ADDIS ABABA INSTITUTE OF TECHNOLOGY (AAiT)

SCHOOL OF MECHANICAL AND INDUSTRIAL ENGINEERING

Parametric Studies on the Fatigue Life Improvement of Passenger Rail Vehicle Axlebox House

A Thesis Submitted to the School of Graduate Studies of Addis Ababa University in Partial Fulfillment of the Requirements for the Degree of Masters of Science in Rolling Stock Engineering (Mechanical Engineering)

Submitted By:

Alem Behaylu

Advisor:

Tollossa Deberie(M.Sc.)

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Declaration

I declare that this work is my own and all sources of materials used for this thesis work have been duly acknowledged. I truly declare that this thesis work is not submitted to any other institution anywhere for the award of any academic degree, diploma, or certificate. With accurate acknowledgment of the source, users are free to use this thesis without special permission.

Name: Alem Behaylu Signature: _________ Date: _________

As thesis research advisor, I hereby approve that I have read and evaluated this thesis prepared, under my guidance, by Alem Behaylu, entitled **"Parametric Studies on the Fatigue Life Improvement of Passenger Rail Vehicle Axlebox House".** I recommend it be submitted as fulfilling the thesis requirement.

Tollossa Deberie	__________________	___________
Name of advisor	Signature	Date

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Tollossa Deberie	____________________	______________
Advisor	Signature	Date
Mr. Fasil Gesessew	____________________	______________
Internal Examiner	Signature	Date
Mr.Haileleul Sahile	____________________	______________
External Examiner	Signature	Date
Dr.Birhanu Beshah	____________________	______________
Railway Head Center	Signature	Date

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Abstract

Failure of passenger rail axlebox house is mainly due to fatigue caused by repeated cyclic loading which cause catastrophic mechanical failure on axleboxes house and bearing assemblies. The general objective of this thesis is to improve the fatigue life of passenger rail vehicle axlebox house by studding influential geometry and fatigue strength factor parameters on the fatigue life of the axlebox house. Primary data such as bogie type and application, axle load, axle journal size, bearing type and size are selected and analysed for modelling the geometry. In addition to primary data, secondary data such as material data, boundary condition, loading data are required for the input of ANSYSI software analysis. Four geometry types are modelled with SolidWorks based on the axlebox house design principles then all models are analysed with FEM ANSYS Workbench 14 fatigue tools under the same non-proportional loading, equivalent dimension and the same material property to investigate which model has a longer fatigue life. This thesis shows by improving the fatigue strength factor can improve the life up to 50% and suitable axlebox house geometry selection can improve the fatigue life up to 70% and it is found that Model II and Model IV have better fatigue life other than other models. Improving the fatigue strength factor and geometry parameters such us geometry model, notches and groves improve the fatigue life of the axlebox house and as the result early failure, operation and maintenance cost, time and risk to catastrophic accident will be minimized. The thesis also recommends that the mass of the heavier models could be farther improved and advanced research on site to be conducted in consideration to environmental factor, microstructure, manufacturing processes, thermal effects, etc since those factors do also have an impact on the fatigue life of the axlebox house.

Keywords: Fatigue; Non-proportional loading; Fatigue strength factor; Microstructure; Thermal effect.

Table of Contents

Contents **Page**

Lists of Figure

Lists of Tables

Nomenclature

σ_a	Stress amplitude
σ_m	Mid range or mean stress
σ_{max}	Maximum stress
σ_{min}	Minimum stress
σ_r	Stress range
σ_s	Steady, or static stress
a_{xc}	Emergency braking rate produced during starting
C	Basic dynamic load rating, kN
c	Point distance from the neutral axis
D	Axle Designation
D_w	Mean wheel diameter, m
f_{ad}	Dynamic axial factor
f_o	Payload factor
F_r	The radial bearing load is given by:
f_{rd}	Dynamic radial factor
f_{tr}	Dynamic traction factor
F_{xc}	Longitudinal force
F_{yc}	Transverse force
F_{zc}	Vertical force
G	Maximum static axlebox load [kN]
G_{00}	Assumed Static axle load [kN]
G_r	Weight of the wheelset [kN]
I	Moment of inertia
J	Polar moment of inertia
k_a	Surface factor
k_b	Size factor
k_b	Size factor for bending
k_c	Load factor
$k_{c,a}$	Load factor for axial
k_d	Temperature factor
k_e	Miscellaneous-effects factor
K_f	Fatigue stress concentration factor
$K_{f\,t}$	Fatigue strength factor for torsion
$K_{f,a}$	Fatigue strength factor for axial
$K_{f,b}$	Fatigue strength factor for bending
K_r	Mean radial load [kN]

l	Distance between the 2 load centres, mm
L_{10}	Basic rating life (at 90% reliability), millions of revolutions
m+	Bogie mass
Mv	Mass of car in running order;
n_b	Number of bogies per body
p	Exponent for the life equation
P	Equivalent dynamic bearing load, kN
P_1	Exceptional payload
P_2	Service (Fatigue) payload
Q	First moment of area
Se	Endurance limit of mechanical element (to be designed)
Se*	Endurance limit of test specimen.
Su	Ultimate strength
Sy	Yield strength, ,
T	The torque
t	Thickness/width
Y	Axial load bearing factor

Chapter One

1. Introduction

1.1. Background of the Project

Since the earliest use in railway applications, axlebox seats including axlebox housing have offered energy and lubrication saving opportunities [35]. Today, most railway vehicles are equipped with increasing advanced designs based on wheelset axlebox assemblies comprising the wheelset bearings or bearing units, the axlebox housing and integrated sensors [35]. Speed has been the essence of railways since the first steam locomotive made its appearance in 1804[35].

Railway axlebox housings are manly cast in Spheroidal Graphite Iron which has a malleable metallic structure with elastic properties able to absorb shock loading over a wide range of operating load and temperatures[18],[19],[33],[34]. The axlebox house is the device that allows the wheelset to rotate by providing the bearing housing and also the mountings for the primary suspension to attach the wheelset to the bogie or vehicle frame. The axlebox house transmits longitudinal, lateral, and vertical forces from the wheelset on to the other bogie elements [6].

Passenger rail vehicle axleboxe house have always been a vital component in the reliability of railway rolling stock and has considerable influence on the operating safety, reliability and economics of railways.

Up to 90% of all structural failure occurs through a fatigue mechanism costs a lot of money and loss life [12]. Failure of axlebox house is mainly due to fatigue caused by repeated cyclic loading. Fatigue analysis procedures for the design of modern structures rely on techniques which have been developed over the last 100 years or so. The first accepted technique was the S-N or stress-life method generally given credit to the German August Woehler for his systematic tests done on railway axles in the 1870's [24].

In this research, the main objective is to improve the fatigue life on the axlebox house of passenger rail vehicle imposed to variable combined load by studding geometry and fatigue strength factor parameters. As the result the axlebox house withstand variable dominating stress and adverse conditions frequently encountered during use. Because the axlebox house is required to withstand high fatigue, variable loads and thermal load with accurate dimensional clearance to safe guard the rail vehicle from catastrophic accident, it has to be designed properly and the fatigue life has to be checked.

The methodology approaches used for this research will discuss the results generated from FEM analysis in scientific manner starting from geometry selection and modelling.

This thesis indicates those geometry and fatigue strength factors are the dominant parameters on the fatigue life the axlebox house. By improving the fatigue strength factor can improve the life up to 50% and suitable geometry selection can improve the fatigue life up to 70% .Those and additional parameters will characterizes the capability of axlebox house material to survive in different cyclic load during its lifetime.

1.2 The Objective of the Research

1.2.1 General Objective

The general objective of this thesis is to improve the fatigue life of passenger rail vehicle axlebox house by studding the geometry and fatigue strength factor parameters with Finite Element Method (FEM) analysis for prolonged, safe and smooth running during its lifetime.

1.2.2 Specific Objective

The specific objective of this thesis will have the following outcomes:-

- To achieve and select appropriate fatigue load resistance passenger rail vehicle axlebox house geometry.
- Reveals the impacts of geometry and fatigue strength factors dominate parameters on fatigue life on the selected or obtained geometry.
- Indicates some of the importance of adopting the result to Ethiopian Railway Corporation (ERC) and compare with other models.

These mentioned aggregated outcomes leads to achieve the general outcomes of the research.

1.3 Statements of the Problem

The main area of axlebox house failure is to its internal bore of the axlebox house body. In recent years and as an attempt to reduce operating costs, some railway operators have embarked upon a policy of axlebox reclamation [20]. The internal bore dimensions have been restored by either the fitting of an internal sleeve, or by the hot spray application of a molten metal to the axlebox bore house. With the latter method of bore restoration, the metal layer is built up to a suitable oversize dimension and the bore is then machined to tolerance. The axlebox house is then returned for normal application use [20].

Fatigue is the initiation and growth of a crack under varying stress. Using the term fatigue to describe structural failure after repeated loadings is believed to have originated in the mid-1800's in the railroad industry [9]. Fatigue is a critical failure for most types of passenger and freight axlebox house due to multitude of reasons such as cyclic and fluctuating loading during service life which induces varying stress. Early fatigue damage is invisible to naked eye and its cracking does not have significant change on the main behaviour of the structural component until the crack has grown to near critical length [12]. Once it happens, it is difficult or impossible to recover and hence catastrophic failure might occur suddenly.

In addition to the above major failure, shock loading on the wheel rail contact, wear, heat and heavy variable loads leads to derailment and catastrophic mechanical failure of axleboxes house and bearing assemblies.

This research aims to improve the fatigue life of passenger rail vehicle axlebox house with FEM which will have more resistance to fatigue loads and will have smooth running survive in different cyclic load during its lifetime. Generally this research will try to answer the following general questions:-

- What are the main design considerations to improve the fatigue life of axlebox house for passenger rail vehicle?
- How can we achieve and analyse the impacts of combined fatigue loads on passenger axlebox houses through scientific methodological research method?

- What are the main input variables which have significant impact on the fatigue life of the axlebox house?
- How the geometry variation and fatigue strength factors parameters affect the fatigue life and the fatigue sensitivity of life in different loading condition?
- What is the significant effect or outcome of the result for ERC?

1.4 Significance of the Research

This research will have the following significance:-

- Improves the fatigue life of passenger rail vehicle axlebox house in consideration to the geometry and fatigue strength factor parameters with FEM approach to reduce the effects different stress and load condition for safe and smooth running survive under cyclic load during its lifetime.
- Improve the safety of operation by improving the early damage of the axlebox house.
- Safes money and time resulted from early failure of bearing assembly units and other attachments.
- Safes importing cost and time of utilization by manufacturing the axlebox units locally by improved design results and the new findings that have significant impact on the development of durable and good failure resistance passenger axlebox house.

The research outcomes will have significant importance for the ERC by avoiding early failures of the axlebox house which leads catastrophic accidents such as derailment, wear, high thermal stress, bearing misalignment as the result of fatigue failure. Furthermore, the research gives significant insight for local manufacturing of the axlebox house for Ethiopian Railway Cooperation (ERC) from the developed design methodology to reduce the purchasing and investment cost.

1.5 Scope and Limitation of the Study

The scope of the thesis is to improve the fatigue life of passenger rail vehicle axlebox house in consideration to geometry and fatigue strength factor parameters that has to withstand different loading and adverse conditions due to fatigue stress with Finite Element Method.

The design will be carried out with respect to the design principle, data collected from standards and different literature review, modelling geometry and analysed with ANSYS software to evaluate the result.

Generally, the main factors that determines the design and construction of the axlebox house is the way it experiences the variable axial, radial and fractional load, fatigue loads and thermal stress [6].On this paper we are very much indebted to improve the fatigue life of the axlebox house since failure due to fatigue has a saviour consequence in comparison to other failures on the axlebox house.

The following major factors delimit this research from farther investigation:-

- There is no suitable working condition to carry out a field and experimental room facility for all loading, environmental, manufacturing and microstructure factors due to The Ethiopian Railway Cooperation project is currently under implementation.
- The axleboxh house is exposed to high temperature especially during braking and the effects of such thermal stresses are ignored.

1.6 Outline of the Thesis

This thesis is organized in five chapters. The first chapter is the introduction part where the background of the project, the objective, statement of the problem, the significance of the research, scope and the limitation of the study are justified.

In the second part, theoretical literatures are reviewed while the third chapter presents the research methodology. The fourth chapter has dealt with fatigue life terminologies and philosophies while the fifth chapter present results, discussion and validation. In its last part conclusion and recommendations are given.

Chapter Two

2. Literature Review

2.1 Fundamentals of Metal Fatigue

Passenger rail vehicle axlebox house is by far the most important component of railway vehicle. The axlebox house has to withstand variable dominating stress and adverse conditions frequently encountered during use and this component is mostly susceptible to fatigue failure. Since the axlebox house is required to withstand high fatigue under variable and combined loads with accurate dimensional clearance to safe guard the rail vehicle from catastrophic accident, it is important to study the property of fatigue on this component and review literatures written within this relevance and it is worth to mention the nature and methods of fatigue analysis done to improve the fatigue life and damage on some parts of railway components and machine elements.

It has been estimated that fatigue contributes to approximately 90% of all mechanical service failures. High-cycle fatigue involves a large number of cycles and an elastically applied stress. High-cycle fatigue tests are usually carried out for 10^7 cycles and sometimes 5×10^8 cycles for nonferrous metals. Although the applied stress is low enough to be elastic, plastic deformation can take place at the crack tip [12].

The article also indicates that one of the most effective methods of improving fatigue life is to induce residual compressive stresses on the surface of the part. This is often accomplished by shot peening or by surface rolling with contoured rollers. In addition the article indicates that when steel parts subjected to wear conditions are often carburized or nitride to harden the surface for greater wear resistance. Other surface-hardening methods, such as flame or induction hardening, produce similar effects. Case hardening is more effective for components that have a stress gradient, as in bending, or where there is a notch.

This article also mentioned the fatigue design methodologies and philosophies such as infinite life design, safe life design, fail safe design and damage tolerance design.

Generally the article tries to indicate the severity of fatigue by describing that the fatigue contributes to approximately 90% of all mechanical service failures. And also mentioned the range of high-cycle fatigue tests is 10^7 cycles to 5 x 10^8 cycles, effective methods of improving fatigue life by inducing residual compressive stresses on the surface of the part, effective methods of improving fatigue life and fatigue design methodologies or philosophies.

Even if this article does not indicate detail analytical procedure how to increase the fatigue life of the component and factor which influence the fatigue life, most of the motioned points will argue theoretically to the analyses of the fatigue life improvement on passenger rail vehicle axlebox house. Some of metallurgy process mentioned above are not the main discussion of the thesis hence the reader can extend farther assessment on it.

[2] The previous experimental work showed that fatigue performance is affected by the alloy system, heat treatment method, and microstructure features of test specimens. The present study will present information concerning the effects of varying the sinter-hardening cooling rate (and subsequent microstructure features) on the mechanical properties sinter-harden steels and the material system. Emphasis will be given to the rotating bending fatigue performance of these systems and how this experimental data correlates with the fatigue performance of the actual component in accelerated life testing.

[30] Stated some of the fundamentals of fatigue and continued their article pointing that fatigue manifests in the form of initiation or nucleation of a crack followed by its growth till the critical crack size of the parent metal under the operating load is reached leading to rupture. This article also defines the type of fatigue loads as load cycles can be of constant amplitude or variable amplitude type. Rotating machines usually operate under pre-decided constant amplitude load cycles whereas aircrafts and ships are subjected to variable amplitude load cycles due to unpredictable and fluctuating wind or sea gusts.

Based on this article the fatigue is classified as load based and environmental based since fatigue characteristics are also affected by operating temperature and aqueous and corrosive environments. And fatigue life is influenced by metal microstructure, manufacturing process, component geometry, type of environment, loading conditioned, etc.

Based on this article the fatigue life is predicted as constant amplitude load and variable amplitude and complex loads.

The above discussion indicate that the crack growth rate decided by the magnitude of load and geometry of the component. This thesis give also high consideration for the effect of combined load and geometry variation on the axlebox house .The thesis design aims to reduce the applied stress to be less than the axlebox house material fatigue limit so that the nucleated crack growth propagate extremely slowly resulting in high fatigue life .

Railway passenger vehicle is rotating machines usually operate under pre-decided constant amplitude load cycles and hence based on this article consideration the type of fatigue load on axlebox house is typed as constant amplitude and the fatigue is classified as load based. To investigate environment based fatigue characteristics, experimental and real life rail vehicle characteristic has to be conducted.

From the stated factors influencing fatigue, component geometry and loading conditions are studied on this thesis to predict the fatigue life due to constant amplitude load. Generally this article argues with the thesis basic concepts but still cannot guide or inefficient to explain how the fatigue life of a certain component is determined and improved analytically.

Service condition and external environment, Metal microstructure and Manufacturing process indicated above are beyond the scope of this thesis .But farther investigation and researched could be made with those factors to improve the fatigue life of the axlebox house.

[8] Indicated that structural failures due to a single static loading are rare. The majority of structural failures are fatigue failures caused by fluctuating loads. Stress-life (SN) and strain-life (EN) are fatigue analysis methodologies that can be used to predict service life in a virtual environment.

Components subjected to stresses less than yield do not experience plastic deformation and have relatively long lives. This type of service is commonly referred to as high-cycle fatigue. For ductile metals, high-cycle fatigue is generally considered to be greater than 100,000 cycles of operation. Components subjected to stresses greater than yield experience plastic deformation and have relatively short lives. This type of service is commonly referred to as low-cycle fatigue.

For ductile metals, low-cycle fatigue is generally considered to be between 100 and 100,000 cycles of operation. This author also indicates that even though EN is applicable to both low-cycle and high-cycle fatigue, SN is still widely used for many reasons such as:

- Most components are designed for high-cycle operation.
- Large amounts of applicable fatigue test data are available.
- Many commercial industrial design codes are based on the SN approach.
- Historical data exists to verify the acceptability of the SN approach.
- SN approach is simpler to implement than EN.

The choice of the appropriate approach depends upon the application. If the component is subjected to fluctuating stresses greater than yield (low-cycle), then EN is probably required. If the component is subjected fluctuating stresses less than yield (high-cycle), then EN is still applicable, but SN is also a valid approach.

[1] Indicate on its page titled fatigue life improvement that electro polishing has become a common metal finishing process used to help improve the life of metal parts that flex, cycle, twist and bend. These components come in many forms, including stampings, wire forms, laser-cut tubes and even machined parts. During the manufacturing process, micro defects on the surface of these components are left behind. These defects, often in the form of micro-cracks or pits, can become initiation sites for crack propagation or corrosion.

[36] Refers treatment method for improving fatigue life and long life metal material treated by using same treatment. A novel treatment method, for improving fatigue life, is provided that aims to resolve problems associated with conventional treatment methods for improving fatigue life of metal by reduction of stress concentration and conventional treatment methods for improving fatigue life of metal by introduction of compressive stress, that is, problems of poor efficiency in work execution, required level of skill of workers, and the impossibility of quality control due to lack of means for measuring the effect after treatment, characterized in that, for locations in metal for which fatigue is a problem, after pre-treatment is performed, ultrasonic

impact treatment is performed, and thereafter, a quality assurance test is performed so as to improve the fatigue life of the metal.

[1,36] These two articles try to indicate that the fatigue life could be also improved by electro polishing and reducing the stress concentration. But this thesis is mainly concerned from the geometry development to the design of fatigue life improvement analysis due to the limitation indicated in sub- section 1.5.

2.2 Fatigue Life Review for Axlebox House

Fatigue life design and improvement of railway vehicle axlebox house is also dependent on the bearing units installed in the axle journal .Therefore, consideration will be given to the type of bearing unit and selection for the axlebox house geometry and load analysis.

With this relevance SKF bearing calculation extracted from the railway technical handbook [34], described that when selecting axlebox bearings, in addition to the bearing rating life calculation, other design elements should be considered as well. These include components associated with the bearing unit and the axlebox such as axle journal, axlebox housing and the interaction of guidance principle, springs and dampers of the bogie design. The lubricant is also a very important component of the bearing arrangement, because it has to prevent wear and protect against corrosion, so that the bearing can reach its full performance potential. The seal performance is of vital importance to the cleanliness of the lubricant. Cleanliness has a profound effect on bearing service life, which is why tapered and cylindrical roller bearing units are mainly used, because they are factory lubricated and have integrated sealing systems.

This article refers basic rating life according to ISO 281. This standard covers the calculation of the dynamic basic load rating and basic rating life of passenger axlebox bearing and gives a good consideration and very good parameters for the design of the axlebox house bearing units which determine the size of the axlebox house.

[33] Point out necessary information from the practical point of view. According to this article the way in which the loading is quantified and applied in exceptional operating scenarios has to correspond with the way in which the permissible stresses are defined. This is particularly important in fatigue analysis; with the endurance limit approach, the stress due to the worst load

combinations should be considered, whereas for a cumulative damage approach, the effective stress spectrum due to the combined effect of the load scenarios is required.

An appropriate failure criterion is chosen for the determination of the stress depending on the type of material. For example, for ductile material, it is common to use the von mises stress criteria .The areas of local plastic deformation associated with stress concentrations shall be sufficiently small so as not to cause any significant permanent deformation when the load is removed. The avoidance of significant permanent deformation can be demonstrated by fatigue strength and endurance limit approach according to this standard. The fatigue strength can be demonstrated by methods like endurance limit approach, cumulative damage approach or other established methods. Fatigue strength can be evaluated using S-N curves, also known as Wöhler curves.

This article also indicates the basic three axlebox design principles applied to different spring and guidance systems such as One-piece housings, Two-piece housings and Three-piece housings.

This article indicates that depending on application and customer specifications, various materials can be used. For housings, mainly spheroidal cast iron (GJS) is used. The FEM results provided as maximum or principal stresses values based on von mises. Based on proven standards and practical experience, like data from the FKM (Forschungskuratorium Maschinenbau) testing institute, statistical post-processing is done. The software uses the fatigue output files (e.g. stress results) to calculate a safety factor. The two load cases that give maximum stress amplitude are further calculated. Several influencing factors, like mean stress, surface condition, and local stress gradient have to be considered. This enables the detection of critical areas that cannot be observed only by FEM calculations.

Eventhogh this article describe most of the basic things which has to be included in the design and fatigue analysis of axlebox house it does not include detail analysis, load analyses and fatigue life analysis and numerical output.

Chapter Three

3. The Research Methodology

Passenger rail vehicle axlebox house is the main component of wheelset axlebox which support the axlebox bearing, protect the axlebox from dusts and foreign derbies, transmit varies loads to the bogie frame, spring and dampers and holds varies accessories such us earth brushes, anti-skid device sensors, speedometer, generator drive gears, hot axlebox sensors, etc. Hence it requires having standard design procedure and methodology.

Thorough the design and analysis of the thesis the best suitable passenger axlebox house geometry will be selected, fatigue life failure modes on the axlebox house will be analysed with Finite Element Method and the result will be discussed based on the design principles and standards.

