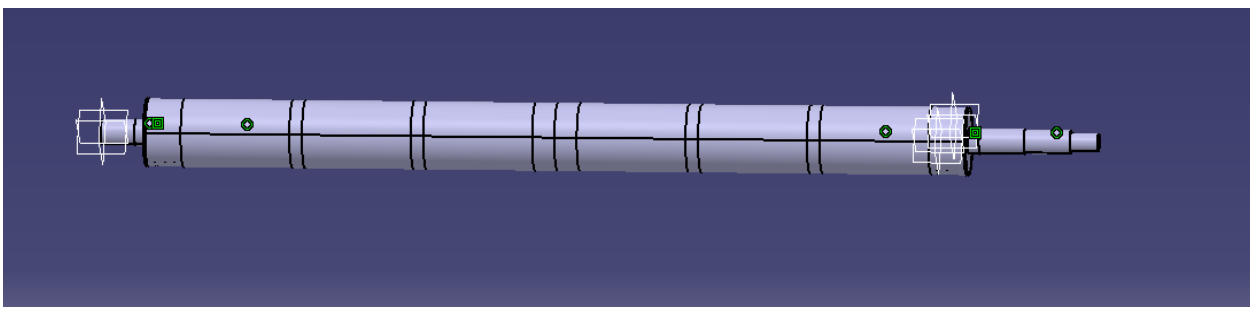

Figure 3.7: 3D model of old feeder table head shaft

3.4.1.3. Modeling of the modified feeder table head shaft

The new modified feeder table head shaft is modeled by CATIA software. This geometry is modified by changing the material from IS2062 steel to AISI 18Ni 350 steel. The material is changed to have a high strength. And the new model of the feeder table head shaft is modified by shortening of the intermediate solid shaft length to avoid the effect of weight. And by calculating the diameter of the solid shaft to resist the applied stress, the diameter of the intermediate solid shaft is increased to 400 mm to have a long life and to reduce the stress concentration occurred on the solid shaft.

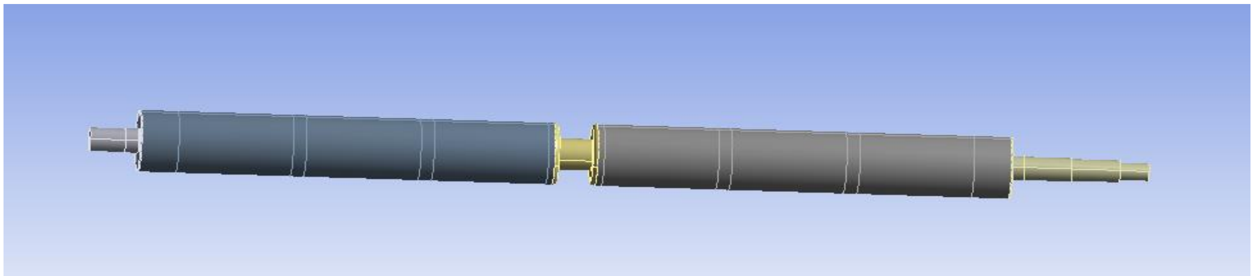

Figure 3.8: 3D model of old feeder table head shaft

3.4.1.4. Meshing of the feeder table head shaft

The CATYA model is imported to ANSYS workbench. Meshing is performed in the same software. Meshing is the process of converting the model in to a number of discrete parts called as an element. The discretization/ mesh geometry is the first step of the finite element. In discretization No. of nodes formed are 598,921 and No of elements are 345,747 using triangle surface mesh.

In this step the component is divided into a number of small parts because the effect of the forces is not the same for each component.

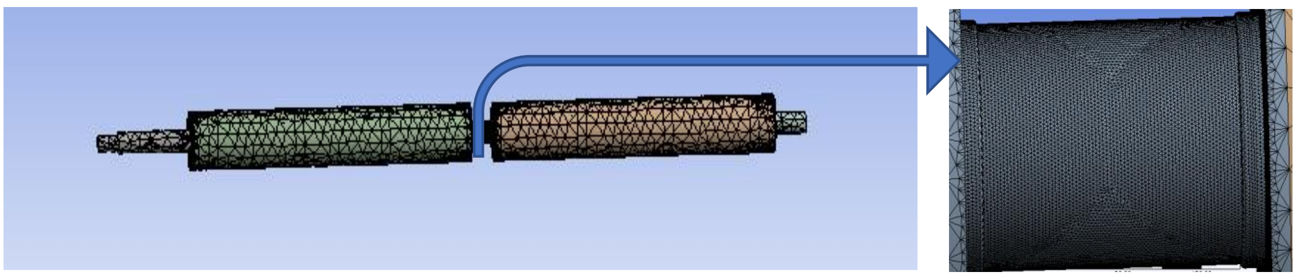

Figure 3.9: meshing of the feeder table head shaft

3.4.1.5. Load and boundary conditions of the feeder table head shaft

The load condition for models of the feeder table head shaft is the same in magnitude and direction. Finite element modelling of the feeder table head shaft consists of geometry generation, applying material properties, meshing the component, defining the boundary constraints, and applying the proper load type. The feeder table head shaft is supported by bearings. And the loads are applied to the feeder table head shaft with the vertical and horizontal components.

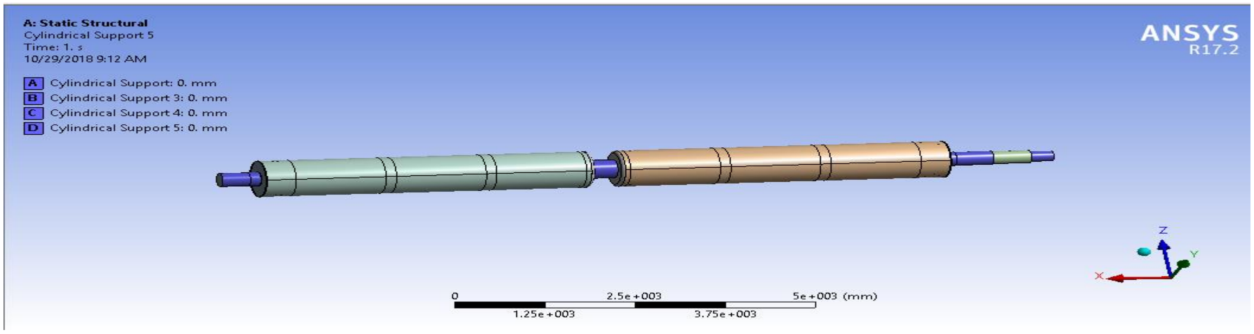

Figure 3.10: Boundary condition of the modified feeder table head shaft

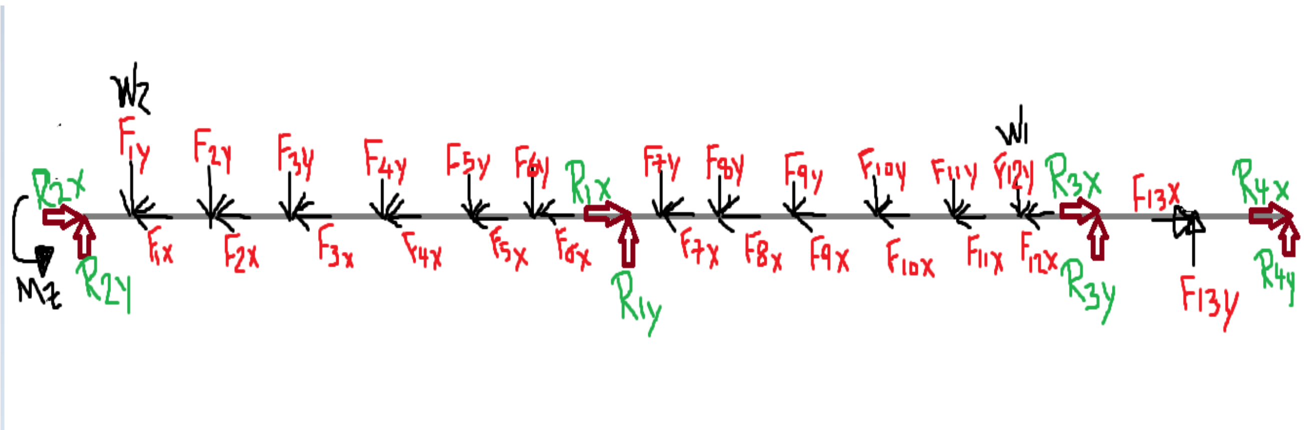

Figure 3.11: Loading and boundary condition of the modified feeder table head shaft

The loads acting on the feeder table head shaft are due to weight of sugar cane, the tension of the chain and the torque. Due to heavy load on the shaft it is necessary to check and calculate the required size of the shaft torsional & bending stresses. The feeder table head shaft is an important item of sugar factory equipment and being subject to heavy loading it must be made to a high standard of quality.

The Speed reduction and torque multiplication are calculated from the data given in wonji-shoa sugar factory from the given power and speed (55 KW and 1500 RPM). The input torque is equal to the power given divided by its angular speed. And the output torque is calculated by multiplying with the speed reduction.

Speed reduction and torque multiplication:

$$P = T\,\omega$$

Where: T = torque

ω = angular speed

P = power

$T1_{input} = P/\,\omega$

$= P/\,(2\pi N/60)$

$= 60 * 55\ KW / (2 * 3.14 * 1500RPM)$

$= 350.1408748\ NM$

Where; $T1_{input}$ = torque input to the gear box from the motor

$T1_{output}$ = speed reduction * $T1_{input}$

$= 142 * 350.1408748$

$= 49720.00422\ NM$

Where; $T1_{output}$ = torque output from the gear box (torque in put to the open gear)

Speed reduction due to open gear = driven gear teeth/ driving gear teeth

= 120/22

= 5.454545455

$T2_{output} = T1_{input\ to\ the\ open\ gear}$ * speed reduction

$= 49745.156\ NM * 5.454545455$

$= 271200.0231\ NM$

Where; $T2_{output}$ = torque output from the open gear AND

$N_{output} = N_{input}$ / speed reduction

= 1500 RPM/ 142 = 10.56338028 RPM

N output = $N_{input\ to\ the\ open\ gear}$/ speed reduction due to the open gear

= 10.56338028 RPM/ 5.454545455

= 1.93 RPM

The chain sprockets transmitting power. The upper part of the chain is in tension and produces the torque on the sprocket. The lower part of the chain, referred to as the slack side, exerts no force on the sprocket. Therefore, the total bending force on the shaft carrying the sprocket is equal to the tension in the tight side of the chain. These two components of the force would cause bending in both the X direction and the Y direction. And the tangential and radial forces of the gear is calculated by considering its torque on the gear and pitch diameter of the gear.

The load at sprocket pitch circle

P.C.D = 882.12 mm

$F_t = 2T/ P.C.D$

$= 2*271200.0231\ NM/ 12*882.12*10^{-3}\ mm$

$= 51240.19\ N$

This is the bending load on the shaft the components are

$F_x = F_t * \cos16.5^0$

$= 49,130.113\ N$

$F_y = F_t * \sin 16.5^0$

$= 14,553.00\ N$

Force on the gear

$F_t = T\ output/r_G$

$= 49720.00422\ NM / 1.080\ M$

$= 46,037.04\ N$

$F_r = F_t * \tan 20^0$

$=46,037.04\ N * \tan 20^0$

$= 16,756.112\ N$

The total capacity of the cane to feed is 6250 TCD. But there are two feeder table head shafts operate in the company to feed the canes. So, the capacity of the cane 6250 tone cane per day is feeds by those two feeder table head shafts. The slope of the inclined portion of the cane carrier in wonji-shoa sugar factory is 16.5^0. Then the weight of the canes is decomposed to X and Y components because of the inclination angle.

Weight of the cane

6250 TCD

6250/24 = 260.4166667 TCH

But the speed of the table is 5 m/min at 1.93 RPM

Then the applied weight of the cane at this 5m/min is 18,993.055 Kg

The distributed weight = 18,993.055 Kg * 9.81 m/s^2 = 186,321.869 N

But the feeder table canes are two

Then the distributed weight for one table is 93,160.93 N

The concentrated load at the center of the shaft is = WL = 93,160.93 N * 9.07 m = 844,969.7 N

And the concentrated loads applied on the two end points of the shaft is:

$$W\ x = WL/2 * \cos 16.5^0 = 405{,}086.8 \text{ N and}$$

$$W\ y = WL/2 * \sin 16.5^0 = 119{,}992.15 \text{ N.}$$

CHAPTER 4

4.Result and Discussions

In this portion all the results of the existing, hollow and modified feeder table head shafts are discussed and compared depending on the results of the feeder table head shaft from ANSYS 17.2 results. To achieve the objective of this paper the static structural and life prediction analysis is done on the different models of the feeder table head shaft.

4.1. Static structural analysis of the feeder table head shaft

Static structural analysis of the feeder table head shaft is done for observing maximum shear stresses, total deformation, total equivalent strain, maximum von-Mises stresses, fatigue life cycle and fatigue factor of safety of the shaft. The static structural analysis is done using software of ANSYS 17.2, workbench.

A static structural analysis is used to determine the displacements, stresses, strains in structures or components caused by loads [43].

4.1.1. Existing feeder table head shaft with the material of IS2062 steel

Table 4.1: Result of existing feeder table head shaft with the material of IS2062

No	IS 2062
1	Ultimate tensile strength = 410 Mpa
2	Yield strength = 240 Mpa
3	Density- 7850 Kg/m³.
4	Modules of elasticity = $2*10^5$ Mpa
5	Poisson ratio = 0.3
6	Von misses stress = 103.49 Mpa
7	Maximum shear stress = 53.6977 Mpa
8	Minimum principal stress = 35.289 Mpa
9	Maximum principal stress = 132.48 Mpa
10	Life cycle = 3.4907e5
11	Fatigue factor of safety = 0.83294
12	Deflection = 0.4499 mm
13	Factor of safety in yield = 2.3191

From the table 4.1: Above the results of the existing feeder table head shaft is discussed with the material of IS2062 carbon steel. The allowable shear stress is less than the design shear stress (53.6977 Mpa < 72 Mpa). The von-misses stress is less than the yield strength (103.49 Mpa < 240 Mpa). The factor of safety in yield is 2.3191 and fatigue factor of safety 0.83294 is less than one.

4.1.2. Existing feeder table head shaft with the material of C45 steel

Table 4.2: Result of existing feeder table head shaft with the material of C45 steel

No	C45 steel
1	Ultimate tensile strength = 600-750 Mpa
2	Yield strength = 380 Mpa
3	Density- 7850 Kg/m³.
4	Modules of elasticity = 190-210 Gpa
5	Poisson ratio = 0 .3
6	Maximum Von misses = 103.49 Mpa
7	Maximum shear stress = 53.697 Mpa
8	Minimum principal stress = 35.289 Mpa
9	Maximum principal stress = 132.48 Mpa
10	Life cycle = 3.4907e5
11	Fatigue factor of safety = 0.83294
12	Deflection = 0.42666 mm
13	Factor of safety in yield = 3.6719

From the table 4.2: Above the results of the existing feeder table head shaft is discussed with the material of C45 carbon steel. The allowable shear stress is 53.697 Mpa. The maximum von-miss stress is 103.49 Mpa. The fatigue factor of safety is 0.83294 and the factor of safety due to yield is 3.6719. The result of this C45 steel is known by its composition using the Spectro test in AKAKI basic metals. The fatigue factor of safety of the C45 carbon steel is less than one (0.83294 < 1).

4.1.3. Hollow feeder table head shaft with the material of IS2062 steel

Table 4.3: Result of hollow feeder table head shaft with the material of IS2062

No	IS 2062
1	Ultimate tensile strength= 410 Mpa
2	Yield strength = 240 Mpa
3	Density- 7850 Kg/m^3.
4	Modules of elasticity = 2*10^5 Mpa
5	Poisson ratio = 0.3
6	Von- misses stress= 114.03 Mpa
7	max. shear stress = 62.514 Mpa
8	Minimum principal stress = 35.23 Mpa
9	Maximum principal stress = 123.86 Mpa
10	Life cycle = 1.998e5
11	Fatigue factor of safety = 0.75593
12	Deflection = 4.2455 mm
13	Factor of safety in yield = 2.1047

Similarly, as shown in the table 4.3: Above the hollow feeder table head shaft is failed with material of IS2062 carbon steel. The fatigue factor of safety is less than one (0.75593 <1) and also, its deflection is very high.

