Addis Ababa Institute of Technology

Analysis of Fatigue Failure on Motor Powered Train Axle of AALRT

SCHOOL OF MECHANICAL AND INDUSTRIAL ENGINEERING

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By

Beza Takele

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Advisor

Mr. Habtamu Tkubet

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ADDIS ABABA UNIVERSITY

ADDIS ABABA INSTITUTE OF TECHNOLOGY

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Approved by:

Mr. Habtamu Tkubet	______________	______________
Advisor	Signature	Date
______________________	______________	______________
Internal Examiner	Signature	Date
______________________	______________	______________
External Examiner	Signature	Date
______________________	______________	______________
Railway Center Head	Signature	Date

Declaration

I hereby declare that the work which is being presented in this thesis entitled as “Analysis of Fatigue Failure on Motor powered Train Axle of AALRT” is originally work of my own, had not been presented for a degree of in any other university, in any project by any means, and all the resource materials used for this thesis had been accordingly acknowledged.

__________________________ __________________________

Beza Takele Date

This is to certify that the above declaration made by candidate is correct to the best of my knowledge.

__________________________ __________________________

Mr. Habtamu Tkubet Date

(Advisor)

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Abstract

Fatigue is one kind of phenomena that results breakdown of such material that are under a cyclic load or repeated stress. Fatigue is a main process resulting in a structural degradation and failure during axles' service life. Railway axles are commonly operated over a service life of 30 years or more which refers to a very high number of loading cycles in the order of $10^{9.}$ This project is all about AALRT train axle failures due to fatigue and discus about how to analyze the fatigue and stress. The reason why the project idea directed to around this area is that nowadays AALRT is operating beyond its design limit. The train designed to load 8 persons in one meter square space in the condition of overload. But practically the train is operating by loading more than 8 passengers in one meter square space. When the train load and operates beyond its design load limit many parts of the train life span will be shorten. One of the sensitive parts of the train that could easily damage and may cause derailment accident is axle. Because the whole train load directs toward to the axle. So the train axle will subject to fatigue and finally failure. The objective of this project is analyzing the axle of AALRT to clearly specify its fatigue life and stress concentration and distribution. This is done by collecting the data from ERC and Internet and analyzing the data that collected using formulas and software. The result will be discussed by comparing the axle fatigue life with the standard axle fatigue life.. And it tried to show that stress analysis by using software like Abacus. According to the data collected and by using approaches and software At Leghar station which is on the worst loading condition the result of the Maximum Principal Stress and the von misses stress are1.72×10^7 N/m^2 and 1.84×10^7 N/m^2 respectively. The result of the pressure is 1.160×10^7 N/m^2. The reaction force is 5.12×10^9 N. The fatigue life is calculated as 3.7×10^7 cycles. . When comparing with the standard axle life it's found that the life of the AALRT axle life has less fatigue life. According to the standard axle life the cycle of the fatigue life reaches more than Giga cycles. The result validates and it is proven. Therefore AALRT should load passengers according to its design load limit and adding additional trailer will make less and disperse the concentrated axle load. Another solution is that using better fatigue resistant material for the axle would make longer the life of the axle.

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Nomenclature

ERA *European Railway Agency*

AALRT *Addis Ababa Light Railway Transit*

3D *Three dimensional*

S-N *Stress Verses with Number of revolution*

ATSB *Australian Transport Station Bureau*

AAU *Addis Ababa University*

ERC *Ethiopia Railway Corporation*

MC *Motor unit installed with powered bogies and pantograph*

Tp *Trailer unit installed with trailer bogie and pantograph*

DC *Direct Current*

S_{ut} *Ultimate tensile strength*

S_y *Tensile yield strength*

W *Weight*

σ_{max} *Maximum Stress*

σ_{min} *Minimum stress*

σ_a *alternative stress*

σ_r *Stress range*

σ'_f *Stress at fracture*

σ_b *Bending Stress*

E *Elastic Modulus*

ν *Poisson's Ratio*

g *Gravity*

V *Velocity*

N *Number of revolution*

S_e *Endurance Limit*

f *Fracture*

AW_n *working conditions*

D *Axle diameter*

$R(r_w)$ *Wheel radius*

D *Axle diameter*

R *Wheel radius*

W *Total Vehicle weight or axle load*

J *Distance between journal center*

G *Wheel trade distance*

A *Distance between journal center and the end of the wheel set*

H *Height from axle to center of gravity*

X *Distance from the outside wheel hub to the contact load*

Y *Distance from the inside wheel hub to the contact load*

I *Addition of x and y*

G *Center of gravity*

a_v *Vertical acceleration coefficient*

M *Factor of safety*

a_l *Lateral acceleration*

Z *Section of Modulus*

μ *Friction Coefficient*

P *Horizontal force*

F	*Force*
ω	*Rotational Velocity*
F_t	*Tangential force*

Chapter One

Introduction

1.1. Background of the study

Fatigue is one kind of phenomena that results breakdown of such material that are under a cyclic load or repeated stress. Fatigue is a main process resulting in a structural degradation and failure during axles' service life. The nominal maximum stress values that cause such damage may be much less than the strength of the material ultimate tensile stress limit. Fatigue start to happening when a material is subjected to cyclic loading and unloading. [1]

Railway axles are commonly operated over a service life of 30 years or more which refers to a very high number of loading cycles in the order of 10^9. Railway axle designed for a long term of operation. However, there are known cases of their failure, which are caused by defects that arise during the operation and serve as a source of origin and development of crack to critical size. During 20th century the number of failures of railway axles was reduced to about 5% due to improved steels and assessment concepts. According to the railway safety performance report of the European Railway agency (ERA) of 2011, the number of broken axle in the European Union between 2006 and 2009 was in total 329. Related to the number of about $1.66*10^{10}$ train kilometers over these four years this refers to one fracture event per 50.45 millions train kilometers. Various reasons are responsible for these differences including the average age of the fleets, maintenance and inspection issues of the rolling stocks, the state of the tracks, different average axle loads, etc., but also the statistically rather small reference period of 4 years only should be kept in mind. Railway axles were investigated at the very beginning of fatigue research and design. Fatigue failure of axle has been a source of difficulty for engineers since the railway service started in the early part of the 19th century. Derailment caused by fatigue of railway axles is very rare in Japan. Railway axles are, however, one of the most important components in railway systems since a failsafe design is not available. [2]

Lately, an unprecedented series of major railway axle fractures with many similarities have occurred. Most of the broken axles were from freight wagons, which usually have higher loads in comparison with passenger wagons. All broken axles were fractured due to the fatigue of material. Even if the relative number of accidents is small in comparison to other causes of derailments it still remains a real possible cause of accidents and cannot be neglected. The typical aspect of a broken axle due to the fatigue of material is cylindrical, with flat, relatively smooth fracture surface, with or without shear lips and with generally horizontal marks. Along with ratchet marks (two distinct ratchet marks are evident along the top edge), these features are consistent with reversed bending fatigue, having multiple origins. [3]

Fig 1. 1: Broken tank wagon axle from a severe derailment [3]

The degree of fatigue damage of railway wheelset axles is one of the rolling stock limiting safety factors (1–5). This is the main reason for developing new concepts of axle design with higher resistance to initiation and propagation of fatigue cracks. Since it has been clearly demonstrated that the final fracture instability can be, in the case of Giga-cycle fatigue, i.e. after a long time in service, initiated under significantly lower stress amplitude than the fatigue limit defined for 10^7cycles, the concept of fatigue limit must always be complemented by fatigue damage tolerance considerations in order to guarantee the axle safety. The fatigue damage tolerance is based on an assumption of existence of fatigue micro-cracks in the axle, which may develop during the service into dangerous long fatigue cracks. Therefore fatigue damage tolerance includes, besides the minimum crack size detectable by NDT, also corresponding intervals of inspections, depending on localization and the time in service. To increase fatigue strength of axles, the technology of surface induction hardening was developed and implemented firstly in Japan. This technology is also deployed by the railway wheel set manufacturer BONATRANS GROUP, the objective being to satisfy requirements of its clients, especially for deliveries of powered axles intended for high speeds, with high degree of safety against fatigue damage and with long service life. The technology of surface hardening allows the operators also to reduce the number of ultrasonic service inspections, thus saving them considerable amounts of money incurred in regular rolling stock sidetracking and inspection. [4]

1.2. Fatigue on Railway Axle

The modern study of fatigue is generally dated from the work of A. W¨ohler, a technologist in the German railroad system in the mid-nineteenth century. Wohler was concerned by the failure of axles after various times in service, at loads considerably less than expected. A railcar axle is essentially a round beam in four-point bending, which produces a compressive stress along the top surface and a tensile stress along the bottom. After the axle has rotated a half turn, the bottom becomes the top and vice versa, so the stresses on a particular region of material at the surface vary sinusoidal from tension to compression and back again. This is now known as fully reversed fatigue loading. [5]

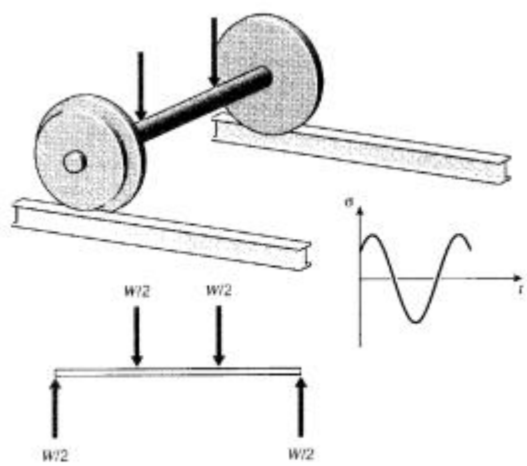


Fig 1. 2: Fatigue in a railcar axle

1.2.1. Factors causing fatigue failure

There three basic factors

1) A maximum tensile stress of sufficiently high value.

2) A large amount of variation or fluctuation in the applied stress.

3) A sufficiently large number of cycles of the applied stress.

Other additional factors are

• Stress concentration

• Corrosion

• Temperature

• Overload

• Metallurgical structure [6]

1.2.2. Stress Cycles

Different types of fluctuating stress

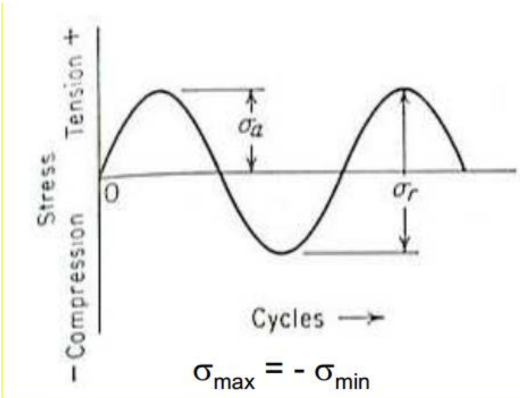


Fig 1. 3: Completely reversed cycle of stress (sinusoidal) [5]

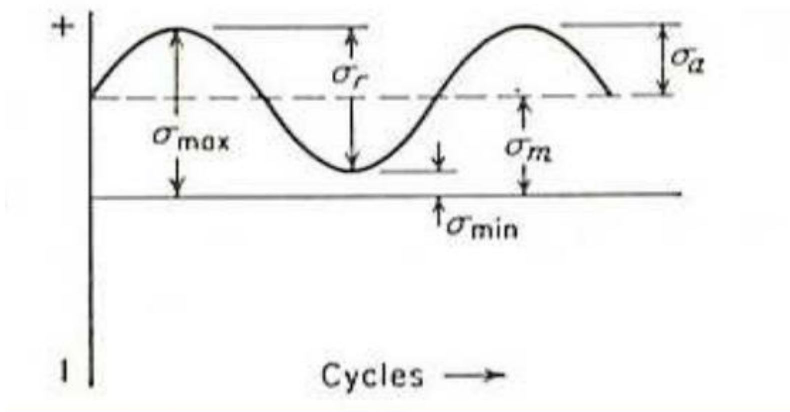


Fig 1. 4: Repeated stress cycle [5]

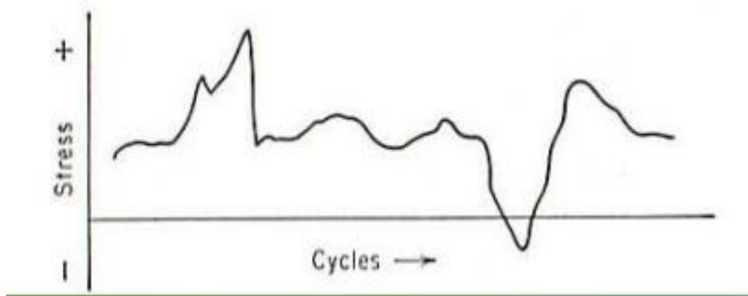


Fig 1. 5: Irregular or random stress cycle [5]

This shows us the addition of tensile stress and compressive stress

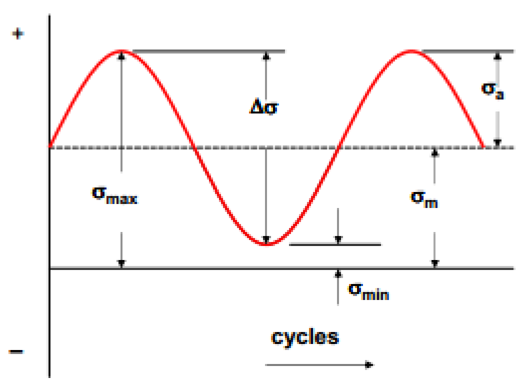


Fig 1. 6: Nomenclature of stress parameter in fatigue loading [5]

Maximum stress, σ_{max}

Minimum stress, σ_{min}

Stress Range

$\Delta\sigma$ or $\sigma_r = \sigma max - \sigma min$ ……………………………………………………..……eq 1.1

Alternating stress

$\sigma_a = \frac{\Delta\sigma}{2} = \frac{\sigma\, max - \sigma min}{2}$……………………………………………………………..eq 1.2

1.2.3. S - N Curve

Well before a micro structural understanding of fatigue processes was developed, engineers had developed empirical means of quantifying the fatigue process and designing against it. Perhaps the most important concept is the S-N diagram, such as those shown in a constant cyclic stress amplitude is applied to a specimen and the number of loading cycles N until the specimen fails is determined. Millions of cycles might be required to cause failure at lower loading levels, so the abscissa in usually plotted logarithmically. [5]

If a plot is made of the applied stress amplitude versus the number of reversals to failure (S-N curve) the following behavior is typically observed:

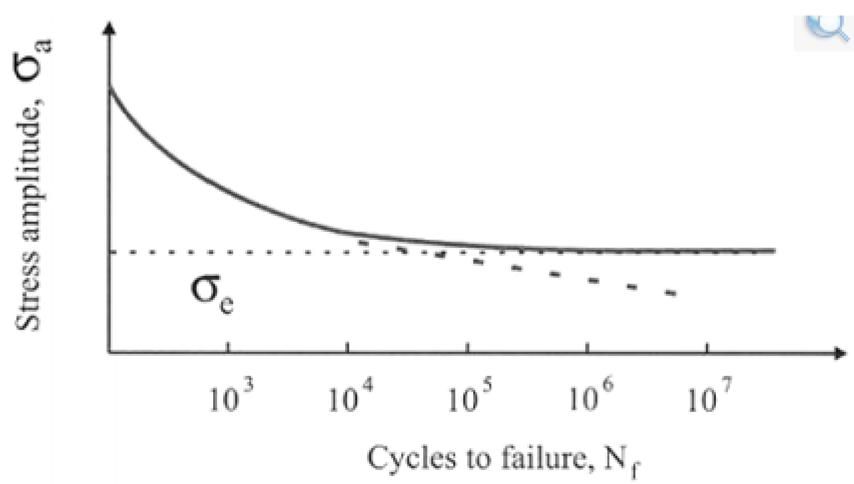


Fig 1. 7: S-N curve [5]

If the stress is below σ_e (the endurance limit or fatigue limit), the component has effectively infinite life.