The main methodological approaches followed to improve the fatigue life of passenger rail vehicle axlebox house are:-

- Data collection and organization from primary and secondary data;
- Data analysis for geometry of axlebox house sizing and load estimation;
- Geometry is modelled with Solidwork software from the above analysed data in consideration to the customized passenger axlebox house types;
- All selected passenger axlebox houses models are analysed by FEM for fatigue life and fatigue sensitivity for life by combining the bending and the torsion loading at different fatigue strength factor and geometry by ANSYS workbench fatigue tool;
- The output result is discussed and significant parameters on the fatigue life are illustrated and validated.

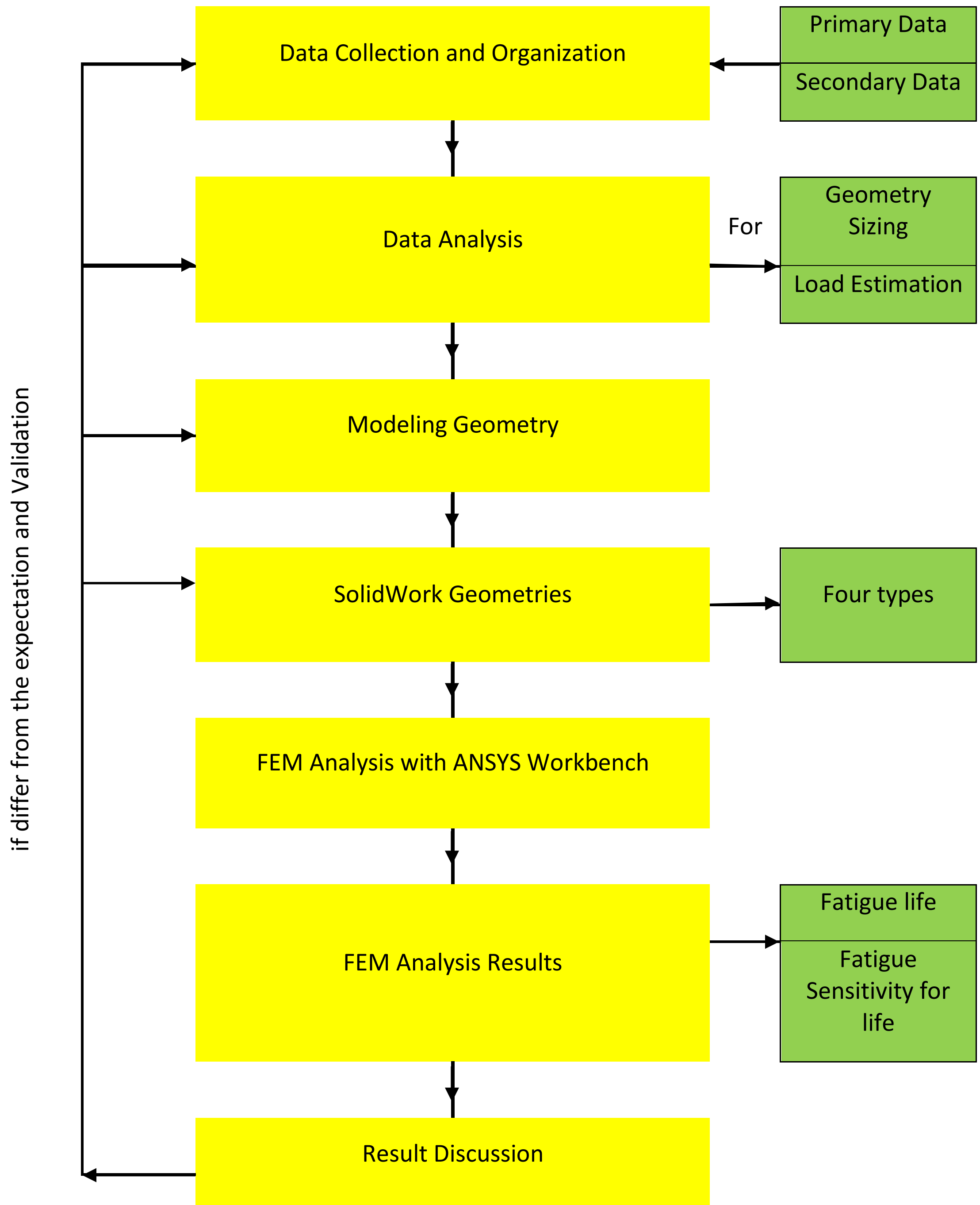


Figure 3-1 Methodology flow Chart

3.1 Data Collection and Organization

3.1.1 Data Collection

The axlebox house geometry is depends on the wheel set journal size, bearing size, arrangement of bogie frame, suspension type, etc. Therefore it is important to consider relevant data equivalent capacity to the desire axlebox house.

Primary as well as secondary data which is needed for the study are collected from different literature reviews and railway standards such as "Railway application wheelsets and bogies methods of specifying the structural requirements of bogies frame UNI EN 13749:2005".

The primary data are those date required for selection and modelling the geometry as well as load analysis includes:-

- Bogie type and application for the axle box house selection
- Type of attachments to the bogie frame
- Axle load
- Axle Journal size
- Bearing type and size, etc.

Secondary data are required for the input of software analysis, in this case for the finite element analysis with ANSYSI software, includes:-

- Material Data
- Boundary data/conditions
- Load data
- Service condition data, etc

A. Primary Data

As per the above discussion primary data required for the geometry modelling and fatigue life load analysis of axlebox house include:

- **Bogie Loading Data:** The load on the axlebox house is transmitted through bogie frame and bearings. Therefore it is important to consider the load characteristic on the bogie frame. Equivalent loading capacity of bogie for light rail vehicle and trams of Category B-IV is selected.[25]

For exceptional loads the effective car body mass m_1, including passengers, corresponding to bogie Category IV is given by[25]:

$$m_1 = (M_v + P_1)\frac{c}{100} - n_b m^+ \quad \text{.. (3.1)}$$

For normal service loads the effective car body mass m_1, including passengers, corresponding to bogie Category IV is given by[25]:

$$m_1 = (M_v + P_2)\frac{c}{100} - n_b m^+ \quad \text{.. (3.2)}$$

Where:

M_v = mass of car in running order;

P_1 = Exceptional payload

P_2 = Service (Fatigue) payload

C= wheel loads of relevant bogie expressed as a %

m+ = bogie mass

n_b= number of bogies per body

Bogie Frame Loads [25]:

Longitudinal force (applied at the centre of gravity) is give by

$F_{xc}=m_1a_{xc}$.. (3.3)

Transverse force (applied at the centre of gravity) is given by

$F_{yc}=m_1(a_{yc}+a_{ycc})$...(3.4)

Vertical force (applied at the centre of gravity) is given by

$F_{zc}=m_1(g+a_{zc})$...(3.5)

Loads in connection between bogie and car body (collision conditions):

$$F_{xb} = m^+ 3\, g \quad \text{.. (3.6)}$$

Where:

a_{xc} = Longitudinal acceleration of the body

a_{yc} = Transverse acceleration of the body

a_{ycc} = Centrifugal acceleration of the body

$a_{zc=}$ Longitudinal acceleration of the body

- **Bogie Technical Specification Data:** Bogie technical specification data is collected from East-West Line (Phase I) AALRT project [7] and equivalent performance bogie data from Siemens Transportation Systems, Austria [29] to meet the objective of the thesis. The vehicles is 70% low-floor articulated 6-axle modern trams, consisting of three modules, bi-directional driving.Two tramcars shall be able to operate with double heading. With maximum operating speed of 70km/h and maximum test speed 80km/h as bellow train formation and articulation.

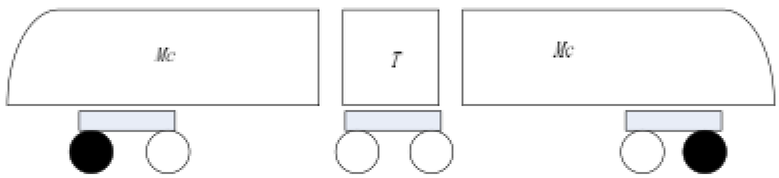


Figure 3-2 Train formation [7]

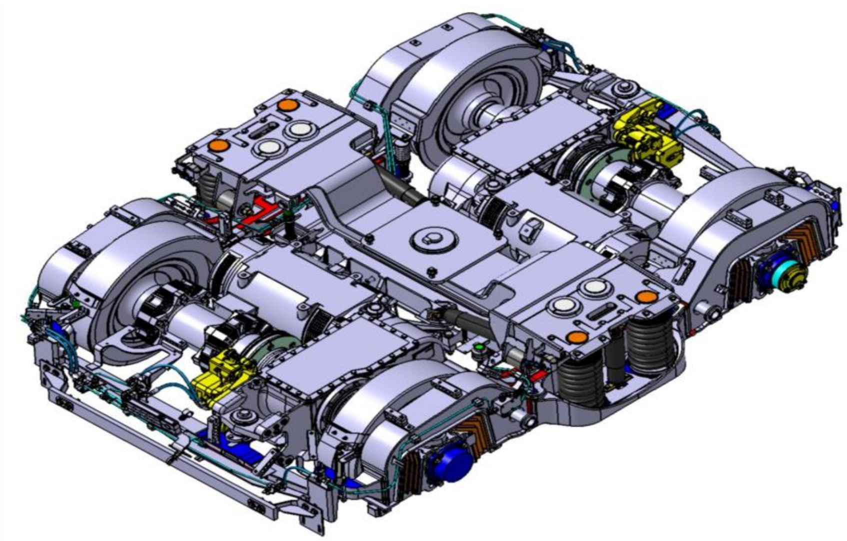


Figure 3-3 AALRT details of designed structure of power bogie [7]

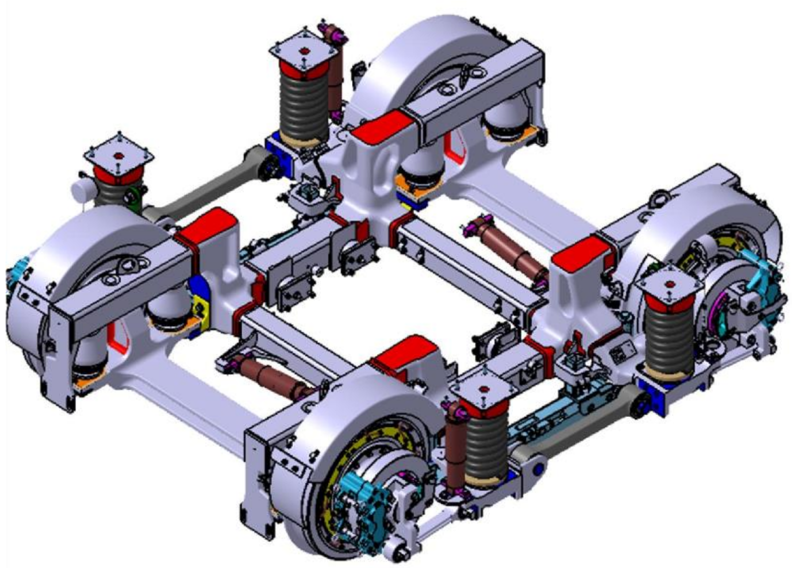


Figure 3-4 AALRT details of designed structure of unpowered bogie [7]

Varies geometries of axlebox house has to be consider to investigate which geometry has a better fatigue life .Therefore, similar capacity of bogie models are taken in consideration .For example, figure 3.5 equivalent SF 1000 Motor and Trailer bogies of Siemens transportation Systems, Austria is taken .Which is designed for light weight metro vehicles with a maximum speed of 80km/h. The wheelbase of 2100mm enables the bogies to be track friendly and particularly suitable for negotiating small radii of curvature.

It is to be noted that the overall dimension and the bearing size for each models are the same.

Figure 3-5 SF 1000 Bogie detail [29]

- **Axle Journal Size Selection Data:** In order to determine and select suitable bearing size so as to determine the internal axlebox house dimension, it is necessary to lock for equivalent axle journal capacity and size relevance to know standards and practice. In this case the Association of American Railroad (AAR), recommended practice provides guidance for specifying the axles for passenger rail cars, whose operations fall under the jurisdiction of Federal Railroad Administration regulation. The designs shown herein are provided as guidelines based on historical experience starting form axle type C [26].

Table 3-1 Passenger car axles load rating [26]

Axle Designation	Size of Journal[in]	Capacity for axle for normal maximum operating speed range of :	
		Up to and including 85mph	86-100mph
C	5x9	28500lbs	27000lbs
D	5 1/2x10	36000lbs	34000lbs
E	6x11	45000lbs	42500lbs
F	6 1/2x12	54000lbs	51000lbs

- **Bearing Selection Data:** Bearings for modern railway systems must offer excellent durability and high speed capability with minimal maintenance requirements. Axle bearings must therefore be designed on the basis of not only dimensional requirements of the axle journal and bearing box geometry, but also these complex load conditions [19], [22] , [23].

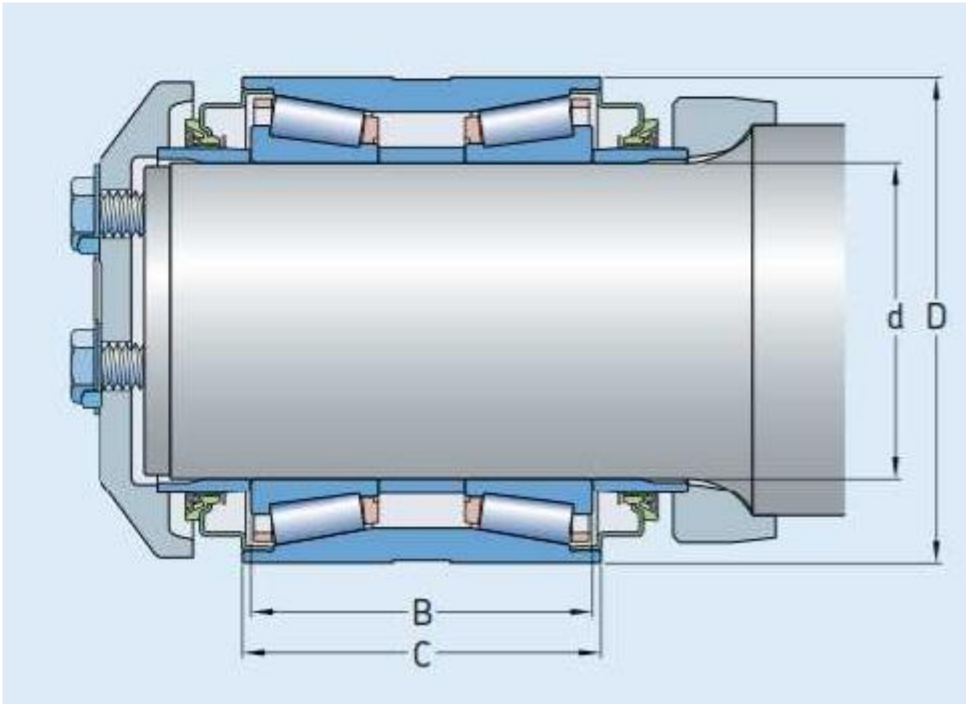


Figure 3-6 Tapper roller bearing assembly on rail axle journal [34]

For locomotives for speed upto 120km/hr, compact TBUs with LL seal in both sides are used. This design requires less space in axial direction compared with a standard TBU design [34].

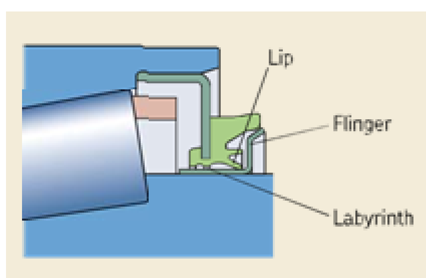


Figure 3-7 LL seal design for compact TBU[34]

The NSK RCT inch series approved by the Association of American Railroads (AAR) provide Sealed-Clean Rotating End Cap Tapered Roller Bearings (RCT) bearings which are highly integrated with surrounding components and incorporate advanced sealing mechanisms. They offer outstanding performance, durability and ease of handling [23].

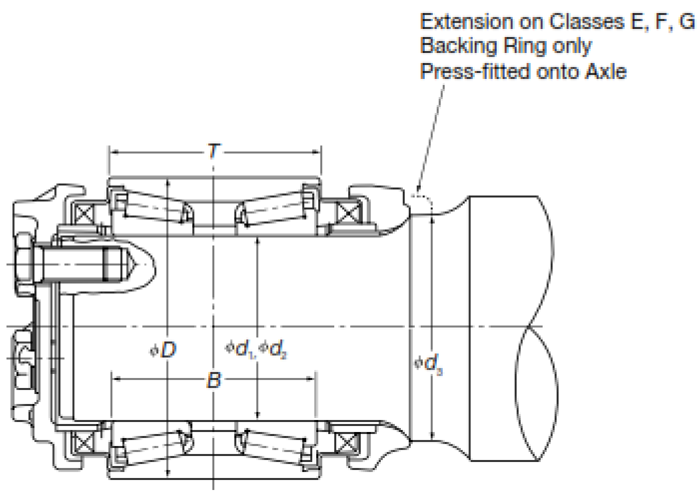


Figure 3-8 RCT Tapper Roller bearings [23]

B. Secondary Data

- **Material Data:** Spheroidal cast iron as it is an iron carbon based material, the carbon being present mainly in the form of spheroidal graphite particles. Due to the nodular shape of the graphite the notch effect is minimized. Thereby high values for tensile strength and elongation at fracture are achieved. EN-GJS-400-15 grade of ductile iron could be used to produce impact-resistant and shock-resistant parts.[8]

Table 3-2 Chemical composition of Shperiodal Cast Iron EN-GJS-400-15[8]

Element Description	Percentage chemical composition
C	2.5-3.8
Si	0.5-2.5
Mn	0.2-0.5
P	≤0.08
S	≤0.02

- **Boundary Condition:** The axlebox house is imposed to different types of combined loading. The loads are assumed to be concentrated at the centre of axlebox and the ends are assumed to be fixed. Deformation and additional loading pressure created on the axlebox house should not be transmitted to the bearings that mean the axlebox house has to withstand those loads by itself.

3.1.2 Data Organization

The primary and the secondary data along with other additional relevant data will be organized for the input of geometry selection, modelling and finite element analysis to achieve the objective of the thesis in consideration to the selected software adaptation or compatibility, real life working environment, boundary condition and limitations.

3.2 Data Analysis

The data will be analysed with Finite Element Method and engineering principles for the axlebox type selected similar to AA LRT passenger vehicle capacity. Then the axlebox house geometry is modelled and analysed based on the design principles and methodology as indicated above. Through those data processing and analysis the fatigue stress and life effect will be revealed on the selected geometry which leads us to achieve improved best design passenger rail vehicle axlebox house.

Load analysis on the axlebox house due to bogie frame and bearing load helps to determine the size of the axlebox geometry model.

3.2.1 Load Analysis on the Axlebox House Due to Bogie Frame

Assumed BO-BO bogie arrangement, rolling coefficient α =0.1 and Bouncing coefficient β=0.2 bogie data for load analysis is as follows [29]:

Table 3-3 Bogie data for numerical analysis [29]

Description	Value
Estimated Vehicle weight,Mv	30000kg
Motor/Powerd bogie mass ,m^+	6700kg
Trailer /unpowered bogie mass,m^+	5000kg
Rolling coefficient, α	0.1
Bouncing coefficient, β	0.2

Loading area of a unit car is approximated=LenghtxCar width=15*2.65=39.75m^2

In reference with Appendix B.1, bogie category B-IV.For maximum of 8 passenger, the exceptional payload will be:P_1=8x39.5x75=23700kg

Therefore, Mass of car from equation 3.1 gives ,m_1=(30000+23700)*1-2*6700=40300kg

And from equation 3.3, $F_{xc}=m_1 x a_{xc}$

Where:

a_{xc}=Emergency braking rate produced during starting /dynamic braking x1.3=a*1.3

Assuming stopping distance,s=700meter at the final zero velocity ,the acceleration of the car body,a is equal to $v^2/2s$.

For exceptional loads for masses attached at the central axis of the axle box the longitudinal acceleration the maximum speed is given as: ± 50 m/s [25].

Hence, $a=50^2/(2*700)=1.78$ and $a_{xc}=1.78*1.3=2.32$m/s2

And

$F_{xc}=m_1 x a_{xc}=40300*2.32=93.5$KN;

$F_{yc}=m_1(a_{yc}+a_{ycc})=40300*(1.3+2)=133$KN and

$F_{zc}= m_1(g+a_{zc})=40300*(9.8+1.6)=451.36$KN

Since we have two bogies per car which are attached to at the center of bolster and the bolsters are sited to the side frame by spring suspension each side frame shares:

$F_{z1} =F_{z2}= F_{zc}/4=112.84$KN;

$F_y= F_{yc}/2=66.5$KN (At Maximum tilting condition) and

$F_x= F_{xc}/4=23.37$KN

The Maximum Load Conditions are selected based on Appendix B.2 and B.3.

The maximum load cases for static test corresponding to vertical and transverse force combinations Appendix B.2 [25] gives:

Loading Case	Fz1 [KN]	Fz2[KN]	Fy [KN]
Case 5:	293.38	248.25	66.5
Case9:	248.25	293.38	-66.5

The maximum load cases resulting from longitudinal forces [25]:

Loading Case	Fz1 [KN]	Fz2[KN]	Fx [KN]
Case 2:	225.68	225.68	46.75
Case 3:	225.68	225.68	-46.75

Assuming the load will be equally shared at each axlebox:

Loading Case	Fz1 [KN]/2	Fz2[KN]/2	Fy [KN]/2
Case 5:	146.69	124.13	33.25
Case9:	124.13	146.69	-33.25

Loading Case	Fz1 [KN]/2	Fz2[KN]/2	Fx [KN]/2
Case 2:	112.84	112.84	23.375
Case 3:	112.84	112.84	-23.375

3.2.2 Load Analysis on the Axlebox House due to Bearing Load

From table (3.1) the axle journal size selected based on the maximum speed 80km/hr and maximum safe axle load 13t(28660.06lbs) is :

Axle Designation =D

Size of Journal [in] =5 1/2x10

Principal dimension according to SKF bearing type TBU and NSK bearing number HM127446R gives the same data :Bore diameter(d) =131.775mm, Outside diameter=

207.963mm, and C(Width) = 152.40mm.And loading data: Basic Dynamic load rating(C) 635KN and Basic Static load rating 1250KN.

The basic rating life of a bearing according to ISO 281 is[34]:

$$L_{10}=(C/P)^{P} \qquad (3.9)$$

Where:

L_{10}= basic rating life (at 90% reliability), millions of revolutions

C = basic dynamic load rating, kN

P = equivalent dynamic bearing load, kN

p = exponent for the life equation

10 = 3 for ball bearings

= 10/3 for roller bearings, as used typically in axlebox applications

For railway applications, it is preferable to calculate the life expressed in operating

Mileage or in million km

$$L_{10s}=\pi D_w/1000(C/P)^{P} \qquad (3.10)$$

Where:

L_{10s}= basic rating life (at 90% reliability),million km

D_w = mean wheel diameter, m

From the above table data of bogie SF1000 D_w=850mm.When determining bearing size and life, it is suitable to verify and compare the C/P value and basic rating life with those of existing similar applications where a long-term field experience is already available .