4.1.4. Modified shaft of feeder table head shaft with the material of AISI 4340 steel

Table4.4: Modified shaft results with the material AISI 4340 steel

No	AISI 4340
1	Ultimate tensile strength = 1110 Mpa
2	Yield strength = 710 Mpa
3	Density- 7850 Kg/m^3
4	Modules of elasticity = 205 Gpa
5	Poisson ratio = 0.29
6	max. shear stress = 40.282 Mpa
7	Von- misses =73.34 Mpa
8	Minimum principal stress = 28.871 Mpa
9	Maximum principal stress = 102.09 Mpa
10	Life cycle = 10^6
11	Fatigue factor of safety = 1.1754
12	Deflection = 0.41212 mm
13	Factor of safety = 9.681

From the above table 4.4: The modified feeder table head shaft with the material of AISI 4340 steel the allowable shear stress 40.282 Mpa is less than the design shear stress 199.8 Mpa. The maximum von-miss stress 73.34 Mpa is less than the yield strength. The fatigue factor of safety and fatigue factor of safety due to yield is greater than one.

4.1.5. Modified shaft of feeder table head shaft with the material of AISI 18Ni 350 steel

Table4.5: Modified shaft results with the material AISI 18Ni 350 steel

No	AISI 18Ni 350
1	Ultimate tensile strength = 2415Mpa
2	Yield strength = 2363 Mpa
3	Density- 7850 Kg/m^3
4	Modules of elasticity = 200 Gpa
5	Poisson ratio = 0.3
6	max. shear stress = 40.171 Mpa
7	Von- misses =72.373 Mpa
8	Minimum principal stress = 30.13 Mpa
9	Maximum principal stress = 102.13 Mpa
10	Life cycle = 10^6
11	Fatigue factor of safety = 1.1911
12	Deflection = 0.42242 mm
13	Factor of safety = 15

From the above table 4.5: The modified feeder table head shaft with the material AISI 18Ni 350 steel is safe. Since, the allowable shear stress 40.171 Mpa is less than the design shear stress. The maximum von-miss stress 72.373 Mpa is less than the yield strength. The fatigue factor of safety and fatigue factor of safety due to yield is greater than one.

S-N curve for the materials of C45 steel

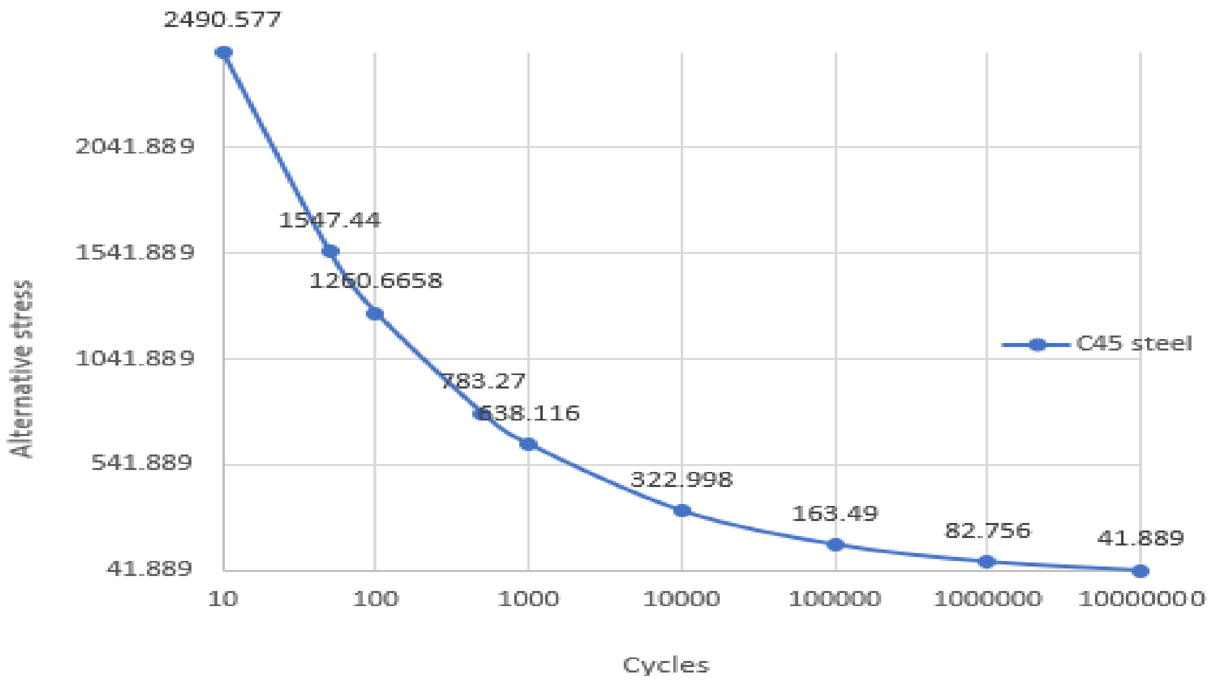

Figure 4.1: Alternating stress parameters of feeder table head shaft with the material of C45 steel

The graph 4.1: Expresses about the alternative stress and the number of cycles of the existing feeder table head shaft with the material of C45 steel. The maximum alternating stress at 10 cycles is 2490.572 Mpa and at 41.889 Mpa is alternating stress the maximum number of cycles is 10^7.

S-N curve for the materials of IS2062 steel

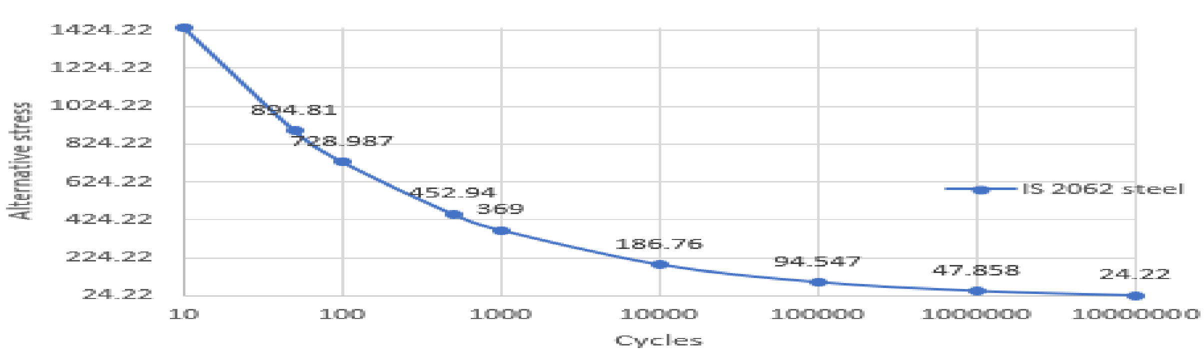

Figure 4.2: Alternating stress parameters of feeder table head shaft with material of IS2062 steel

The graph 4.2: Expresses about the alternative stress and the number of cycles of the existing feeder table head shaft with the material of IS2062 steel. The maximum alternating stress at 10 cycles is 1440.15 Mpa and at 24.22 Mpa is alternating stress the maximum number of cycles is 10^7.

S-N curve for the materials of AISI 4340 steel

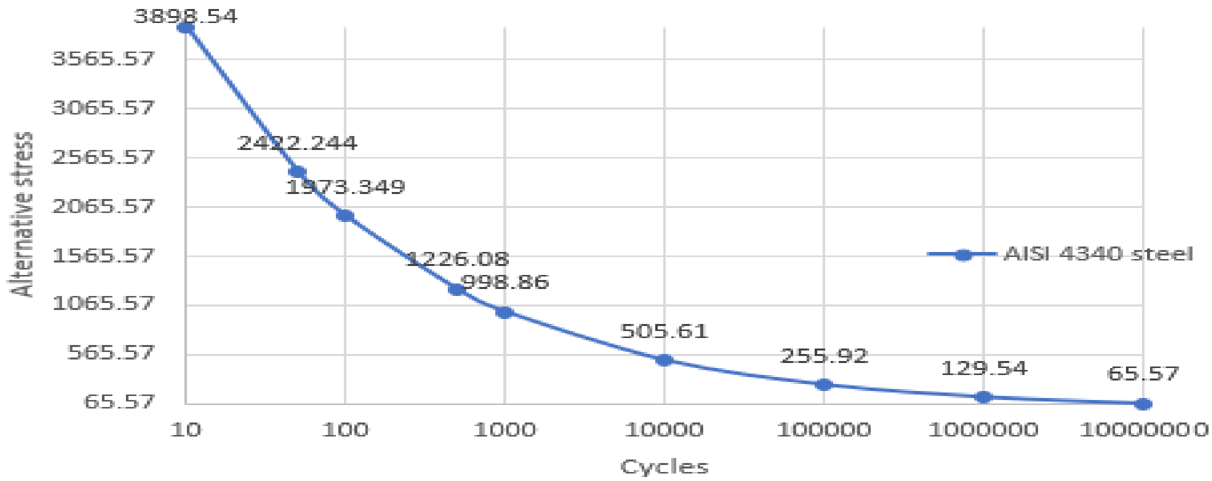

Figure 4.3: Alternating stress parameters of feeder table head shaft with the material of AISI 4340 steel

The graph 4.3: Expresses about the alternative stress and the number of cycles of the modified feeder table head shaft with the material of AISI 4340 steel. The maximum alternating stress at 10 cycles is 3898.54 Mpa and at 65.57 Mpa is alternating stress the maximum number of cycles is 10^7.

S-N curve for the materials of AISI 18Ni 350 steel

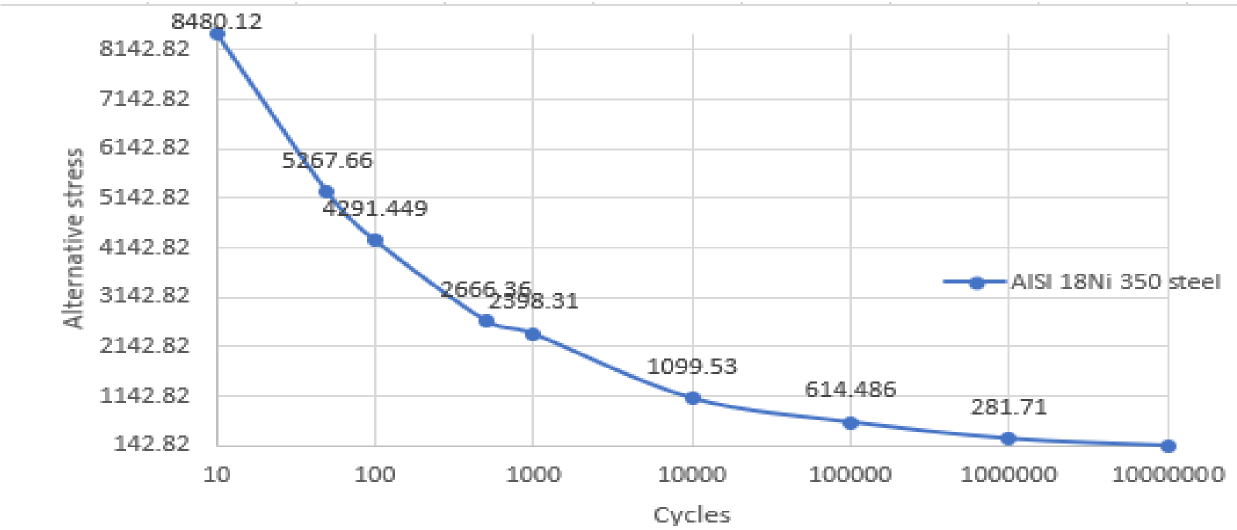

Figure 4.4: Alternating stress parameters of feeder table head shaft with the material of AISI 18Ni 350 steel

The graph 4.4: Expresses about the alternative stress and the number of cycles of the modified feeder table head shaft with the material of AISI 18Ni 350 steel. The maximum alternating stress at 10 cycles is 8480.12 Mpa and at 142.82 Mpa is alternating stress the maximum number of cycles is 10^7.

Comparison of S-N curve for the materials and life prediction of the modified feeder table head shaft.

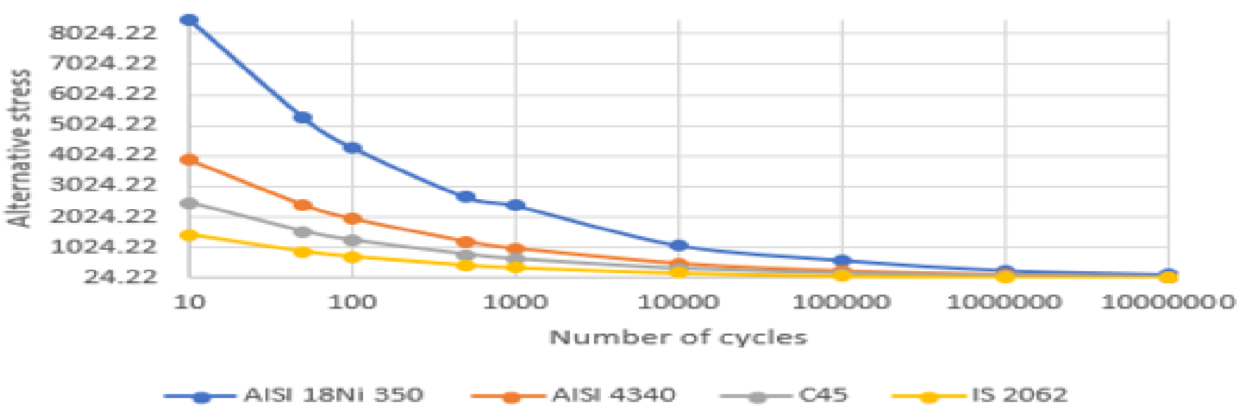

Figure 4.5: alternating stress parameters of feeder table head shaft

As shown in the figure 4.5: Above the feeder table head shaft as the number of cycles per revolutions goes increasing, alternating stress increases proportionally as shown in the graph. The graph expresses about the alternative stress and the number of cycles of the modified and existing feeder table head shaft with the different four materials. The modified feeder table head shaft with the material AISI 18Ni 350 steel is selected from the result. Because the life cycle of the AISI 18Ni 350 steel is better than the other materials listed above. Materials have a fatigue limit or endurance limit which represents a stress level below which the material does not fail and can be cycled infinitely. If the applied stress level is below the endurance limit of the material, the structure is said to have an infinite life. AISI 18Ni 350 has more advantages when compared to IS 2062, C45, AISI 4340 steel. So, it is selected for the feeder table head shaft to have long life time.

Accurate prediction of life is very important to insure safety of the feeder table head shaft and its reliability. The life of the modified feeder table head shaft with the material of AISI 18 Ni 350 steel at 10^7 cycles the stress is 142.82 Mpa based on banding load and 74.40 Mpa based on torsional load.

And the maximum von- misses stress and maximum shear stress of the modified feeder table head shaft is 72.373 Mpa and 40.171 Mpa.

The feeder table head shaft works for 22 hours per day and it works for 9 months per year and the left 3 months use to overall maintenance.

Life cycle = 10^7

The speed of the shaft = 1.93 RPM

So, the life prediction is;

$10^7/1.93*60*22*30*9 = 14.538$ year

Then, the life prediction for the new modified feeder table head shaft is 14.5 years.

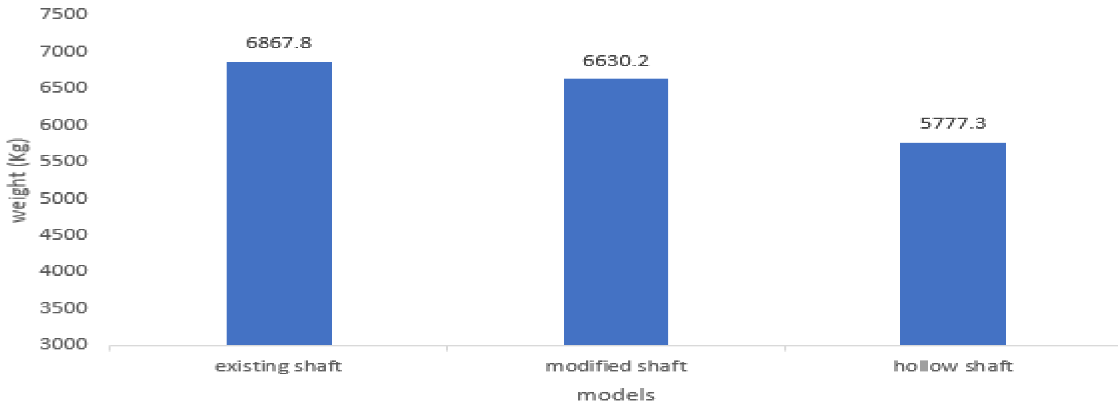

Figure 4.6: weight comparison of the existing, hollow shafts with the new modified shaft

The figure 4.6: Above shown that the compares the weight of the model's feeder table head shaft. The original weight of the feeder table head shaft is 6867.8 Kg. The modified shaft weight is reduced by 3.45% from the existing shaft. since, its diameter is increased and its length is shortened. And the hollow shaft is reduced by 14.76% from the existing feeder table head shaft. But the hollow feeder table head shaft deflection, maximum shear stress and maximum von-misses stress is large compared to the existing and modified feeder table head shaft. And its fatigue life cycle and fatigue factor of safety is very small comparing to the existing and modified feeder table head shaft.