$\sigma e \approx 0.35\sigma_{TS}$ -$0.50\sigma_{TS}$ for most steels and copper alloys. If the material does not have a well defined σ_e is arbitrarily defined as the stress that give s $N_f=10^7$ [5]

1.2.4. Fatigue design approaches

The total fatigue life of a component can be considered to have two parts, the initiation life and the propagation life, as depicted below. [5]

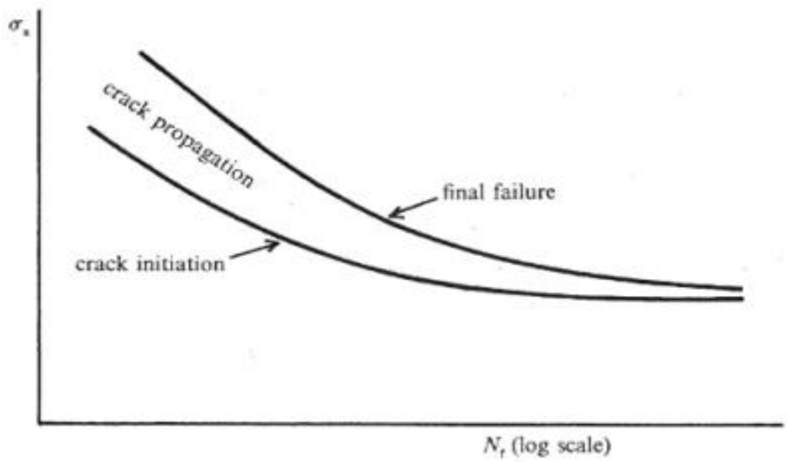


Fig 1. 8: S-N curve (relationship between stress and number of cycle) [5]

The preceding approach to calculate the lifetime assumes fully reversed fatigue loads, so that the mean stress σ_m is zero. [5]

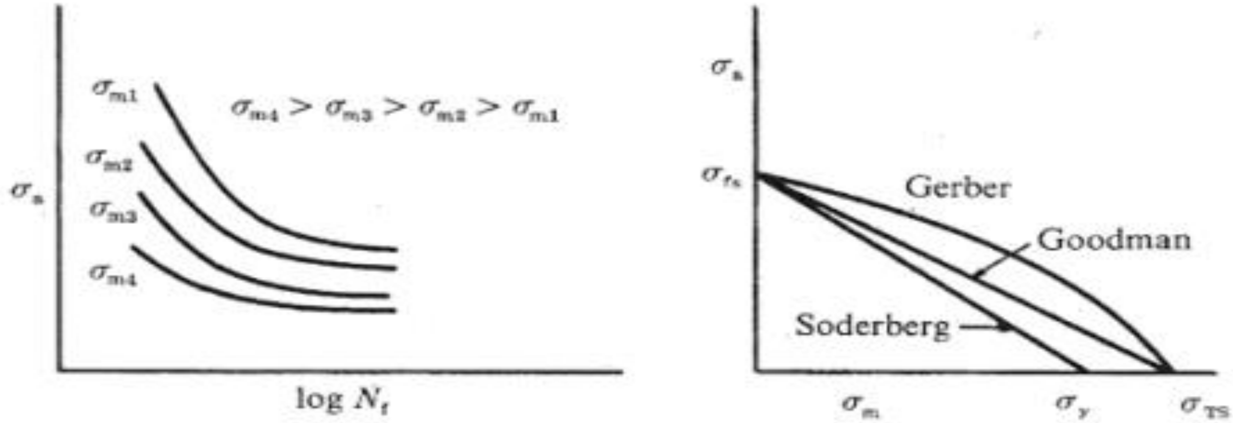


Fig 1. 9: S - N Curve [5]

$\sigma_a = \sigma_a|_{\sigma a} = 0\left\{1- \frac{\sigma m}{\sigma y}\right\}$ (Soderberg)……………………………..…………………… eq 4.16

$\sigma_a = \sigma_a|_{\sigma a} = 0\left\{1- \frac{\sigma m}{\sigma TS}\right\}$ (Goodman) ……………...……………..…………………… eq 4.17

$\sigma_a = \sigma_a|_{\sigma a} = 0\left\{1-(\frac{\sigma m}{\sigma TS})^2\right\}$ (Gerber) [5] ……………………..……………………...… eq 4.18

1.3. Problem Statement

Now days it is observed that the AALRT trains are loading more than the designed loading limit. This causes damages to parts of the trains. For example the wheels are already exposed to damage. Therefore for the coming years there is no reason that the rest parts of the train will survive without damage, such as the axle. This will be due to fatigue and overload condition. This over load facilitates the failure due to fatigue. The train designed to load 8 persons in one meter square space. But practically the train is operating by loading more than 8 passengers in one meter square space. When the train load and operates beyond its design load limit many parts of the train life span will be shorten. One of the sensitive parts of the train that could easily damage and may cause derailment accident is axle. Because the whole train load directs toward to the axle. So the train axle will subject to fatigue and finally failure. The worst part is train accident is not like car accident. It may take many lives and damage much amount of property, especially if it derails. As this technology is new for our country it may also destruct the image of the company and business. Therefore the company may loss its customers. This problem initiates to do the research around train axle failure in order to analyze its fatigue failure and its life.

1.4. Objectives

1.4.1. General Objective

The general objective is analyzing the axle of AALRT to clearly specify its fatigue life and stress concentration and distribution. The result will be discussed by comparing the axle fatigue life with the standard axle life.

1.4.2. Specific Objective

The stress analysis will be calculated on worst cases at selected station. The maximum pressure that develops on the axle will be also analyzed. The fatigue life of maximum loading station will be calculated using the maximum von misses stress.

1.5. Scope of the Project

Damage on the axle due to fatigue is not a common damage. It may not occur usually in any train routes. But when it happens, the damage that follows is very harsh and massive. That's why everybody has to give attention for this problem to find a way to resolve the situation on the overloading. The main reason for this damage is cyclic overloading condition. Therefore the scope of the project is to make awareness for the corporation workers to think any other way to handle the over loading condition before massive accidents happen. If derailment happens, the crises will not stop only by damage the axle part. It may take many lives and damage a lot of property. In this study the fatigue definitions, theories, fatigue analysis and approaches to resolve the problem are covered.

Chapter Two

2. Methodology

2.1. Data collection

These data are collected from ERC Kality Depot. Based on these fatigue analysis of the axle is done. The train is consisted of three articulated modules: ＋Mc1－Tp－Mc2＋

- Mc 1: motor unit installed with powered bogies and driver's cab
- T p: trailer unit installed with trailer bogie and pantograph
- + : embedded fold car couplers －: articulated devices

Table 2. 1: Basic Technical Parameters of the AALRT train

Basic Technical Parameter	**Parameter**
Track gauge	1435mm
Minimum radius of vertical curve	1000m(auxiliary line)
Minimum radius of horizontal curve	25m
Axle load	≤11(1+3%)t
Maximum gradient	5%$_0$
Rated voltage	DC750V
Train length (exclusive of train couplers)	28800mm

Table 2. 2: Basic Technical Parameter of the Train

Basic technical parameter	Parameter value
Maximum width	2650mm
Height	3610mm
Floor level to rail level at low-floor area	380mm (350mm at door areas)
Floor level to rail level at high-floor area	655mm
Center-to-center distance between both bogies	10400mm
Axle distance of the motor unit	1900mm
Axle distance of the trailer unit	1800mm
Clear height in passenger compartment at low-floor area	2255mm
Clear height in passenger compartment at high-floor area	1980mm

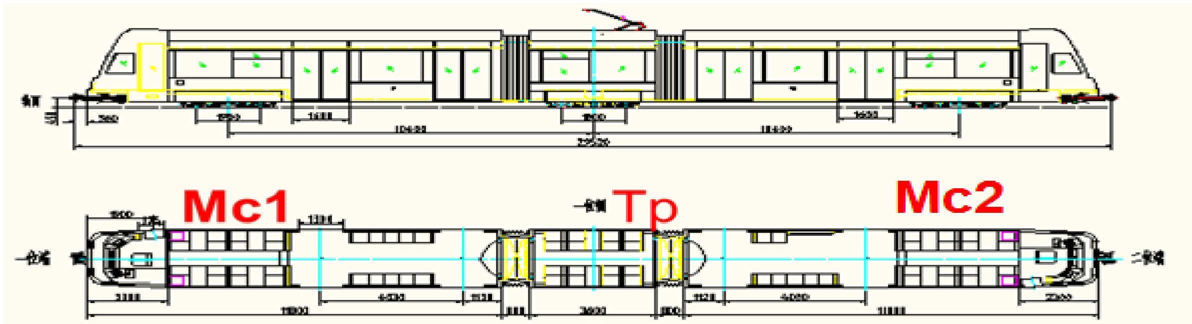
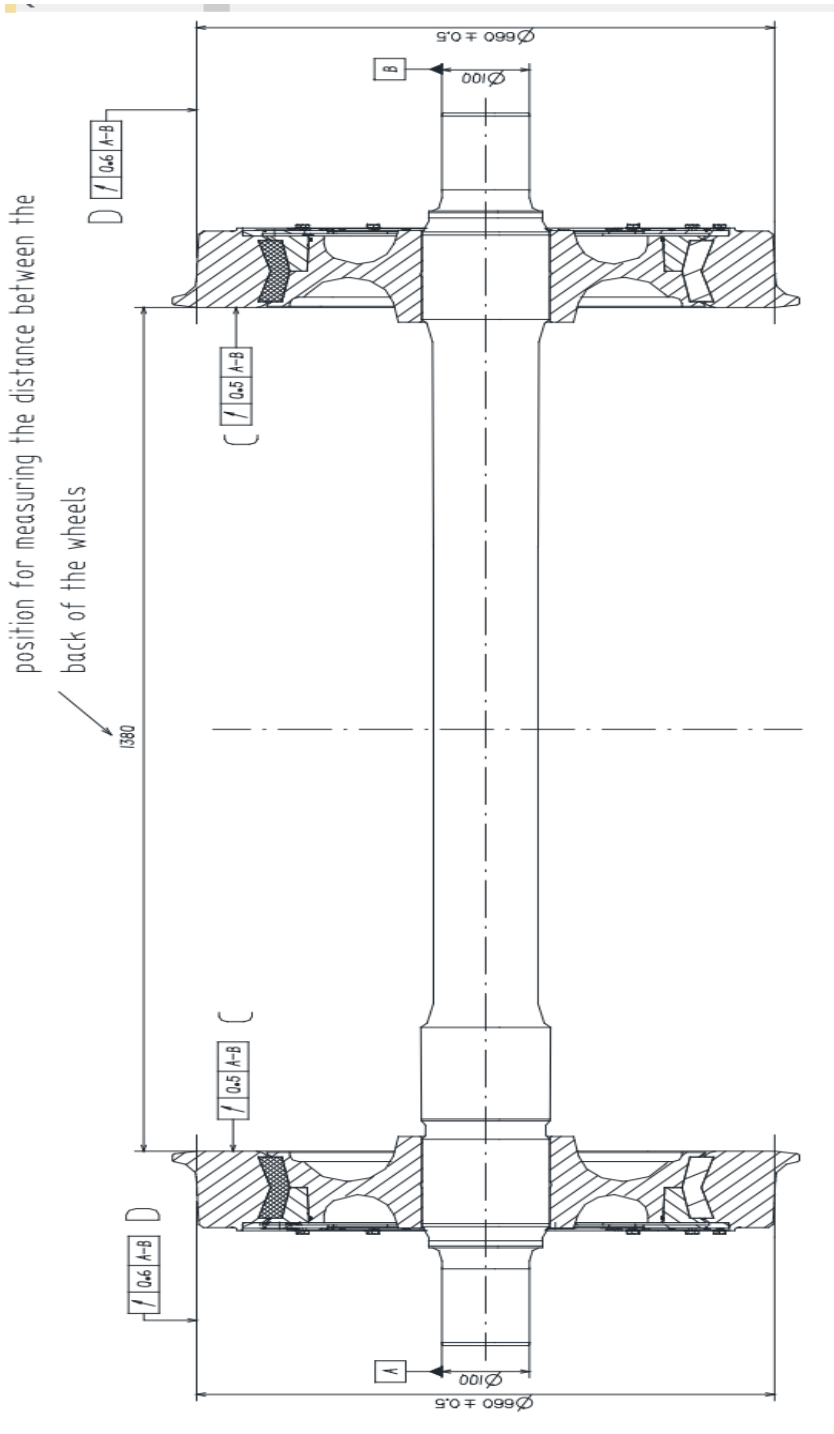