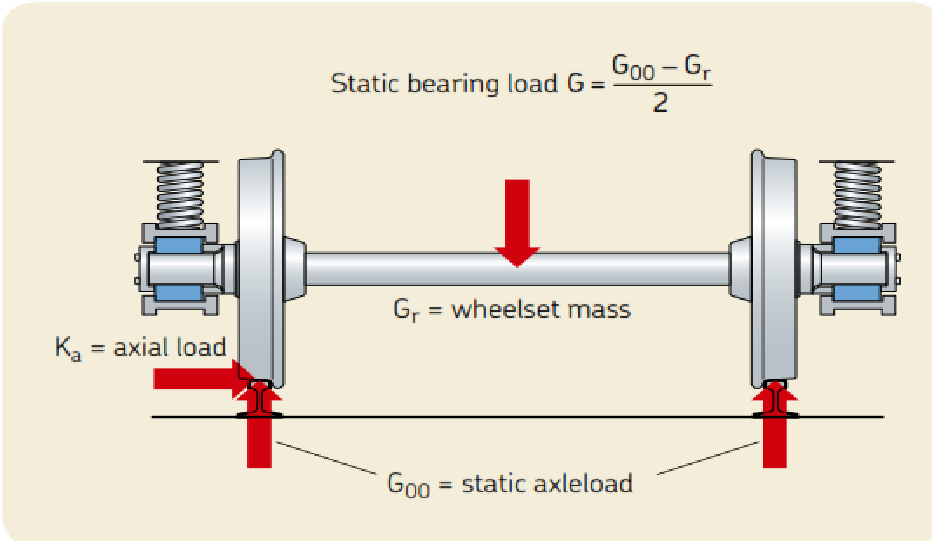


Figure 3-9 Simplified acting bearing loads principles [34]

The maximum static axlebox load per bearing arrangement/unit is obtained from:

$$G=(G_{oo}-G_r)/2 \quad \text{...(3.11)}$$

Where:

G = maximum static axlebox load [kN]

G_{oo} = Assumed Static axle load [kN]

G_r = weight of the wheelset [kN]

From the SF 1000 bogie data the maximum static axlebox load is 13ton or 13000kg (127.53KN) on the two bearings and estimated wheel set Gr weight taken as 1500kg (14.715KN).

For Wheel diameter =0.85, Maximum Speed=80km/hr, formula (3.11) gives:

$$G=[\ G_{00}-G_r]/2=[127.53-14.71]/2=56.41KN$$

Based on the static axlebox load, the mean radial load is calculated by considering any variations in the payload as well as any additional dynamic radial forces.

$$K_r = f_0\, f_{rd}\, f_{tr}\, G \quad \text{... (3.12)}$$

Where:

K_r = mean radial load [kN]

f_o= payload factor

f_{rd}= dynamic radial factor

f_{tr}= dynamic traction factor

Considering variation load in the static loading f_o (payload factor)=0.9,considering the maximum dynamic radial factor due to guasi –static effects like rolling, pitch and dynamic effect from the wheel rail contact frd(dynamic radial factor)=1.3,for powered vehicles with a non-elastic drive system which account additional radial load coused by the drive system factor ftr(dynamic traction factor)=1.1 are used to calculate the mean radial load as of equation (3.12) [34]:

$K_r=f_o f_{rd} f_{tr} G_{max}.=0.9*1.3*1.1*56.41=72.60KN$

In most cases, the radial load is acting symmetrically either on top of the axlebox both sides .In case of such symmetrical axlebox designs:

$$F_r=K_r \qquad (3.13)$$

Where:

- F_r = radial load [kN].In this case F_r= 72.60KN

The mean axial load is calculated by considering the dynamic axial forces on the Axlebox as:

$$K_a=f_o f_{ad} G \qquad (3.14)$$

Where:

- f_{ad} = dynamic axial factor, Hence formula (3.14) gives;

$K_a=f_o f_{ad} G=0.9*0.08*56.41=4.06KN$

And in this case;

$F_a = K_a$= 4.06KN

The radial bearing load is given by:

$$F_r = K_r + 2f_cK_a \quad (3.15)$$

Where:

$f_c = h\,Da / l$

D = shaft diameter, mm

l = distance between the 2 load centres, mm

h = 0,25 if the load acts at the top or at the bottom of the housing

h = 0,10 if the load acts near to the central plane of the bearing

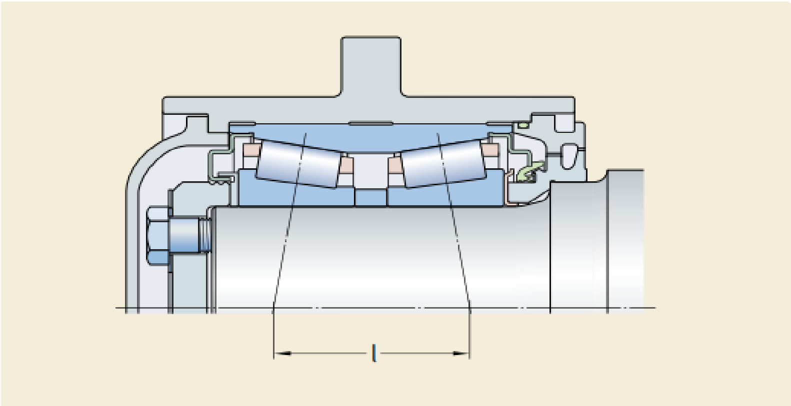


Figure 3-10 Distance between two roller load center [34]

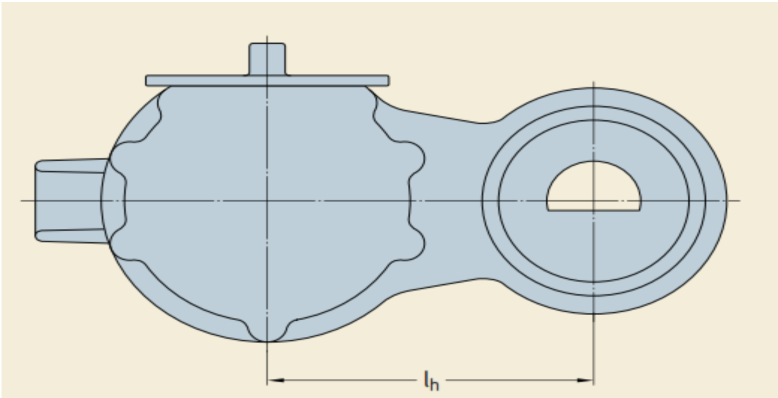


Figure 3-11 Loading center distance [34]

The Geometry factor f_c=0.1*131.775/112=0.12 and taking the maximum load condition;

$F_r=K_r + 2f_cK_a$=72.6 +2*0.12*4.06=73.57KN and Mean bearing axial load $F_a=K_a$=4.06KN

Where;

f_c is the geometry factor

Equivalent dynamic bearing load P is used for basic rating life calculation as as follows:

$P = F_r + YF_a$(for tapered roller bearings and spherical roller bearings)

$P = F_r$(for cylindrical roller bearings)

Where;

P = equivalent dynamic bearing load [kN]

Y = axial load bearing factor

$Y\alpha F_a/F_r$ for F_a/F_r=2.11/38.22=0.06,Y is taken as approximately 2.

Hence , $P = F_r + YF_a$=73.57 + 2*4.06=81.69KN.

From equation (3.9) and (3.10) the basic rating life is calculated as:

$L_{10}=[C/P]^{10/3}$ in millions of revolution.

$=[635/81.69]^{10/3} = 930*10^6$ revolution

$L_{10s}=\pi L_{10}D_w/1000=3.14*930*10^6*0.85/1000$=2.5 million km which is greater than 1.5 million km basic rating life as per the reference [34].This indicates that the selected bearing is safe under the given load condition and can serve more than 10years for 8 working hours per day.

Table 3-4 Load summary on the axlebox house

Load transferred by bogie frame		**Load transferred by bearing**	
Maximum in the vertical direction:	146.69KN	Maximum Radial load:	72.6KN
Minimum in the vertical direction:	124.13KN	Maximum axial load:	4.06KN
Lateral force :	+/-33.5KN		

The axlebox house has to transfer its load to the bearing unit without deformation and without additional stress on the bearing unit. That means it should beer the maximum load created by the side frame.

3.3 Modelling Geometry

Basically there are three axlebox house design principles applied with relevance to different rail vehicle spring and guidance systems, those are [33]:-

- One-piece housings that have to be mounted axially onto the wheelset with the mounted bearing unit.
- Two-piece housings that are split designs. This enables a radial mounting of the axlebox. The main advantage is that it is easily dismount the complete wheelset by unscrewing and removing the upper part of the axlebox and exchange of wheelsets in workshops is much easier.
- Three-piece housings are split designs as well, but have an additional sleeve to protect the bearing arrangement or unit.

Indicative geometry dimension of the axlebox house is referred from railway bearing manufacturer, KINEX [14], which offers complex services in the field of research, development and production of the rolling bearings and rolling elements so as to use as a boundary over all dimension for all models.

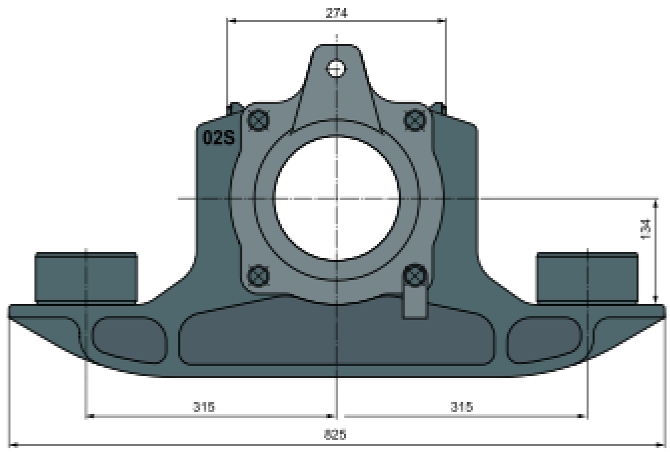


Figure 3-12 Typical one-piece axlebox house dimension [17]

Varies referral dimensional drawings cannot be achieved for this article and this thesis will try to compare different axlebox house drawing models done by SolidWork modelling software based on constrained and obtained dimension from the bearing data to compare the best performed axlebox house for our future use and development. The Solidwork models are prepared considering the reputable design model pricinciples, mass reduction, surface finish and actual working conditions to demonstrate the best output data relevant to the actual operation condition.

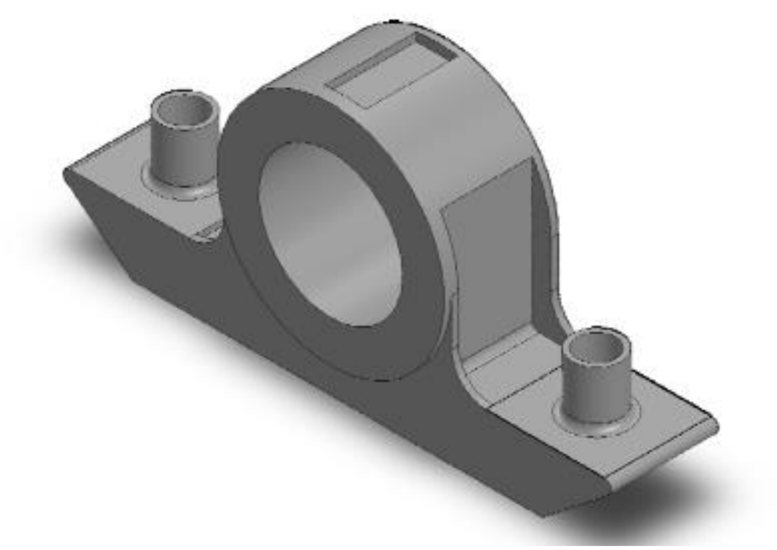


Figure 3-13 Solid work geometry Model I

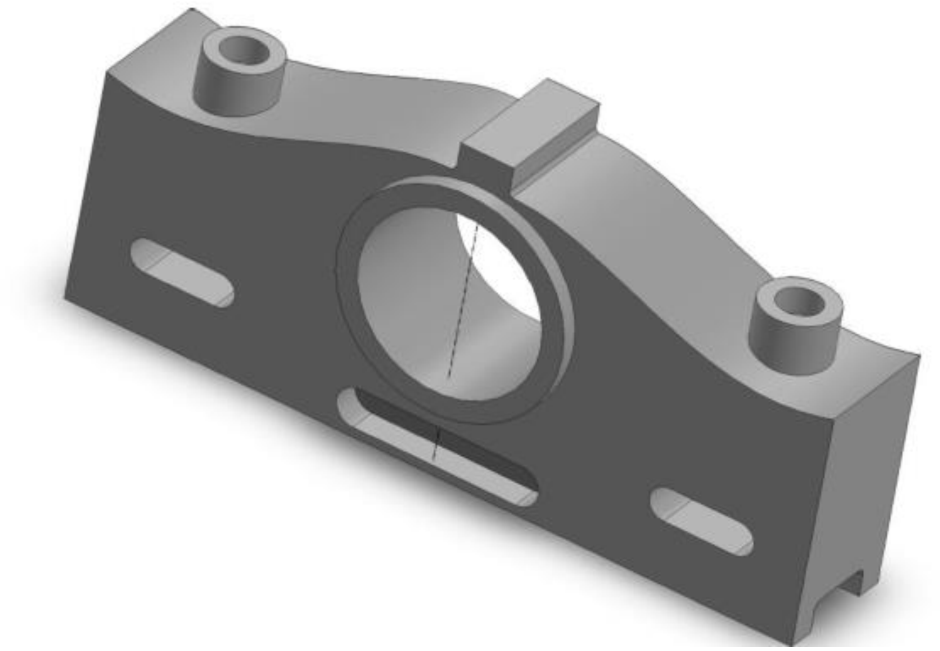


Figure 3-14 Solid work geometry Model II

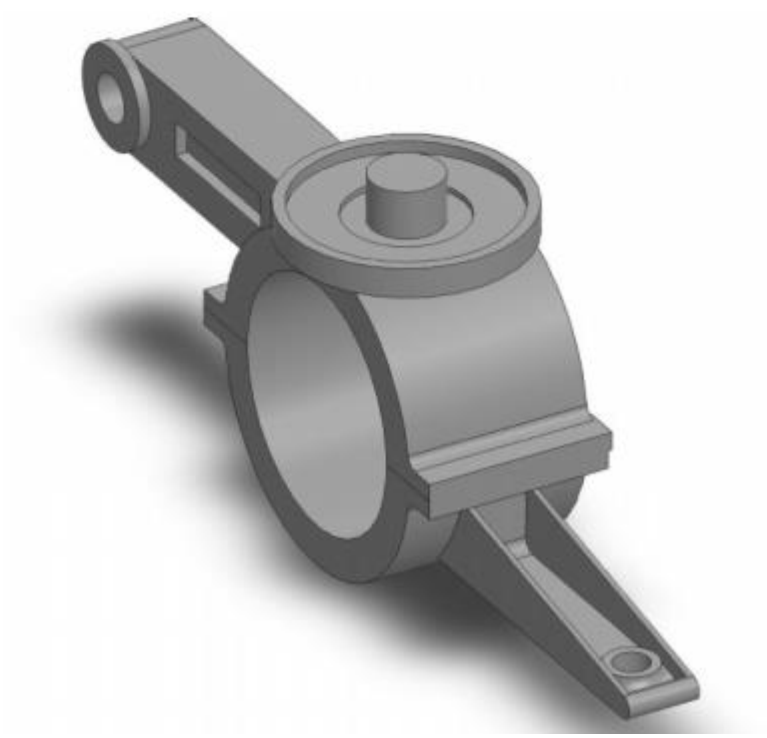


Figure 3-15 Solid work geometry Model III

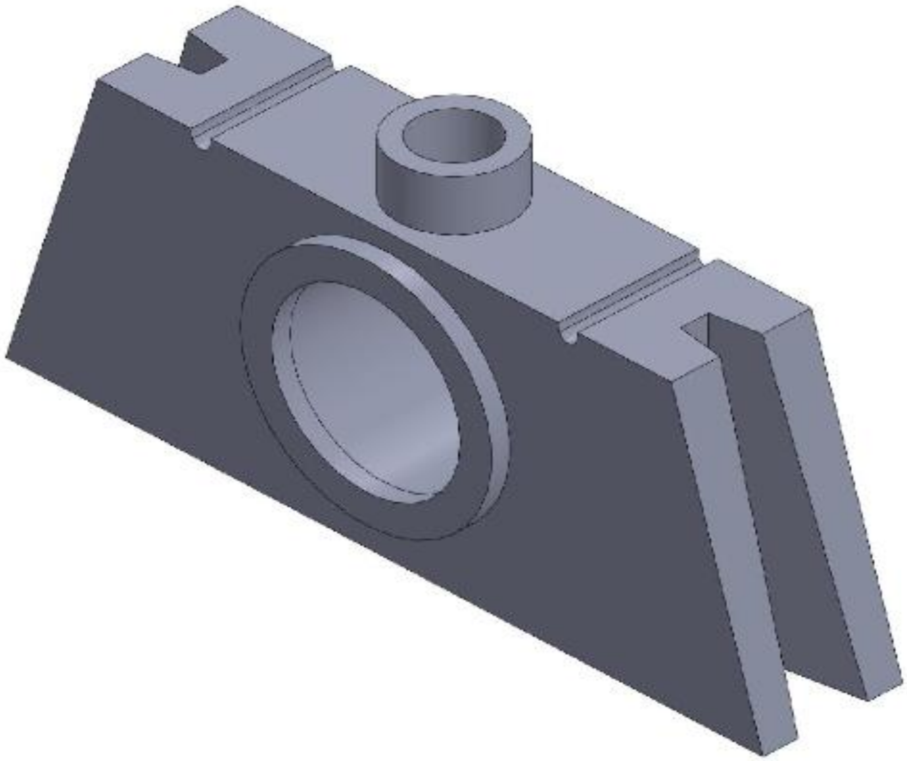


Figure 3-16 Solid work geometry Model IV

Chapter Four

4.1 Fatigue Life Terminologies and Philosophies

A fatigue failure almost always begins at a local discontinuity such as a notch, crack, or other area of stress concentration. When the stress at the discontinuity exceeds the elastic limit, plastic stain occurs. For fatigue fracture to occur there must exist cyclic plastic strains and it is affected by:

- Nature and type of loading: Axial Tension, bending, torsion and combined reversible, Repeated, Fluctuating and Alternating, Mean and Variable components Frequency of loading and rest periods.
- Geometry: size effects and stress concentration
- Material: Composition, structure, directional properties and notch sensitivity
- Manufacturing: Surface finish, heat treatment, residual stresses
- Environment: Corrosion, high temperature, radiation

The reader need to have a clear understanding and knowhow to the following terms before we go to the detail FEM fatigue life analysis for the selected geometries [3], [4], [10], [11], [13],[21],[35],[37] ,[39]:

a. Endurance Limit and Endurance Strength

b. Endurance Limit Modification Factors

c. High and Low cycle fatigue

d. Finite and Infinite life

e. Fatigue stress concentration factor

f. Fluctuating Stresses

g. Failure Criteria

h. Combination of loading modes

i. Fatigue strength and finite life design

a) Endurance Limit and Endurance Strength:

In the case of the steels, a knee occurs in the graph, and beyond this knee failure will not occur, no matter how great the numbers of cycles are. The strength (stress amplitude value) corresponding to the knee is called the endurance limit (Se) or the fatigue limit. Finding the Endurance Limit using the rotating beam experiment is time consuming where it requires testing many samples and the time for each test is relatively long. Therefore they try to relate the endurance limit to other mechanical properties which are easier to find (such as the ultimate tensile strength).

Steel [16]:

$$Se=\begin{cases} 0.5S_{ut} & ,S_{ut}\leq 200ksi(1400MPa) \\ 100ksi & ,S_{ut}>200ksi \\ 700MPA & ,S_{ut}>1400MPa \end{cases} \quad \text{...}(3.16)$$

Cast Iron [16]:

$$Se=\begin{cases} 0.4S_{ut} & ,S_{ut}\leq 60ksi(400MPa) \\ 24ksi & ,S_{ut}>60ksi \\ 160MPA & ,S_{ut}>1400MPa \end{cases} \quad \text{...}(3.17)$$

Se is the endurance limit value obtained for the test specimen (modifications are still needed).

b) Endurance Limit Modification Factors:

The most important factors that affect the fatigue performance (strength) are presented above. To account for these conditions a variety of modifying factors, each of which is intended to account for a single effect, is applied to the endurance limit value of test specimen obtained under laboratory conditions. Consequently we may write ;

$$S_e = Se^* k_a k_b k_c k_d k_e \qquad (3.18)$$

Where;

S_e = endurance limit of mechanical element (to be designed)

Se^* = endurance limit of test specimen.

k_a = surface factor

k_b = size factor

k_c = load factor

k_d = temperature factor

k_e = miscellaneous-effects factor

- **Surface Factor, k_a:** The surface of the rotating-beam specimen is highly polished, with final polishing in the axial direction to smooth out any circumferential scratches. For other conditions the modification factor depends upon the quality of the finish and upon the tensile strength.

  Based on surface finish the surface factor is given by $= A (S_{ut})^{b}$ ....(3.19)

  For forged surface finish A=272mpa and b=-0.995mpa

- **Size factor K_b:** The rotating beam specimens have a specific (small) diameter. Parts of larger size are more likely to contain flaws and to have more non-homogeneity.

The size factor is given as:

$$K_b = \begin{cases} 1.24d^{-0.107} & , 2.79 \le d \le 51\text{mm, for bending} \\ 1.51\, d^{-0.157} & , 51 \le d \le 254\text{mm, for torsion} \end{cases} \qquad (3.20)$$

Where d is the diameter and ,K_b=1 for axial loading.Both of the above apply strictly only to fully reversed rotating bending or torsion of round shafts. For other situations, you may use an equivalent diameter de in the above relations.For a rectangular section in bending, use de = 0.808 √hb where h,b are the height and breadth of the section.For a circular section in non rotating bending, use de = 0.370d

- **Loading Factor (Kc):**

$$\text{Loading Factor } (K_c): \begin{cases} 1 & \text{bending} \\ 0.85 & \text{axial} \\ 0.59 & \text{torsion} \end{cases} \qquad \text{(3.21)}$$

The loading factor is 1 for combined load and if von mises stress is used.

- **Temperature Factor (Kd):** When the operating temperature is below room temperature the material becomes more brittle .When the temperature is high the yield strength decreases and the material becomes more ductile and creep may occur.

$$K_d = 1 \text{ for } t < 450^{o}C$$

- **Reliability Factor Ke:**The endurance limit obtained from testing is usually reported at mean value .