4.1.5. Maximum shear of the feeder table head shaft

According to ASME code the maximum shear stress on the shaft is 0.3* S_{yt} and 0.18* S_{ut}. Or, ASME Code for steel under definite specifications 30% of the yield strength but not over 18% of the ultimate strength in tension for shafts [7].

For the material, "IS 2062" (S_{yt} = 240 Mpa and S_{ut} = 410 Mpa)

0.3 * 240 Mpa = 72 Mpa and 0.18 * 410 Mpa = 73.8 Mpa

The smaller design shear stress of the material is 72 Mpa.

For the material 45C8 steel (S_{yt} = 380 and S_{ut} = 710)

0.3 * 380 = 114 Mpa and 0.18 * 710 = 127.8 Mpa

The smaller is 114 Mpa.

For the material AISI 4340 steel, S_{yt} = 710 and S_{ut} = 1110

0.3 * 710 = 213 Mpa

0.18 * 1110 = 199.8 Mpa

The smaller is 199.8 Mpa

For the material AISI 18Ni 350 steel, S_{yt} = 2363 and S_{ut} = 2415

0.3 * 2363 = 708.9 Mpa

0.18 * 2415 = 434.7 Mpa

Then the smaller design shear stress of the material is 434.7 Mpa.

4.1.5.1. Maximum shear stress of the existing feeder table head shaft (IS 2062)

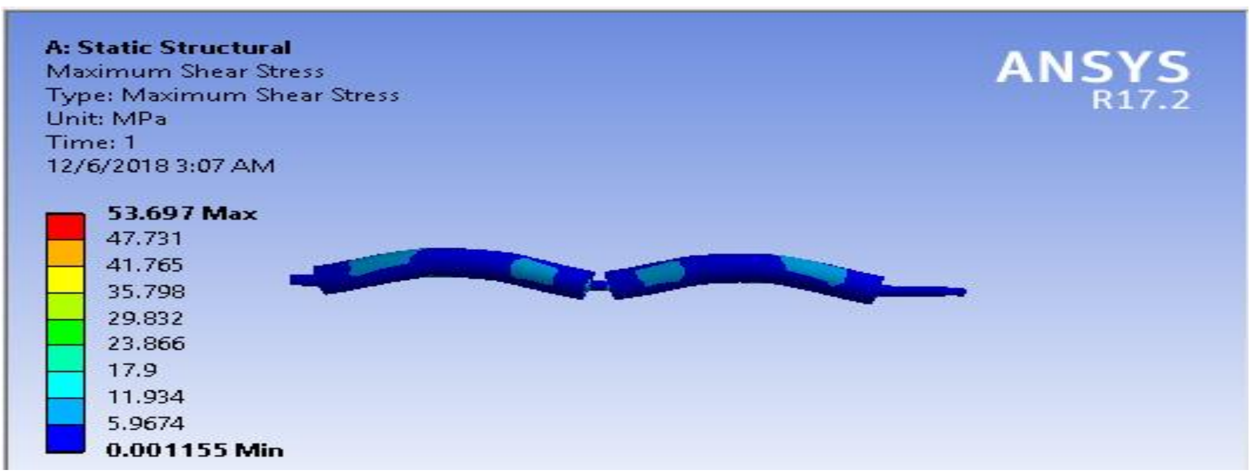

Figure 4.7: maximum shear stress of the existing feeder table head shaft

From the figure 4.7: Above the maximum shear stress value of the existing feeder table head shaft is 53.697 MPa and the minimum value of shear stress is 0.001155 Mpa.

4.1.5.2. Maximum shear stress of the hollow feeder table head shaft

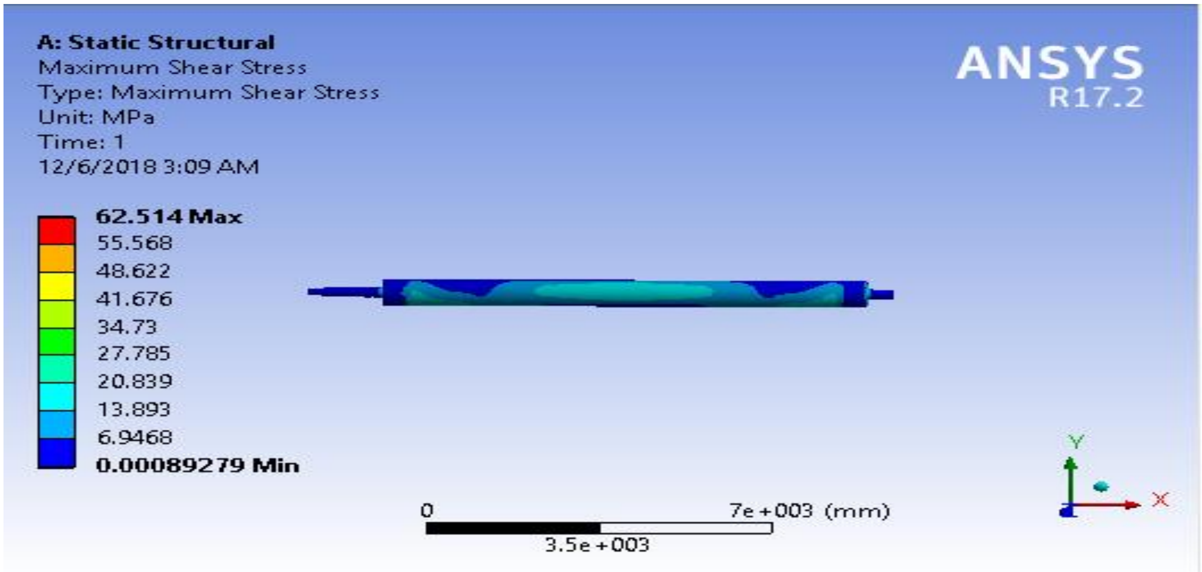

Figure 4.8: maximum shear stress of the hollow feeder table3 head shaft

From figure 4.8: Above the maximum value of shear stress is 62.514 MPa and minimum value of shear stress is 0.00089279 Mpa.

4.1.5.3. Maximum shear stress of the modified feeder table head shaft (AISI 18Ni 350)

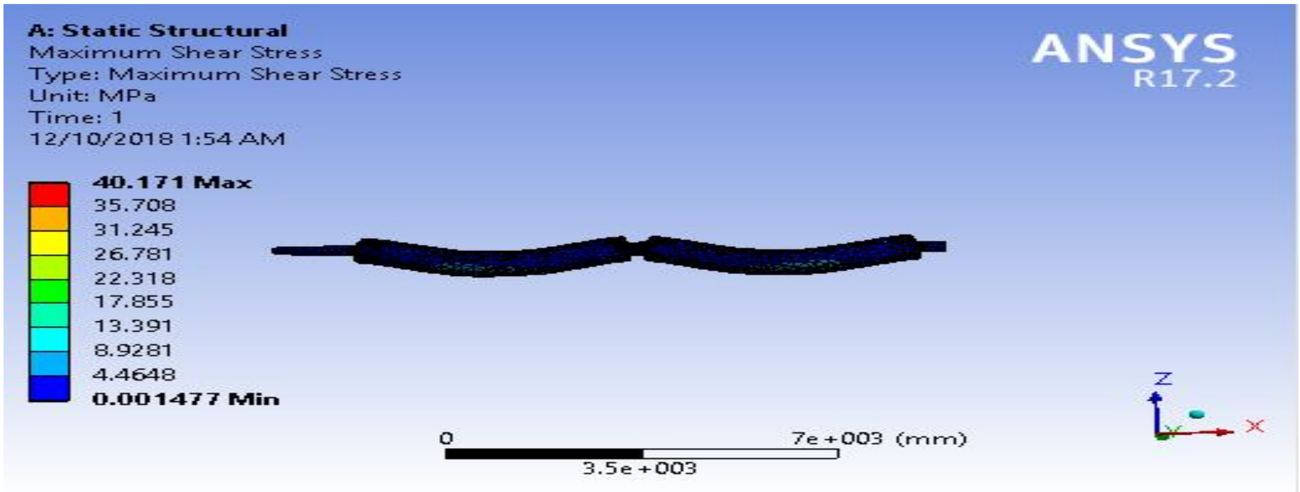

Figure 4.9: maximum shear stress of the modified feeder table head shaft

As observed from figure 4.9: Above the maximum value of shear stress is 40.171 MPa and minimum value of shear stress is 0.001477 Mpa.

Table 4.6: Comparison and discussion of the feeder table head shaft based on Maximum shear stress.

Feeder table head shaft	Maximum shear stress in (MPa)
Existing shaft (IS 2062)	53.697
Modified shaft (AISI 4340)	40.282
Hollow shaft (IS 2062)	62.514
Modified shaft (AISI 18Ni 350)	40.171

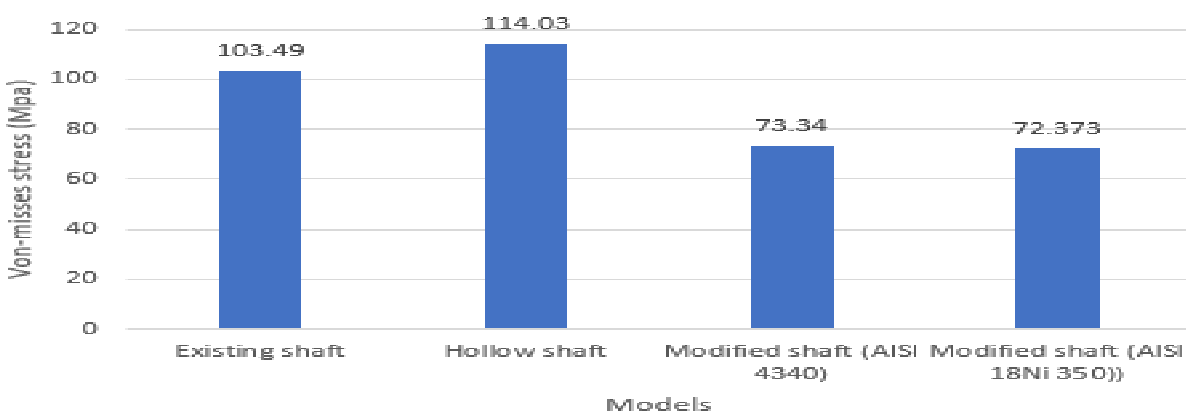

Figure 4.10: comparison of maximum shear stress of the shaft models existing, hollow and modified feeder table head shafts

As shown from the figure 4.10: and table 4.6: The maximum shear stress induced on the modified feeder table head shaft with the material AISI 18Ni 350 is less than the existing and hollow feeder table head shafts. The maximum shear stress of the hollow feeder table head shaft is more than the modified and existing shafts.

4.1.6.von-Mises stress of the feeder table head shaft

4.1.6.1. von-Mises stress of the existing feeder table head shaft (IS 2062)

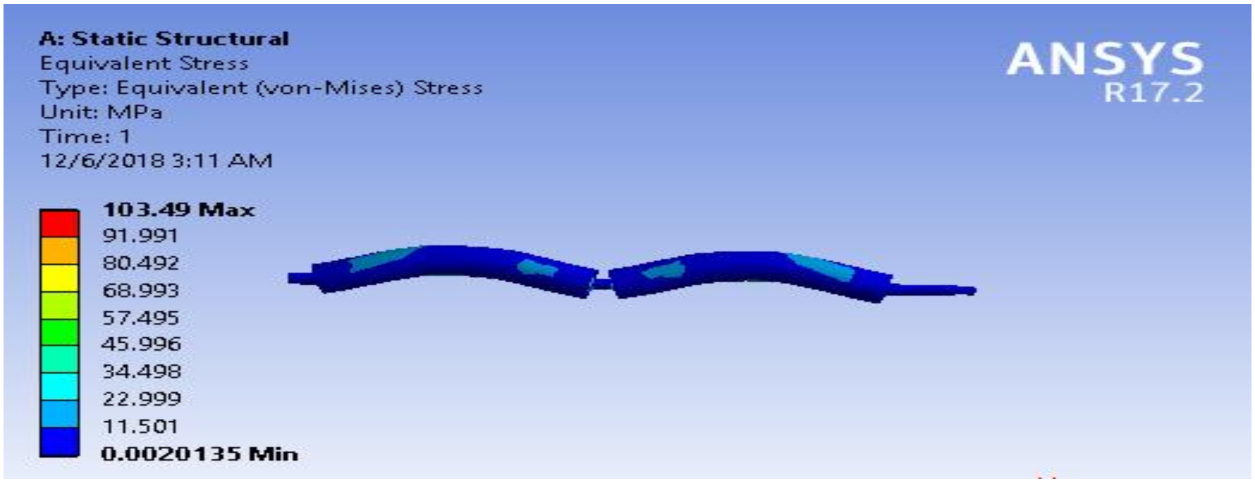

Figure 4.11: von-misses of the existing feeder table head shaft

From figure 4.11: Above shown the maximum value of the von-Mises equivalent stress is 103.49 Mpa the and minimum value of the von-Mises stress is 0.0020135 Mpa.

4.1.6.2. von-Mises stress of the hollow feeder table head shaft

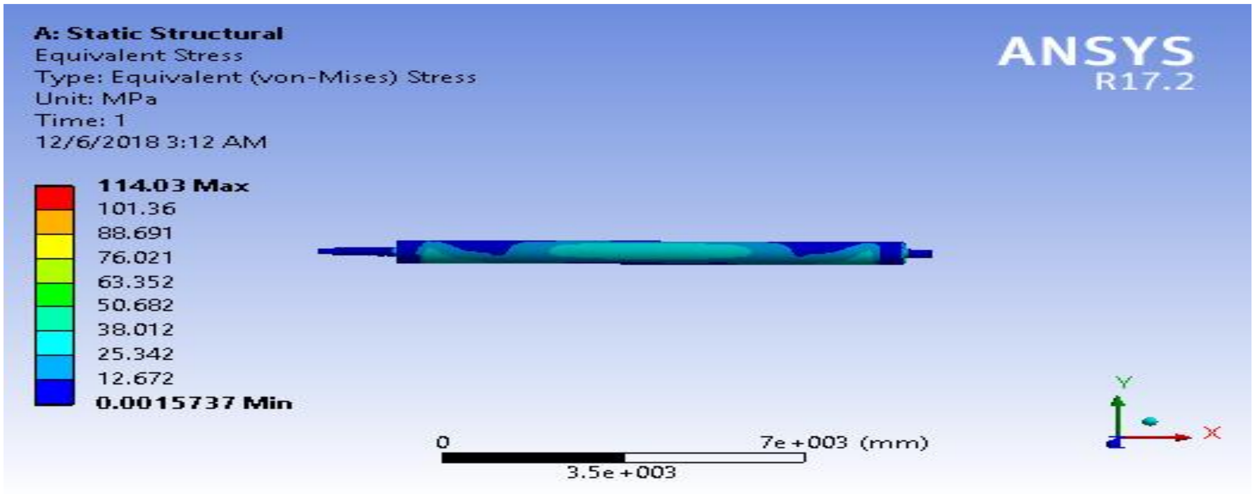

Figure 4.12: von-misses of the hollow feeder table head shaft

As shown from figure 4.12: Above the maximum value of the von-Mises equivalent stress is 114.03 Mpa the and minimum value of the von-Mises stress is 0.0015737 Mpa.

4.1.6.3. von-Mises stress of the modified feeder table head shaft (AISI 18Ni 350)

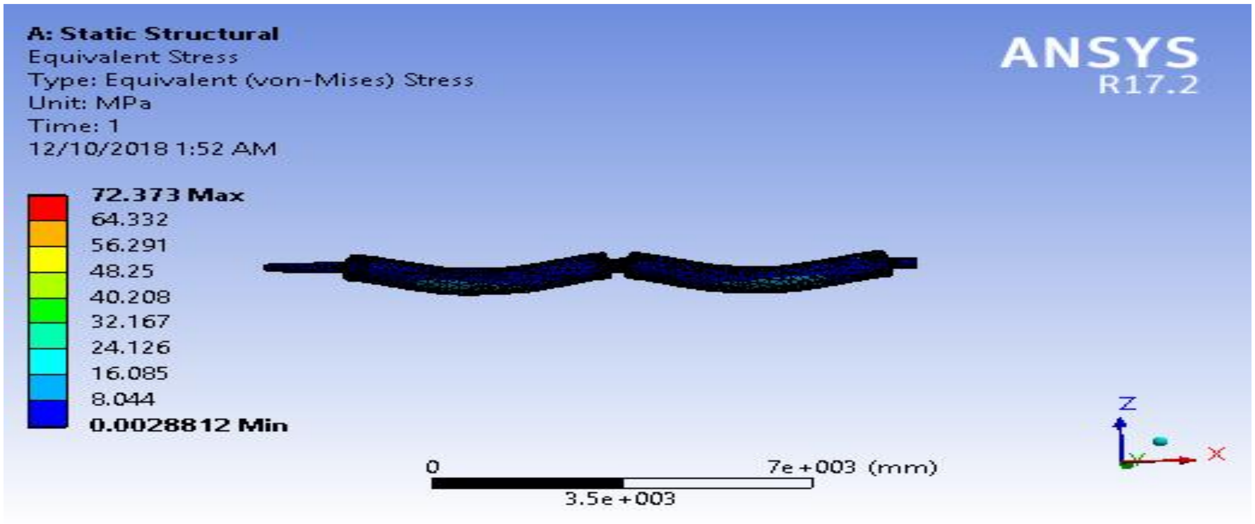

Figure 4.13: von-misses of the modified feeder table head shaft

The figure 4.13: Above expresses the stress acting on the feeder table head shaft. The maximum stresses on the feeder table head shaft is 72.373 Mpa and the minimum stress (von-misses) is 0.00288121 Mpa.