Fig 2. 1: AALRT Rolling stock

In calculation the occupied area of standing passengers is 6 persons / m^2 for rated passenger capacity, or 8 persons / m^2 for over-crowed capacity; the average weight of the passenger is 60Kg/person. [6]

Table 2. 3: Axle type

Axle type	Material	Axle load	Wheel Diameter	Bogie weight	Train loading capacity
Hollow	Steel(AlN)	11.5ton	660 mm	4 ton	317/vehicle


position for measuring the distance between the back of the wheels
1380
Ø660 ± 0.5
Ø100
A
B
C
D
0.6 A-B
0.5 A-B

(a) [7]

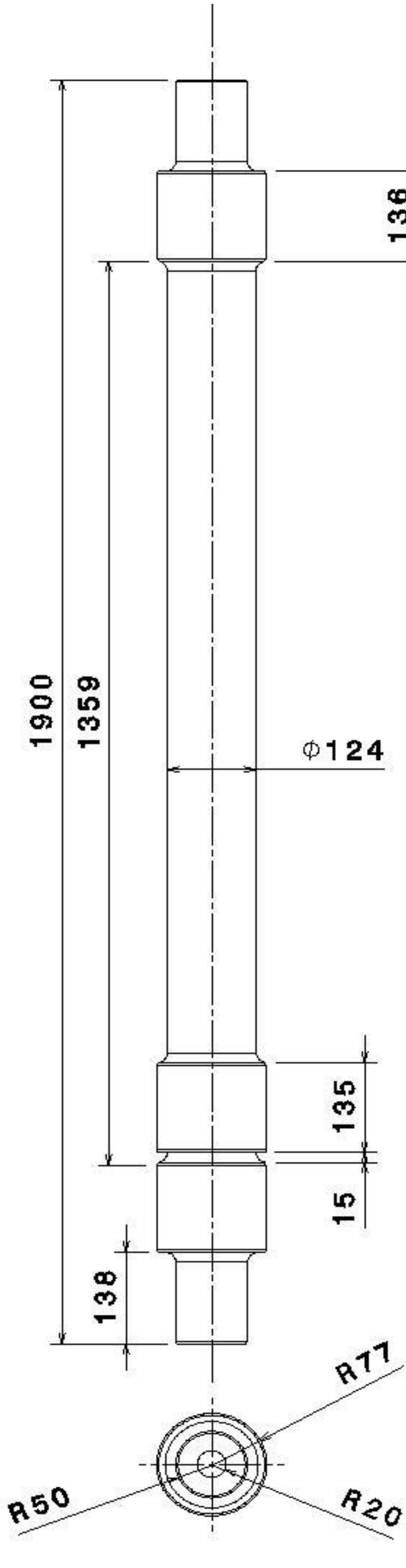


Fig 2. 2: (a) and (b) AALRT axle in 2d Drawing[7]

Table 2. 4: (a) and (b) Technical Specifications of the Axle

(a)

Steel Grade	S_y(MPa)	S_{ut}(MPa)	Elongation (%)	Minimum impact energy at 20-C (J)
AIN	320	550-650	22	25

(b) [3] and [8]

Mechanical Properties of AIN Steel	**Value**
Young's Modulus	200Gpa
Elastic Modulus	210,000 N/m
Poisson's Ratio	0.26
Shear Modulus	7.69×10^{10} N/m^2
Density	7,300 kg/m^3
Ultimate Tensile Strength	550 MPa
Compression strength	550 MPa
Yield strength	320 MPa

S.N	Gauge(mm)	Axle load (t)	Length (mm)
1	1,435	11.5	1900

Table 2. 5: Specification of AALRT Axle

Max. operating speed	70km/h
Distance between backs of wheel flanges	1,377.7 (-1,+2) mm
Wheel base	18000 mm
Axle load	11.5t
Wheel diameter(new/worn)	660/580mm
Bogie weight	≤4t
Vertical damper quantity	2 pieces
Lateral damper quality	3 Pieces

2.2. Processing the Data

Process implies editing, coding, classification and tabulation (putting in the form of table). The data that gathered should arrange according to publishing time and closeness or relativity. Data should arrange according to their role in the research. There the data that collected from ERC and Internet collected and arranged properly. Most of them tabulated and edited.

2.3. Analyze the Data

Analysis refers to computation of certain measures along searching for patterns of relationship that exist among data groups. This helps to answer the research question of the failure behind the train axle. This is done by modeling the train axle on Catia and Abacus software to do some force and stress analysis by applying the axle load on the tip of the axle (where the axle box found) and the reaction force on the axle where the wheels positioned in the reverse direction. Then after the analysis recording the results should not be missed. Then after doing some numerical calculation using different formulas and software, comparing the axle fatigue life with the standard axle life will be followed. The final conclusion decision will be made.

2.4. Interpret the findings

This is done by interpreting AALRT axle analyzed results and numerical calculation value and by giving the proper suggestion and conclusion. At the result discussion and conclusion, the result that found from the numerical calculation and software analysis interpreted and briefly discussed.

Chapter Three

3. Literature Review

It is said that the railway traffic system as we know today dates back to 1825 when the Stockton and Darlington Railway of England started commercial operation. In Japan, the first railway line was operated between Shimbashi and Yokohama in 1872. At that time, the maximum speed of trains was not higher than 60 km/h. However, with the progress of railways, train speeds have been increasing. [9]

In particular, since the opening of the Tokaido Shinkansen in 1964, train operations have been markedly accelerated. Today, thanks to the remarkable progress of railway technology, the maximum speed of trains has reached more than 300 km/h. Steel products have largely been used for many of the railway system components such as rails, wheels, axles, and bogies. Considering that the railway traffic system has an enormous influence on the public, the strength reliability of these components is an important matter associated with the security and safety of the public. As for the strength reliability, one of the important study themes is the strength or fatigue characteristic of each railway system component that is subject to repeated loads during use. [9]

August Wohler, German railway engineer started his well known test series, the results of which were published between 1858 and 1871. [10]

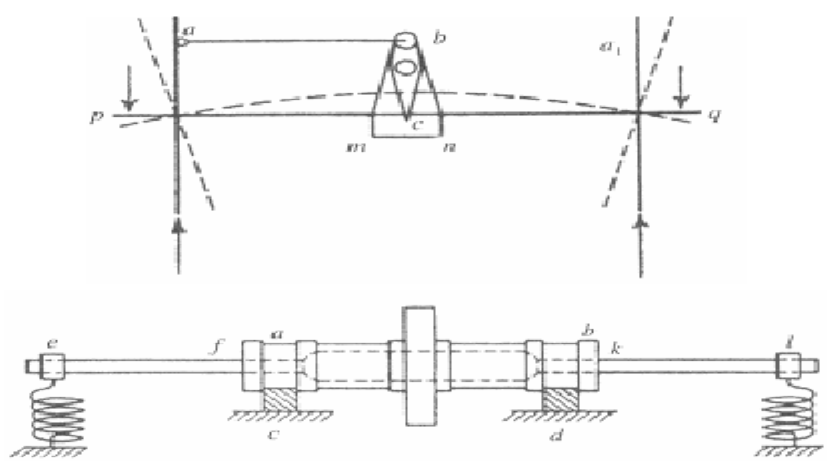


Fig 2. 3: Wholer's apparatus fatigue testing machine [10]

The upper drawing shows Wohler's apparatus for the measurement of service strains on railway axles. The dashed line indicates the deflected position, and the below drawing shows Wohler's purpose built fatigue testing machine to apply reserved bending to axle-like specimen.[10] He conducted a systematic experiment to evaluate the fatigue of steam locomotive axles and studied the stress (S)-life (N) relationship, which is used even today as the most basic fatigue characteristic data. He established the foundation for the study of fatigue. [9] The stress locally in the axle to initiate a fatigue crack. Wohler led to the identification of the fatigue limit for steels. Despite its long use, there is growing evidence that for lives longer than the conventional 10^6 to 10^7 cycles, at which the fatigue limit is determined, the safe stress range continues to be eroded down to 10^9 cycles and more, which is at the very long lives typical of that required of axles and wheels. [10]

From the beginning of 1997, axle failures have occurred on rail vehicles of the Istanbul Transportation Co. that have been in service for Istanbul city rail transportation. Finally, in 2007 after the vehicle entered the station in Istanbul Esenler, the axle of the vehicle broke and the wheel fell between the rail and the switch as a result of fatigue fracture. [10]

Meral Bayraktar, **Necati Tahrali** and **Rahmi Guclu** Turkish mechanical engineering department crew write a paper try to investigate the reason behind the train axle failure in February, 2009. The experts reported that the fracture occurred at the transition section having radius between the wheel and gear box. From the result of examination of fracture surface is 10% less than the total surface. Also, it is clear that two different regions have been detected on surface. While 90 % of the section surface has shown the growth of fracture by the time, 10% of the section surface (smooth part) has been pointed out as instantaneously broken. The forces affecting the axle are tension and low stress. The edges of the fracture move along rapidly because of the poor but effective environmental notch impact. After many investigations they conclude that the reason behind the axle failure is that the fractures result from fatigue according to the "LRT Axle Report" prepared by the establishment under the responsibility of the firm. The rail vehicle is subjected to different occupancy rates at the different times during the day. So, it leads to handle the situation for three loading configuration; full, medium and low load. [10]

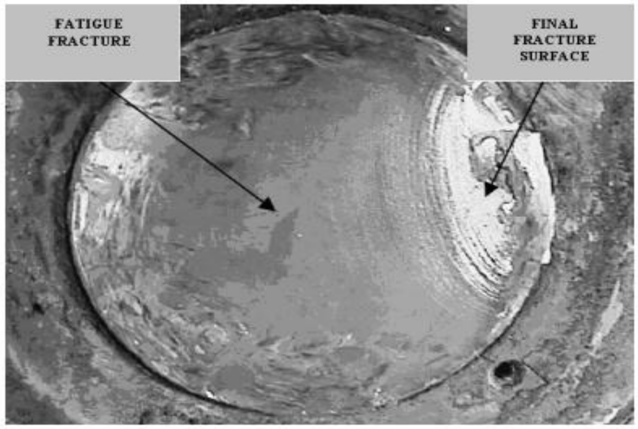


Fig 2. 4: The Failed Axle[10]

In 2011 the wagon axle of the wagon of TCDD (Turkish Republic State Railways) with serial number 8000 suburban train travelling Sirkeci-Halkali route has been examined. It is observed that it is the axle of trailing wagon which is subjected to the most forcing. Related to the static load and the dynamic forces, which are functions of speed, critical section of the axle is determined by calculating the minimum safety factor along the axle. Therefore, the location of the fracture is the area between the wheel and the gear in which safety factor is less than 1. [11]

In 2010, In Germany Bundesanstalt für Arbeit state there were train accident because of train axle failure. And the customers accused the manufacturers that produce that product. But the court ordered to investigate the problem first by using different scientific techniques.

C.Klinger, **D.Bettge**, **R.Hacker**, **T.Heckel**, **D.Gohlke** and **D.KlingBeil** different interdisciplinary Failure Analysts involve investigating the complex accidents in October, 2010. Mechanical engineers, Material testers, Numerical analysts, Microstructure investigators and Corrosion analysts get-together and started to solve the problem. The broken railway axle model was ICE3. The axle failures cause derailment of the train and property damage. [12]

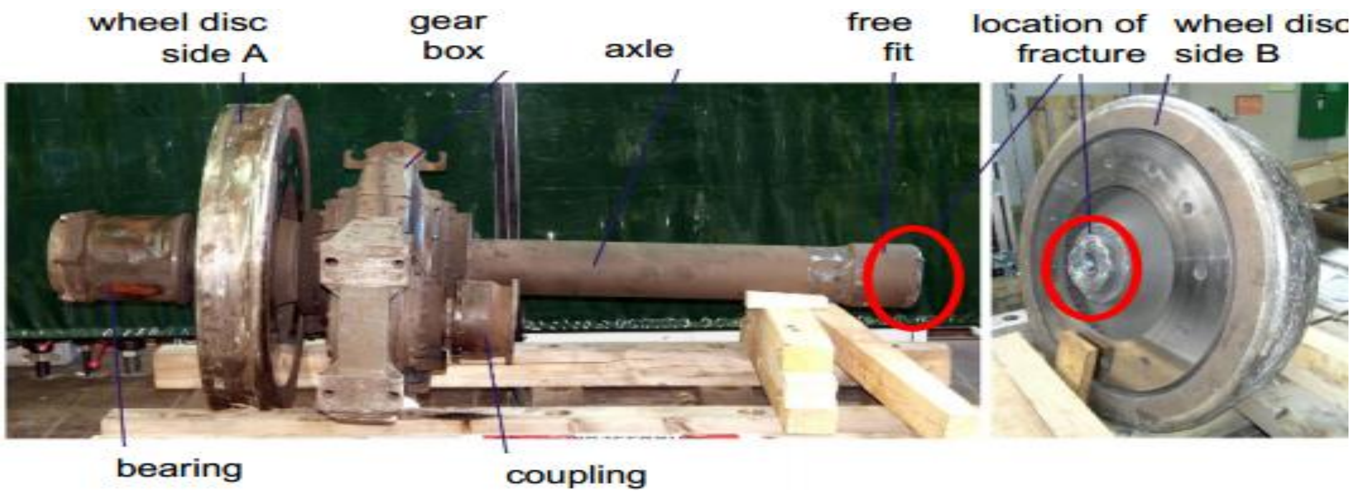


Fig 2. 5: : Broken Wheelset Gear box [12]

They first separate the broken wheelset ant they tested mechanical test like tensile and impact tests. They did the chemical analysis and follows fractography. Then they do the metallographic and identification of purity. [12]

The result of fractography test was

- Fatigue crack caused by rotation bending
- Residual fracture about 1/5th of fracture surface
- Successful allocation of the fracture surface to each other

Metallographic investigation

- Micro structure – OK
- Hardness – OK
- Microstructure – OK
- Grain Size – OK
- Surface Roughness – OK