Reliability	Ke
90%	0.897
95%	0.868
99%	0.814
99.9%	0.753

c) High and low cycle fatigue

The body of knowledge available on fatigue failure from N=1 to N=1000 cycles is generally classified as low-cycle fatigue. High-cycle fatigue, then, is concerned with failure corresponding to stress cycles greater than 10^3 cycles.

d).Finite and Infinite Life

We also distinguish a finite-life and an infinite-life region. Finite life region covers life in terms of number of stress reversals upto the knee point

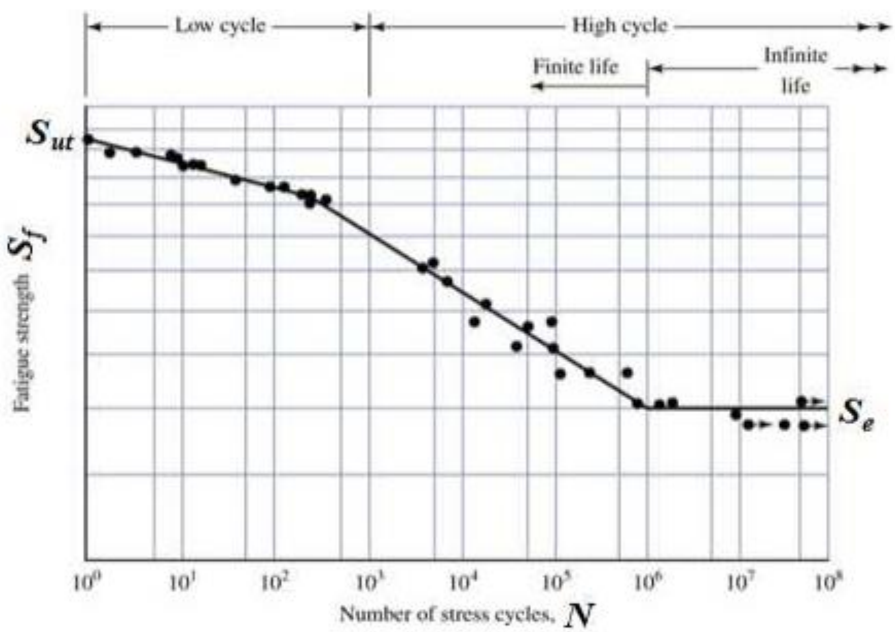


Figure 3-1 Typical S-N diagram for steel in log-log scale[27]

e) Fatigue stress concentration factor,K_f

The existence of irregularities or discontinuities, such as holes, grooves, or notches, in a part increase the magnitude of stresses significantly in the immediate vicinity of the discontinuity.

Fatigue stress concentration factor, K_f = Maximum stress in notched specimen/Maximum stress in notch free specimen.

f) Fluctuating Stresses:

Quite frequently it is necessary to determine the strength of parts corresponding to stress situations other than complete reversals. Many times in design the stresses fluctuate without passing through zero. Relations among them indicated as follows [16]:

For fluctuating stress;

The mean stress is given by :

$$\sigma_m = \frac{\sigma_{max} + \sigma_{min}}{2} \quad \text{...} \quad (3.22)$$

And the alternating stress:

$$\sigma_a = \frac{\sigma_{max} - \sigma_{min}}{2} \quad \text{...} \quad (3.23)$$

Where;

σ_{min} = minimum stress

σ_{max} = maximum stress

g) Failure criteria:

A Haigh diagram plots the mean stress, usually tensile, along the x-axis and the oscillatory stress amplitude along the y-axis. Lines of constant life are drawn through the data points. The infinite life region is the region under the curve and the finite life region is the region above the curve.

The two most widely accepted methods are those of Goodman and Gerber. Experience has shown that test data tends to fall between the Goodman and Gerber curves. Goodman is often used due to mathematical simplicity and slightly conservative values. For design applications the stresses σ_a and σ_m can replace Sa and Sm in the above equations and each strength is divided by a factor of safety N. The resulting equations are:

Soderberg criteria (line)

$$K_f \frac{\sigma_a}{S_e} + \frac{\sigma_m}{S_{yt}} = \frac{1}{N} \quad \ldots\ldots (3.24)$$

Goodman relation or criteria

$$K_f \frac{\sigma_a}{S_e} + \frac{\sigma_m}{S_{ut}} = \frac{1}{N} \quad \ldots\ldots (3.25)$$

Gerber parabolic relation:

$$K_f \frac{N\sigma_a}{S_e} + \left(\frac{N\sigma_m}{S_{ut}}\right)^2 = 1 \quad \ldots\ldots (3.26)$$

S_e is corrected endurance limit values and K_f factor accounts for stress concentration effects. Good predictor of failure in ductile materials experiencing fluctuating stress is illustrated as bellow diagram:

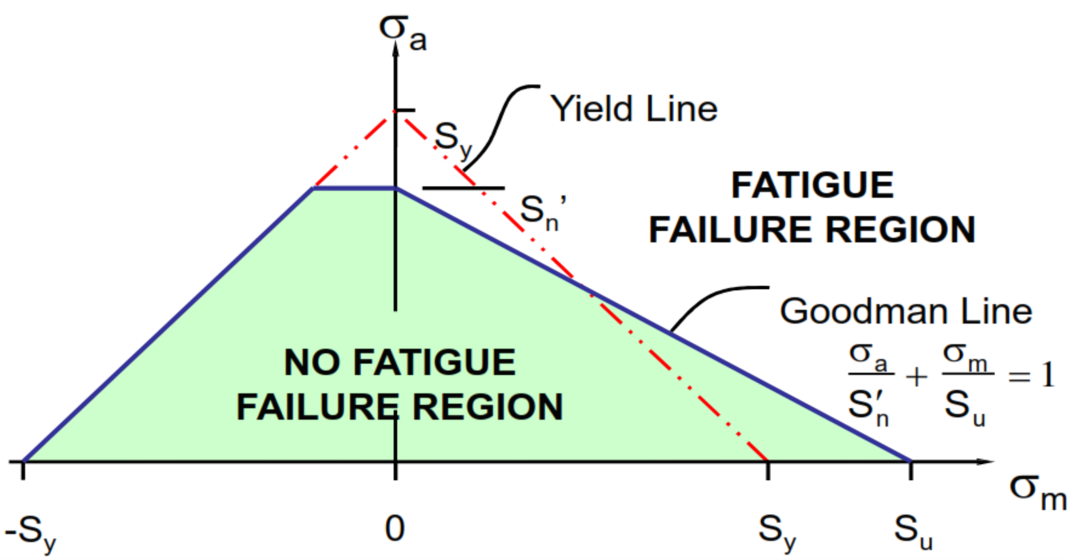


Figure 3-2 Goodman fatigue failure region[14]

Where: S_n' = actual endurance strength, σ_a = alternating stress and σ_m = mean stress

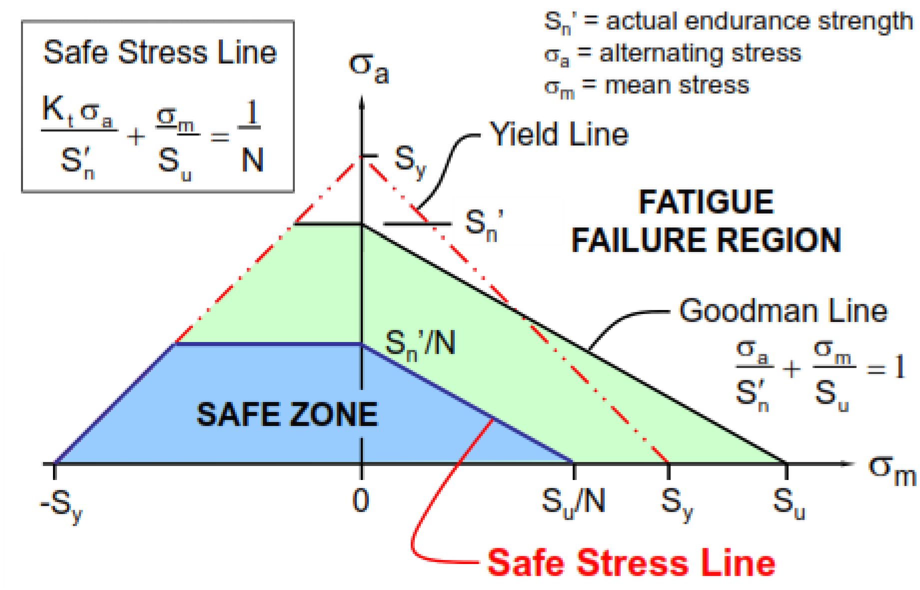


Figure 3-3 Goodman safe zone region[14]

h) Combination of loading modes

This section answers the questions, how do we proceed when the loading is a mixture of say, axial, bending, and torsional loads.

To begin a combined loading fatigue analysis you must first start with a stress analysis at the point of interest. This analysis will yield a maximum and minimum stress for each type of stress: axial, bending, and torsion. This process might involve moving an applied force to the point of interest and creating resulting moments and torques [15].

After calculating the maximum and minimum for each stresses the alternating and mean effective stresses must be calculated. The alternating stress must then have various size, load, and stress concentration factors applied to it. This is necessary because these values are different for each loading mode. In addition, because these factors are applied to each stress they are not factored into endurance limit in the Marin equation. The following table shows how to apply these factors to each type of stress. [15]

Table 3-1 Stress factor for different stress [15]

Axial	$\sigma_{alt}=K_{f,a}*\sigma_{alt,a}/k_{c,a}$
Bending	$\sigma_{alt}= K_{f,b}*\sigma_{alt,b}/k_b$
Torsion	$\tau_{alt}= K_{f\,t}*\tau_{alt\,t}/k_b$

Where:

$K_{f,a}$ = Fatigue strength factor for axial

$K_{f,b}$ = Fatigue strength factor for bending

$K_{f\,t}$= Fatigue strength factor for torsion

$k_{c,a}$ = Load factor for axial

k_b= Size factor for bending

Therefore, generally the fatigue strength factor is mainly depends on endurance limit and endurance strength, fatigue stress concentration factor and combination of loading modes.

4.2 FEM Analysis

The created geometry model requires ANSYS software for theoretical analysis, simulation and analysis of given data to validate the axlebox house at field of application in this case fatigue life validation.

ANSYS software is considered to be the best optimization tool for the design of passenger axlebox house due to the validation of the design at the calculated life and avoiding sample testing in many cases. The software analysis is used to make sure or validate that no permanent deformation, instability or fracture of the axlebox house will occur in its life time due to fatigue under an exceptional design loads.

4.2.1 Overview of Fatigue Life Analysis In ANSYS Workbench

The ANSYS Fatigue Module has a wide range of features for performing calculations and presenting analysis results.[4]

Before going through the detail fatigue life analysis of the axlebox house it is important to define some of the listed technical terms used in this fatigue module analysis [3], [4],[8], [21], [28]

,[37]:

Classes of fatigue loading: Unlike static stress, which is analyzed with calculations for a single stress state, fatigue damage occurs when stress at a point changes over time. There are essentially four classes of fatigue loading:

i. Constant amplitude, proportional loading

ii. Constant amplitude, non-proportional loading

iii.Non-constant amplitude, proportional loading

iv.Non-constant amplitude, non-proportional loading

The second identifier, proportionality, describes whether the changing load causes the principal stress axes to change. If it does not change, then it is proportional loading. If the principal stress axes do change, then the cycles cannot be counted simply and it is non-proportional loading.

The non-proportional constant amplitude loading type for models that alternate between two completely different stress states (for example, between bending and torsional loading). Problems such as an alternating stress imposed on a static stress can be modeled with this feature. Non-proportional loading is applicable on fatigue tools under Solution Combination where exactly two environments are selected

Fatigue Strength Factor: Fatigue material property tests are usually conducted under very specific and controlled conditions. If the service part conditions differ from the as tested conditions, modification factors can be applied to try to account for the difference. The fatigue alternating stress is usually divided by this modification factor and can be found in design handbooks. (Dividing the alternating stress is equivalent to multiplying the fatigue strength by K_f.) Fatigue Strength Factor (K_f) reduces the fatigue strength and must be less than one. This setting is used to account for a "real world" environment that may be harsher than a rigidly-controlled laboratory environment in which the data was collected. Common fatigue strength reduction factors to account for such things as surface finish can be found in design handbooks.

Specify a modification factor of 0.8 since material data represents a polished specimen and the in-service component is cast. Modification factors to account for the differences between the in service part from the as tested conditions in design handbooks, the fatigue alternating stress is usually divided by this modification factor in the ANSYS Fatigue Module, the Fatigue Strength Factor (K_f) reduces the fatigue strength and must be less than one.

Fatigue Sensitivity: Fatigue Sensitivity shows how the fatigue results change as a function of the loading at the critical location on the model. This result may be scoped. Sensitivity may be found for life, damage, or factor of safety. The user may set the number of fill points as well as the load variation limits.

Fatigue Material Properties: Engineering data contains example materials which may include fatigue curves populated with data from engineering handbooks. New material data also could be added. In this case spheroidal cast iron EN-GJS 400-15, with EN1563 standard is used for the axlebox house.

Table 4-2 Spheroidal cast iron EN-GJS 400-15 material data[8]

Engineering Data Toolbox	Property	Value	Unit
Isotropic secant coefficient of thermal expansion	Density	7100	Kg/m3
	Coefficient of thermal expansion	1.1E-05	1/C
	Reference temperature	22	C
Isotropic Elasticity	Young's Modulus	1.69E11	pa
	Poisson's Ratio	0.275	
	Bulk Modulus	6.76E10	pa
	Shear Modulus	4.05E8	pa
Alternating Stress mean stress	Interpolation	Cycles Alternating Stress 10 1.018e9 20 8.7e8 50 7.08e8 100 6.05e8 200 5.17e8 2000 3.08e8 10000 2.14e8 20000 1.83e8 $1x10^5$ 1.27e8 $2x10^5$ 1.09e8 $1x10^6$ 7.58e7	pa
	Scale	1	
Strain-life parameters	Strength coefficient	11.99E08	
	Strength exponent	-0.183	
	Ductile exponent	0.202	
	Cyclic strength coefficient	1.061E09	
	Cyclic strain hardening exponent	0.114	
Strength	Tensile yield strength	2.5E08	pa
	Compressive yield strength	2.5E08	pa
	Tensile ultimate strength	4E08	pa

Chapter Five

5. Result, Discussion and Validation

5.1 FEM Analysis ANSYS Workbench Result

The result obtained from the primary and the secondary data with the application of Finite Element Method analysis software under different loading condition are illustrated by figures and graphs.

The maximum loading conditions are taken from calculated load summary table result which gives:

Maximum load in the vertical direction: 146.69KN

Minimum load in the vertical direction: 124.13KN

Lateral force: +/-33.5KN

Static structural toolbox analysis systems selected and the material date is inserted in the engineering data. The model geometry is selected and meshed with advance size proximity and curvature with fine relevance centre to have a better accurate solution.

The two ends are fixed supported and 124.13KN and 146.69KN vertical forces acted at the top center since the axlebox has to withstand the load by itself without any deformation. The second alternating force 33.5KN which act on the periphery of the axlebox house acted on the direction of y-axis also result total deformation and equivalent stress.

The bending and the torque force output will be combined in the solution combination to find the fatigue life and fatigue sensitivity in the fatigue tool. The result of each model evaluated by ANSYS Workbench at the finite life cycle is as follows:

Table 5-1 Model I geometry property

Bounding Box	
Length X	0.78768 m
Length Y	0.37337 m
Length Z	0.163 m
Properties	
Volume	1.6416e-002 m³
Mass	116.55 kg

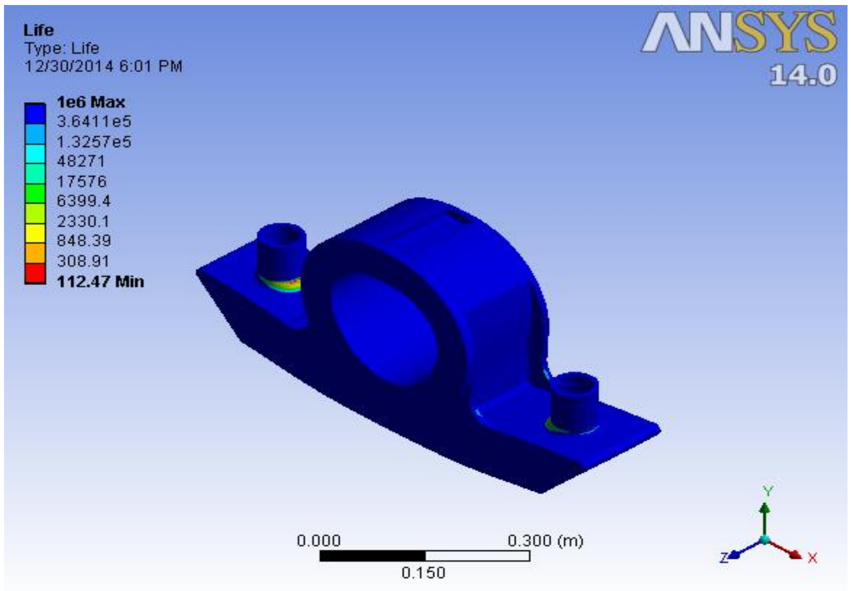


Figure 5-1 Model I finite life result with 0.5 fatigue strength factor

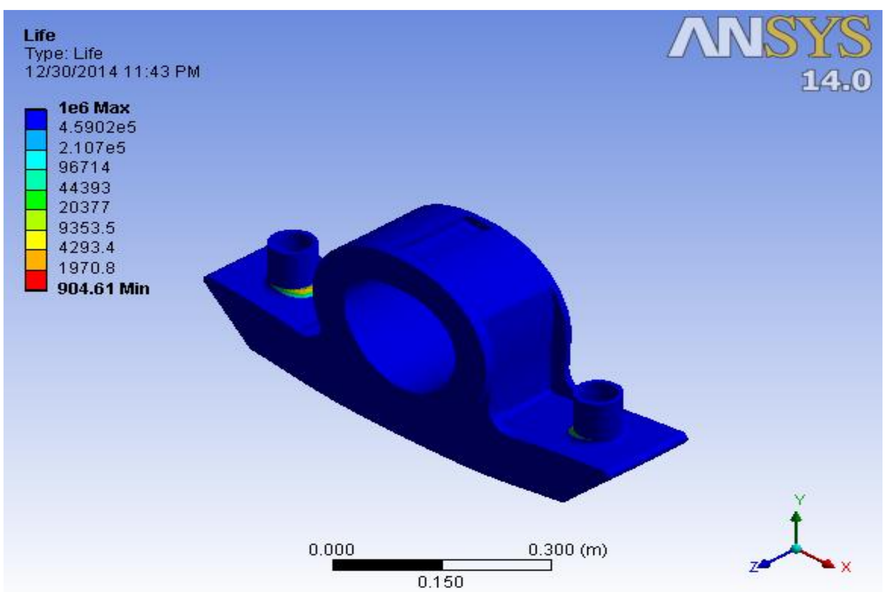


Figure 5-2 Model I finite life result with 0.8 fatigue strength factor

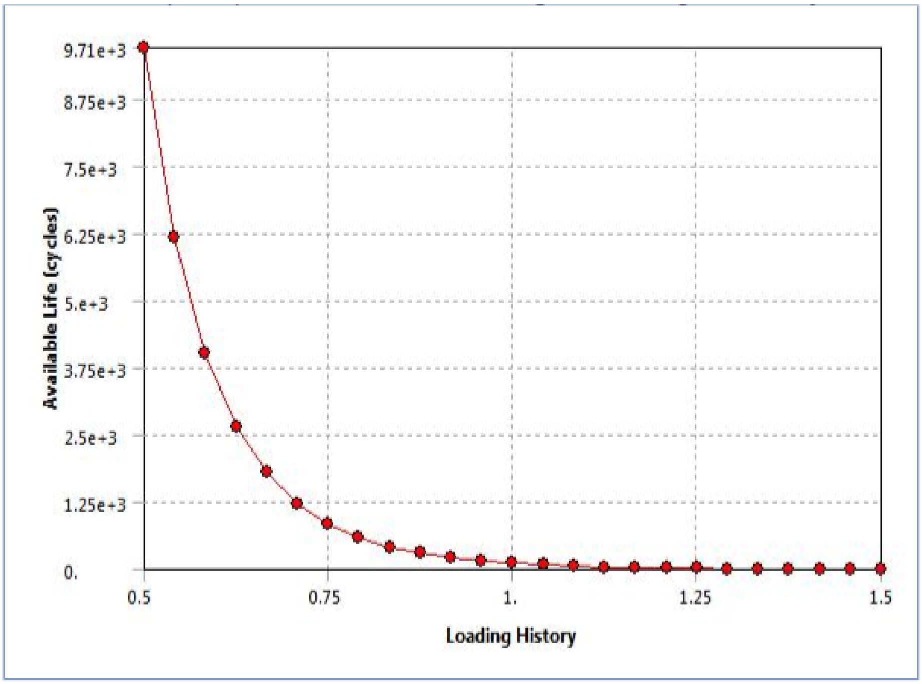


Figure 5-3 Model I fatigue sensitivity for life curve with 0.5 fatigue strength factor

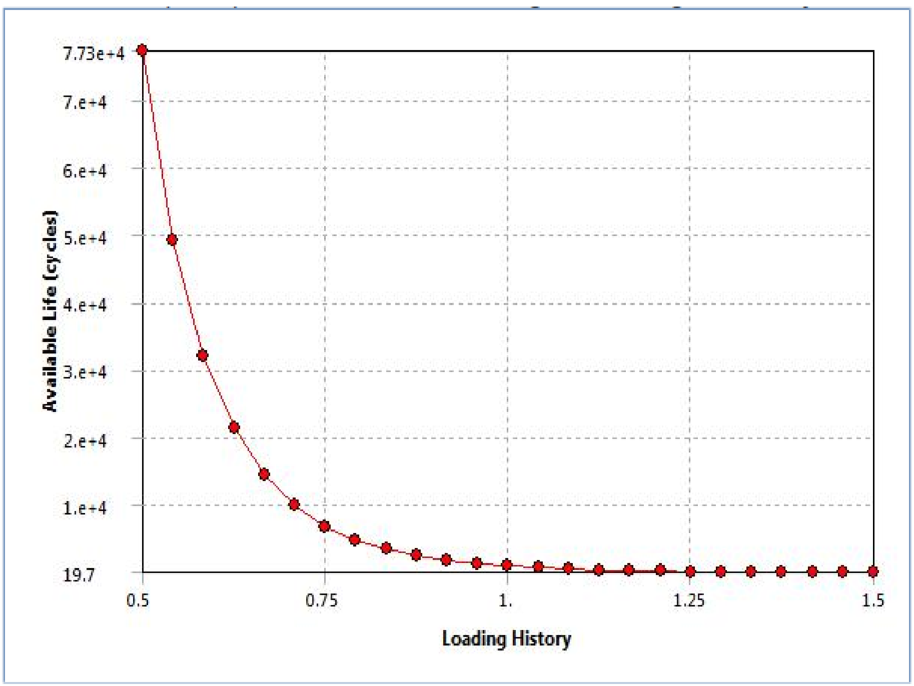


Figure 5-4 Model I fatigue sensitivity for life curve with 0.8 fatigue strength factor

Table 5-2 Modified Model I geometry property

Bounding Box	
Length X	0.78768 m
Length Y	0.37337 m
Length Z	0.163 m
Properties	
Volume	1.6511e-002 m³
Mass	117.23 kg