Table 4.7: Comparison and discussion of the feeder table head shaft based on maximum von-Misses stress.

Feeder table head shaft	Maximum von-Misses stress in (MPa)
Existing shaft (IS 2062)	103.49
Modified shaft (AISI 4340)	73.34
Hollow shaft (IS 2062)	114.03
Modified shaft (AISI 18Ni 350)	72.373

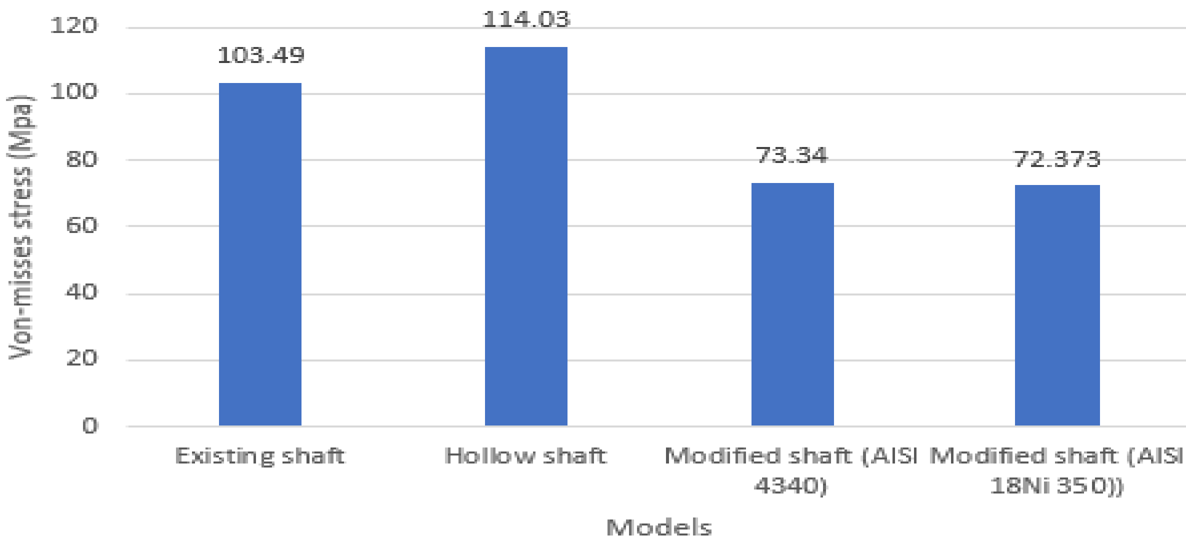

Figure 4.14: comparison of maximum von-misses of the models existing, hollow and modified feeder table head shaft.

From the figure 4.14: and table 4.7: Above the von-misses stress on the modified feeder table head shaft with the material AISI 18Ni 350 is less than the existing and hollow feeder table head shaft. So, the modified feeder table head shaft (AISI 18Ni 350) is safe and, it is better than the existing and hollow feeder table head shafts when it compared based on their results.

4.1.7. Total deformation of the feeder table head shaft

4.1.7.1. Total deformation of the existing feeder table head shaft (IS 2062)

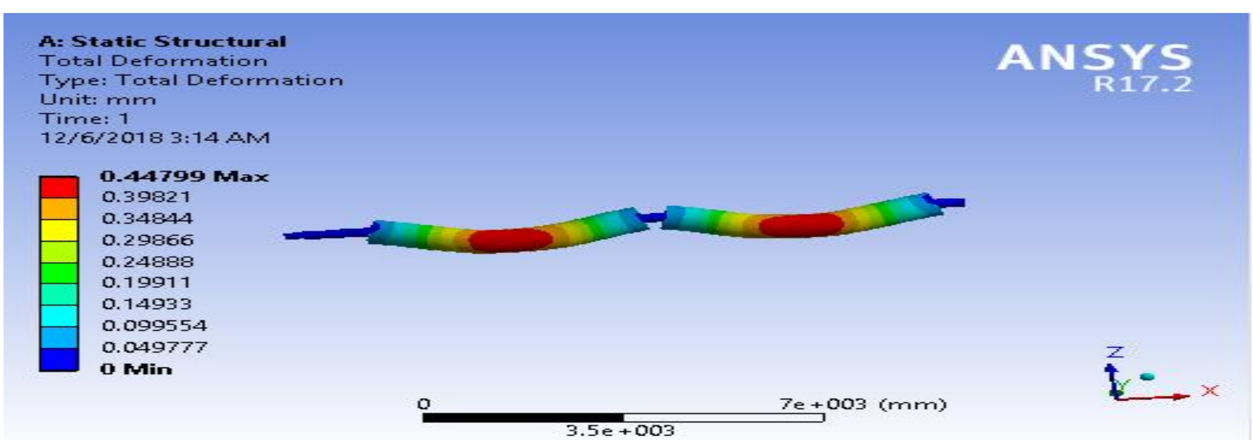

Figure 4.15: deformation of existing feeder table head shaft

The total deformation of the feeder table head shaft is shown in the figure 4.15: The deformation in the feeder table head shaft is not the same throughout. The portion in red color shows that the deformation in that region is maximized and the portion in blue color shows that the deformation is minimized in that region. The maximum deformation is 0.44799 mm. And the total equivalent strain of the existing feeder table head shaft is 1.1139e-8.

4.1.7.2. Total deformation of the hollow feeder table head shaft

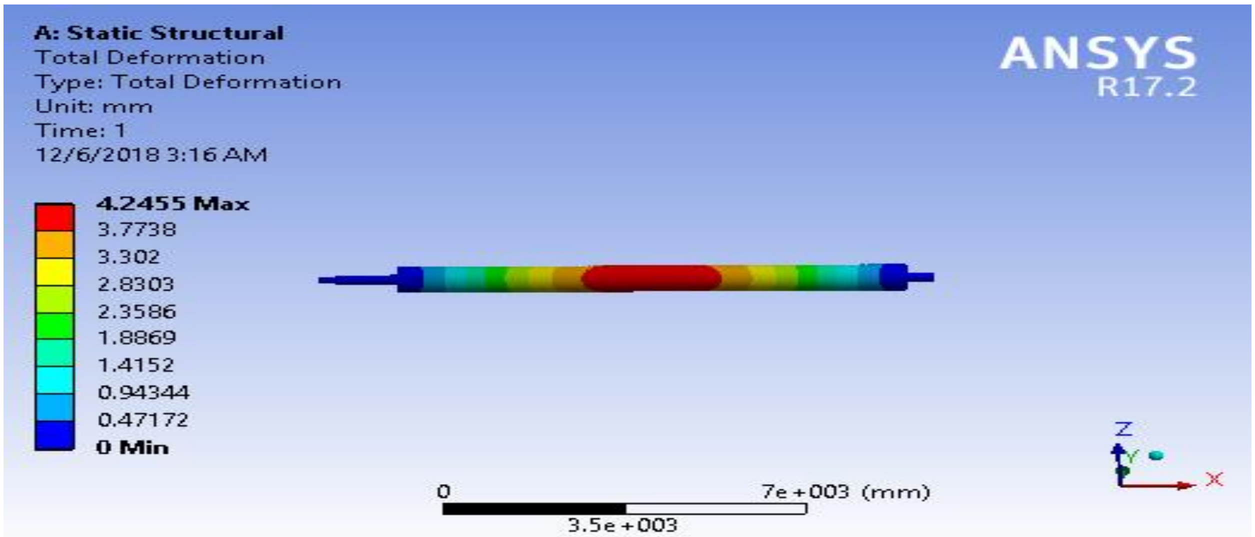

Figure 4.16: deformation of hollow feeder table head shaft

From the figure 4.16: as shown the total deformation of the feeder table head shaft is not the same throughout. The portion in red color shows that the deformation in that region is maximized and the portion in blue color shows that the deformation is minimized in that region. The maximum displacement is 4.2455 mm. And the equivalent total strain of the hollow feeder table head shaft is 2.1542e-8.

4.1.7.3. Total deformation of the modified feeder table head shaft (AISI 18Ni 350)

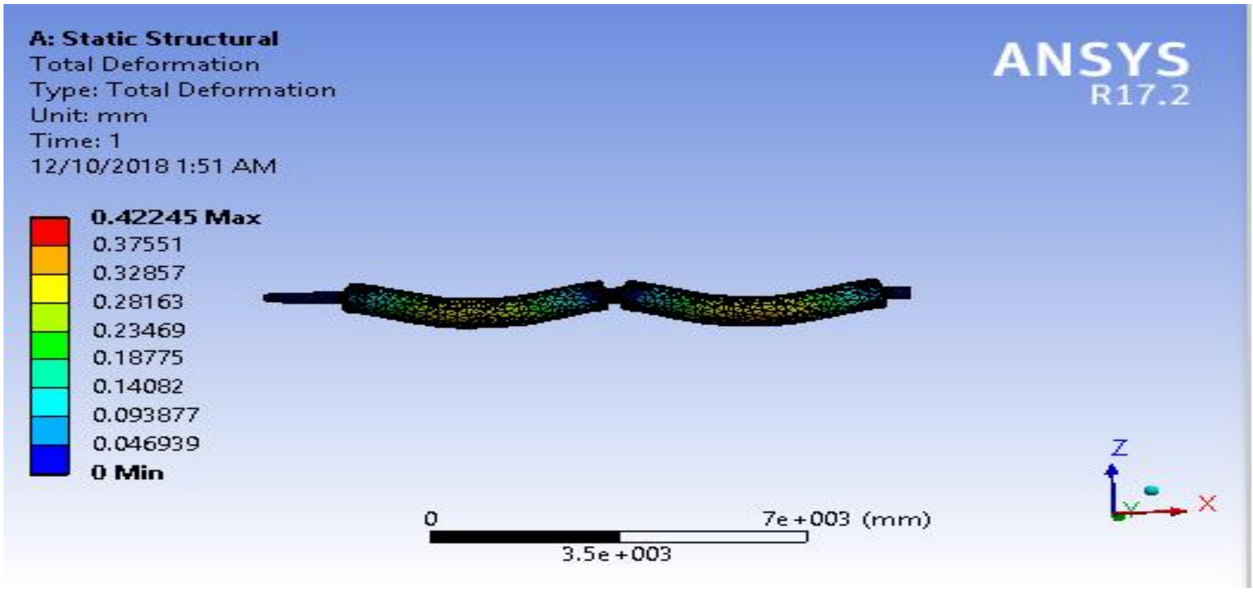

Figure 4.17: deformation of modified feeder table head shaft

The total deformation of the feeder table head shaft is shown in the figure 4.17: the minimum deformation is 0 and the maximum deformation is 0.42245 mm. The portion in red color shows that the deformation in that region is maximized and the portion in blue color shows that the deformation is minimized in that region. And the equivalent total strain of the modified feeder table head shaft is 1.6229e-8

4.1.8. Fatigue life cycle of the feeder table head shaft

Fatigue is caused by a repeated cycle load. It is a progressive localized damage and the fatigue is when a motion is repeated the object that is doing the work becomes weak. Fatigue occurs when a material is subject to alternating stresses, over a long period of time. The fatigue analysis study is important in the case of the feeder table head shaft. Because one of the most common causes of shaft failure is due to fatigue.

Two basic regimes are low-cycle fatigue (generally below 1000 stress cycles) and high-cycle fatigue (more than around 1000 but less than one million stress cycles). The slope of the line is much lower in low-cycle fatigue than in high-cycle fatigue. Low-cycle fatigue is any loading that causes failure below 1000 cycles.

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.

4.1.8.1. Fatigue life cycle of the existing feeder table head shaft (IS 2062)

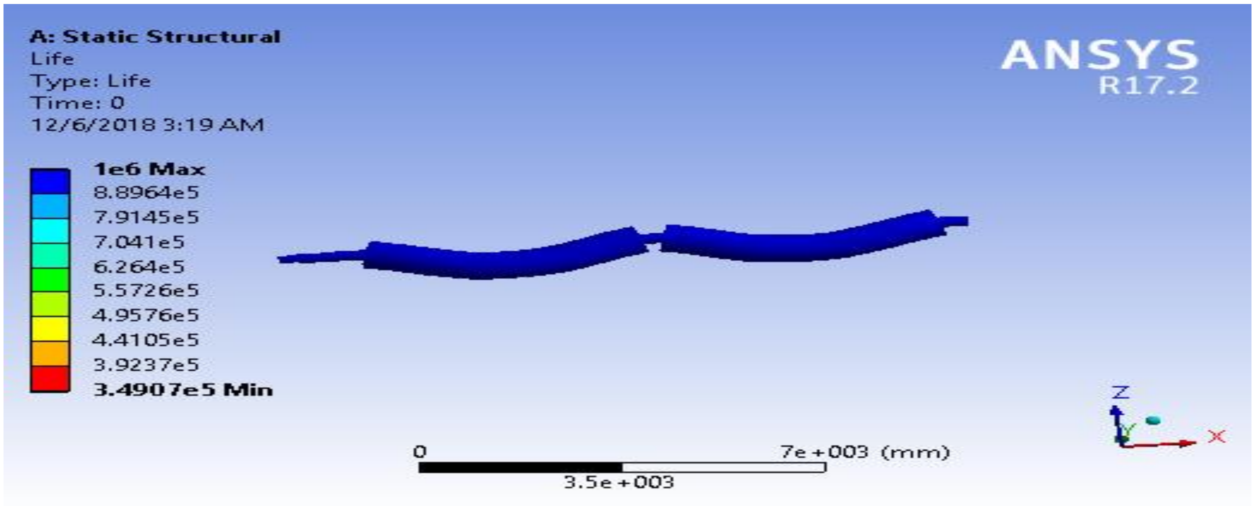

Figure 4.18: fatigue life cycle of existing feeder table shaft

As it observed from the figure 4.18: Above the fatigue life cycle of the existing feeder table head shaft is 3.4904e5 cycles.

4.1.8.2. Fatigue life cycle of the hollow feeder table head shaft

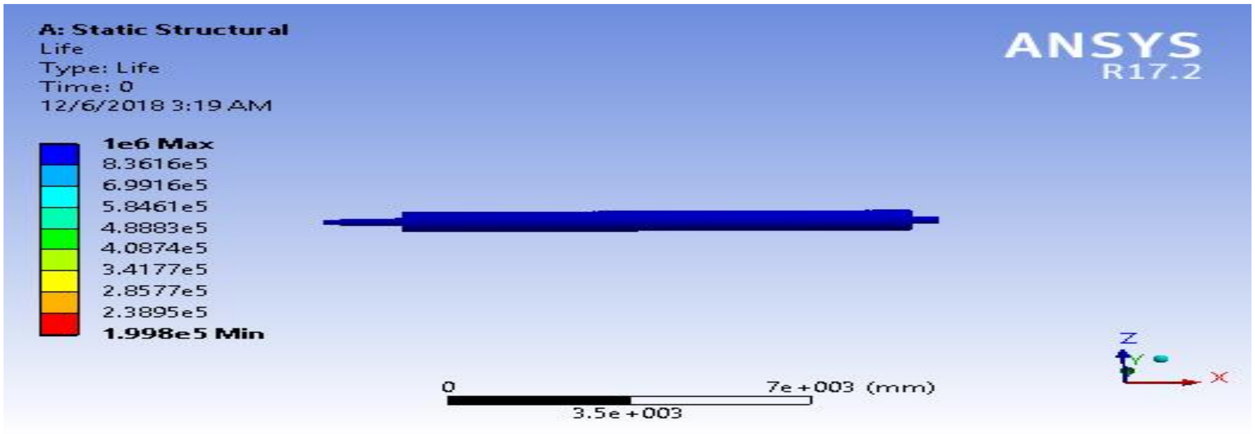

Figure 4.19: fatigue life cycle of hollow feeder table shaft

As it observed from the figure 4.19: Above the fatigue life cycle of the hollow feeder table head shaft is 1.998e5 cycles.