Then they finally hypothesized the root cause for the fracture. The axle was nonmetallic inclusion of unacceptable size from the production. It was operating with a very high cycles in service $3*10^6$ km~10^9 cycles. And the initiation of fatigue crack in the recess (three center curve) shows it was highly utilized axle. They investigated also there were fast crack propagation due to high stresses. [12]

On Thursday 9 February 2006 at about 0351 an XPT passenger train travelling from Melbourne to Sydney derailed near Harden in New South Wales. An inspection by the driver found one wheel on the trailing bogie of the leading power car had derailed. During recovery operations the axle of the derailed wheel was found to have completely sheared with a crack in the radius relief area between the gear and wheel seats. The ATSB's investigation concluded that impacts from track ballast from unknown location(s) had led to the formation of the cracks in the axles. The investigation also concluded that routine testing of the axles carried out by the operator's maintenance contractor, using magnetic particle inspection (MPI), was ineffective and resulted in the fatigue cracks going undetected for a considerable period of time. A number of safety actions have been undertaken by Rail Corp and the Independent Transport Safety and Reliability Regulator of New South Wales which include measures aimed at the early detection and prevention of axle fatigue cracks in XPT and other diesel fleet rail vehicles to limit the risk of further axle failures. Additionally, the Australian Transport Safety Bureau has issued a safety advisory notice to all rail vehicle operators in Australia that they should consider the risks associated with axle failures as a result of fatigue cracks initiated by ballast strikes and review their maintenance practices accordingly. [13]

Fig 2. 7: Fractured surface of an axle [13]

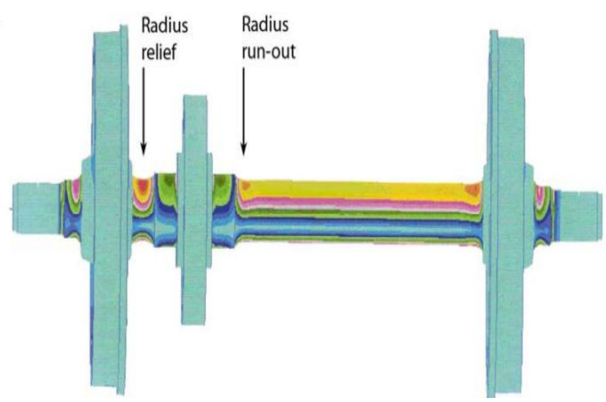


Fig 2. 6: Finite Element Model of an XPT Power Car Axle [13]

Cosmin Locovei, **Aurel Răduŀă**, **Mircea Nicoara** and **Laurentiu Roland Cucuruz** write a paper about Analysis of Fatigue Fracture of Tank Wagon Railway Axles in 2011 in Romania. The paper is focused on the fracture mechanism of railway axles due to the fatigue of material. The purpose of the present article is to numerically predict the number of cycles (or kilometers) to fracture of tank wagon railway axles in various theoretical conditions. The stresses in the axles were calculated by finite element methods. The number of cycles to fracture was calculated using closed form solution of NASGRO equation for fatigue crack development starting from an initial crack detectable by means of non-destructive testing. In order to demonstrate the deep negative impact of forbidden thermal treatments and operations applied to railway axles, residual stresses of these treatments were calculated and new numerical predictions of number of cycles to fracture have been made. [14]

G. Fischer, V. Grubisic and **M. Grosse-Hovest** write a research paper about 'Methodology for Reliable Durability Validation of Wheelset-Axles'. In their study they conclude that an "infinite life" design of wheelset axles is not possible based on practical experience and scientific investigations. And much greater vertical and lateral forces acting on axles were often found by the service measurements, compared with the values on which different standards for strength assessment of axles were based. In particular as a result of modified characteristics of rail vehicles, such as higher speed and adaptive matching to the track section (tilt technique), the stresses and number of cycles reached are much greater than those covered by the standards. An estimate of the maximum allowable nominal stresses according to the standard with the help of Woehler curves up to 10 and the properties determined with small specimens as proposed in some norms does not constitute a reliable basis for assessing fretting fatigue of higher strength new materials. Suitable validation tests are necessary for this purpose. The experience with regard to load values and permissible stresses on which the standards are based should be validated using corresponding safety factors. The large number of fractures in operation resulting from fatigue –particularly due to fretting fatigue –and re-call actions and inspections this entails, call for revision of the design concepts used so far and adaptation to the state of technology. Optimum and reliable design of wheel set axles requires determination of the operational loads and of the fatigue strength of axles. A corresponding procedure must ensure that these decisive influencing factors are appropriately considered for the design of wheel set axles as proposed in this paper. [15]

Kidanemariam G/tsadik is a former 2014 batch ERC (AAU) institute student. He worked on the fretting damage on train wheel set when the train is on the condition of empty. He tried to analyze the fretting fatigue life of the wheel set. [8]

Now in Ethiopia, The AALRT is giving service to its customers. But the service is not safe due to improper train loading condition. Different parts of the train are facing failure (like the wheel and rail) before there service life span. At the stations a repetitive loading and unloading condition and at traction time the way the axle subjected to rolling contact applied load creates a situation which is called fatigue, reason for the failure. The reason behind the failure will be identify clearly in this study using analysis. This research is done according to AALRT operating loading condition and tried to give other alternative ways to minimize the over loading condition. Here different approaching and new analyzing ideas are used to analyze the stress and force and fatigue life of the AALRT axle.

ERC employees are interviewed and explained that the trains are operating beyond their operating load limit. According to the data collected the trains design to load only 317 per vehicle. But they are loading beyond that. This will shorten the life span of the train axle. Therefore in this study other direction to increase the life span of the axle is stated. Like

- Using hollow shaft
- Using additional support like bearings to distribute the axle load and minimize the applied stress
- Using bolster less bogie

Chapter Four

4. Numerical Calculation and Modeling on AALRT Axle

4.1. Rout Selection

To calculate the fatigue life of the axle life of AALRT, first it is better to select the traction root as a sample. For this analysis from Kality to Menilik square root taken to analyze the fatigue life of the axle by calculating the nominal bending moment of the axle in each station to find the fatigue life. Note that all the stations are not taken to analyze the maximum bending stress. Stations are selected according to the much variance of loading and unloading. This is one journey test that is done catching the train from Kality depot by making the arrival Menlik square. Root selected – From Kality to Menelik square.

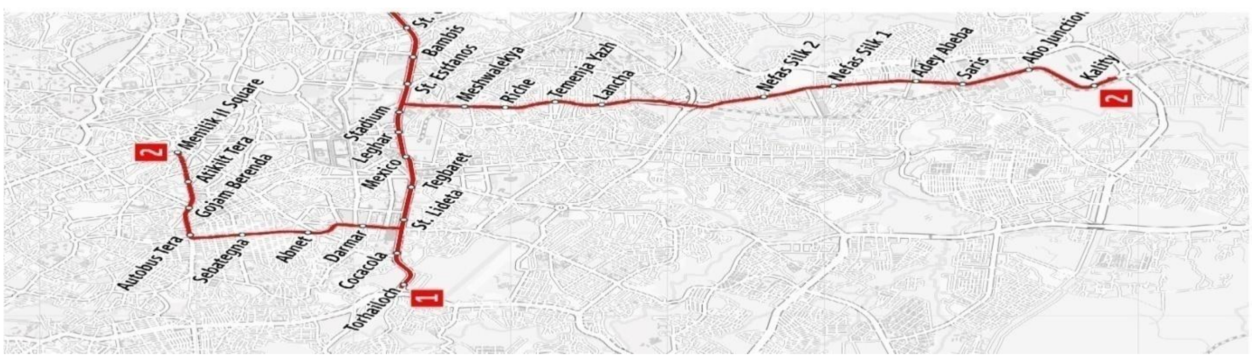


Fig 4. 1: Selected root (From Kality to Menilik square[16]

Stations selected to analyze the maximum bending stress

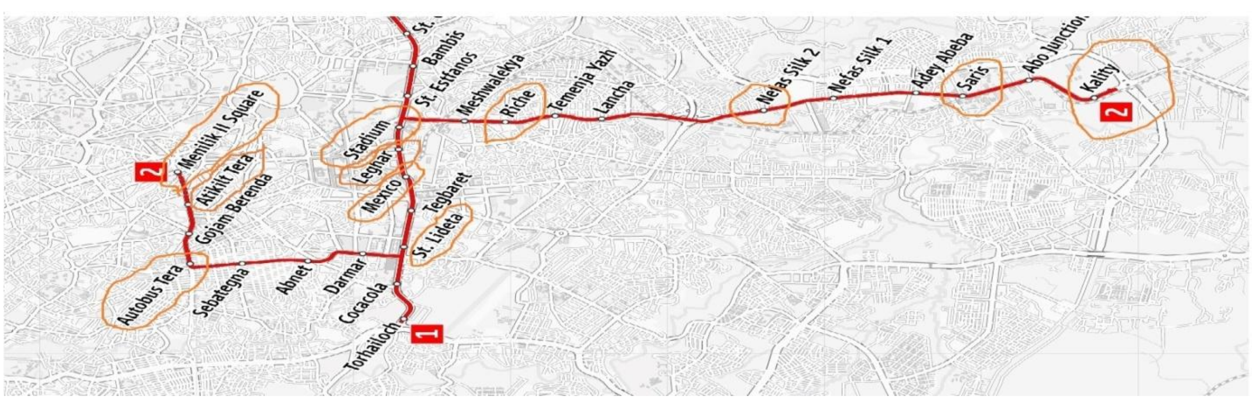


Fig 4. 2: Selected stations from Kality to Menilik square [16]

Station that selected for the maximum bending stress analysis is as follows.

1. Kality
2. Saris
3. Nefas Silk 2
4. Riche
5. Stadium
6. Leghar
7. Mexico
8. St. Lideta
9. Autobus Tera
10. Atikilit Tera
11. Menelik Square

These stations are selected based on the load variance. It is the nominal or maximum stress that cause fatigue failure that nominal or maximum bending stress is calculated for each station as follows.

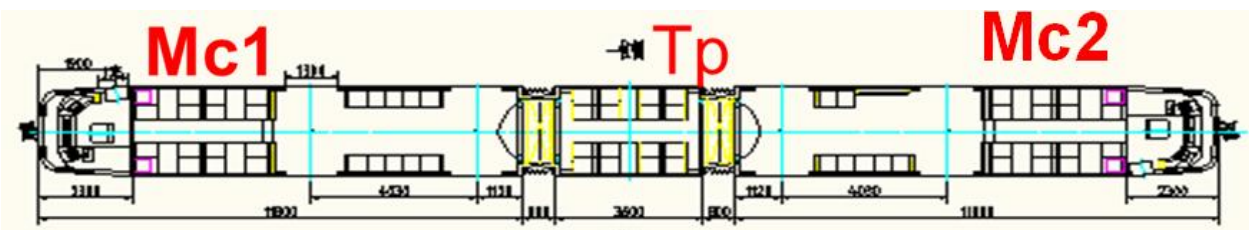


Fig 4. 3: Three Section of the train[6]

The axle load of empty train is 11.5 ton. AALRT train is a 70% low floor train of Chinese brand at 750 v DC, with a capacity of 317 passenger per train for average weight of 60 kg (standing capacity : 8 person/m^2), and the maximum speed is 70 km/h. [17]

4.2. The total axle load calculations for each train stations

To find the total passenger number in the train;

Total number of passenger = standing passenger + sitting passenger

From ERC data,

For rated passenger loading condition - 6 persons/m^2

For over-crowed loading condition - 8 persons/m^2

Table 4. 1: Weight of the train when in empty, normal, full and over loading condition [17]

Working condition	Seating capacity	Standing capacity	Total capacity (persons)	Total weight (t)
AW0	0	0	0	About 44
AW1	65	0	65	About 47.9
AW2	65	189	254	About 59.2
AW3	65	252	317	About 63

Therefore from this it is possible to find the area of the standing passenger no by taking the over-crowed passenger loading condition, the amount of passenger loading capacity is 317. [17]

Therefore to calculate the area of standing passenger

Standing passenger number ÷ Person/m^2 = Area for standing passenger (m^2) …………… eq 4.1

$252 \div 8 = 31.5\ m^2$

When calculating the total passenger number the extra sitting passengers also considered.

11t is the maximum limit of the axle that can carry.

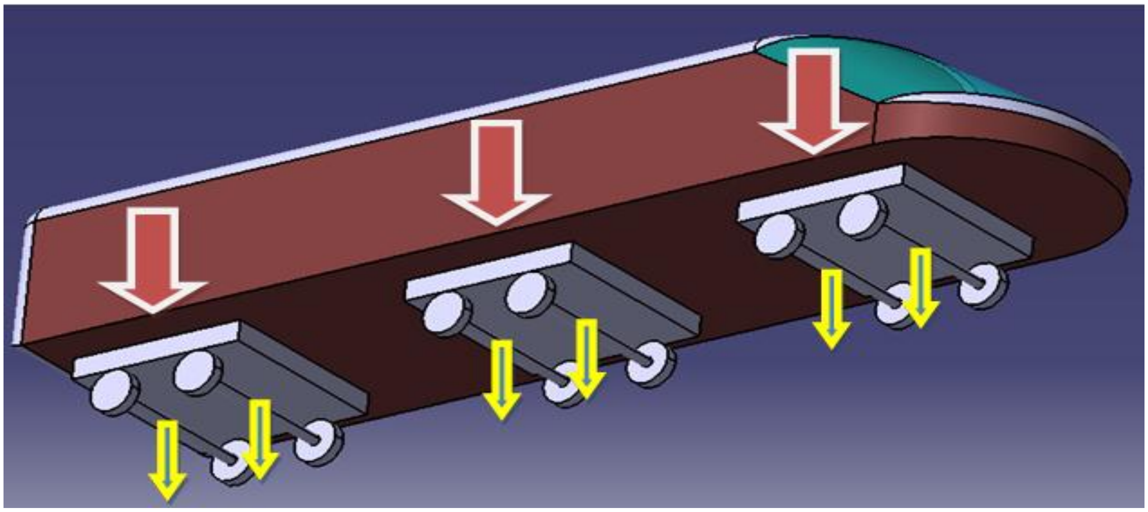


Fig 4. 4: Load Transfer from the car body to the wheels

To find the axle load that exerts by the empity vehicle weight is taken, 44 t.