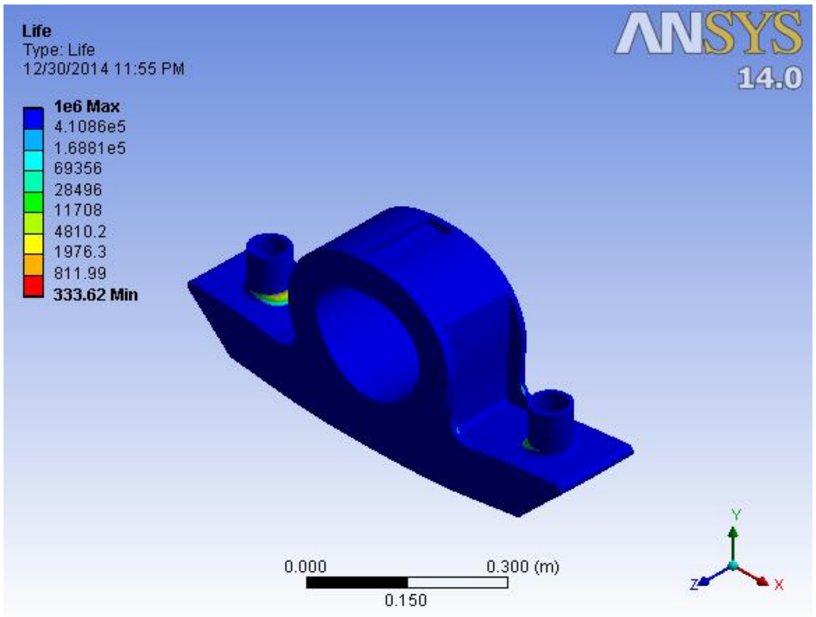


Figure 5-5 Modified Model I fatigue life with fatigue strength factor 0.5

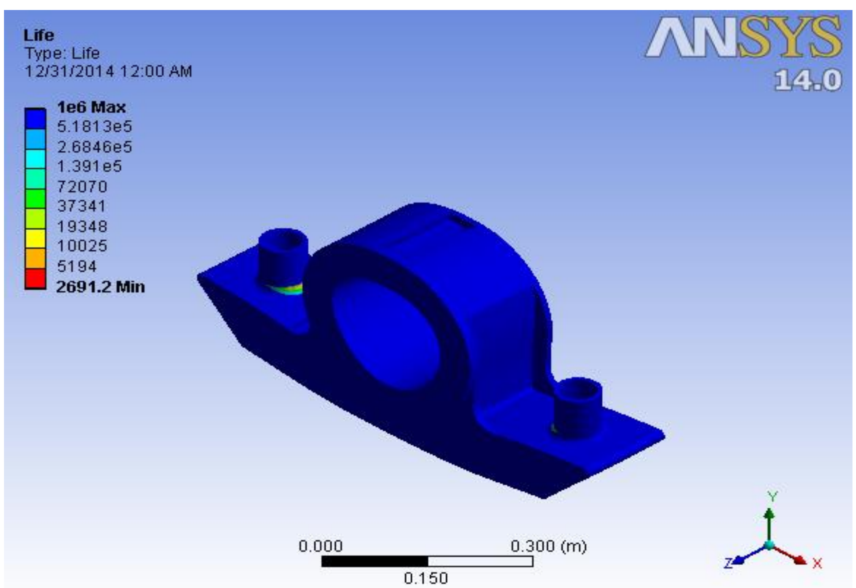


Figure 5-6 Modified Model I fatigue life with fatigue strength factor 0.8

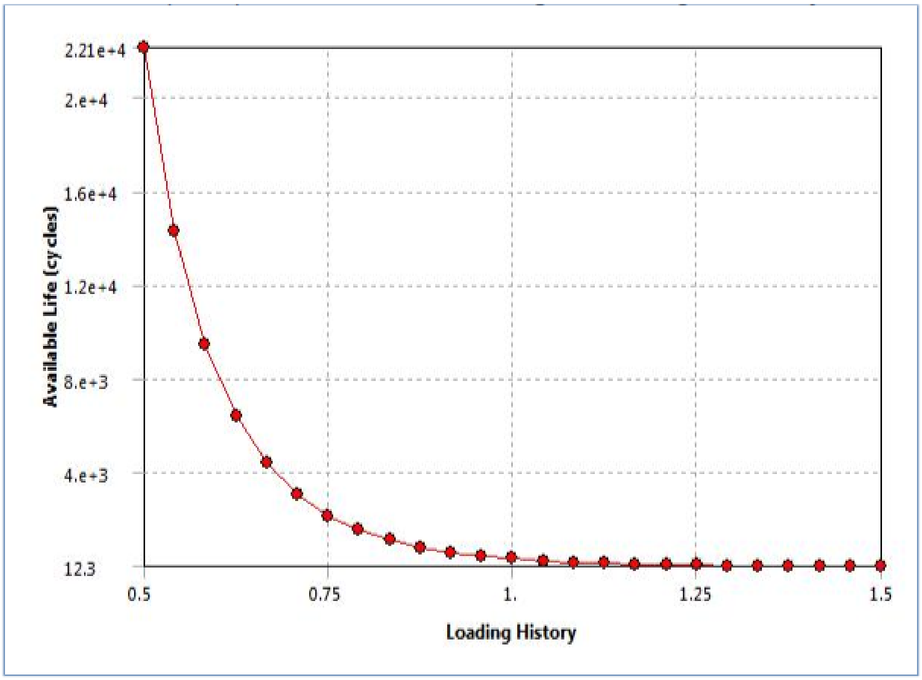


Figure 5-7 Modified Model I fatigue sensitivity of life result with fatigue strength factor 0.5

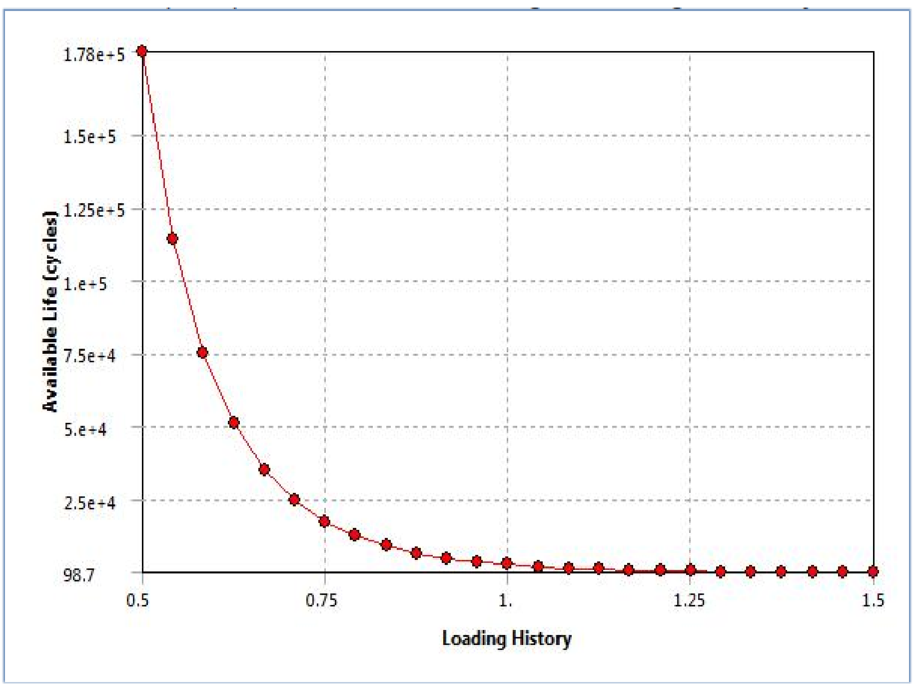


Figure 5-8 Modified Model I fatigue sensitivity of life curve with fatigue strength factor 0.8

Table 5-3 Model 2 geometry property

Bounding Box	
Length X	0.825 m
Length Y	0.3755 m
Length Z	0.173 m
Properties	
Volume	2.8559e-002 m³
Mass	202.77 kg

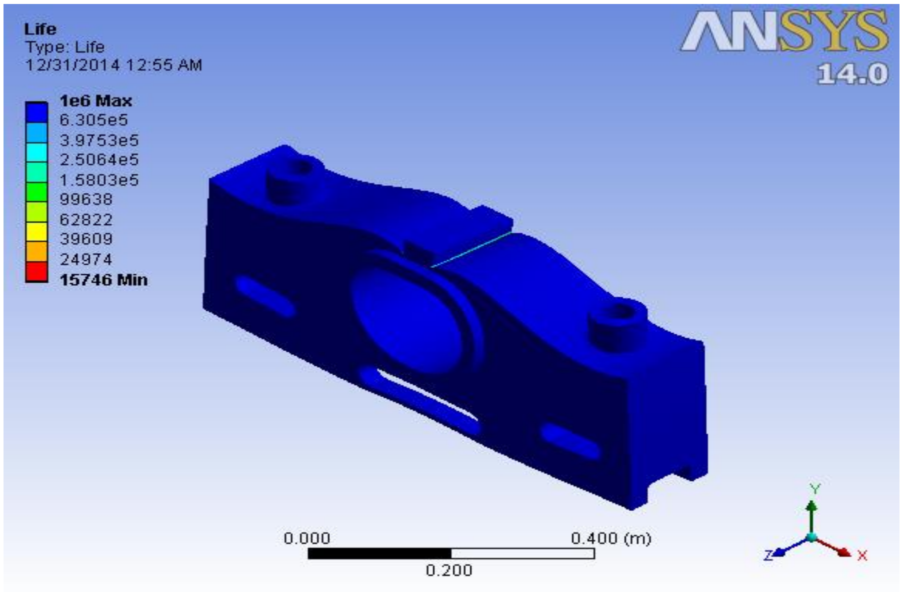


Figure 5-9 Model 2 finite life result with 0.5 fatigue strength factor

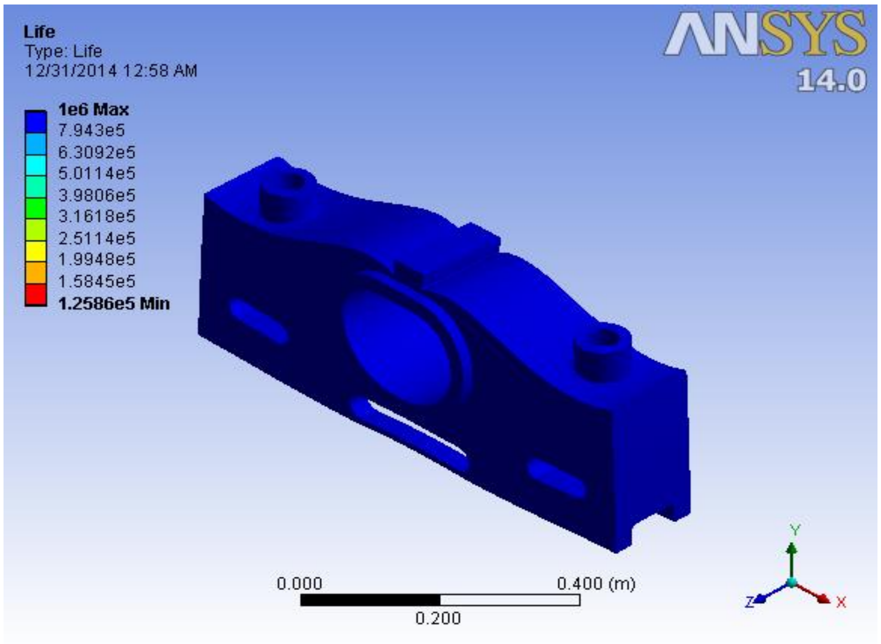


Figure 5-10 Model 2 finite life result with 0.8 fatigue strength factor

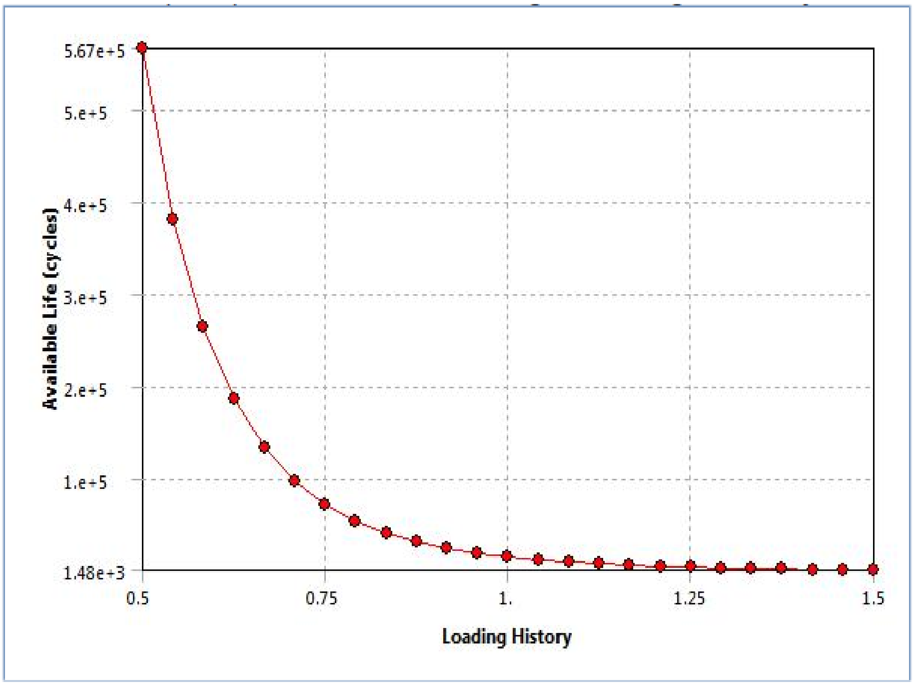


Figure 5-11 Model 2 fatigue sensitivity for life curve with 0.5 fatigue strength factor

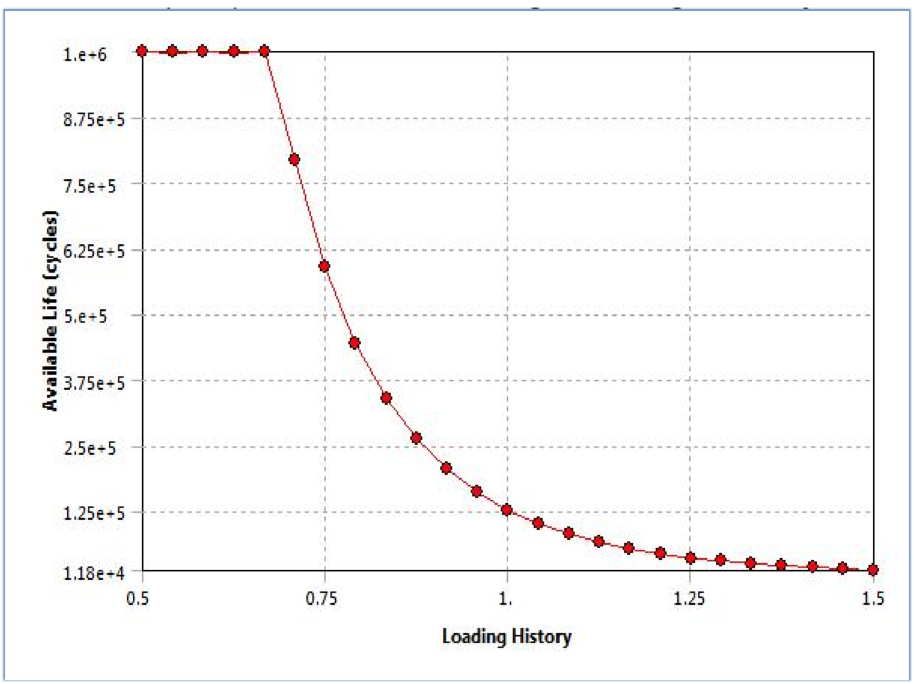


Figure 5-12 Model 2 fatigue sensitivity for life curve with 0.8 fatigue strength factor

Table 5-4 Modified Model II geometry property

Bounding Box	
Length X	0.825 m
Length Y	0.3855 m
Length Z	0.173 m
Properties	
Volume	2.8714e-002 m³
Mass	203.87 kg

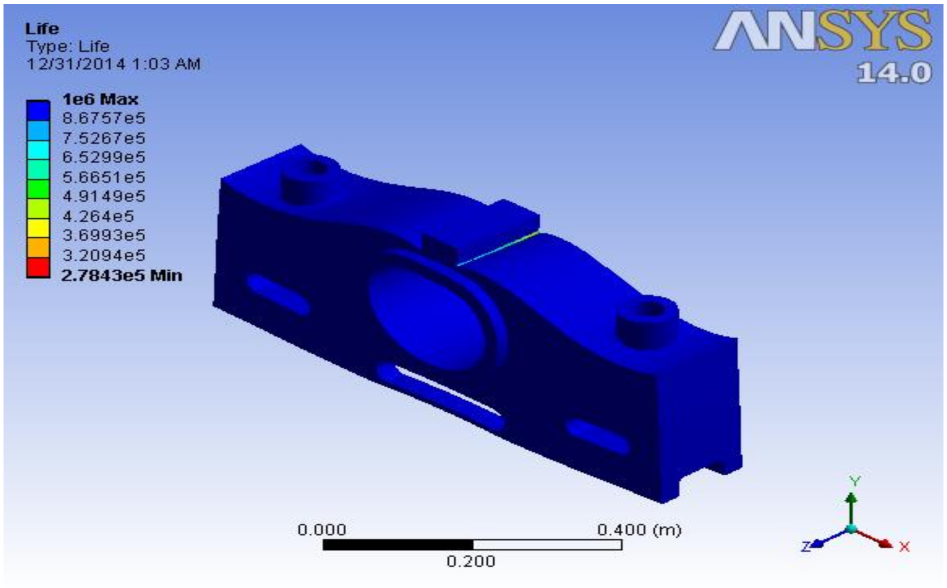


Figure 5-13 Modified Model II finite life result with 0.5 fatigue strength factor

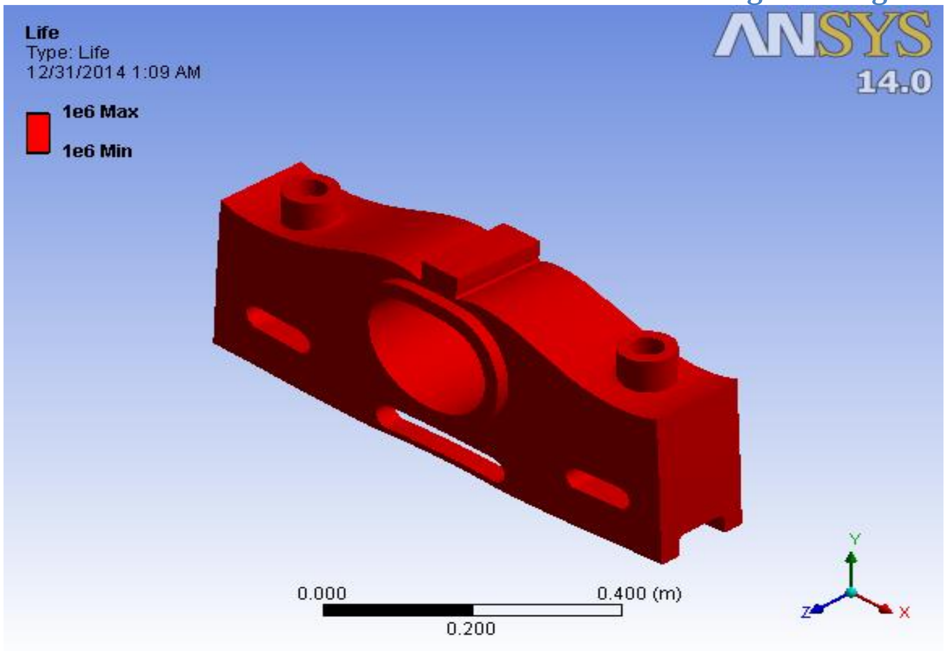


Figure 5-14 Modified Model II finite life result with 0.8 fatigue strength factor

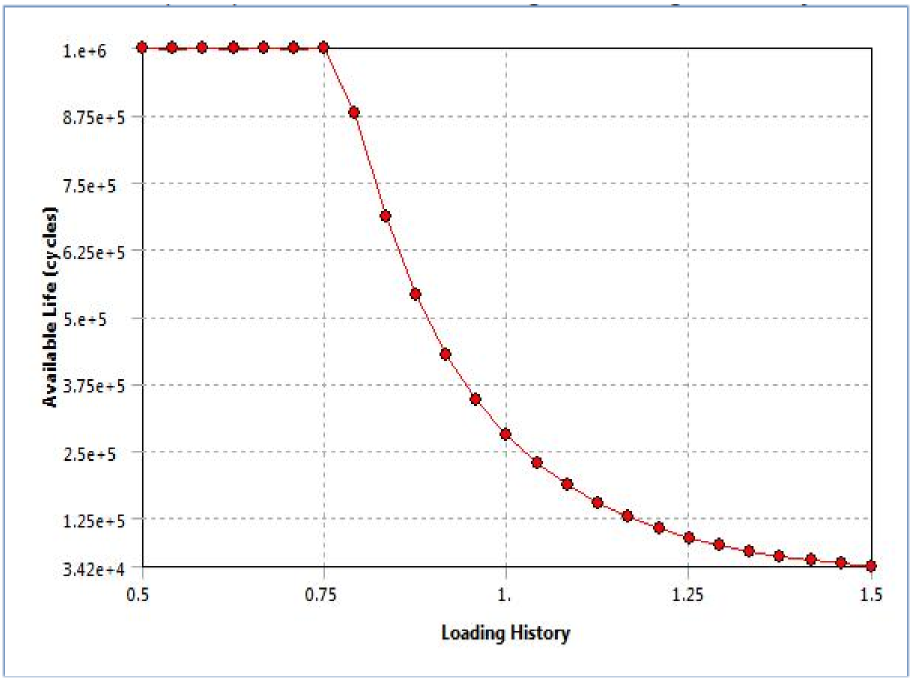


Figure 5-15 Modified Model II fatigue sensitivity for life curve with 0.5 fatigue strength factor

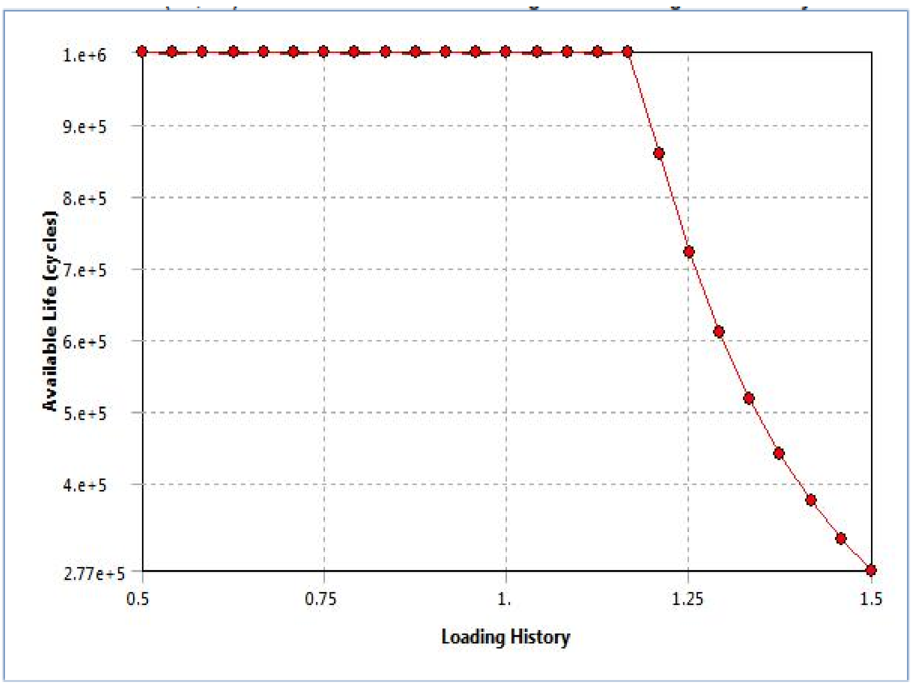


Figure 5-16 Modified Model II fatigue sensitivity for life curve with 0.8 fatigue strength factor