4.1.8.3. Fatigue life cycle of the modified feeder table head shaft (AISI 18Ni 350)

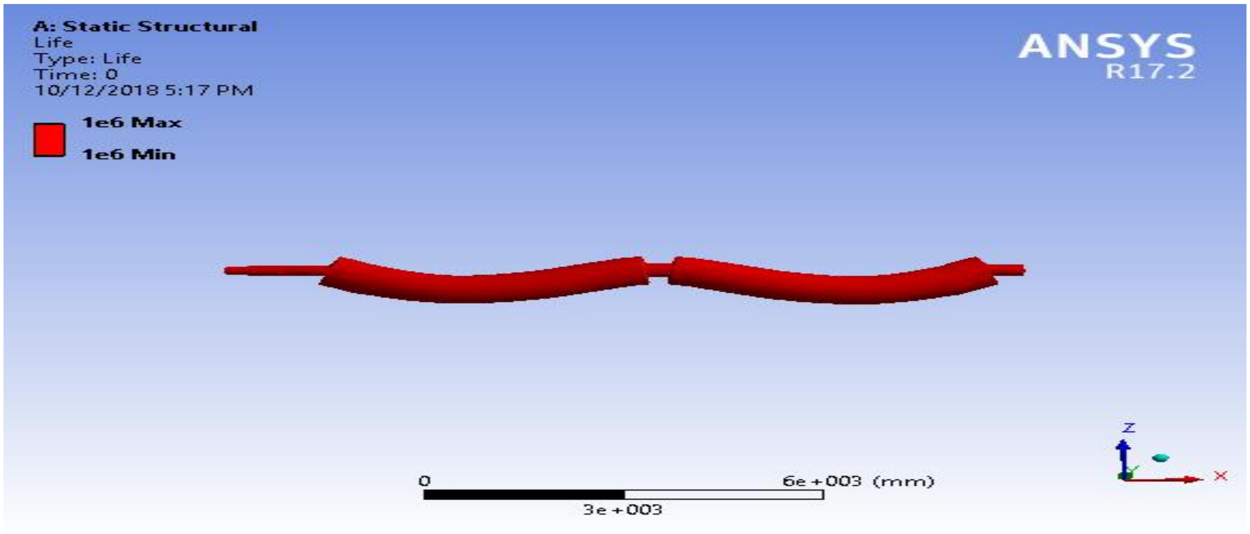

Figure 4.20: fatigue life cycle of modified feeder table shaft

As it observed from the figure 4.18: Above the fatigue life cycle of the new modified feeder table head shaft is 10^6 cycles.

Table 4.8: comparison and discussion of fatigue life and total deformation with given loads

	existing shaft (IS 2062)		New modified shaft (AISI 4340)		Hollow shaft (IS 2062)		New modified shaft (AISI 18Ni 350)	
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
140%	0.4823	2.6999e5	0.4462	10^6	4.3944	1.3821e5	0.4573	10^6
120%	0.4647	3.0669e5	0.4288	10^6	4.3191	1.6541e5	0.4395	10^6
100% (Designed load)	0.4479	3.4907e5	0.4121	10^6	4.2455	1.998e5	0.4224	10^6
75%	0.4282	4.1153e5	0.3923	10^6	4.1573	2.9207e5	0.4022	10^6

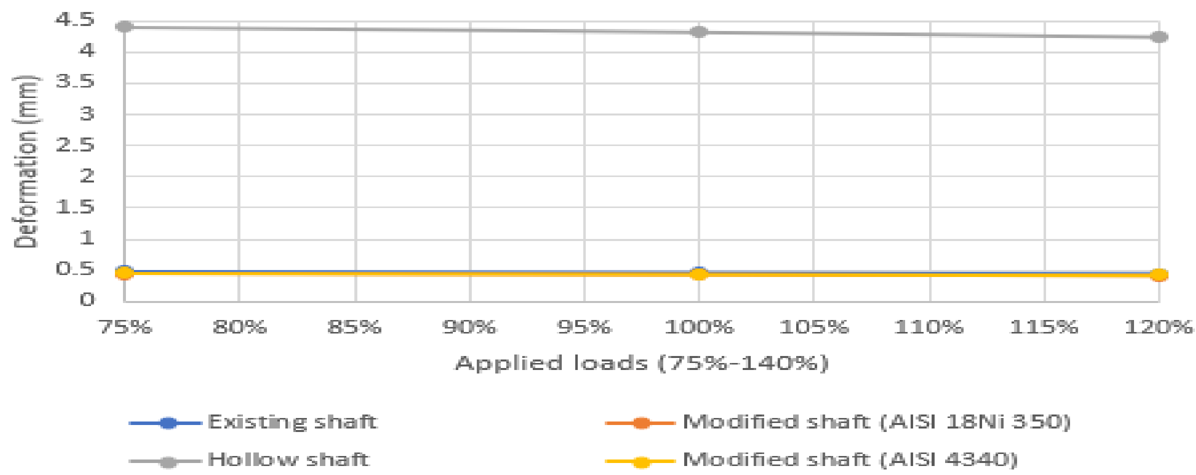

Figure 4.21: deformation comparison of the existing, hollow shafts with the new modified shaft

From the above table 4.8: and figure 4.21: The comparison of the existing, hollow and modified shafts based on the deformation and fatigue life results is very different. The deflection of the existing shaft with the material IS 2062 steel is 0.4479 mm at the designed load applied (100%). The modified new shaft deflection with the material AISI 18Ni 350 steel is 0.4224 mm at 100% applied load. The number of cycles of the modified shaft is 10^6 it has infinite life and the number of cycles of the existing shaft is 3.4907e5 cycles. The deflection of the hollow feeder table head shaft is 4.2455 and the number of cycles is 1.998e5 at 100% load applied.

4.1.9. Factor of safety cycle of the feeder table head shaft

Factor of safety is a term that describes the load carrying capacity of a system beyond the expected loads. And it describes how much stronger the feeder table head shaft that resists to the applied load. For most materials, it is often required that the factor of safety be checked against both yield and ultimate strengths. The yield calculation to determine the safety factor until the part starts to plastically deform and the ultimate calculation to determine the safety factor until failure.

Fatigue factor of safety is a contour plot of the factor of safety with respect to a fatigue failure of a given design life. For fatigue factor of safety values less than one indicate failure before the design life is reached. The factor of safety for high cycle fatigue should be more than one to resist the stress applied to the shaft. (factor of safety > 1 design is safe) [44].

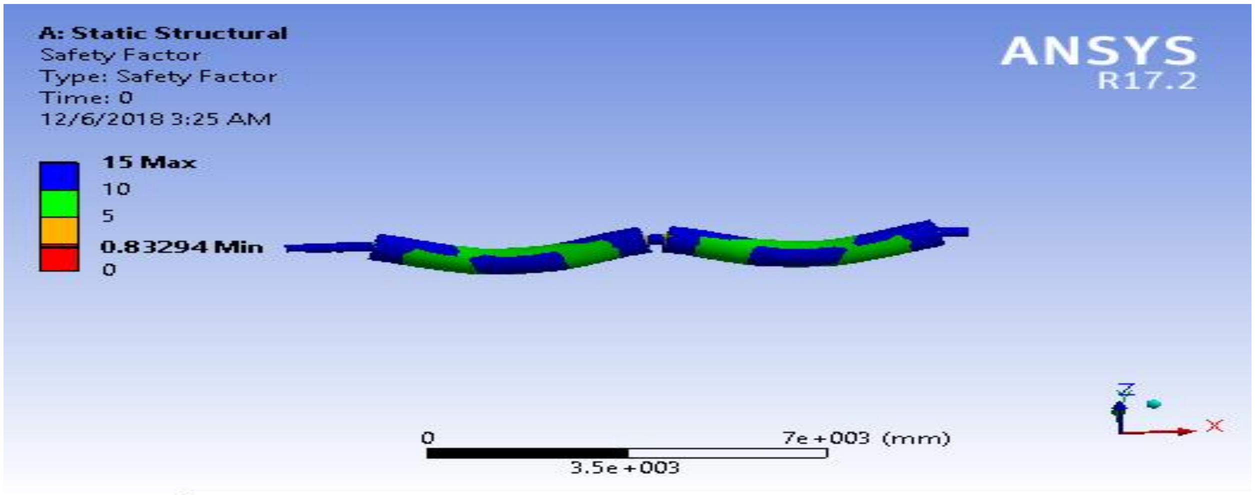

Figure 4.22: Factor of safety of existing feeder table head shaft

As shown from the figure 4.22: Above the fatigue factor of safety due to its cycles of the existing feeder table head shaft is 0.83294 and the factor of safety due to the applied load is 2.3191.

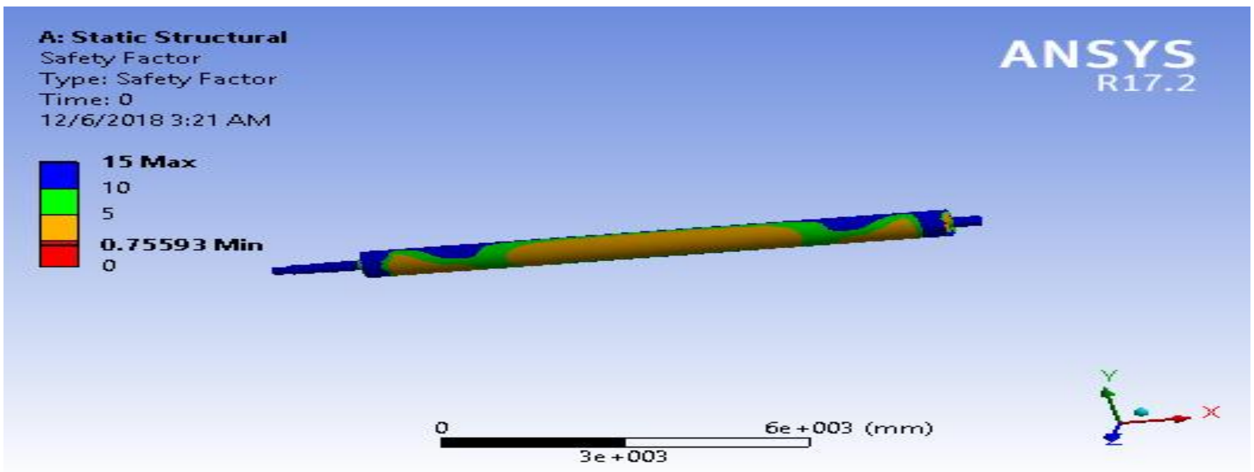

Figure 4.23: Factor of safety of hollow feeder table head shaft

From the figure 4.23: Above the fatigue factor of safety due to its cycles of the existing feeder table head shaft is 0.75593 and the factor of safety due to the applied load is 2.1047. The portion in red color shows that the maximized stress. Since the stress in the feeder table head shaft is not the same throughout.

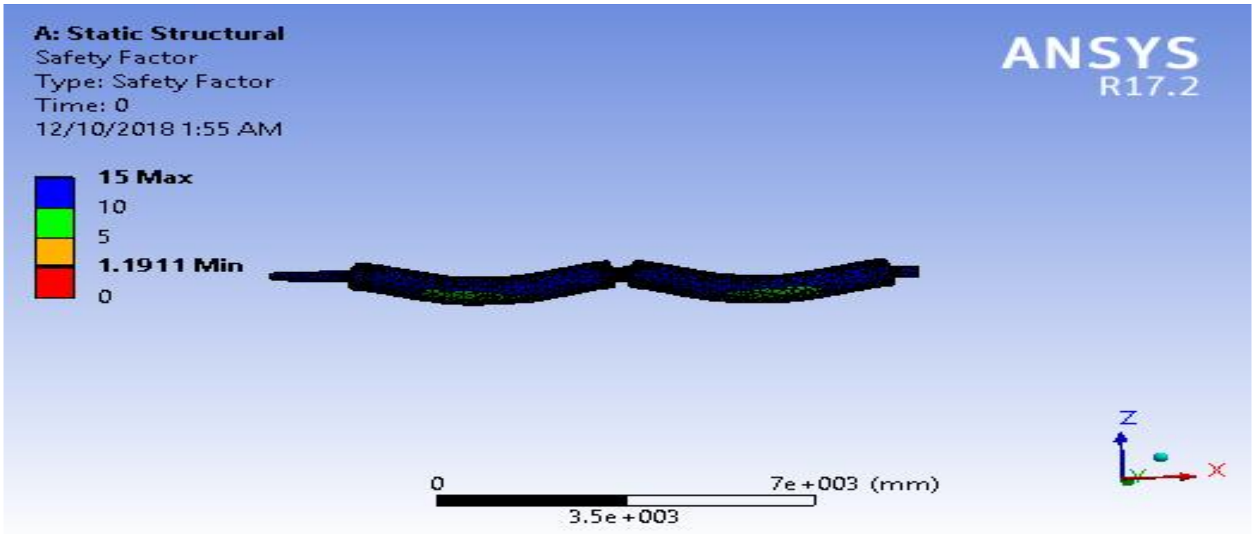

Figure 4.24: factor of safety of modified feeder table shaft

From the figure 4.24: Above the fatigue factor of safety due to its cycles of the existing feeder table head shaft is 1.1911 and the factor of safety due to the applied load is 4.8552.

Table 4.9: Comparison and discussion of fatigue safety factor with different applied loads of all models of the feeder table head shaft

Applied load	Fatigue Safety factor of existing shaft (IS 2062)	Fatigue Safety factor of Modified shaft (AISI 4340)	Fatigue Safety factor of Modified shaft (AISI 18Ni 350)	Fatigue Safety factor of hollow shaft (IS 2062)
75%	0.8571	1.2123	1.2278	0.80754
100%	0.83294	1.1754	1.1911	0.75593
120%	0.81442	1.1469	1.1627	0.71758
140%	0.79659	1.1194	1.1352	0.68292

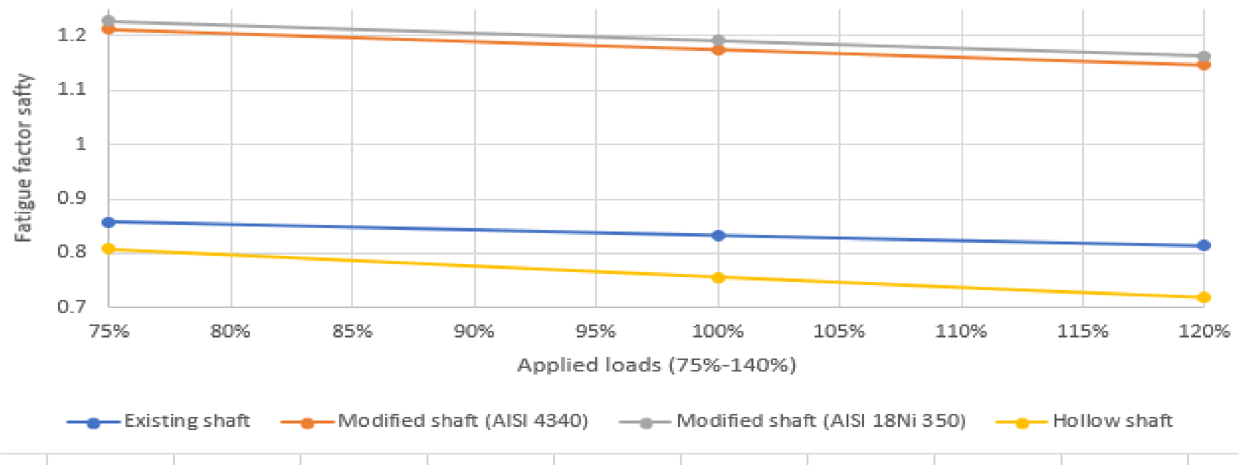

Figure 4.25: comparison of fatigue factor of safety with applied different loads

As shown from the figure 4.25: The result of the fatigue factor of safety of the existing and hollow feeder table head shafts are failed because the factor of safety is less than one. The fatigue factor of safety at 100% load applied to the existing shaft with the material IS 2062 steel is 0.83294 and the fatigue factor of safety of the hollow shaft at 100% applied load is 0.75593. But the fatigue factor of safety of the modified feeder table head shaft with the material of AISI 18Ni 350 steel at 100% load applied is 1.1911. It is greater than one, so the shaft is safe.

4.1.10. Result and discussion of fatigue sensitivity of the feeder table head shaft

Fatigue Sensitivity, shows how the fatigue results change as a function of the loading at the critical location on the model of the feeder table head shaft. Sensitivity may be found in life, damage, or factor of safety. It can estimate the fatigue sensitivity of this study used the load varying from 50% to 150% of the cur1rent load is applied. A value of 100% corresponds to the life at the current load on the feeder table head shaft.