Axle load = $\frac{Empty\ weight\ of\ the\ vehicle}{Number\ of\ Axle}$ ……………………………………………… eq 4.2

Axle load = $\frac{44t}{6}$ = 7.333t = 7,333 kg

Axle load = $7{,}333 \times 9.8 = 71{,}863.4$ N

Ten passengers per m^2 and over seated passenger are assumed for train over loading. This unacceptable load condition is based on the Addis Ababa city bus overloading trend. There is another unacceptable load condition based on the new Addis Ababa city locomotive train which has three five-in-one (five passengers seat on one long chair as indicated on figure 3.5) and twenty seven two- in-one chairs. One passenger seat Chair has 50cm width. The average human being width is 36cm. So in time of overloading it is assumed that three passenger seat on two chairs of the train and eight passengers seat on five in one chairs. Ten passengers are assumed also to stand per m^2. [18]

Table 4. 2: Number of Passengers loading in each station and their amount of weight

Stations	Number of seats	Person/m^2	No of sitting passengers (with extra passengers)	The area for standing passenger (m^2)	Standing passenger number	Total passenger number
Kality	65	8	73	31.5	252	325
Saris	65	8.5	70	31.5	268	338
Nefas Silk	65	9	70	31.5	284	354
Riche	65	9	72	31.5	284	356
Stadium	65	9.5	73	31.5	299	372
Leghar	65	10	72	31.5	315	387
Mexico	65	8	73	31.5	252	325
Sr. Lideta	65	7.5	71	31.5	236	307
Autobus tera	65	8.5	71	31.5	268	339
Atikilit tera	65	7	65	31.5	221	286
Menelik square	65	-	-	31.5	-	-

Therefore the passenger load and the total axle load for each station is calculated as follows in table. This table also shows the load transfer from the vehicle to the bogie and the axles.

Table 4. 3: Load transfer table for each station

No	Stations	No of people loaded In one train	Passenger load (Kg)	Passenger load per bogie (kg)
1	Kality	325	325 × 60 = 19,500	19,500 ÷ 3 = 6,500
2	Saris	338	338× 60 = 20,280	20,280 ÷ 3 = 6,760
3	Nefas Silk 2	354	354 × 60 = 21,240	21,240 ÷ 3 = 7,080
4	Riche	356	356 × 60 = 21,360	21,360÷ 3 = 7,120
5	Stadium	372	372 × 60 = 22,320	22,320 ÷ 3 = 7,440
6	Leghar	387	387 × 60 = 23,220	23,220 ÷ 3 = 7,740
7	Mexico	325	325 × 60 = 19,500	19,500 ÷ 3 = 6,500
8	St. Lideta	307	307 × 60 = 18,420	18,420 ÷ 3 = 6,140
9	Autobus Tera	339	339 × 60 = 20,340	20,360 ÷ 3 = 6,780
10	Atikilit Tera	286	268 × 60 = 16,080	16,080 ÷ 3 = 5,360
11	Menelik Square	-	-	-

No	Passenger load per axle	Total axle load (Kg)	Weight = mg (N)	Total axle load per wheel (N)
1	6,500 ÷ 2 = 3,250	7,333 + 3,250 = 10,583	10,583 × 9.8 = 103,713.4	103,713.4 ÷ 2 = 51,856.7
2	6,760 ÷ 2 = 3,380	7,333 + 3,380 = 10,713	10,713 × 9.8 = 104,987.4	104,987.4 ÷ 2 = 52,493.7
3	7,080 ÷ 2 = 3,540	7,333 + 3,540 = 10,873	10,873 × 9.8 = 106,555.4	106,555.4 ÷ 2 = 53,277.7
4	7,120 ÷ 2 = 3,560	7,333 + 3,560 = 10,893	10,893 × 9.8 = 106,751.4	106,751.4 ÷ 2 = 53,375.7
5	7,440 ÷ 2 = 3,720	7,333 + 3,720 = 11,053	11,053 × 9.8 = 108,319.4	108,319.4 ÷ 2 = 54,159.7
6	7,740 ÷ 2 = 3,870	7,333 + 3,870 = 11,203	11,203 × 9.8 = 109,789.4	109,789.4 ÷ 2 = 54,894.7

7	6,500 ÷ 2 = 3,250	7,333 + 3,250 = 10,583	10,583 × 9.8 = 103,713.4	103,713.4 ÷ 2 = 51,856.7
8	6,140 ÷ 2 = 3,070	7,333 + 3,070 = 10,403	10,403 × 9.8 = 101,949.4	101,949.4 ÷ 2 = 50,974.7
9	6,780 ÷ 2 = 3,390	7,333 + 3,390 = 10,723	10,723 × 9.8 = 105,085.4	105,085.4 ÷ 2 = 52,542.7
10	5,360 ÷ 2 = 2,680	7,333 + 2,680 = 10,013	10,013 × 9.8 = 98,127.4	98,127.4 ÷ 2 = 49,063.7
11	-	7,333 + 0 = 7,333	7,333 × 9.8 = 71,863.4	71,863.4 ÷ 2 = 35,931.7

Where

Passenger load = No of people loaded × 60 kg (because average weight of one passenger is 60kg) …………………………………………………………………………………..… eq 4.3

Passenger load per bogie = passenger load ÷ 3 (because one train have three bogies) …..… eq 4.4

Passenger load per axle = Passenger load per bogie ÷ 2 ………………………………....… eq 4.5

(because the bogie have two axles)

Total axle load = empty train axle load + Passenger load per axle ……………………..….. eq 4.6

Total axle load per wheel = Total axle load ÷ 2 ………………………..……….……..… eq 4.7

1 ton = 1,000Kg

4.3. Calculating the Nominal Bending Moment on the axle

AS Kidanemariam Gebretsadik calculated the nominal bending stress on the wheel set, for the Calculation of the nominal bending moment of the axle, two parameters are observed. They named as Independent (Fixed) and Dependent (Variable) parameters. [8]

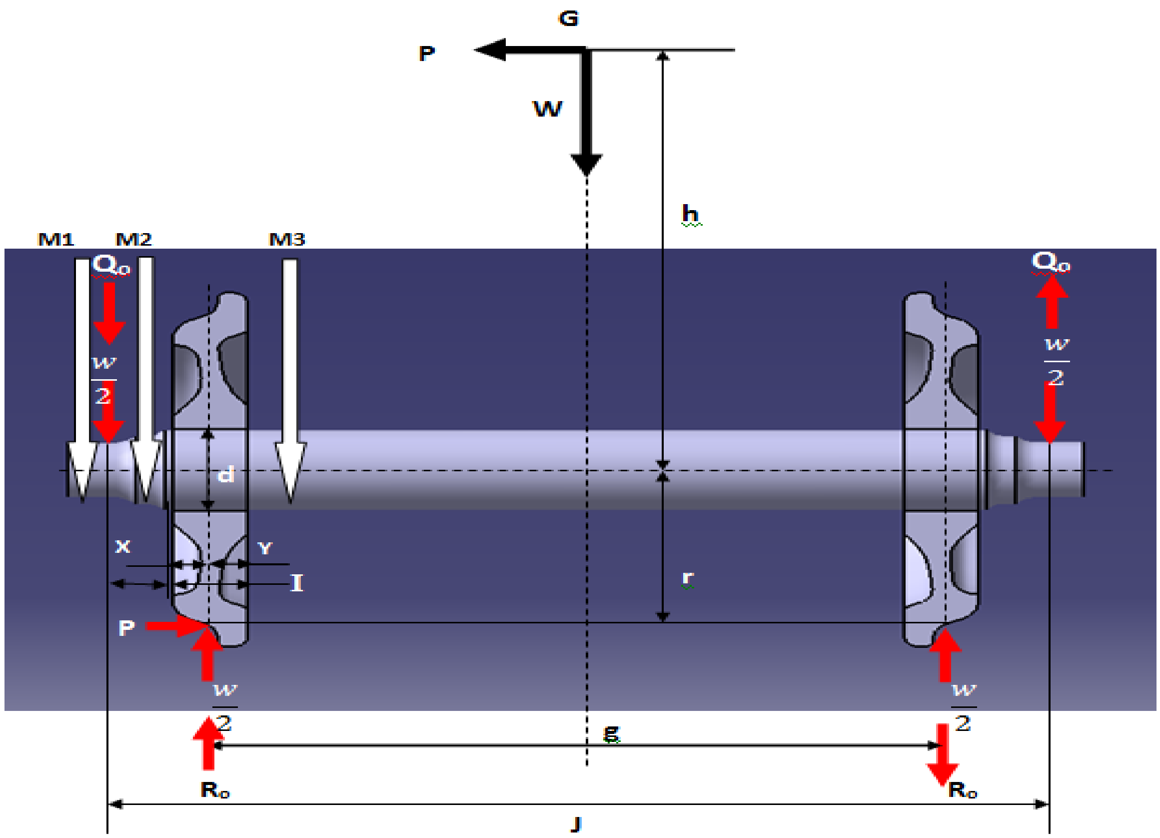


Fig 4. 5: Applied alternate forces on the axle[8]

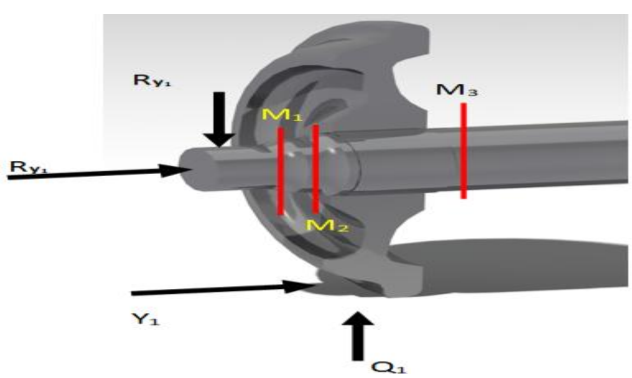


Fig 4. 6: Applied alternative forces on the wheel and developed momentum through the axle [8]

Table 4. 4: Independent (Fixed) parameters [8]

D	Axle diameter	124.14mm
R	Wheel radius	0.33 m
W	Total Vehicle weight or axle load	11.5 t
J	Distance between journal center	1.853 m
G	Wheel trade distance	1.551 m
A	Distance between journal center and the end of the wheel set	0.0715 m
H	Height from axle to center of gravity	1.024 m
X	Distance from the outside wheel hub to the contact load	0.067 m
Y	Distance from the inside wheel hub to the contact load	0.067m
I	x + y	0.134m
G	Center of gravity	No value
a_v	Vertical acceleration coefficient	1.31
M	Factor of safety	1
a_l	Lateral acceleration	2.04
Z	Section of Modulus = $\pi(D^4 – d^4) \div 32D$	$0.18579m^3$
μ	Friction Coefficient	0.16

Section of modulus $z = \frac{\pi(D^4 - d^4)}{32D}$ …………………………………………………….… eq 4.8

$$Z = \frac{\pi(124.14\text{mm}^4 - 40\text{mm}^4)}{32D} = 185{,}792.54\text{mm}^3 = 0.18579\text{m}^3$$

4.4. The Nominal Bending Stress Calculation of the Axle for each Stations

4.4.1. The Nominal Bending Stress Calculation for Kality Station

To calculate the maximum bending stress first it is better to calculate the input parameter like , Horizontal force where α_L mean lateral accelerate (P), Vertical force of journal bearing by P (Q_o) and Vertical force of threads by P (R_o). After that we can calculate the maximum bending stress.

Table 4. 5: Value of P, Qo and Ro for Kality Station

Horizontal force where α_L mean lateral acceleration	P	α_{LW}	$2.04 \times 103{,}713.4$ N	211,575.34 N
Vertical force of journal bearing by P	Q_o	$\frac{Ph}{j}$	$\frac{211{,}575.34\ \mathrm{N} \times 1.024\mathrm{m}}{1.853\ \mathrm{m}}$	116,920.2 N
Vertical force of threads by P	R_o	$\frac{P(h+r)}{g}$	$\frac{211{,}575.34\ \mathrm{N}(1.024\mathrm{m} + 0.33\mathrm{m})}{1.551\ \mathrm{m}}$	184,702.13 N