Table 5-5 Model III geometry property

Bounding Box	
Length X	0.78448 m
Length Y	0.314 m
Length Z	0.2 m
Properties	
Volume	6.4888e-003 m³
Mass	46.071 kg

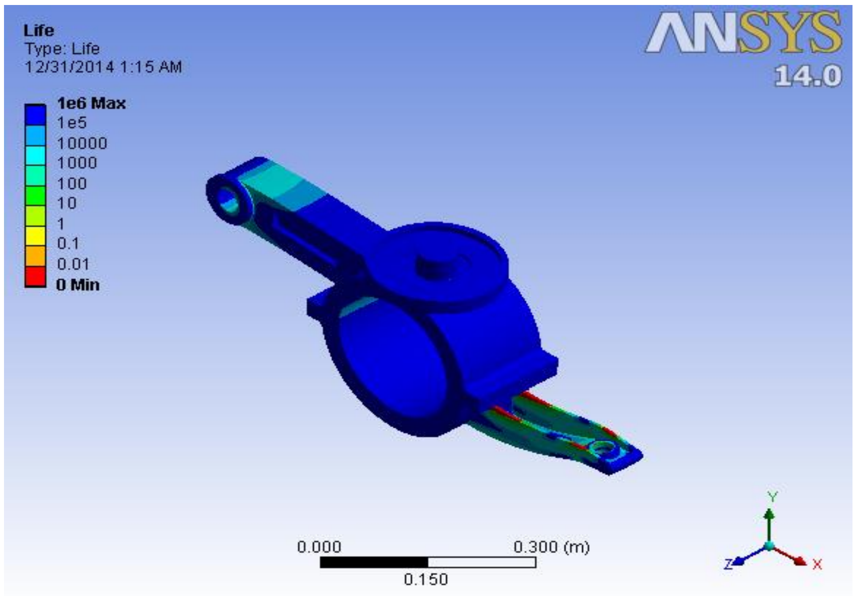


Figure 5-17 Model III finite life result with 0.5 fatigue strength factor

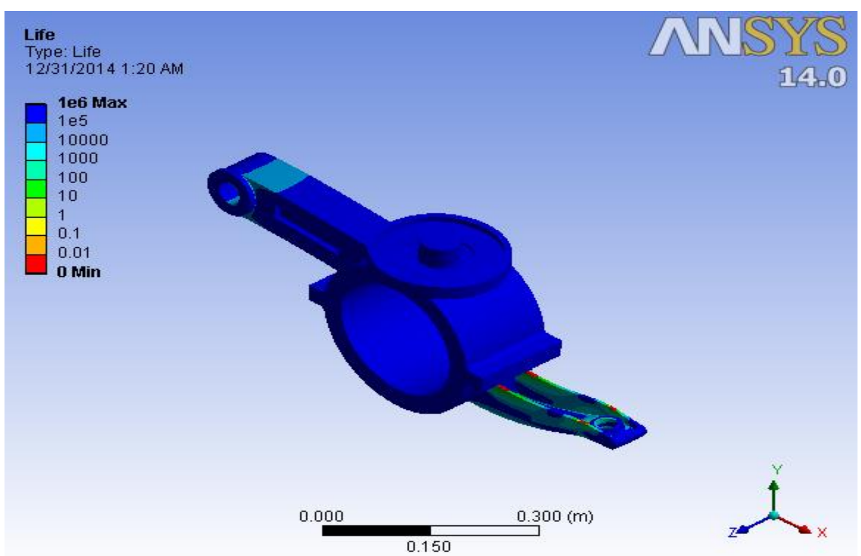


Figure 5-18 Model III finite life result with 0.8 fatigue strength factor

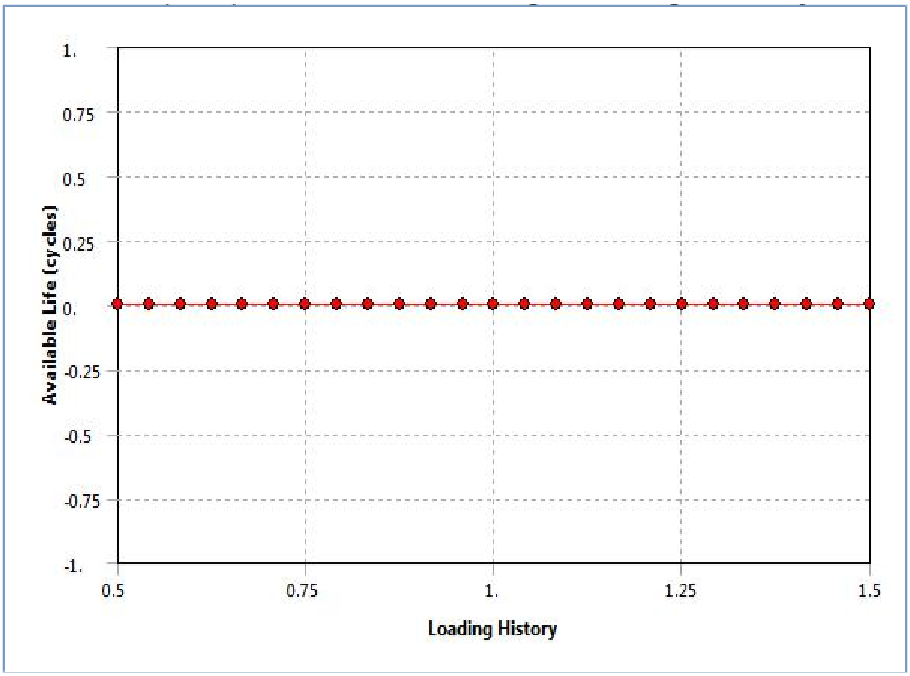


Figure 5-19 Model III fatigue sensitivity for life curve with 0.5 fatigue strength factor

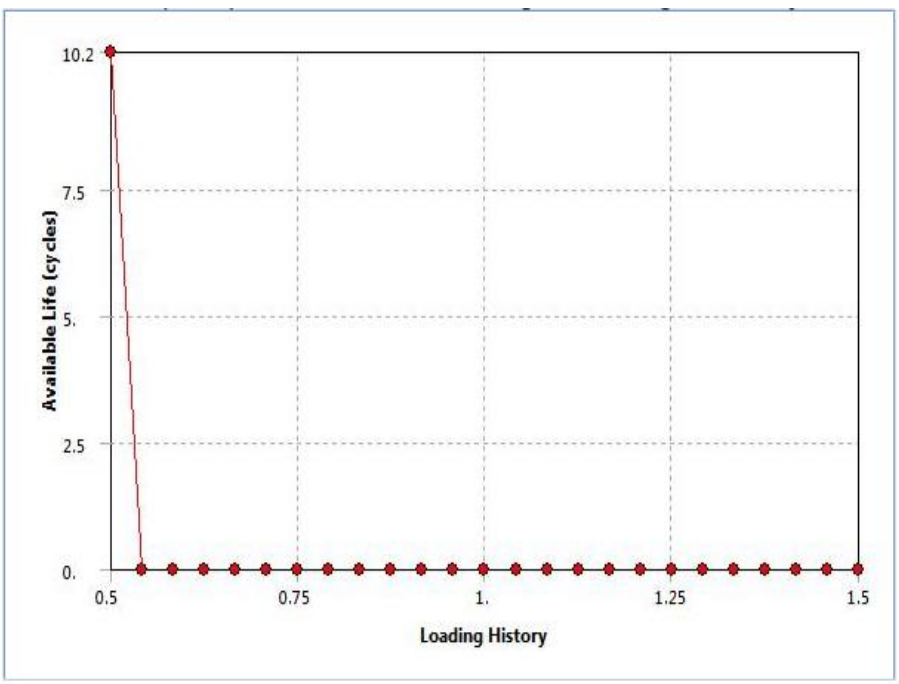


Figure 5-20 Model III fatigue sensitivity for life curve with 0.8 fatigue strength factor

Table 5-6 Modified Model III geometry property

Bounding Box	
Length X	0.78448 m
Length Y	0.314 m
Length Z	0.2 m
Properties	
Volume	6.907e-003 m³
Mass	49.04 kg

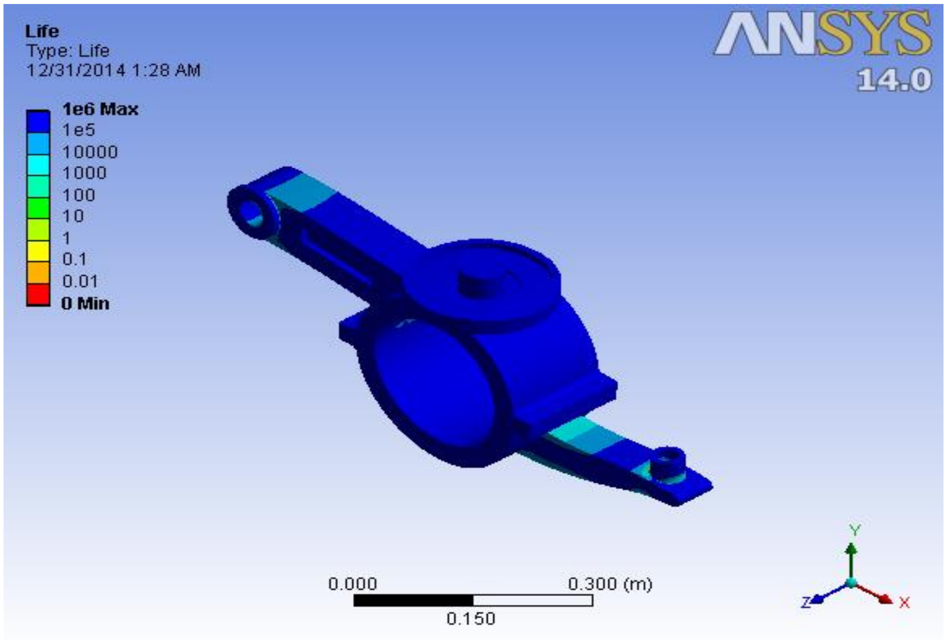


Figure 5-21 Modified Model III finite life result with 0.5 fatigue strength factor

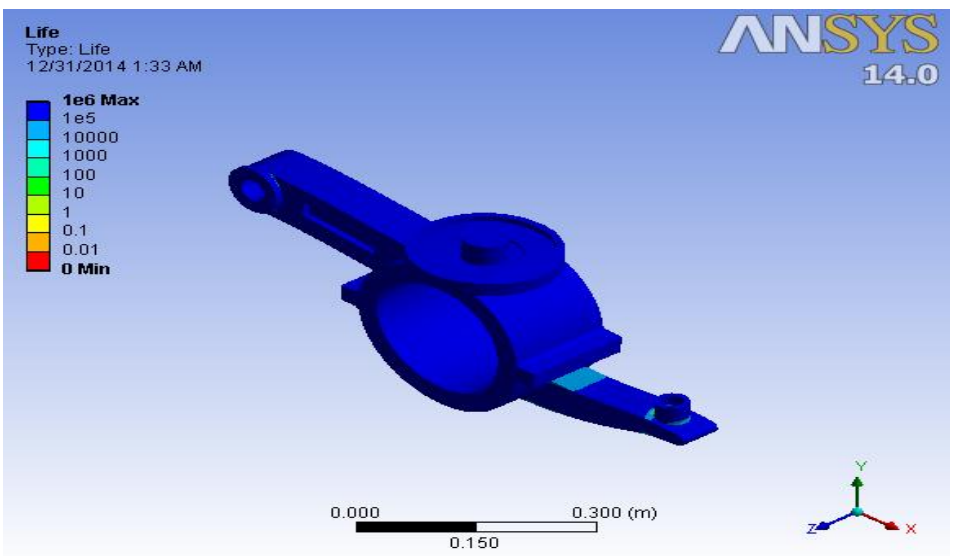


Figure 5-22 Modified Model III finite life result with 0.8 fatigue strength factor

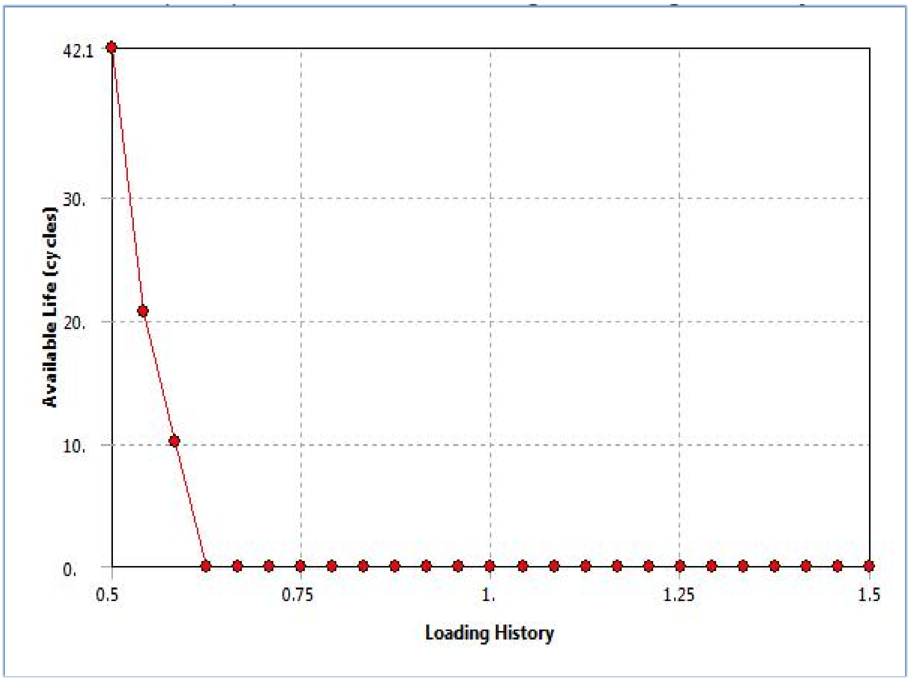


Figure 5-23 Modified Model III fatigue sensitivity for life curve with 0.5 fatigue strength factor

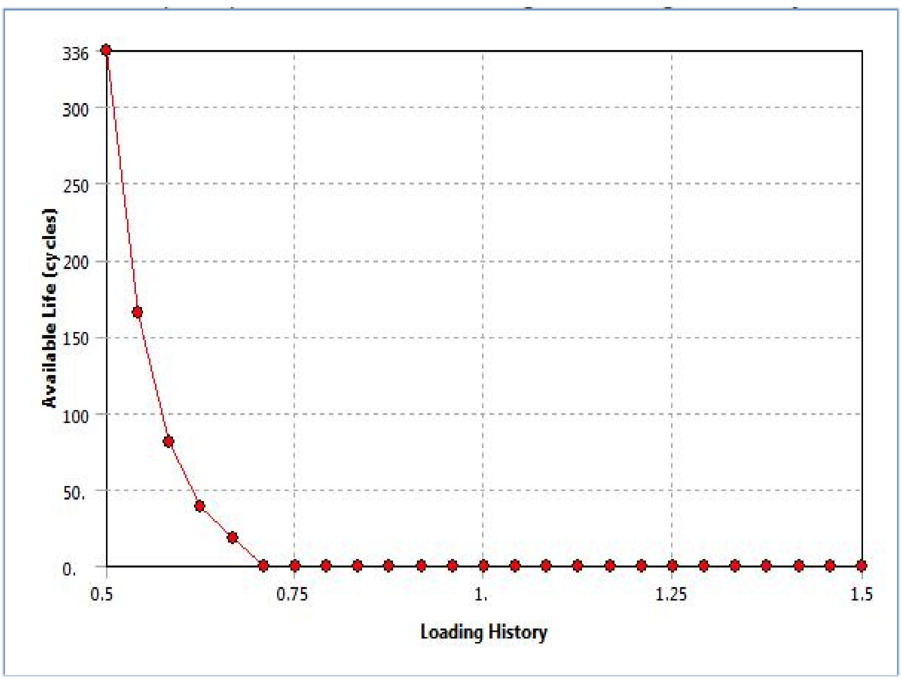


Figure 5-24 Modified Model III fatigue sensitivity for life curve with 0.8 fatigue strength factor

Table 5-7 Model IV geometry property

Bounding Box	
Length X	0.825 m
Length Y	0.397 m
Length Z	0.1724 m
Properties	
Volume	2.9723e-002 m³
Mass	211.03 kg

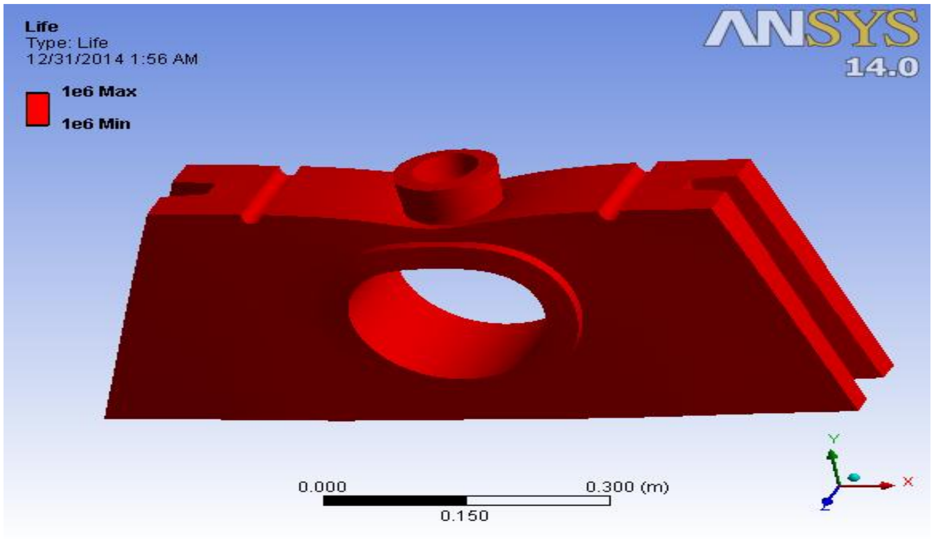


Figure 5-25 Model IV finite life result with 0.5 fatigue strength factor

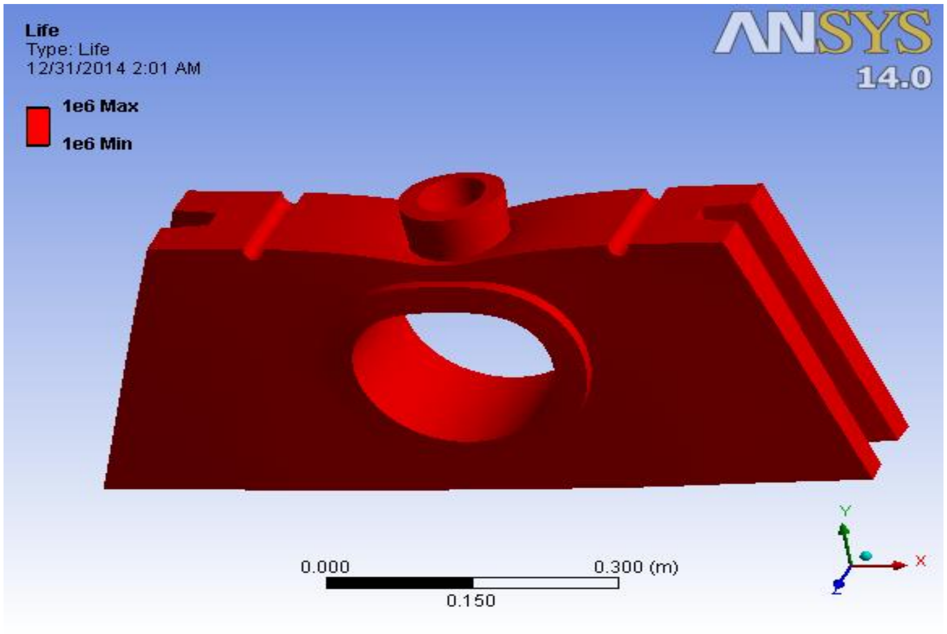


Figure 5-26 Model IV finite life result with 0.8 fatigue strength factor

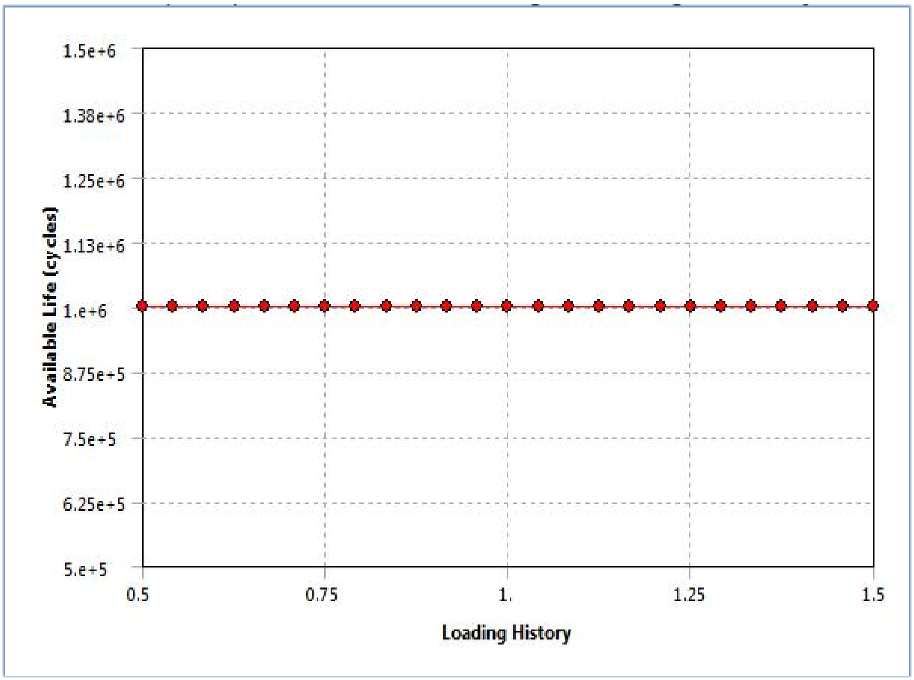


Figure 5-27 Model IV fatigue sensitivity for life curve with 0.5 fatigue strength factor