Table 4.10: Results of fatigue sensitivity with varying loads (50%-150%)

loades	life of hollow shaft	life of existing shaft	life of modified shaft AISI 18Ni 350)	life of modified shaft (AISI 4340)
0.5	1.00E+06	1.00E+06	1.00E+06	1.00E+06
0.542	1.00E+06	1.00E+06	1.00E+06	1.00E+06
0.625	1.00E+06	1.00E+06	1.00E+06	1.00E+06
0.667	1.00E+06	1.00E+06	1.00E+06	1.00E+06
0.708	1.00E+06	1.00E+06	1.00E+06	1.00E+06
0.75	1.00E+06	1.00E+06	1.00E+06	1.00E+06
0.792	7.66E+05	1.00E+06	1.00E+06	1.00E+06
0.875	4.31E+05	7.53E+05	1.00E+06	1.00E+06
0.917	3.30E+05	5.76E+05	1.00E+06	1.00E+06
0.958	2.55E+05	4.46E+05	1.00E+06	1.00E+06
1	2.00E+05	3.49E+05	1.00E+06	1.00E+06
1.042	1.72E+05	2.76E+05	1.00E+06	1.00E+06
1.083	1.49E+05	2.20E+05	1.00E+06	1.00E+06
1.125	1.30E+05	1.85E+05	7.57E+05	7.02E+05
1.208	1.01E+05	1.43E+05	6.27E+05	5.81E+05
1.25	88805	1.26E+05	5.22E+05	4.84E+05
1.292	78740	1.12E+05	4.37E+05	4.05E+05
1.417	56108	80090	3.68E+05	3.41E+05
1.458	50448	72011	3.12E+05	2.89E+05
1.5	45495	64940	2.65E+05	2.46E+05

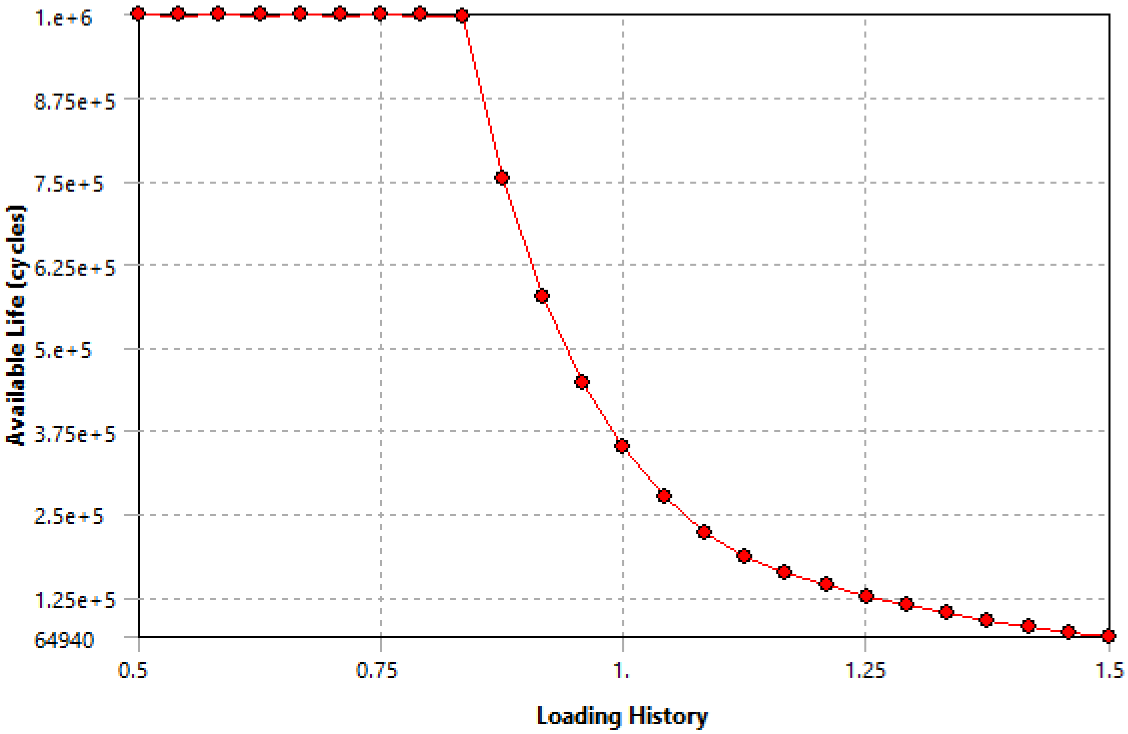

Figure 4.26: fatigue sensitivity of the existing feeder table head shaft

From the figure 4.26: The fatigue sensitivity of the existing feeder table head shaft at 1 or 100% load applied the fatigue life cycle is 3.49e5. That means it has a finite life. And at 1.5 or 150% the fatigue sensitivity is 64940.

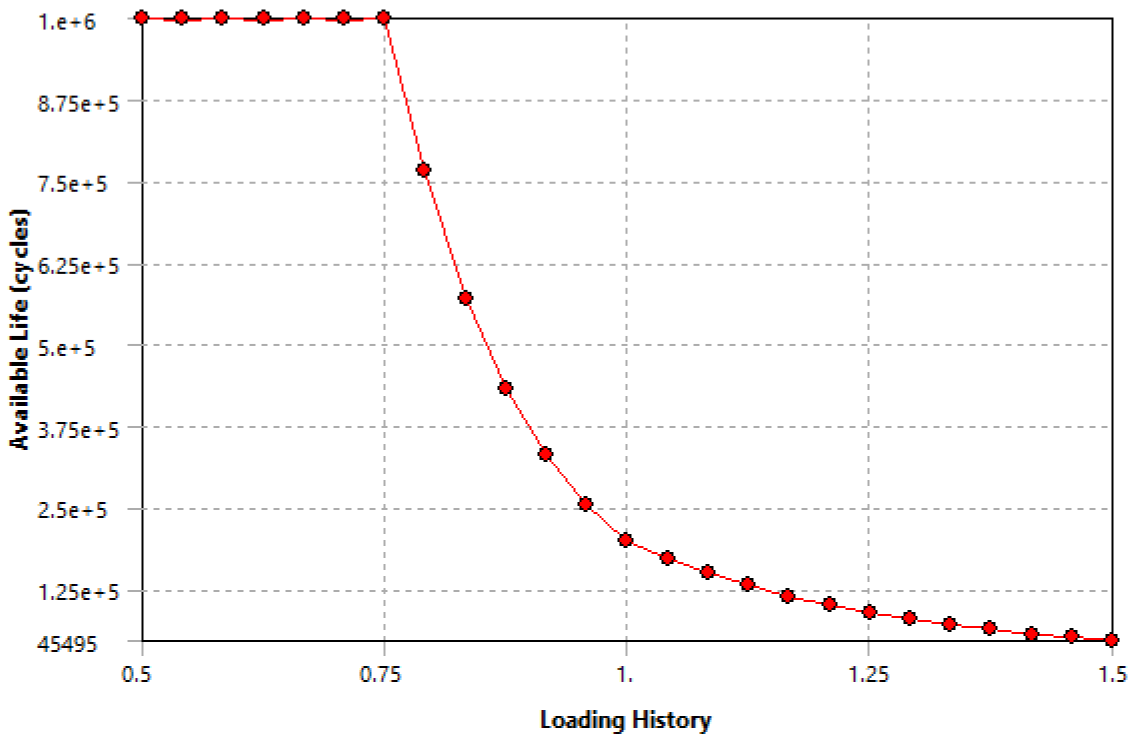

Figure 4.27: fatigue sensitivity of the hollow feeder table head shaft

From the figure 4.27: Shown that the fatigue sensitivity of the hollow feeder table head shaft at 1 or 100% load applied the fatigue life cycle is 1.99e5. That means it has a finite life. And at 1.5 or 150% the fatigue sensitivity is 45495.

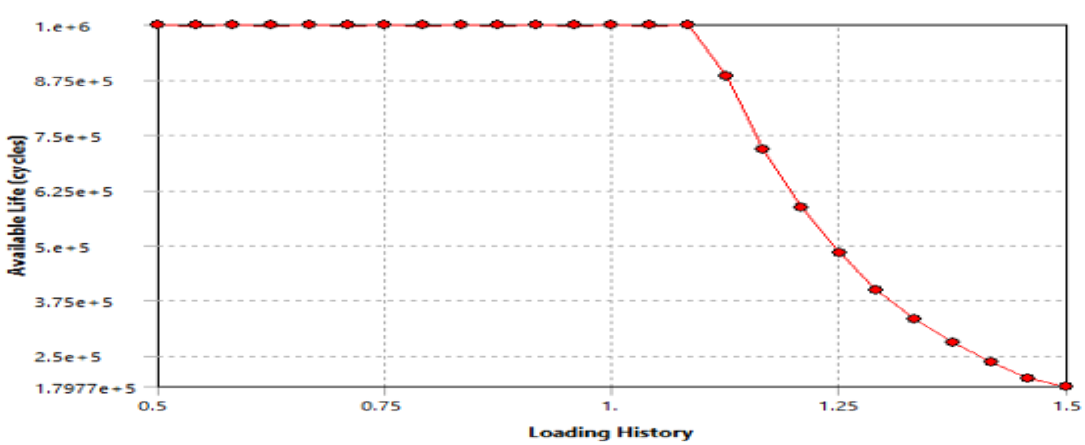

Figure 4.28: fatigue sensitivity of the modified feeder table head shaft

From the figure 4.28: Shown the fatigue sensitivity of the modified feeder table head shaft with the material AISI 4340 steel at 1 or 100% load applied the fatigue life cycle is 10^6. That means it has infinite life. And at 1.5 or 150% the fatigue sensitivity is 2.46e5.

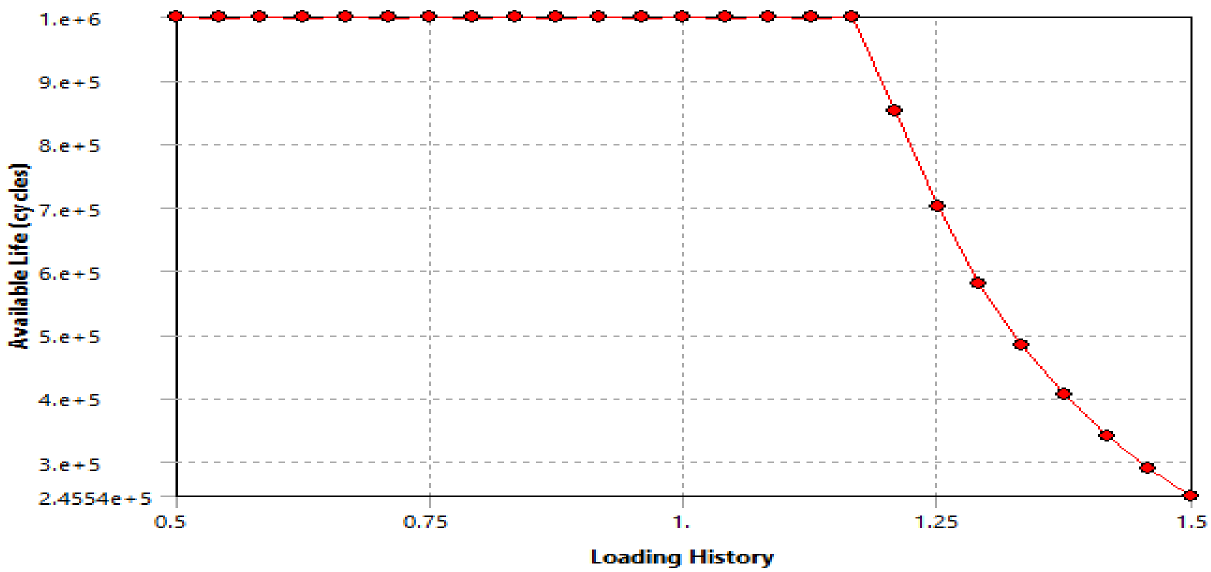

Figure 4.29: fatigue sensitivity of the modified feeder table head shaft

From the figure 4.29: Shown that the fatigue sensitivity of the modified feeder table head shaft with the material at 1 or 100% load applied the fatigue life cycle is 10^6. That means it has a finite life. And at 1.5 or 150% the fatigue sensitivity is 2.65e5.

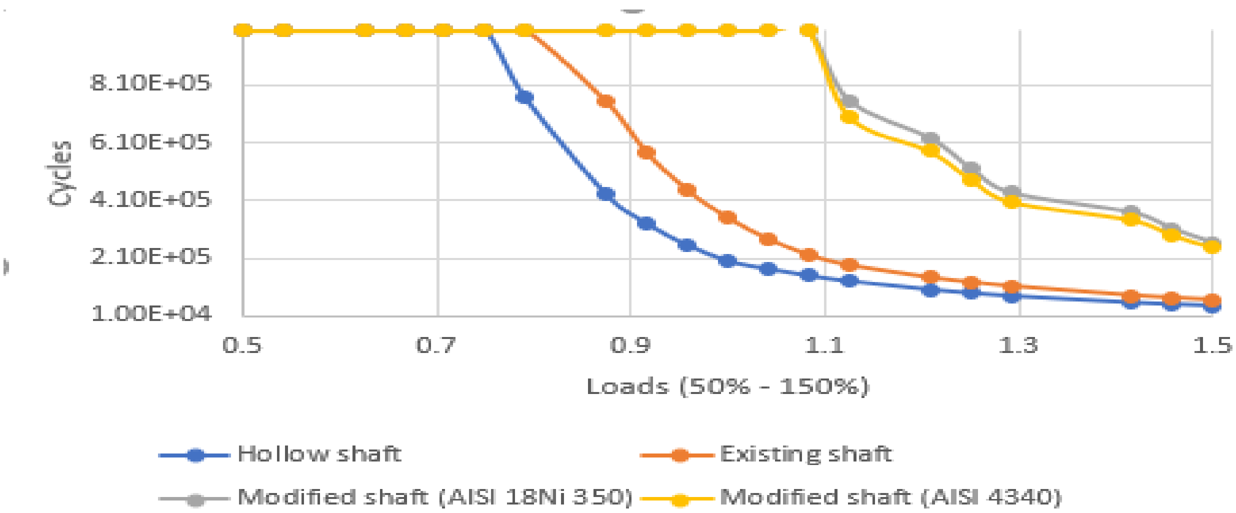

Figure 4.30: fatigue life cycle comparison of the existing, hollow shaft with the new modified shaft

The figure 4.30: Above show that, the fatigue sensitivity of models of existing, hollow and modified feeder table head shaft. The loads applied to the shaft from 50% - 150%. The existing feeder table head shaft with the material IS 2062 steel at 100% load applied the life cycle is 3.49e5, the hollow feeder table head shaft at 100% applied load the life cycle is 1.99e5 and the new modified feeder table head shaft with both AISI 4340 and AISI 18Ni 350 steel has 10^6 cycles at 100% applied load. That means if the applied load on the feeder table head shaft is less than the designed load, the feeder table head shaft has not failed before 10^6 cycles or it has infinite life. So, the fatigue life of the new modified feeder table head shaft is better than the existing and hollow feeder table head shaft. Its fatigue life cycle is infinite life (10^6) So it is safe and has a good life time.

CHAPTER 5

5. Conclusion, Recommendation and Future Works

5.1. Conclusion

The static structural and life prediction is performed based on finite element analysis using ANSYS 17.2 software. The mechanical design and material selection of the feeder table head shaft was not appropriate for its intended service. The existing feeder table head shaft failed due to stress concentration at the intermediate shoulder solid shaft where the diameter of the shaft was small to carry the loads. And the existing and hollow shafts are failed due to fatigue. It avoided its failure by modifying of the feeder table head shaft with increase its diameter and change the material. The total deformation, total equivalent strain, maximum von-misses stress and maximum shear stress of the modified feeder table head shaft is smaller than existing and hollow shafts. The fatigue factor of safety and fatigue life cycle of the modified feeder table head shaft is better than the existing and hollow feeder table head shafts. Therefore, the modified feeder table head shaft with the material AISI 18Ni 350 steel is safe and it has a good life time.

5.2. Recommendation and Future works

In this thesis the feeder table head shaft is studied with different models and with different materials using finite element method by ANSYS software. To avoid the failure of the feeder table head shaft it is recommended to modified the shaft by changing its diameter from 300 mm to 400 mm and by changing the material from IS 2062 carbon steel to AISI 18Ni 350 steel and it needs to study other source of failures. There are no titles about feeder table head shaft that has been published. So, it needs to study more papers about the feeder table head shaft to operate the feeder table head shaft safely for a long period of time without failure. And it needs more research by considering its vibration and dynamics effect of the feeder table head shaft.

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Appendix I

The result of maximum Von-Misses stress, maximum shear stress, total deformation, fatigue life cycle, factor of safety and total equivalent strain are done by using ANSYS workbench 17.2 for all models of the feeder table head shaft.