Bending Moment 1

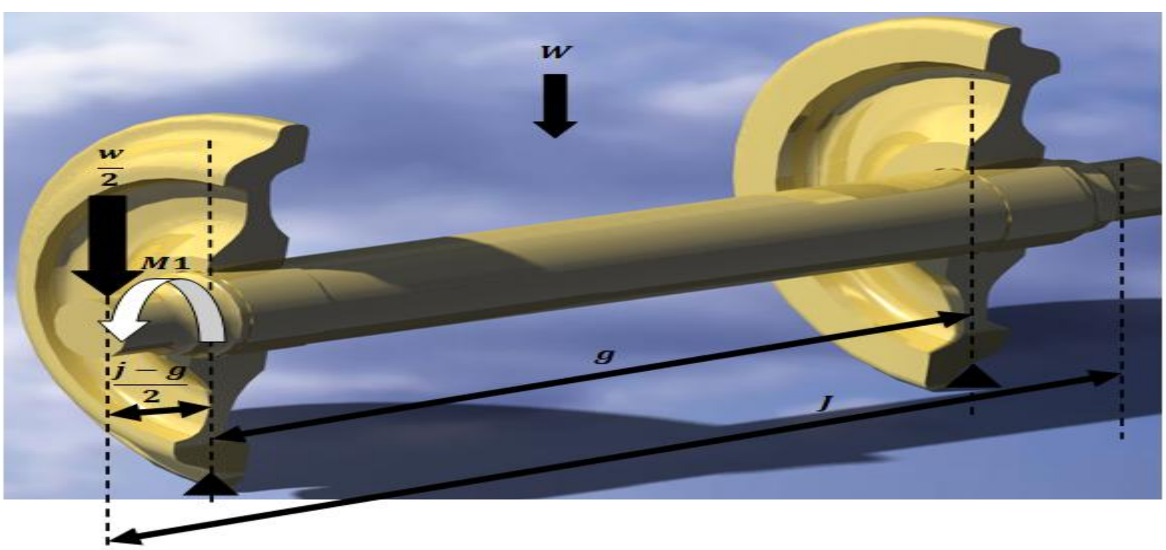


Fig 4. 7: Bending moment 1

$M_1 = \frac{(j-g)W}{4}$ …………………………………………………………………...………..… eq 4.9

Bending Moment 2

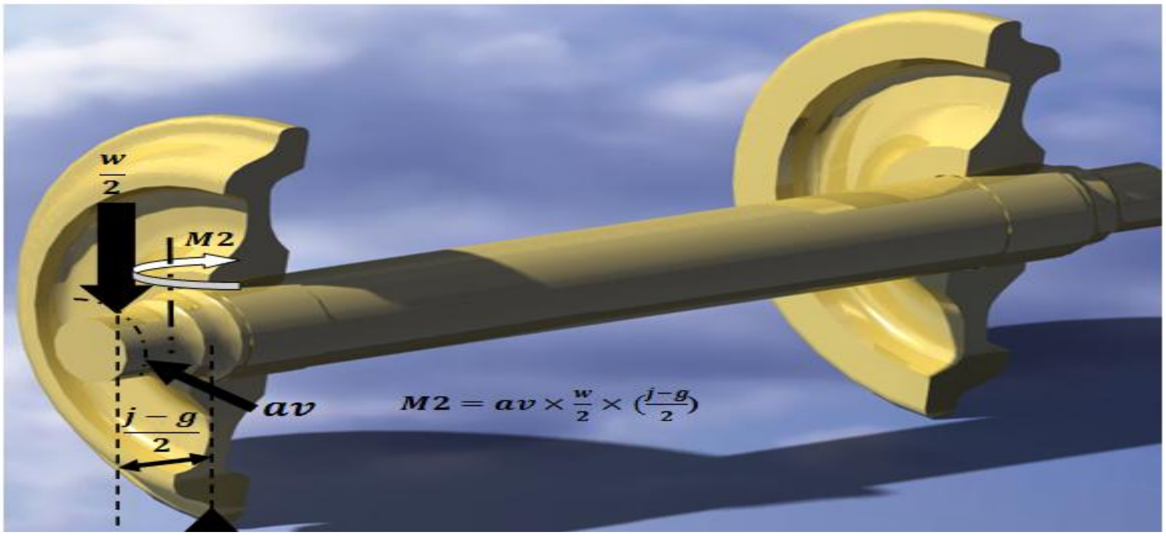


Fig 4. 8: Bending Moment 2

$M_2 = a_v M_1$……………………………………………….……………………...………..….eq 4.10

Bending Moment 3

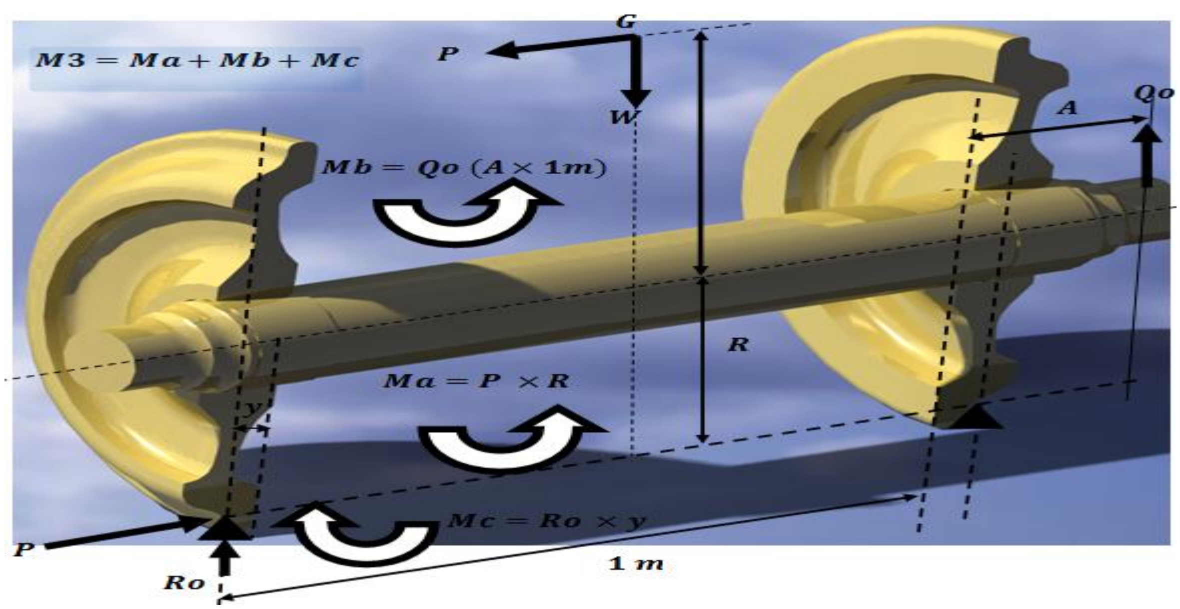


Fig 4. 9: Bending Moment 3

$M_3 = P.r + Q_o(a+1) - R_o y$………………………………………………………..……….… eq 4.11

Substituting the values in the formula

$M_1 = \frac{(1.853-1.551)103{,}713.4\ \text{N}}{4} = 7{,}830.36$ Nm

$M_2 = 1.31 \times 7{,}830.36$ Nm $= 10{,}257.77$ Nm

$M_3 = (211{,}575.34\ \text{N} \times 0.33\text{m}) + 116{,}920.2\ \text{N}\ (0.0715\text{m}+1) - (184{,}702.13\ \text{N} \times 0.067)$

$= 182{,}724.81$ Nm

The Maximum or Nominal Bending Moment

$\boldsymbol{\sigma_b = m\frac{(M1+M2+M3)}{z}}$ ……………………………………………………..………..… eq 4.12

The Maximum or Nominal Bending Moment

$\sigma_b = \frac{1(7{,}830.36 + 10{,}257.77 + 182{,}724.81)\text{Nm}}{0.18579} = 1{,}080{,}859.8\ \frac{\text{N}}{m^2} = 1{,}080.85\frac{\text{KN}}{m^2}$

Table 4. 6: Value of P, Qo, Ro and Moments for each Station

Stations	P(N)	Q_o (N)	R_o (N)	M_1(**Nm**)	M_2(**Nm**)	M_3(**Nm**)	$\sigma_b(\frac{N}{m^2})$
Kality	211,575.34	116,920.2	184,702.13	7,830.36	10,257.77	182,724.81	1,080,859.8
Saris	214,174.29	118,356.43	186,970.97	7,926.54	10,383.77	184,969.38	1,094,136.9
Nefas Silk 2	217,373.02	120,124.02	189,763.4	8,044.9	10,538.8	187,731.83	1,110,477.04
Riche	217,772.86	120,345.06	190,112.48	8,059.73	10,558.25	188,077.24	1,112,520.7
Stadium	220,971.58	122,112.73	192,904.9	16,683.35	21,855.19	190,839.78	1,234,610.7
Leghar	223,970.37	123,769.92	195,522.81	8,289.09	10,858.7	193,429.71	1,144,181.6
Mexico	211,575.34	116,920.2	184,702.13	7,830.36	10,257.77	182,724.8	1,080,859.7
St. Lideta	207,976.77	114,931.6	181,560.6	7,697.18	10,083.3	179,616.98	1,062,476.24
Autobus Tera	214,374.2	118,466.9	187,145.5	14,129.49	18,509.63	185,142.03	1,173,522.6
Atikilit Tera	200,179.9	110,622.9	174,754.08	7,408.6	9,705.3	172,883.28	1,022,644.8
Menelik Square	146,601.3	81,014.45	70,724.4	5,425.69	7,107.65	130,446.78	1,539,158.4

Calculating the Angular Velocity

$$\omega = \frac{v}{r} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \text{eq 4.13}$$

The speed of the train is 70 km/h. when converting it in to m/s

$$70 \times \frac{1000}{3600} = 19.44 \frac{m}{s}$$

The radius of the wheel is 330 mm or 0.33m. Therefore

$$\omega = \frac{19.44}{0.33} = 58.9 \text{ rad/s}$$

4.5. Distance between Stations

Table 4. 7: Distance between each station (information gathered from ERC)

Stations	**Distance between each Station (Km)**	**Selected Stations**	**Distance between Selected Stations (Km)**
Kality – Abo Mazoria	0.95	Kality – Saris	1.795
Abo Mazoria – Saris	0.845		
Saris – Adey Abeba	0.535	Saris - Nifas Silk 2	2.664
Adey Abeba - Nifas Silk 1	0.995		
Nifas Silk 1 - Nifas Silk 2	1.134		
Nifas Silk 2 – Lancha	1.969	Nifas Silk 2 - Riche	3.134
Lancha - Temenja Yazh	0.555		
Temenja Yazh - Riche	0.610		
Riche – Meshualekia	0.463	Riche – Stadium	1.371
Meshualekia - Stadium	0.908		
Stadium – Legar	0.445	Stadium – Legar	0.445
Legar – Mexico	0.560	Legar – Mexico	0.560
Mexico – Tegbared	0.688	Mexico - S. Lideta	1.423
Tegbared – S. Lideta	0.735		
S. Lideta – Darmar	0.591	S. Lideta - Autobus tera	2.878
Darmar – Abnet	0.739		
Abnet – Sebategna	0.881		
Sebategna – Autobus tera	0.667		
Autobus tera – Gojam b.	0.613	Autobus Tera - Atikilit Tera	1.558
Gojjam B. – Atililit Tera	0.945		
Atililit Tera - D.Menilik	0.743	Atikilit Tera - D.Menilik	0.743

4.5.1. Calculation of Train Traction Time between Stations

1 revolution = $2\pi r$ = Circumference of the wheel = $2 \times \pi \times 0.33$ m = 2.07 m

Where r is the radius of the wheel

Therefore in order to find the number of revolution (N) between each station

N between stations = distance between stations ÷ $2\pi r$ (Circumference of the wheel) ……. eq 4.14

= distance between stations ÷ 2.07 m

Table 4. 8: Number of revolution between each station

Selected Stations	**Distance between stations (S in M)**	**Taken Time between Stations (T = S ÷ V), Where V is in m/s**	
		S ÷ V	**T (in s)**
Kality – Saris	1,795	1,795 ÷ 19.44	92.335
Saris - Nifas Silk 2	2,664	2,664 ÷ 19.44	137.037
Nifas Silk 2 – Riche	3,134	3,134 ÷ 19.44	161.213
Riche – Stadium	1,371	1,371 ÷ 19.44	70.524
Stadium – Legar	445	445 ÷ 19.44	22.89
Legar – Mexico	560	560 ÷ 19.44	28.806
Mexico - S. Lideta	1,423	1,423 ÷ 19.44	73.199
S. Lideta - Autobus tera	2,878	2,878 ÷ 19.44	148.04
Autobus tera - Atikilit Tera	1,558	1,558 ÷ 19.44	81.68
Atikilit Tera - D.Menilik	743	743 ÷ 19.44	38.22

Among all the stations Legar station found loaded more passengers than the rest stations. Therefore for the analysis Legar station selected to further calculation.

4.5.2. Calculating the Tangential Force of the Train

AALRT train has 4 motors. Each of them generates 184.6 KW. [19]

From this it is easy to find the tangential Force of the train.

$\text{Power} = F_t \times V$ …………………………………………………………...………..… eq 4.15

Where V is the velocity of the train

$F_t = 4(\text{Power}) \div V$ ……………………………………………………..…..……….… eq 4.16

Where the trains have four motors

$Ft = 4(184.6KW) \div 70km/h$

$$Ft = \frac{4(184.6\text{KW})}{70 \times \frac{1000}{3600}} = 37.94 \text{ KN}$$

4.5.3. Calculating the Rotational velocity of the Train

$\omega = \frac{v}{rw}$ ……………………………………..…………………………...………..… eq 4.17

$$\omega = \frac{70 \times \frac{1000}{3600}}{0.33} = 58.9 \text{ rad/s}$$

4.5.4. Calculating acceleration of the Train

$a = \frac{Ft}{m}$ ………………………………………………………..……………...………..… eq 4.18

Where m is mass of the train

$$a = \frac{37940 \text{ N}}{15{,}104.54 \text{ kg}} = 2.511 \text{ m/s}^2$$

Moment of rotation or torque

$M = F_t \times r$ …………………………………………………………………..…..……….… eq 4.19

$M = 37940 \text{ N} \times 0.33\text{m} = 12{,}520.2 \text{ Nm}$

4.5.5. Fatigue Strength – Calculating Fatigue life of the Axle

In some design applications the number of load cycles the element is subject to, is limited(less than 10^6) and therefore there is no need to design for infinite life using the endurance limit. In such cases we need to find the fatigue strength with the desired life. For the high cycle fatigue (10^3-10^6), the line equation is $S_f = aN^b$ where the constants "a" (y intercept) and "b" (slope) are determined from the end point $(S_f)10^3$ and $(S_f)10^6$ as : [20]

$a = \frac{(s_f)^2\ 10^3}{s_e}$ and $b = -\frac{\log\left(\sigma_f'/s_e\right)}{\log(N_e)}$ ……………………………………… eq 4.20

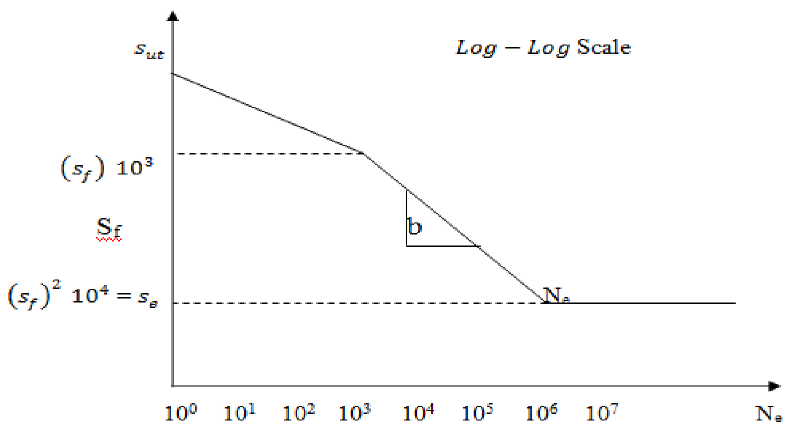


Fig 4. 10: S- N Curve [20]

Fatigue life (N) at a given fatigue stress (σ) is found as:

$N = (\frac{\sigma}{a})^{\frac{1}{b}}$, where $b = -\frac{1}{3}\log(\frac{fS_{ut}}{S_{ut}})^2$ ………………………………..…………………… eq 4.21

S_{ut} = ultimate tensile strength

Se = endurance limit, where $S_e = 0.5\ S_{ut}$ ……………………………..…………………… eq 4.22

Where $a = (\frac{fS_{ut}}{S_e})^2$ and $f = \left(\sigma_f'/S_{ut}\right)(2\times10^3)^b$ Where, ……………………………..……… eq 4.23

σ_f' is true stress fracture.