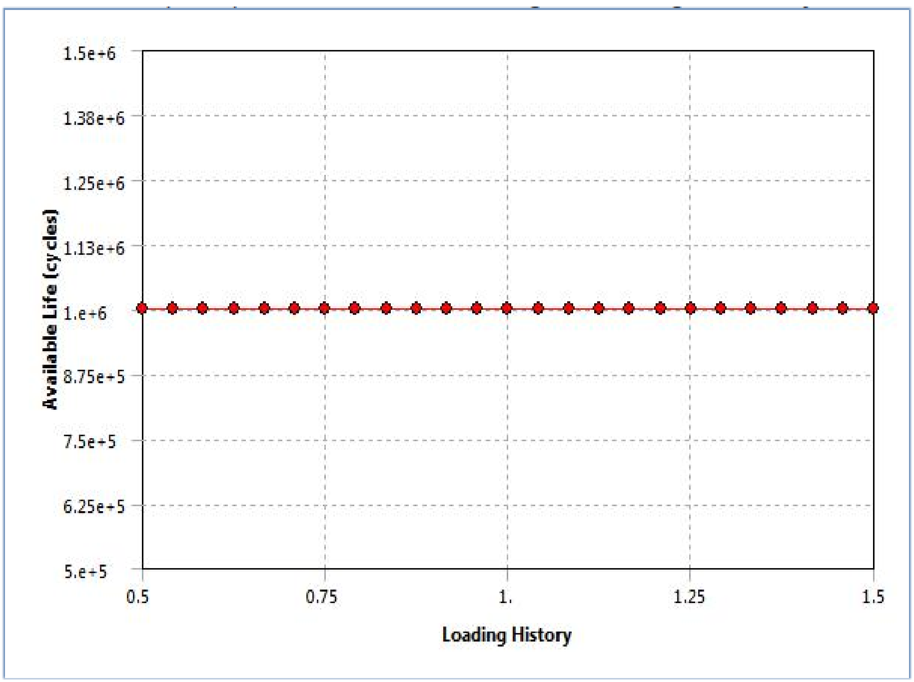


Figure 5-28 Model IV fatigue sensitivity for life curve with 0.5 fatigue strength factor

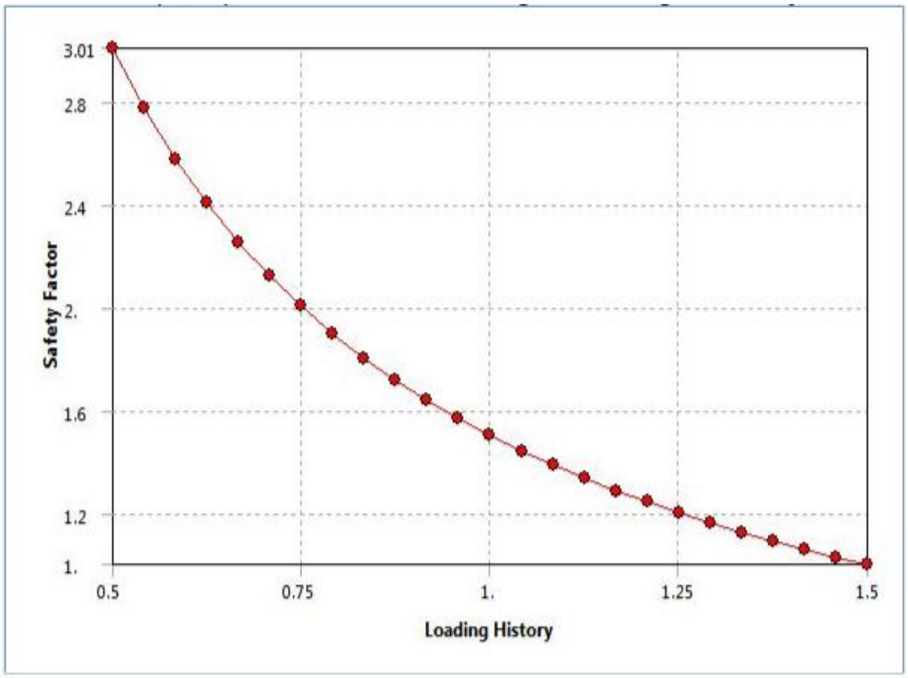


Figure 5-29 Model IV fatigue sensitivity for safety curve with 0.5 fatigue strength factor

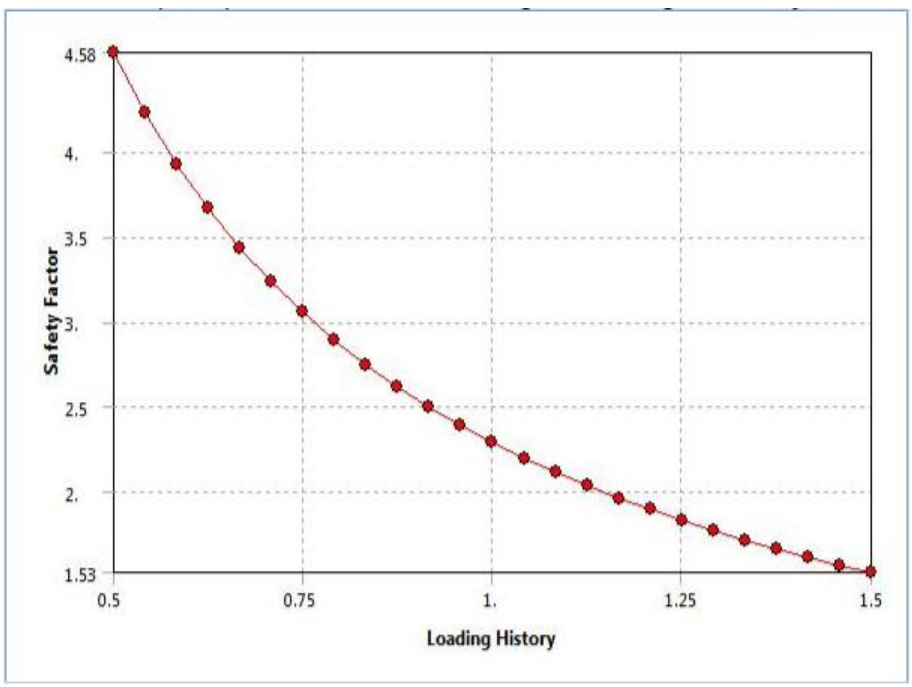


Figure 5-30 Model IV fatigue sensitivity for life curve with 0.8 fatigue strength

5.2 Result Discussion

In FEM analysis the geometry models are analysed under the same non-proportional loading, almost equivalent dimension and the same material property to investigate which model has a longer fatigue life.

Table 5-8 Ansys fatigue tools output result summary

Output Result	**Geometry Model**													
	Model I		Improved Model I		Model II		Improved Model II		Model III		Improved Model III		Model IV	
	at Kf 0.5	at Kf 0.8	at Kf 0.5	at Kf 0.8	at Kf 0.5	at Kf 0.8	at Kf 0.5	at Kf 0.8	at Kf 0.5	at Kf 0.8	at Kf 0.5	at Kf 0.8	at Kf 0.5	at Kf 0.8
Mass [kg]	116.55	116.55	117.23	117.23	202.77	202.77	203.87	203.87	46.07	46.07	49.04	49.04	211.03	211.03
Minimum Fatigue Life[cycle]	112.47	904.61	333.62	2691.2	15746	1.2586×10^5	2.7843×10^5	1×10^6	0	0	0	0	1×10^6	1×10^6
Fatigue sensitivity for life at 50% lower variation loading	9.71×10^3	7.73×10^4	2.21×10^4	1.78×10^5	5.67×10^5	1×10^6	1×10^6	1×10^6	0	0	42.1	336	1×10^6	1×10^6
at 150% upper variation loading	0	19.7	12.3	98.7	1.48×10^3	1.18×10^4	3.42×10^4	2.77×10^5	10.2	0	0	0	1×10^6	1×10^6
% of life improve due to geometry variations	99		99		98		0		100		100		0	
% Minimum Life Improved due to K_f	87.6		87.6		87.4		87.4		0		0		0	

As we can see in the result output fatigue strength factor, geometry variations, geometry improvement at critical cross-section are parameters generally dominate on the fatigue life of axlebox house and their relationships with the fatigue life are summarized in Table 4.11 and independently in the following charts:

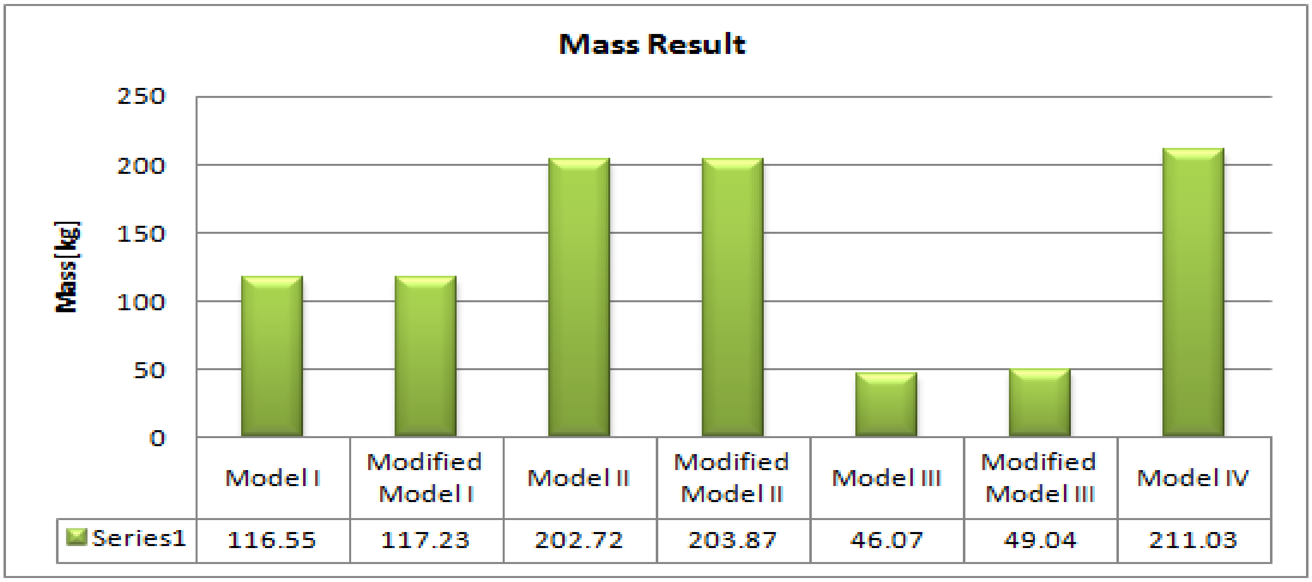


Figure 5-31 Mass result chart

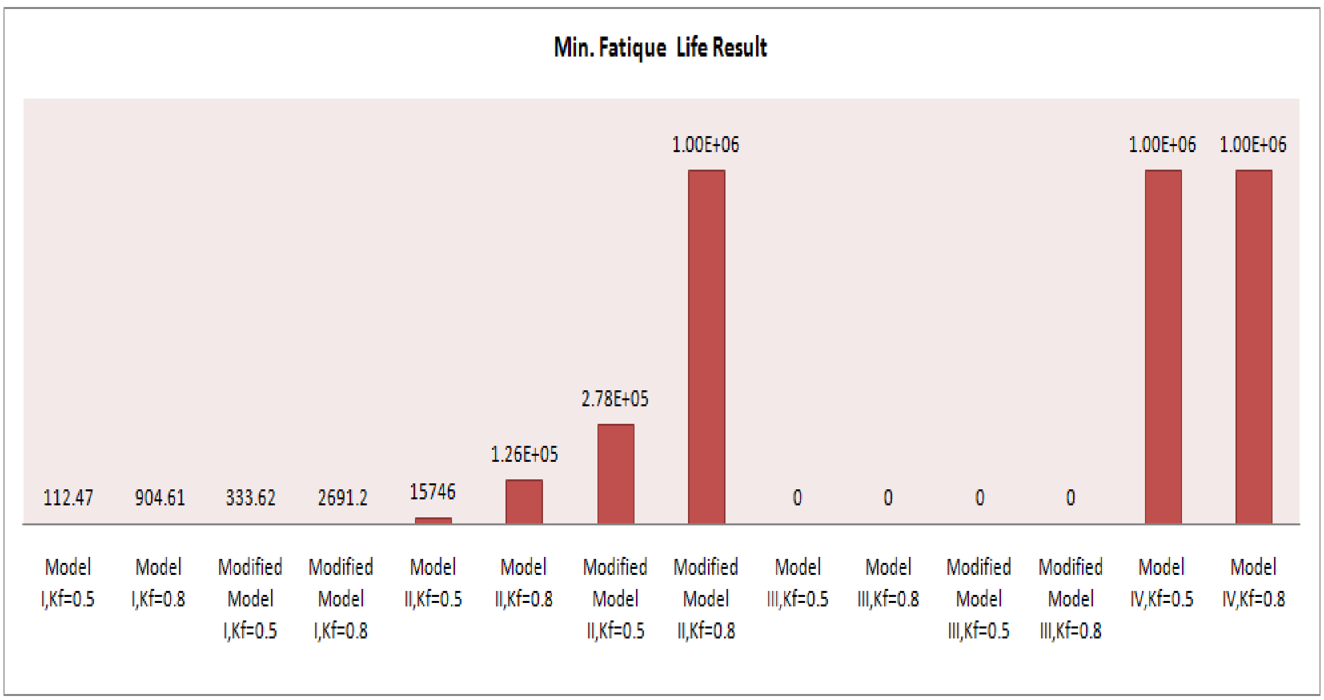
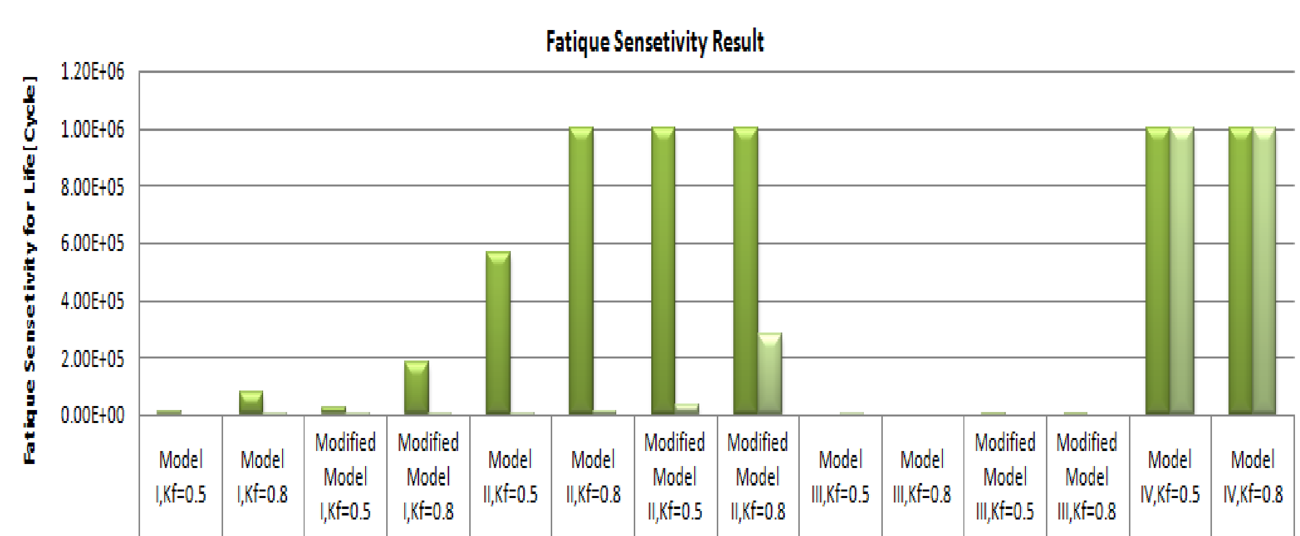


Figure 5-32 Minimum Fatigue life result for each Model



	Model I,Kf=0.5	Model I,Kf=0.8	Modified Model I,Kf=0.5	Modified Model I,Kf=0.8	Model II,Kf=0.5	Model II,Kf=0.8	Modified Model II,Kf=0.5	Modified Model II,Kf=0.8	Model III,Kf=0.5	Model III,Kf=0.8	Modified Model III,Kf=0.5	Modified Model III,Kf=0.8	Model IV,Kf=0.5	Model IV,Kf=0.8
50% loading	9.70E+03	7.73E+04	2.21E+04	1.78E+05	5.67E+05	1.00E+06	1.00E+06	1.00E+06	0	0	42.1	33.6	1.00E+06	1.00E+06
150% loading	0	19.7	12.3	98.7	1.48E+03	1.18E+04	3.42E+04	2.77E+05	10.2	0	0	0	1.00E+06	1.00E+06

Figure 5-33 Fatigue sensitivity for life result

Improved Model II at fatigue strength factor K_f, 0.8 and Model IV have good fatigue life and fatigue sensitivity for life compared to other models due to their geometry even if their mass is higher than other models. The result clearly shows that improving the fatigue strength factor and geometry variance improve the fatigue life of the axlebox house .In this thesis scenario by improving the fatigue strength factor can improve the life up to 50% and suitable geometry selection can improve the fatigue life up to 70%.

It is also worth to mention from the result that in comparison to the geometry mass and minimum fatigue life, as the geometry improved for minimum fatigue life, the mass is increased by 0.016% which is very small in considerations to our desire life but the mass value has significant effect the railway operation.

AALRT axlebox type Model IV have the best fatigue life even though its mass is higher than the other model .The service life of this model could also be farther improved by correcting the fatigue strength factor .As we can see also from the result by improving the K_f from0.5 to 0.8 for this model, the fatigue sensitivity for safety factor is increased from 1 to 1.53 at a150% upper loading variation.

5.3 Result Validation

The FEM using ANSYS software workbench result shows minimum life at notched geometry area and at stress concentrated cross-sectional area.

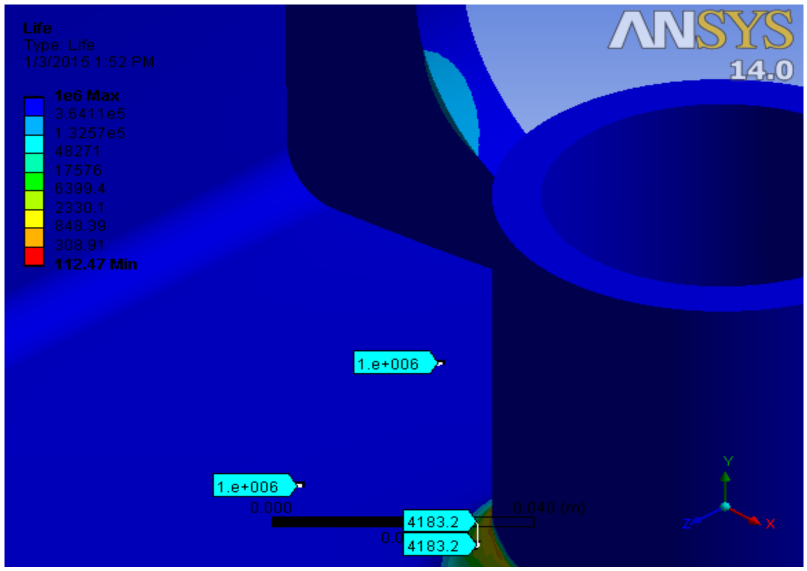


Figure 5-34 Magnified Model I around some cross section

FEM result extracted from SKF, Railway technical handbook [33] also demonstrates that stress is concentrated around notched area where minimum fatigue life is manifested. The experimental test result also shows an inverse relationship response of life to load.

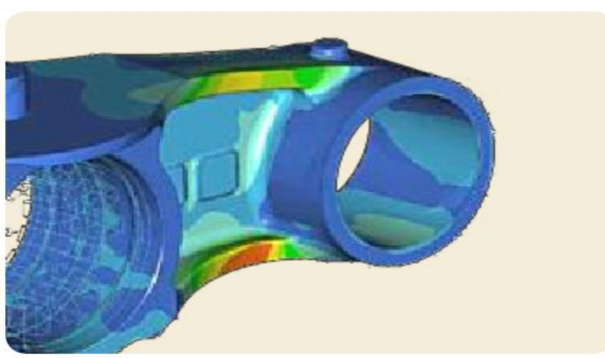


Figure 5-35 FEM results for a link arm axlebox housing [33]

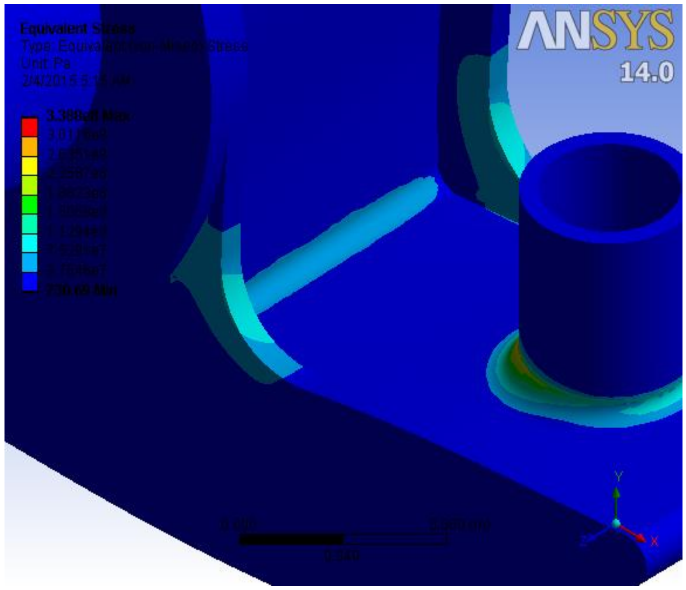


Figure 5-36 Combined Von Mises stress result of Model I

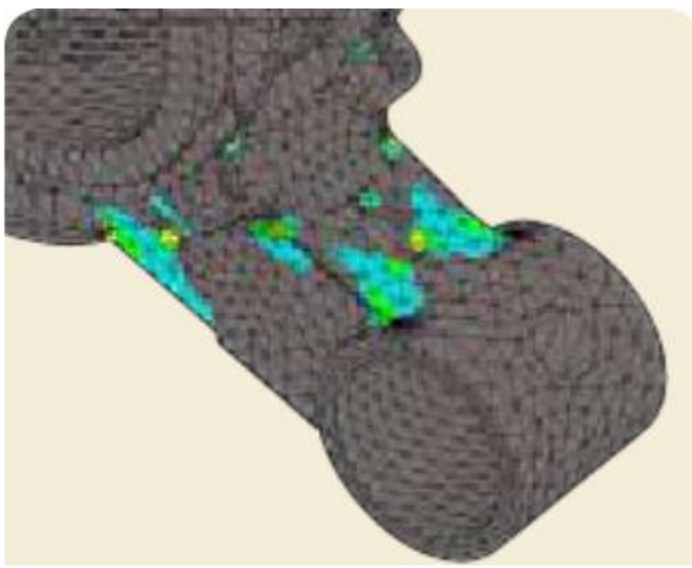


Figure 5-37 Visualization of the fatigue post processing results[33]

It is to be noted that the experimental model indicated on figure 5-37 is not quite similar to the model I, but could demonstrate similar future with relevance to the study.