Appendix A

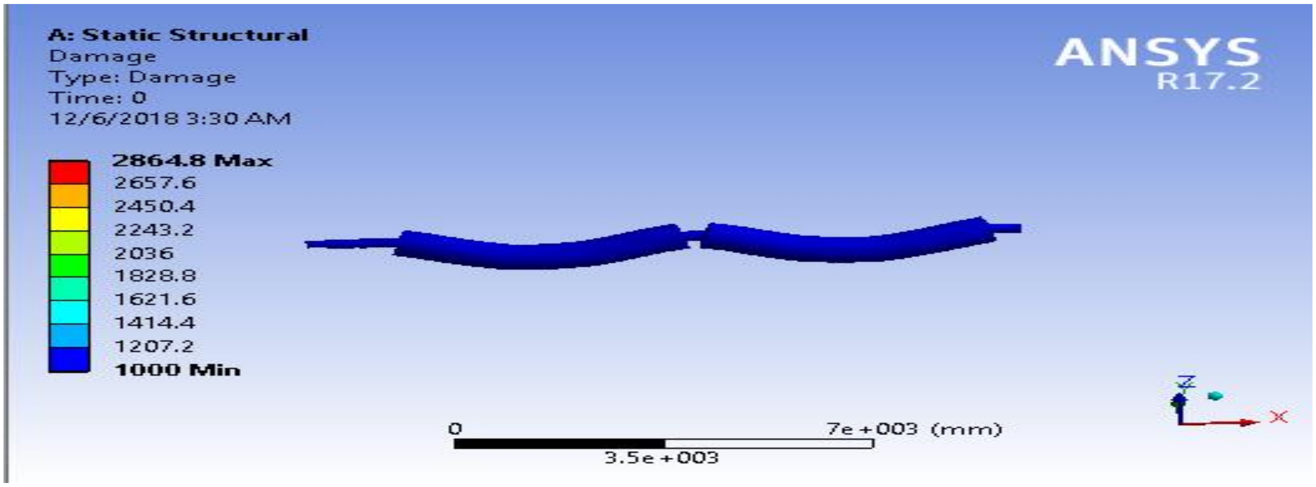

Figure A1: Damage of the existing feeder table head shaft

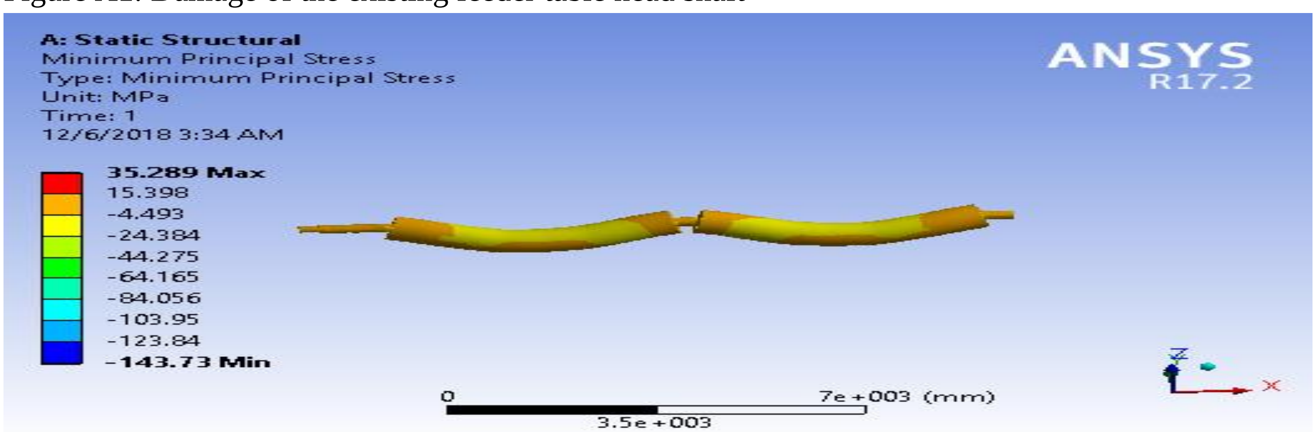

Figure A2: Minimum principal stress of the existing feeder table head shaft

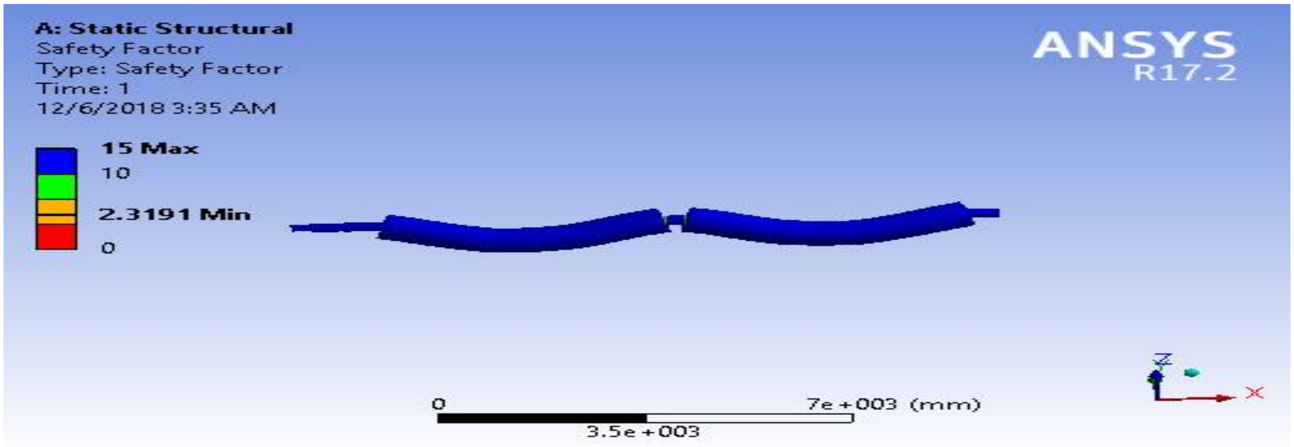

Figure A3: Safety factor in yield of the existing feeder table head shaft

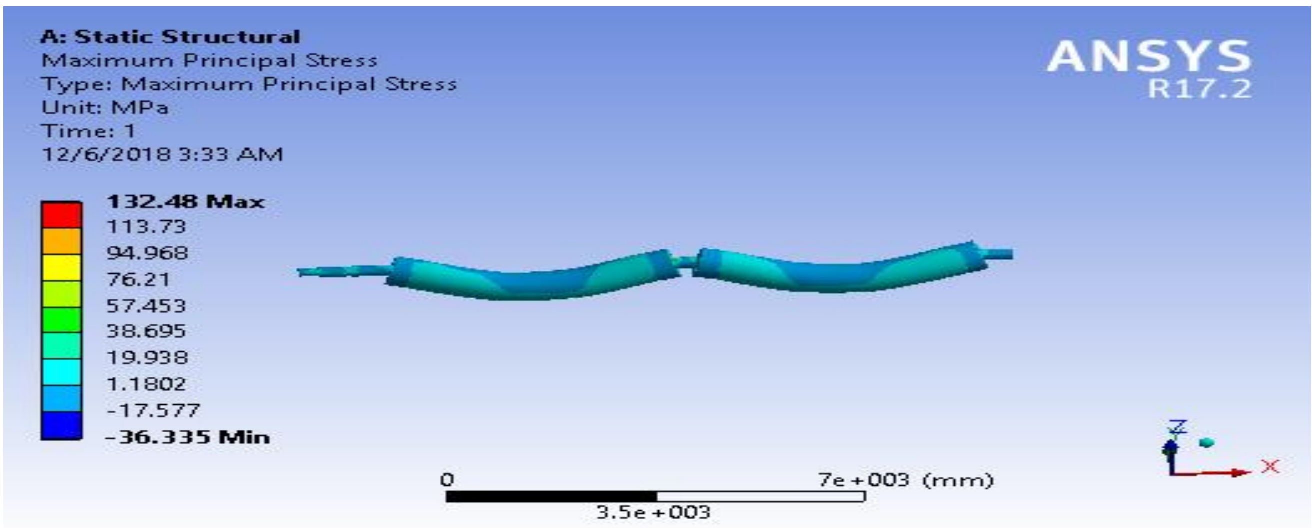

Figure A4: Maximum principle stress of existing of the existing feeder table head shaft