$\sigma_f' = S_{ut} + 345$ Mpa ………………………………………………...…………………… eq 4.24

For S_{ut} values less than 490Mpa, use f = 0.9 to be conservative [20]

Fatigue life Calculation from Abaqus Von misses stress result on Leghar Station

Fatigue life (N) at a given fatigue stress (σ) is found as:

$N=(\frac{\sigma}{a})^{\frac{1}{b}}$ …………………………………………………………………………… eq 4.25

, where $a =\frac{(S_f)^2}{S_e} 10^3$ but $(s_f)10^3 = fs_{ut}$, where f is called fraction.

$a = \left(\frac{fS_{ut}}{S_e}\right)^2$ …………………………………………………………………………… eq 4.26

and $b = \frac{\log\left({\sigma'_f}/{s_e}\right)}{\log(2Ne)} = -\frac{1}{3}\log(\frac{fS_{ut}}{S_e})^2$ …………………………………………………… eq 4.27

Ne = endurance limit of number of cycle (For the high-cycle fatigue 10^3 -10^6)

S_{ut} = ultimate tensile strength

S_f = Fatigue strength

Se = endurance limit, where

$S_e = 0.5\ S_{ut}$ …………………………………………………………………………… eq 4.28

The ultimate tensile strength for the AIN Steel is

S_{ut} = 550Mpa, where $S_e = 0.5S_{ut}$

$S_e = 0.5\times550$Mpa = 275Mpa

$\sigma'_f = S_{ut} + 345$ Mpa …………………………………………………………………… eq 4.29

Where σ'_f is called stress at fraction

$\sigma'_f = 550 + 345$ Mpa = 895 Mpa

$b= - \frac{log\left({\sigma'_f}/{s_e}\right)}{\log(2N_e)}$

$$b = \frac{\log\left(\sigma_f^{'}/s_e\right)}{\log\left(2\times10^3\right)} = \frac{\log\left(895/275\right)}{\log\left(2\times10^3\right)} = -0.1552$$

$$f = \frac{\sigma_f^{'}}{S_{ut}}(2 \times 10^3)^{b}$$

$$f = \frac{895}{550}(2 \times 10^3)^{-0.1552} = 0.5001$$

$$a = \left(\frac{f S_{ut}}{S_e}\right)^2$$

$$a = \left(\frac{0.5001 \times 550 \text{ Mpa}}{275}\right)^2 = 275.2 \text{ Mpa}$$

The value of Von misses stress result from Abaqus software is 5.8Mpa.Therefore the fatigue life of the axle is;

The fatigue life Calculation from Abaqus Von misses stress result based on the worst Condition

The value of Von misses stress result from Abaqus software is 18.4Mpa.Therefore the fatigue life of the axle is;

$$N = \left(\frac{\sigma}{a}\right)^{\frac{1}{b}}$$

$$N = \left(\frac{18.4 \text{ Mpa}}{275.2\ Mpa}\right)^{\frac{1}{-0.1552}} = 3.7 \times 10^7 \text{ Cycles}$$

4.5.6. Modeling the axle

To model the axle we used Catia v5 software. By using the data that gathered from ERC documents, practical measuring and Kiadnemariam G/Tsadik project, modeling the axle is done. The Analysis Part is done using Abaqus Software.

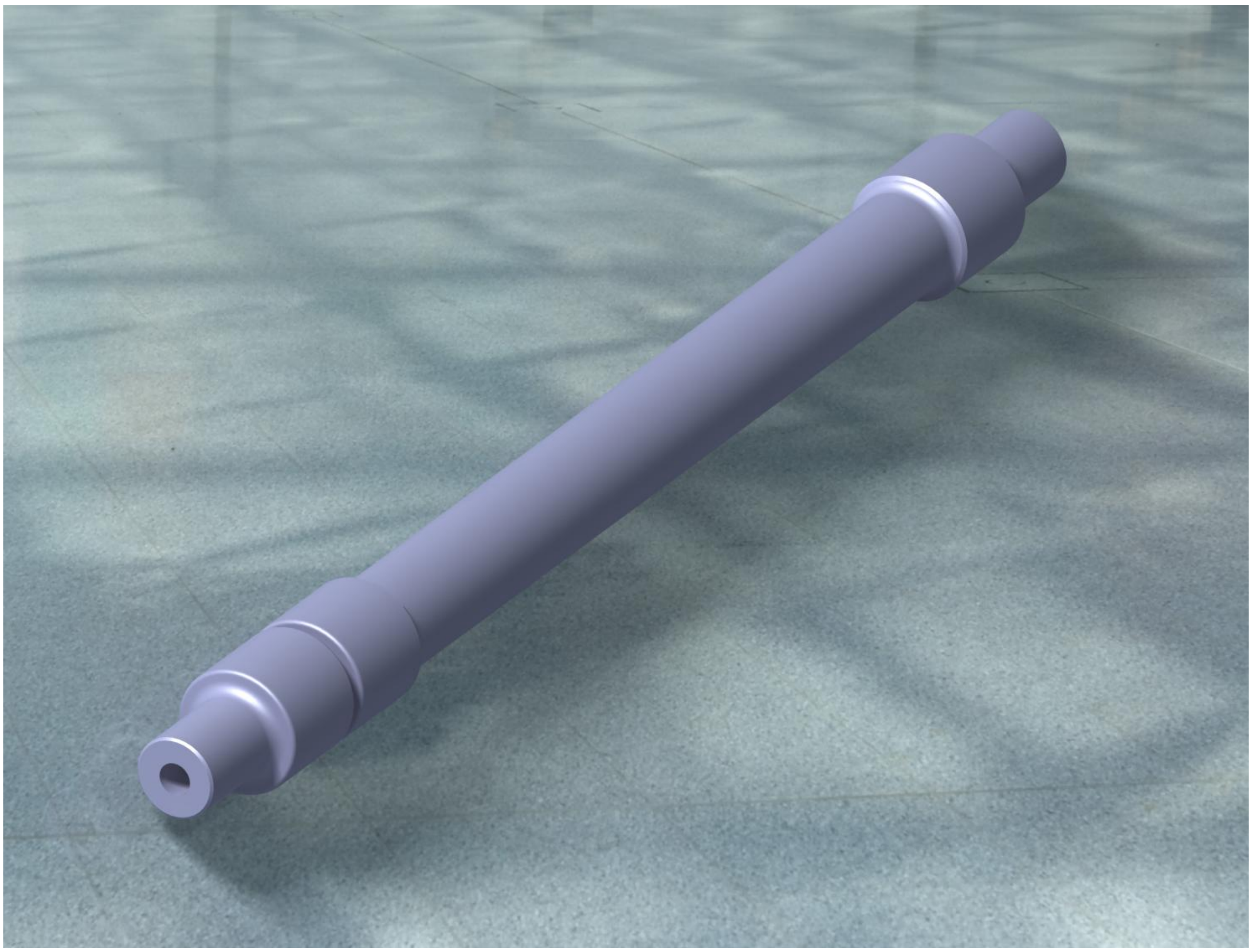


Fig 4. 11: 3d model axle designed on Catia V5

Chapter Five

5. Result Discussion

5.1. The Von Messes Stress at Legehar station (At the worst Loading condition)

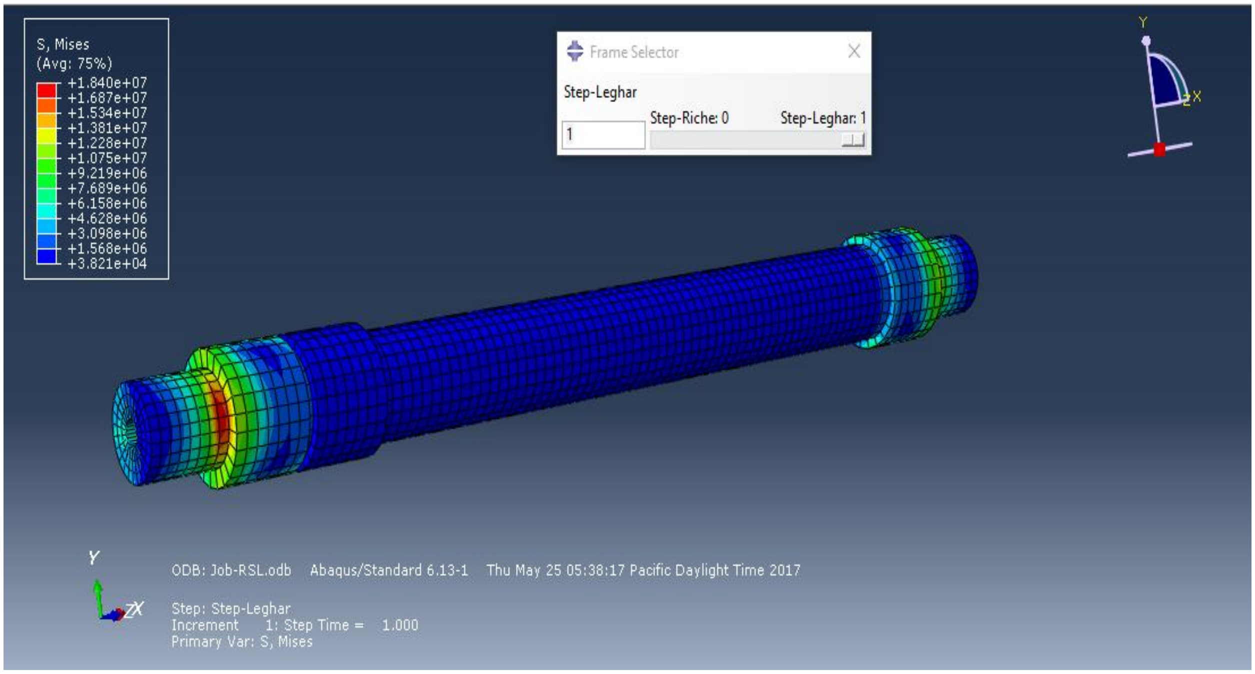


Fig 5. 1: Von-Misses Stress

At this station, it is where the maximum passenger load recorded. Therefore the value of the stress, pressure and reaction force is comparatively high other than the rest of the stations. The picture above shows that the value of the von misses stress that develops on the axle at maximum loading conditions. The red color shows that maximum stress develops on that area. It is about 18.4 Mpa. The reason why that the maximum stress generates on that area is because it is located between where the load applies and support and compression and tension forces develops on it. The rainbow or the composition of the colors shows that the amount of the stress develops on the axle. The stress rises when the colors observe from cold to warm. That means the red color shows the maximum stress and the blue color shows the minimum stress develops on the axle. The minimum stress that develops on the axle is that 0.03821 Mpa. It is the part of the axle where almost load is not applied. By using this Von Misses stress the life time or the number of the axle can be calculated. The red color shows that there is high stress at that region that can develop crack at the final destination of the axle after number of cyclic loading.

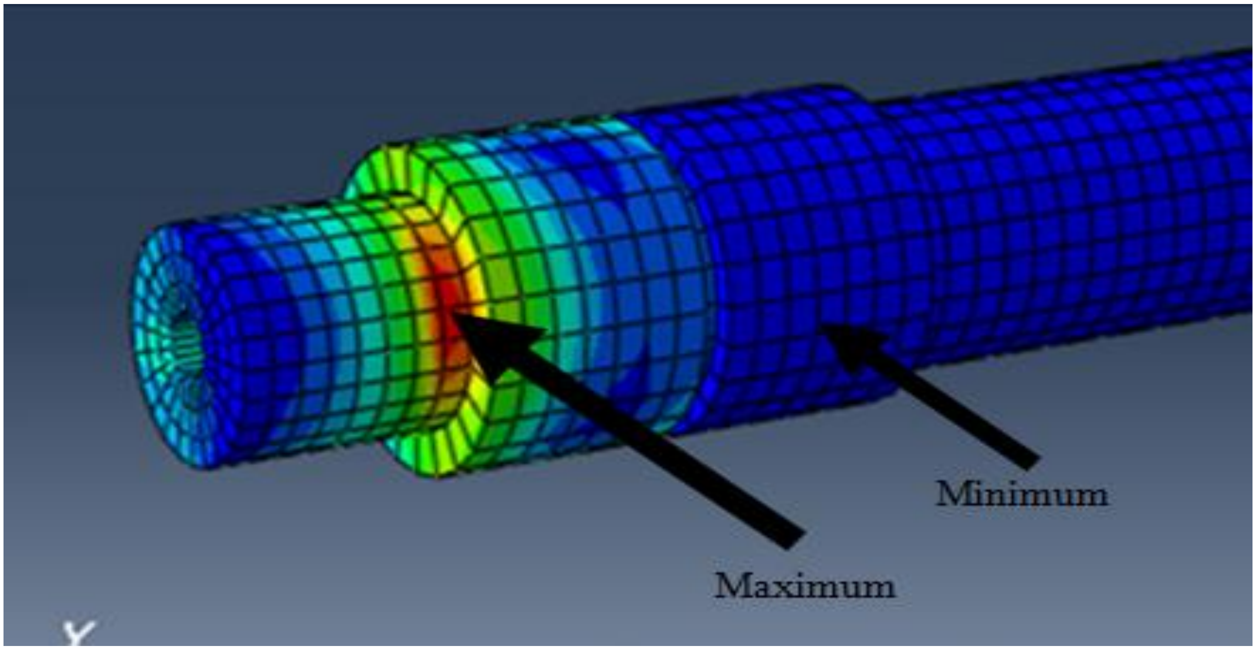


Fig 5. 2: Maximum and Minimum point of Von Mosses Stress

Table 5. 1: Maximum and Minimum Result of Von Misses Stress

Colors	**Von Misses Stress (N/m^2)**	
	1.840×10^7	Maximum
	1.687×10^7	
	1.534×10^7	
	1.381×10^7	
	1.228×10^7	
	1.075×10^7	
	9.219×10^6	
	7.689×10^6	
	6.158×10^6	
	4.628×10^6	
	3.098×10^6	
	1.568×10^6	
	3.821×10^4	Minimum

When the stress develops on the axle increases, the Fatigue life or the life span of the axle decreases.