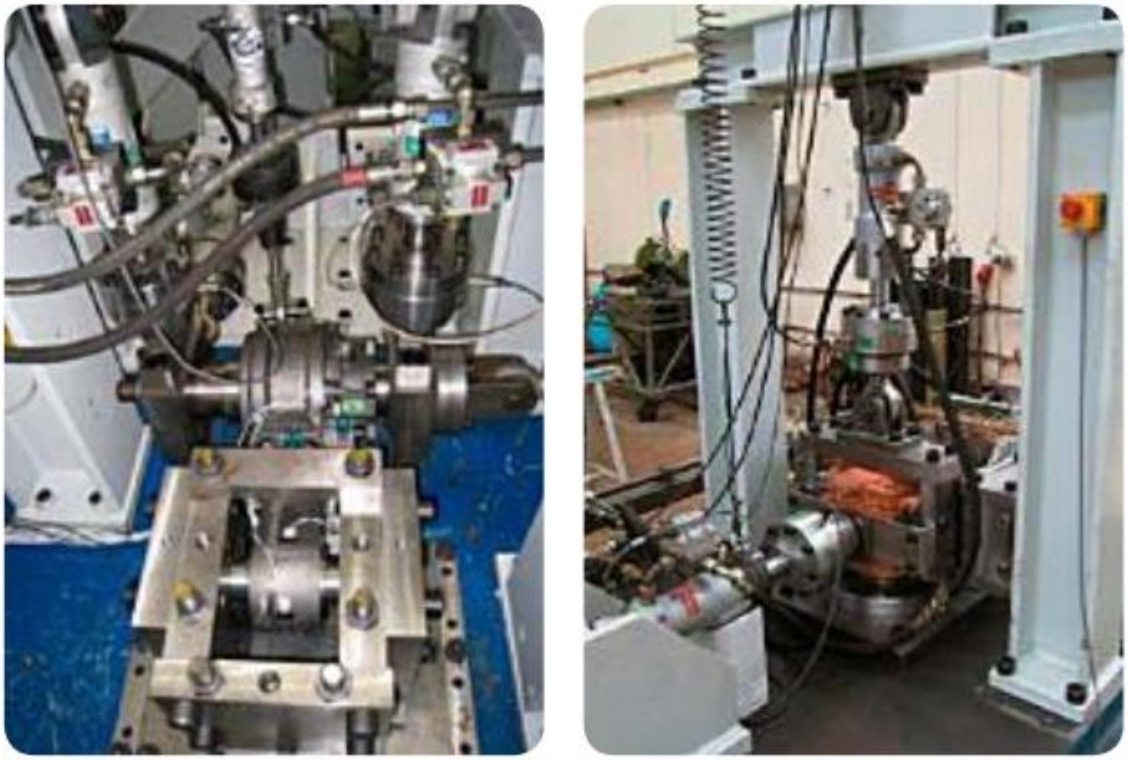


Figure 5-38 Axlebox house fatigue testing bench [33]

General, axlebox testing is in accordance with UIC 615-4 is also shown on fig.5-39 explains that as the magnitude of load increase the working cycle of the axlebox house decrease:

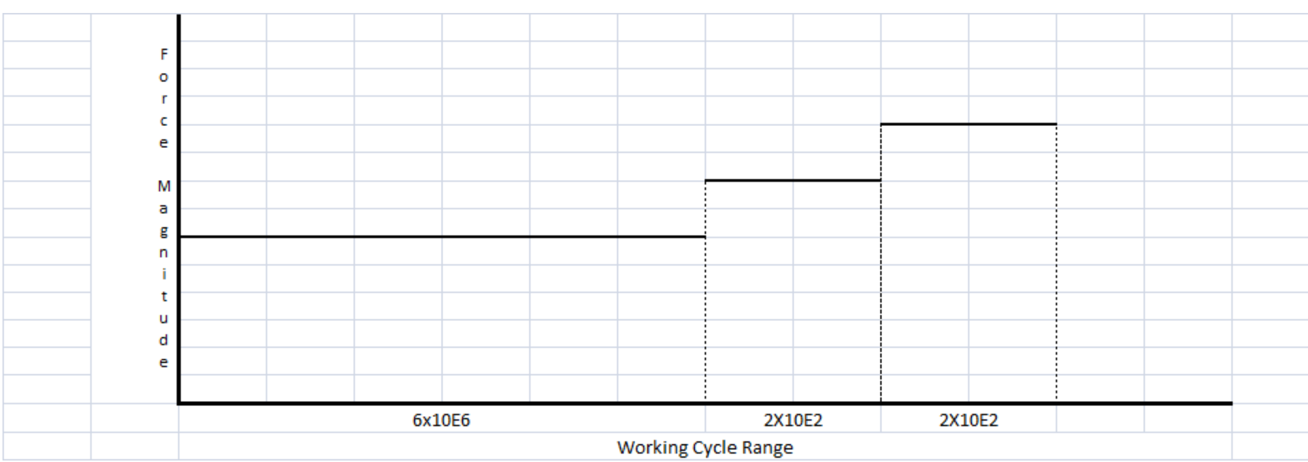


Figure 5-39 Quasi- static load cycles normally reversed every 10 to 20 dynamic cycles[33]

The total deformation of each model also checked and does not have significant effect on the bearing and the wheel rail clearance. And it is shown that Model IV is by far the most durable and high loading resistance axlebox house on this scenario. This means that the other models are not disregarded. Those models could have efficient performance when the speed is increased and compatible bogie frame structure and attachments altered.

A similar approach of Railway technical hand book [33] indicates the results of equivalent stress history and fatigue post presses figural representation on the selected axlebox house similar to Model I also demonstrate that the stress are concentrated on notched and maximum loading cross-section.

Chapter Six

6. Conclusion and Recommendation

6.1 Conclusion

Passenger rail vehicle axleboxe house have always been a vital component in the reliability of railway rolling stock and has considerable influence on the operating safety, reliability and economics of railways.

It has been also described in the introductory part that up to 90% of all structural failure occurs through a fatigue mechanism costs a lot of money and loss of life. Therefore the research is focused on to improve the fatigue life of passenger rail vehicle axlebox house in consideration to geometry and fatigue strength factor parameters.

The research is done by considering various adapted models of axlebox house on passenger rail vehicle .The models are developed by Solidwork software then each model is analysed with ANSYS workbench by correcting some parts of the stress concentration zone and improving the fatigue strength factor.

The first chapter is the introduction part where the background of the project, the objective, problem of the study, and the significance of the research, scope of the study and limitation of the study is justified. In the second part, theoretical literatures are reviewed while the third chapter presents the research methodology. The fourth chapter has dealt with fatigue life terminologies and philosophies and the fifth chapter presents results, discussion and validation. In its last part conclusion and recommendations are given. This outlined structure is selected for the parametric study to improve the fatigue life of passenger rail vehicle axlebox house.

Even if unavailability of experimental room facility and additional environmental factors renders the study for farther investigation the thesis gives a significant contribution by indicating that improving the geometry model, fatigue strength factors and eradicating stress concentration zone improves the fatigue life of passenger rail vehicle axlebox house for safe and smooth running survive under cyclic load during its lifetime. In addition the research outcomes have a significant importance for the AA LRT project and for other extension ERC projects by avoiding early failures of the axlebox house which leads catastrophic accidents such as derailment, wear, high thermal stress, bearing misalignment and operation and maintenance cost could also be reduced by locally manufacturing in the future.

6.2 Recommendation

This thesis shows that the passenger axlebox house fatigue life is improved by modelling suitable axlebox house, improving geometry factors such as the stress concentration area on notches and grooves and improving the fatigue strength factors.

But additional factors such as environmental conditions, the effect of thermal creep, contact property between the axlebox house attachments like dumping units, material microstructure and manufacturing process do have also an influence on the fatigue life of the axlebox house. By research and implementing those additional factors, loading variable and boundary condition farther research could be made to select the best suitable and durable axlebox hose for passenger rail vehicle axlebox house for the application of Ethiopian Railway Corporation future demand. In addition the ERC can have a capacity to manufacture this component and by doing so saves operating cost, maintenance cost and time.

Moreover, the mass effect of the Model IV shall be farther improved since it has significant loading impact regardless to its fatigue life on the axle as well as on the wheel rail interaction and across the slipper.

Therefore, others could extend this research by modelling different and suitable axlebox house in consideration to all the additional variables stated above i.e environmental conditions, the effect of thermal creep, contact property between the axlebox house attachments, material, material microstructure, mass, etc by conducting software simulation, experimental and on site testing.

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Appendices

Appendix A. East-west and South-north line AALRT and SF1000 Technical Data [7], [29]

A.1 Basic technical conditions of vehicles

Vehicle Parameter Description	Parameter Value
Train formation:	Mc+Tp+Mc
Mc module:	Motor car with driver's cab
Tp module:	Trailer without driver's cab with pantograph
Track gauge:	1435mm
Minimum radius of horizontal curve:	50m
Minimum radius of vertical curve:	1000m
Maximum gradient:	55‰
Axle load:	11(1+3%）t
Length of Car body:	≤3000mm
Height of vehicle from the top of rail excluding pantograph	≤3700mm
Maximum Width of car body	2650mm
Wheelbase(power bogie)	1900mm
Wheelbase(unpowered bogie)	1800mm
Wheel diameter(new wheel)	≤660mm
Wheel diameter(Max.wear)	≤600mm

A.2 Rated passenger capacity

Number of passengers (persons)	Seated	Standing	Total
Seats (AW_1)	65	0	65
Rated passenger capacity (AW_2) (standing: 6 persons/m^2)	65	189	254
Overload capacity (AW_3)(standing: 8 persons/m^2)	65	252	317

A.3 Weights of vehicles

Loads	Car body weight	Passenger weight	Total weight
Empty vehicle (t)	44	0	44
Rated passenger capacity (t)	44	15.24	59.24
Overload capacity (t)	44	19.02	63.02

- 60kg as average weight of each passenger is taken

A.4 Power bogie data

Parameter Description	**Parameter Value**
Maximum Operating Speed	70km/hr
Maximum Tested Speed	80km/hr
Axle load	11(1+3%)t
Track gauge	1435mm
Wheel diameter	660mm new wheel/580 full wear
Axle box bearing	Taper roller bearing unit
Motor car drive mode	Double reduction gear
Brake rigging	Disc brake+magnetic track brake
Suspension	Primary rubber spring, secondary steel spring
Damping	Lateral and vertical oil damper

A.5 SF1000 Technical Data

Description	Technical Data value
Bogie	SF 1000
Type	Motor and trailer bogies
Running speed	80km/h
Axle load	13t
Continuous power per wheelset	14/190kw
Wheelbase	2100mm
Track gauge	1435mm`
Wheel diameter new/worn	850/770mm
Smallest radius of curvature in service /workshop	90/70mm
Weight	6.7/5.0t
Bogie height	840mm

Appendix B. Loading condition of Passenger Rail Vehicle[25]

B.1 Loading Condition of Vehicles

Category	Exceptional payload P_1	Service (Fatigue) payload P_2
B-I - Main Line Passenger Rolling Stock	1 passenger per seat. 4 passengers / m^2 in passageways, access and service areas.	1 passenger per seat. Up to 2 passengers / m^2 in passageways, access and service areas.
B-II - Inner and Outer Suburban Passenger Rolling Stock	1 passenger per seat. 5 to 10 passengers / m^2 in passageways and access areas. 300 kg/m^2 in luggage areas.	1passenger per seat. Up to 6 passengers / m^2 in passageways and access areas. 300 kg/m^2 in luggage areas.
B-III - Metro and Rapid Transit Rolling Stock	1 passenger per seat. 5 to 10 passengers / m^2 in passageways and access areas.	1 passenger per seat. Up to 6 passengers / m^2 in passageways and access areas.
B-IV - Trams	1 passenger per seat. 6 to 8 passengers / m^2 in passageways and access areas. Passenger mass = 70 kg to 75 kg.	1 passenger per seat. Up to 6 passengers / m^2 in passageways and access areas. Passenger mass = 70 kg to 75 kg.
B-V & B-VI - Freight Rolling	Maximum payload.	Maximum payload.
B-VII - Locomotives	Zero payload, i.e. vehicle in normal working order with all	Zero payload, i.e. vehicle in normal working order with all supplies.
NOTE Passenger mass includes hand luggage.		

B.2 Load cases for static test corresponding to vertical and transverse force combinations

Load case	F_{z1}	F_{z2}	F_y
1	$F_z / 2$	$F_z / 2$	0
2	$(1 + \alpha - \beta) F_z / 2$	$(1 - \alpha - \beta) F_z / 2$	0
3	$(1 + \alpha - \beta) F_z / 2$	$(1 - \alpha - \beta) F_z / 2$	$+ F_y$
4	$(1 + \alpha + \beta) F_z / 2$	$(1 - \alpha + \beta) F_z / 2$	0
5	$(1+ \alpha + \beta) F_z / 2$	$(1 - \alpha + \beta) F_z / 2$	$+ F_y$
6	$(1 - \alpha - \beta) F_z / 2$	$(1 + \alpha - \beta) F_z / 2$	0
7	$(1 - \alpha - \beta) F_z / 2$	$(1 + \alpha - \beta) F_z / 2$	$- F_y$
8	$(1 - \alpha + \beta) F_z / 2$	$(1 + \alpha + \beta) F_z / 2$	0
9	$(1 - \alpha + \beta) F_z / 2$	$(1 + \alpha + \beta) F_z / 2$	$- F_y$

B.3 Load cases resulting from longitudinal forces

Load case	F_{z1}	F_{z2}	F_x
1	$F_z / 2$	$F_z / 2$	0
2	$F_z / 2$	$F_z / 2$	$+ F_{x1}$
3	$F_z / 2$	$F_z / 2$	$- F_{x1}$

B.4 Loads cases for tests under normal service loads resulting from bogie running

Load Case	Force on side pad 1	Force on pivot	Force on side pad 2	Transverse force
1	0	F_z	0	0
2	0	$(1 + \beta) F_z$	0	0
3	0	$(1 - \beta) F_z$	0	0
4	0	$(1 - \alpha)(1 + \beta) F_z$	$\alpha(1 + \beta) F_z$	F_y
5	$\alpha(1 + \beta) F_z$	$(1 - \alpha)(1 + \beta) F_z$	0	$- F_y$
6	0	$(1 - \alpha)(1 - \beta) F_z$	$\alpha(1 - \beta) F_z$	F_y
7	$\alpha(1 - \beta) F_z$	$(1 - \alpha)(1 - \beta) F_z$	0	$- F_y$

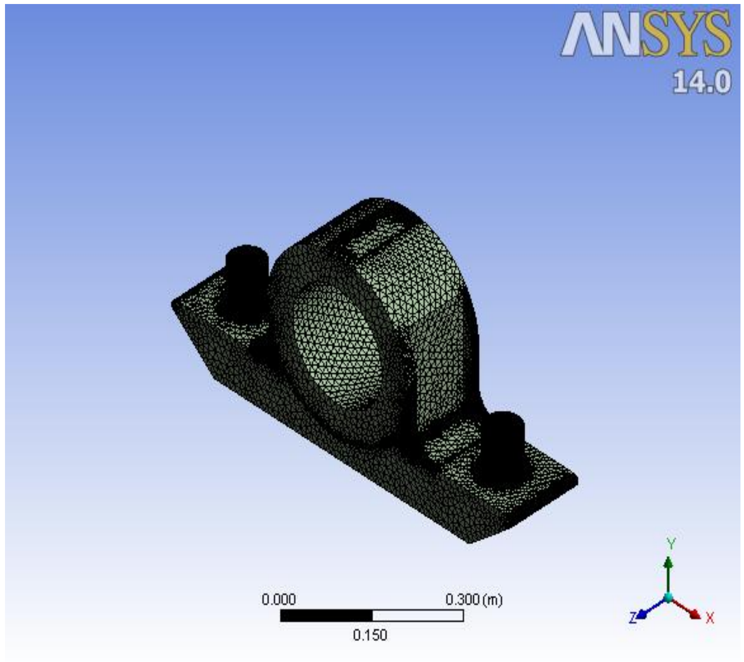
B.5 Exceptional loads

Load case	Vehicle body masses					Bogie masses			
	a_{zc} (m/s²)	a_{yc} (m/s²)	a_{ycc} (m/s²)	a_{xc} (m/s²)	q (N/m²)	a_{zb} (m/s²)	a_{yb} (m/s²)	a_{ycb} (m/s²)	a_{xb} (m/s²)
Switches	3,2	2,2	—	Emergency	600 [a]	30	16	—	Emergency
Running through	1,6	1,3	2,0	Emergency braking rate	600 [a]	12	6,5	2	Emergency braking rate
[a] Wind speed of 105 km/h									

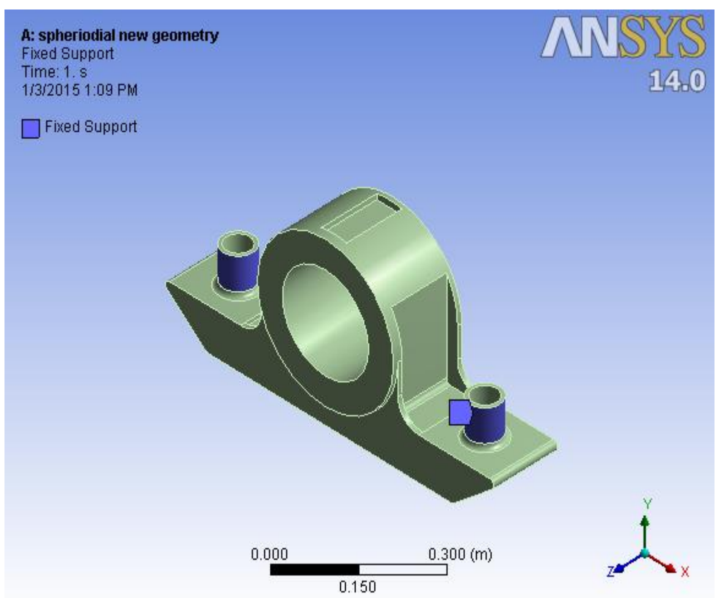
B.6 Normal service loads

Load case	Vehicle body masses					Bogie masses			
	a_{zc} (m/s²)	a_{yc} (m/s²)	a_{ycc} (m/s²)	a_{xc} (m/s²)	q (N/m²)	a_{zb} (m/s²)	a_{yb} (m/s²)	a_{ycb} (m/s²)	a_{xb} (m/s²)
Switches	2,4	1,6	—	—	200 [a]	25	12	—	—
Straight track	1,2	0,9	—	Service braking	—	8,5	4,5	—	Service braking
Running through	1,2	0,9	1,0	Service braking	—	8,5	4,5	1,0	Service braking
[a] Wind speed of 60 km/h									

Appendix C. Similar ANSYS Input Future Result for Model I



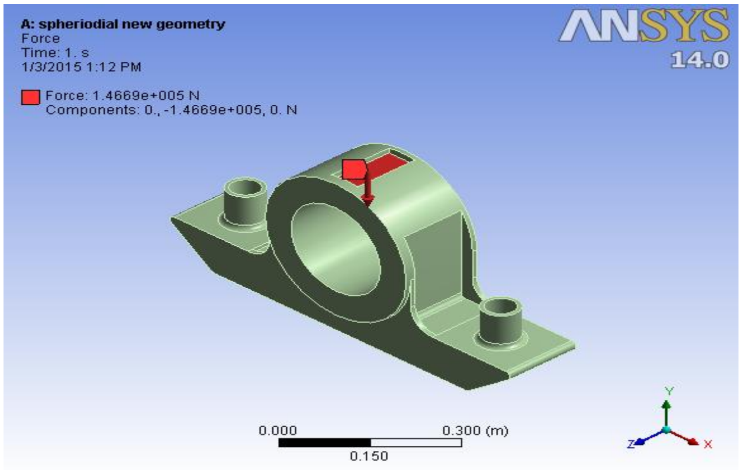
C.1 Advanced size function on proximity and curvat fine meshing of Model 1



C.2 Fixed support

C.3 Bending load component

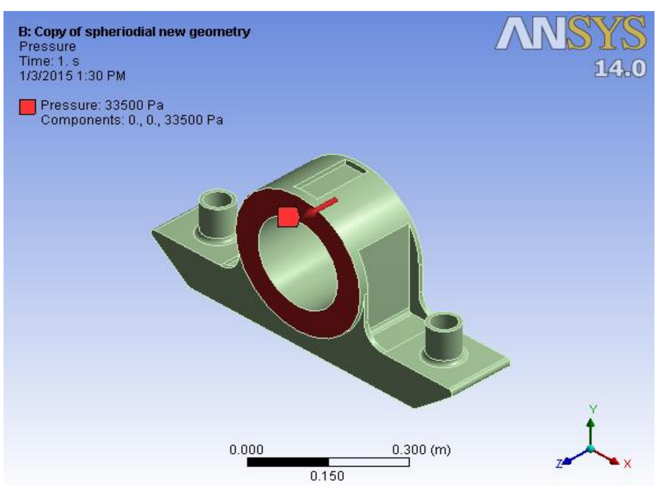
Steps	Time [s]	X [N]	Y [N]	Z [N]
1	0.	0.	-1.2413e+005	0.
	1.		-1.4669e+005	



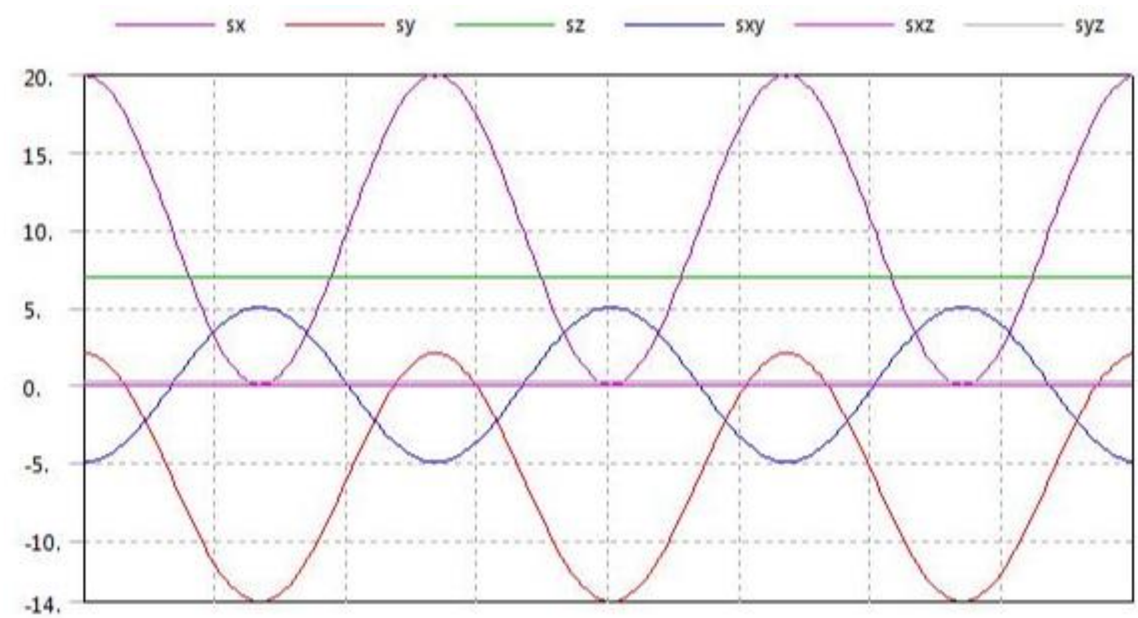
C.4 Bending load at the selected location

C.5 Torsional loading component

Steps	Time [s]	X [Pa]	Y [Pa]	Z [Pa]
1	0.	0.	0.	-33500
	1.			33500



C.6 Torsion load at the selected location



C.7 Non-proportional loading representative factor