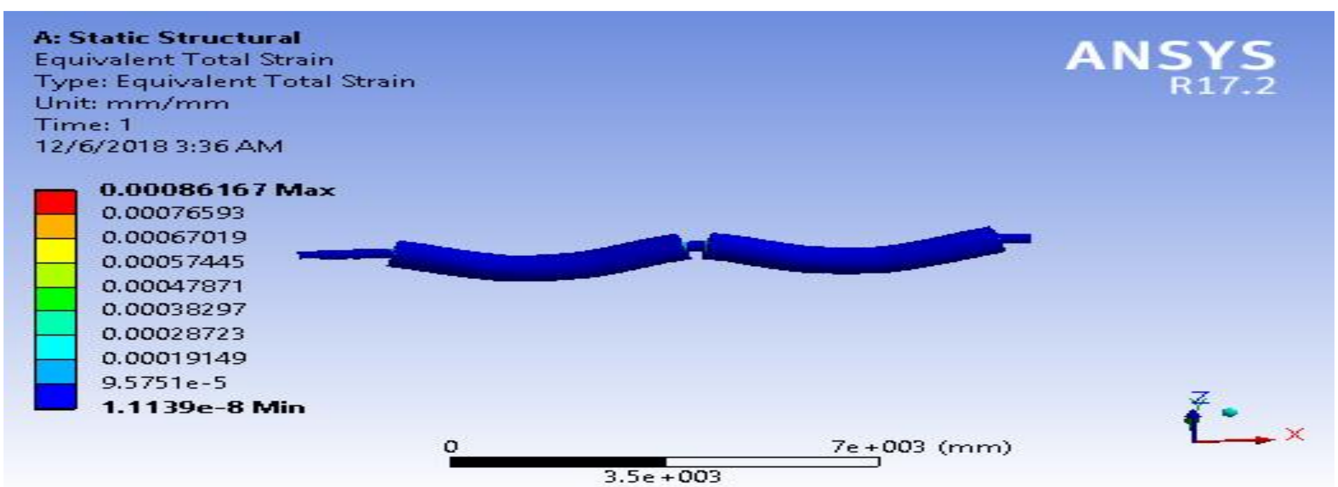

Figure A5: equivalent total strain of existing feeder table head shaft

Appendix B

Static structural result of hollow feeder table head shaft

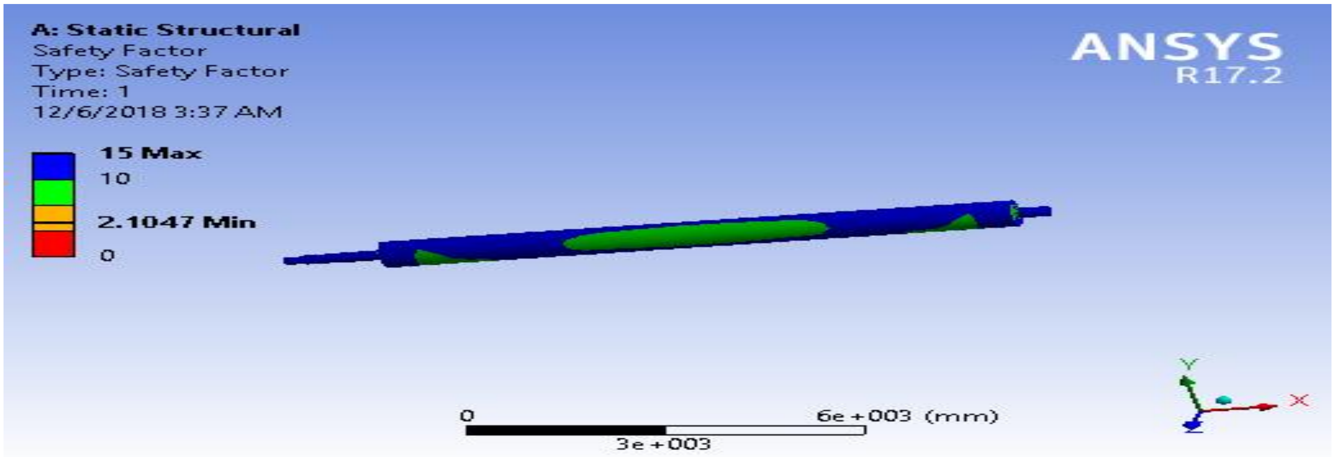

Figure B1: Safety factor in yield of hollow feeder table head shaft

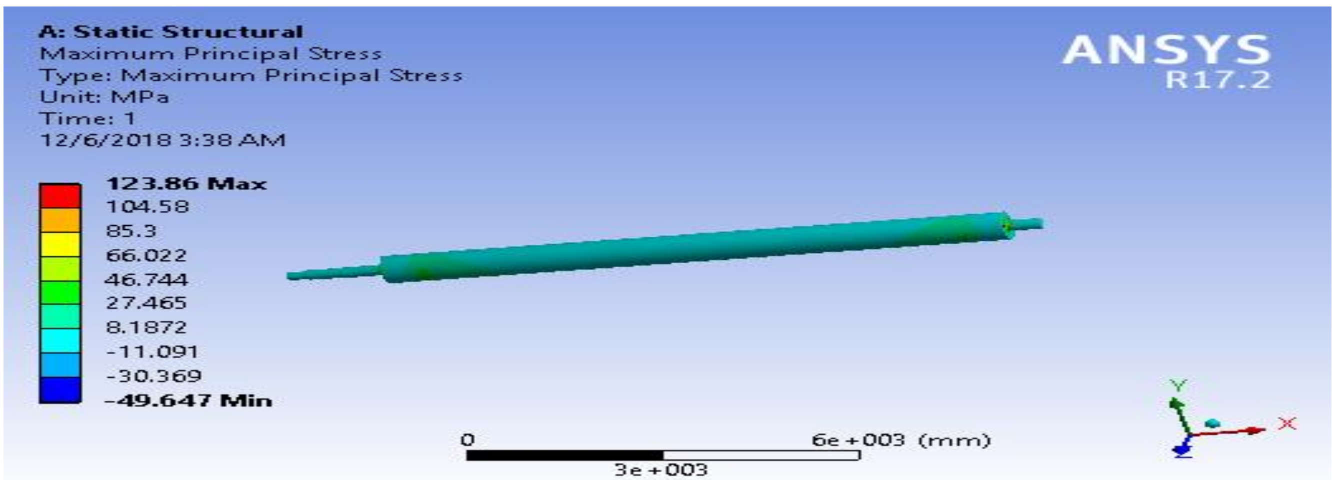

Figure B2: Maximum principal stress of the hollow feeder table head shaft

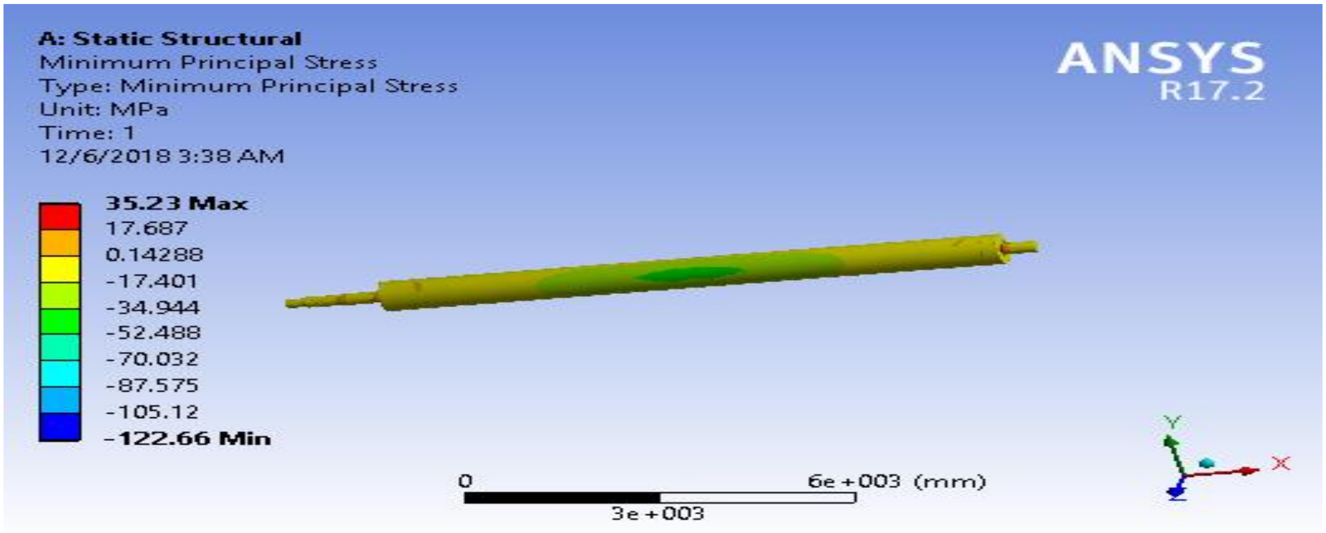

Figure B3: Minimum principal stress of the hollow feeder table head shaft

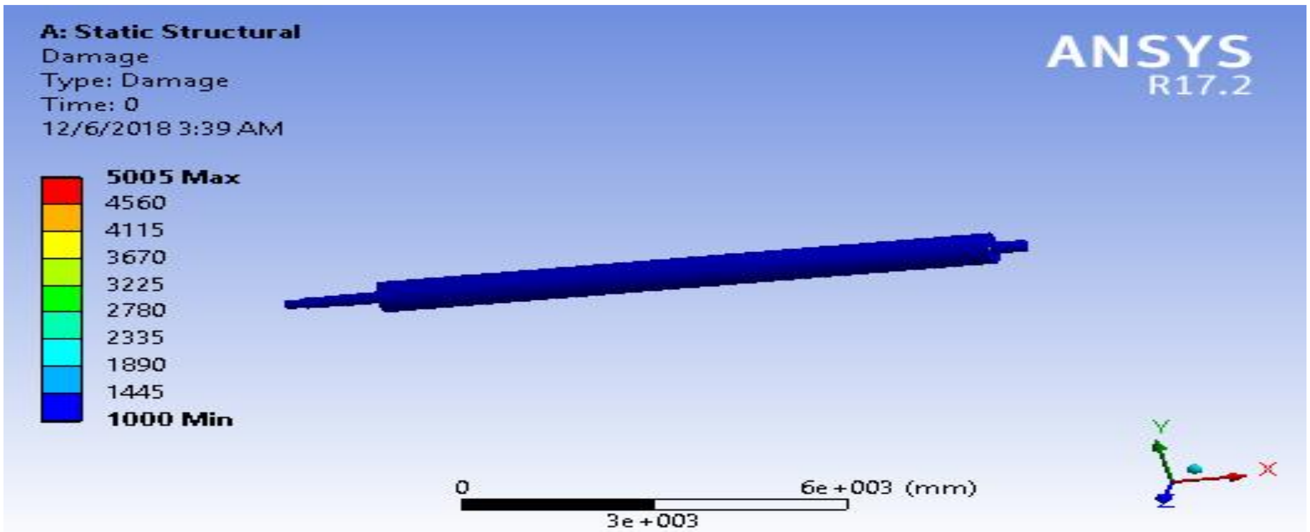

Figure B4: Damage of the hollow feeder table head shaft

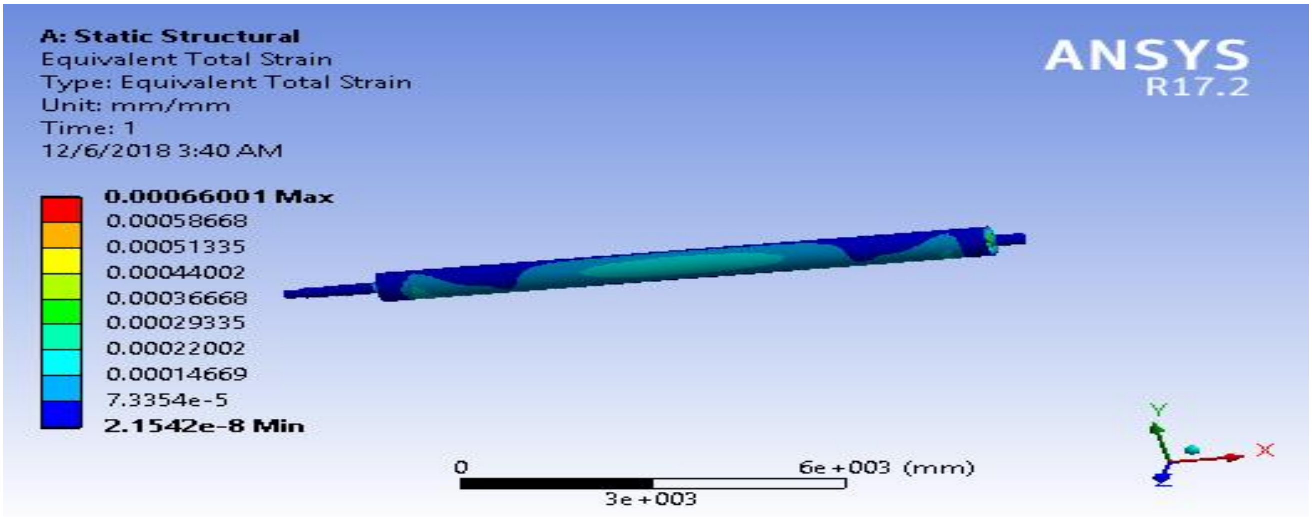

Figure B5: equivalent total strain of the hollow feeder table head shaft

Appendix C

Static structural result of the modified shaft

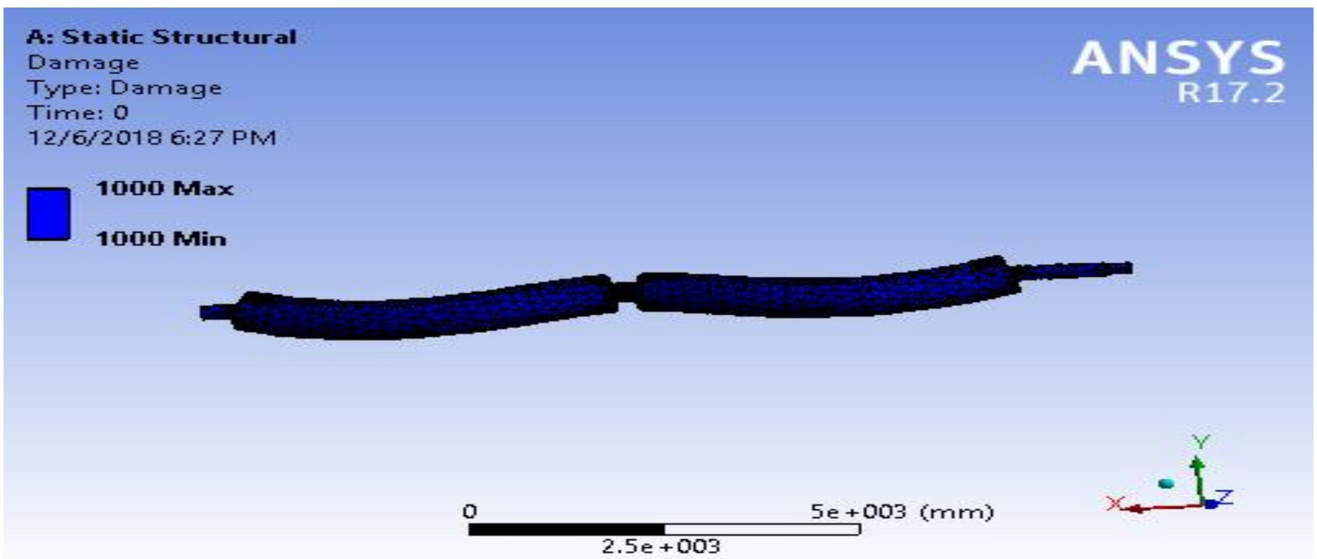

Figure C1: Damage of the modified feeder table head shaft

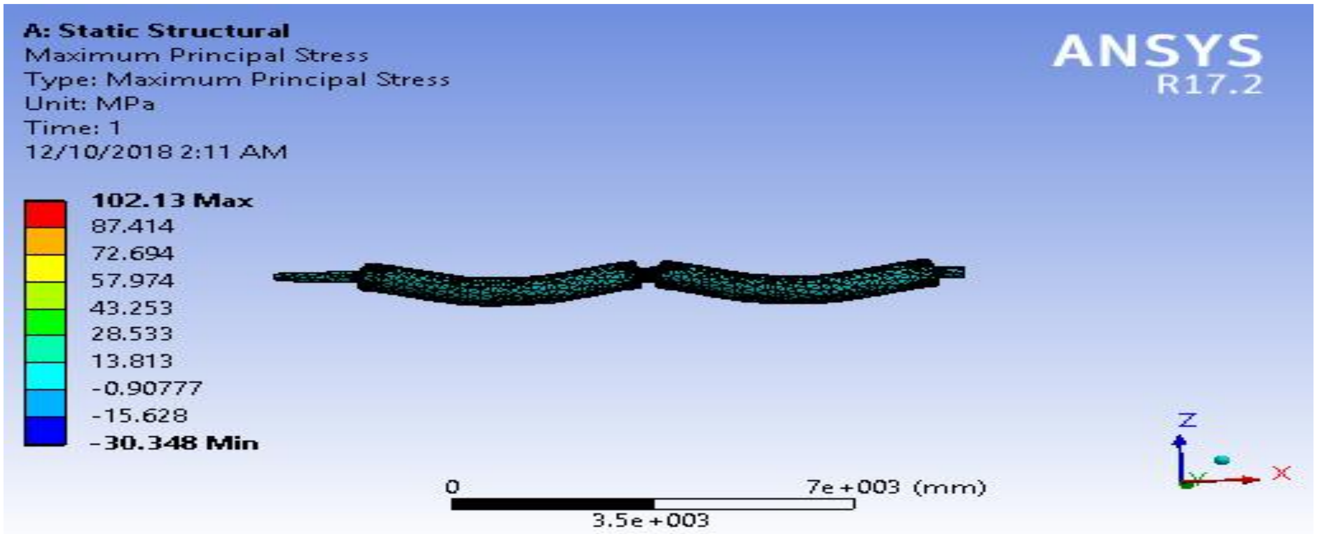

Figure C2: Maximum principal stress of the modified feeder table head shaft

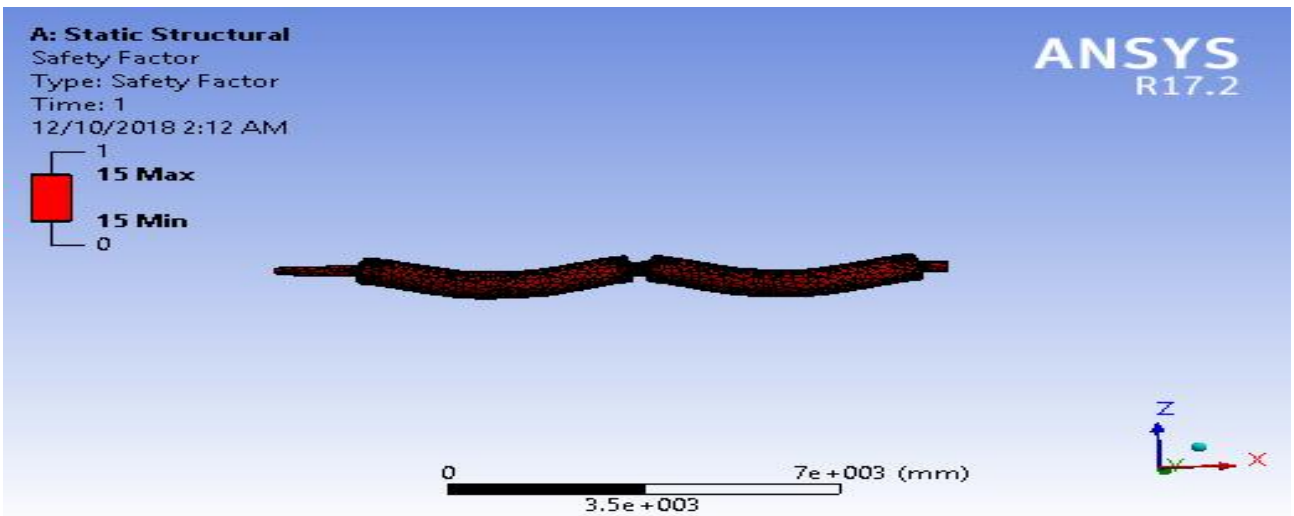

Figure C3: Safety of factor in yield of the modified feeder table head shaft

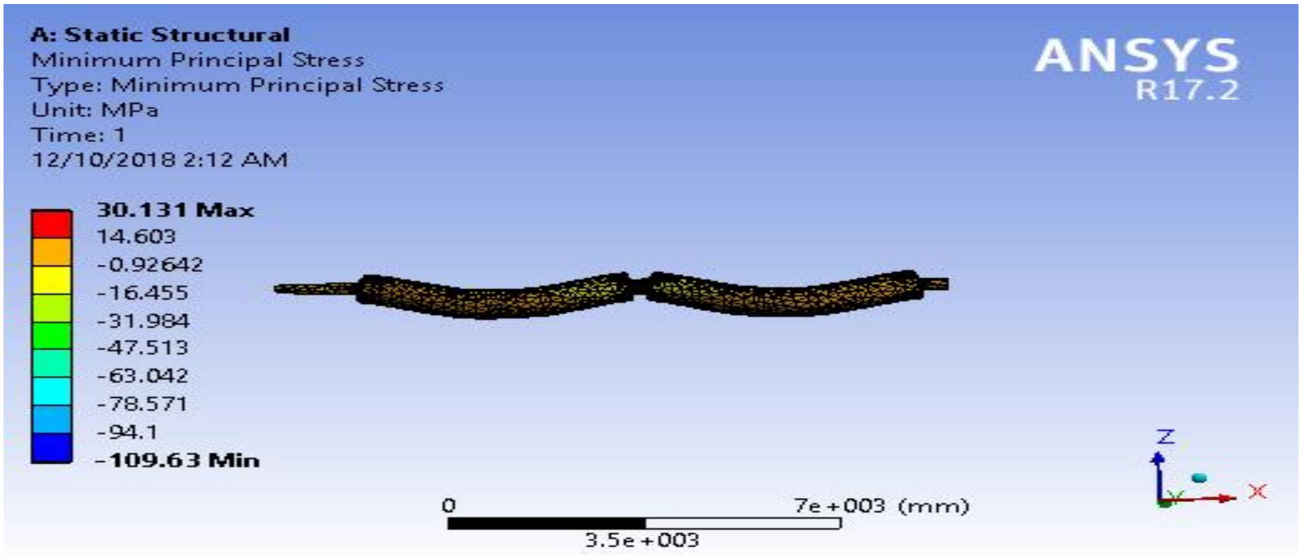

Figure C4: Minimum principal stress of the modified feeder table head shaft

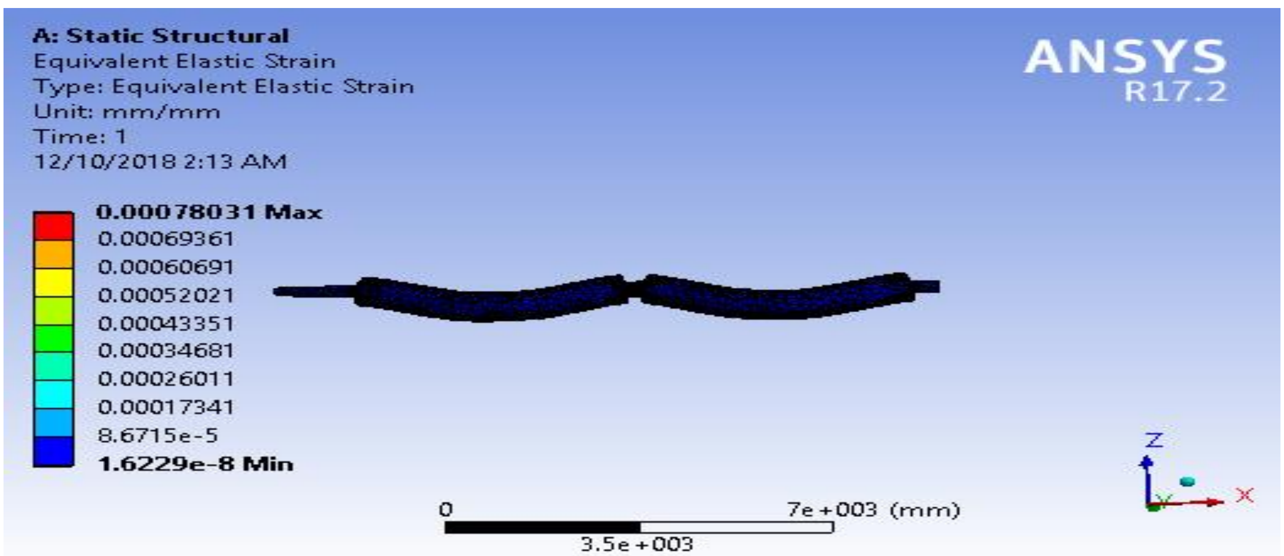

Figure C5: Equivalent total strain of modified feeder table head shaft

Appendix II

Input data and dimension of the feeder table head shaft from wonji-shoa sugar factory.

Table 1: The input data and dimensions of the feeder table head shaft

Overall length of the shaft = 11.877 mm
Outer diameter of the hollow shaft = 790 mm
Inner diameter of the hollow shaft = 750 mm
Diameter of the intermediate solid shaft = 300 mm
Degree of inclination of the conveyor feeder table = 16.5^0
Capacity of the cane feeder table per day = 6250 TCD
Drive motor = 55 Kw
Motor speed = 1500 RPM
Reduction gear box = 142:1
Open gear = 120/22 teeth*18 module*375 face width
Sprocket = 9 teeth with circular pitch 2*/153.18(882.12 p.c.d)
Chain = 153.18 pitch
Number of strands =12 Nos
Speed of the table = 5m/min
Length of the table = 28.88 m