5.2. The Reaction Force developed on the Axle at Worst condition (at Leghar Station)

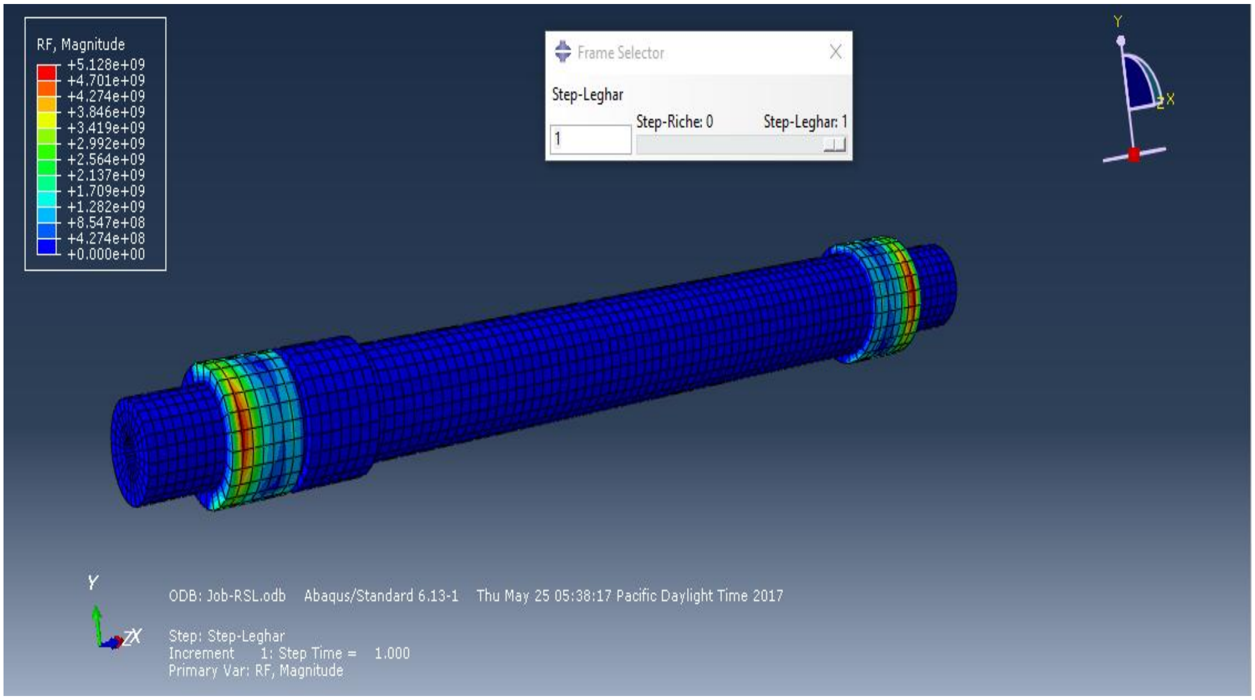


Fig 5. 3: Reaction Forces

If there is action force there will be a reaction force. The picture shows the values of the reaction forces that develop on the axle where the wheels are located on it. The red colors that shown on the above picture is where the wheels are located. The amount of the maximum reaction force is 5.128×10^9 N. The cyclic action and the reaction forces are the reason for the fatigue failure for the axle.

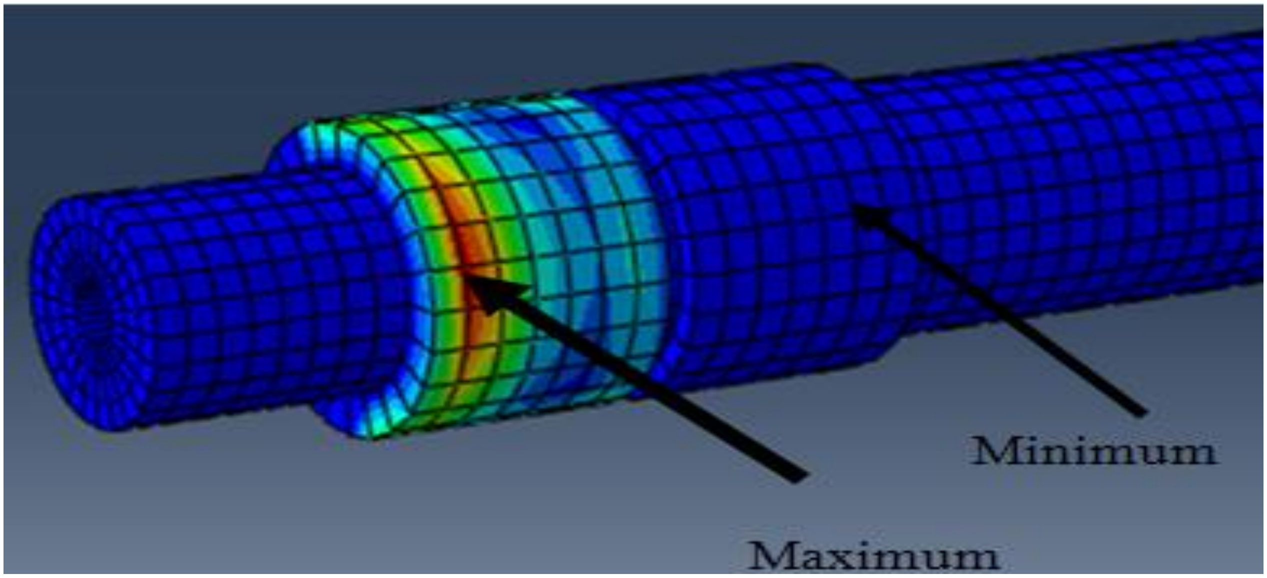


Fig 5. 4: Maximum and Minimum point of Reaction Forces

The reason behind why the maximum reaction force developed on this region is because the wheels are located on that region. The wheels are also directly connected with the rails. And the rail is fitted with the ground. Therefore the reaction force is reacted through the wheels.

Table 5. 2: Maximum and Minimum Result of Reaction Forces

Colors	Reaction Forces(N)	
	5.128×10^9	Maximum
	4.701×10^9	
	4.274×10^9	
	3.846×10^9	
	3.419×10^9	
	2.992×10^9	
	2.564×10^9	
	2.137×10^9	
	1.709×10^9	
	1.282×10^9	
	8.547×10^8	
	4.274×10^8	
	0	Minimum

The reaction force and the fatigue life of the axle are inversely proportional. When one of them increases, the other decreases and vice versa.

5.3. Pressure Developed on the Axle at worst loading Condition

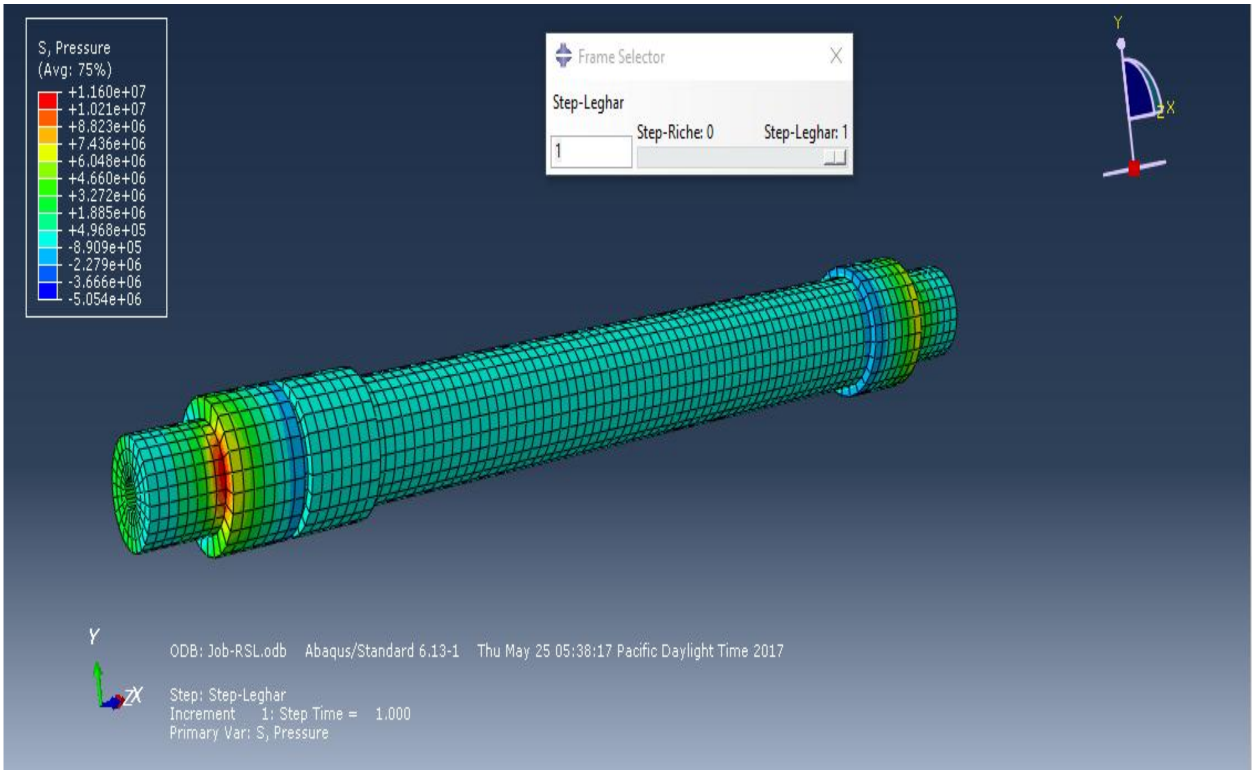


Fig 5. 5 : Pressure

Pressure is the force on some certain area that develops externally. On the figure it is shown that the pressure developed on the axle. This pressure is generates because of the reaction forces of the wheels. The pressure rises on the stepped part of the axle where the wheels are located. The applied loads also create pressure on the axle. But the maximum one is shown on the figure on the region where the wheels are located.

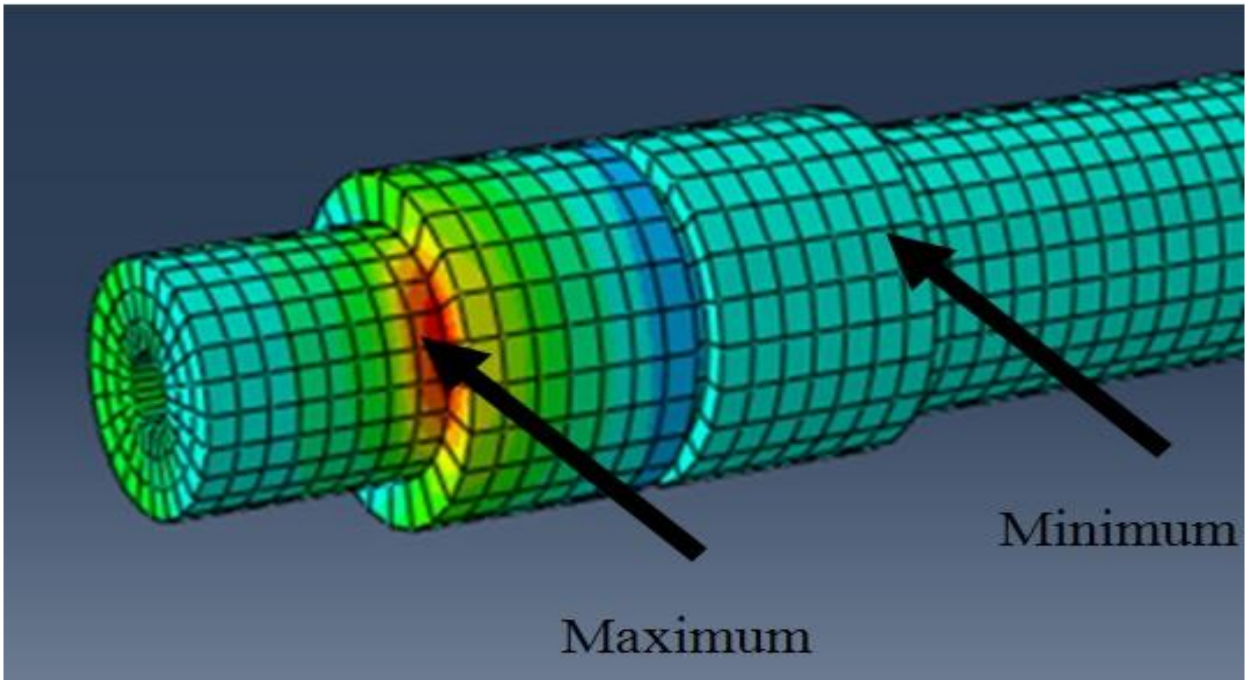


Fig 5. 6: Maximum and Minimum point of Pressure Developed on the Axle

Table 5. 3: Maximum and Minimum Result of Pressure Develops

Colors	**Pressure(N/m^2)**	
	1.16×10^7	Maximum
	1.021×10^7	
	8.823×10^6	
	7.436×10^6	
	6.048×10^6	
	4.660×10^6	
	3.272×10^6	
	1.885×10^6	
	4.968×10^5	
	-8.909×10^5	
	-2.279×10^6	
	-3.667×10^6	
	-5.054×10^6	Minimum

5.4. Maximum Principal Stress

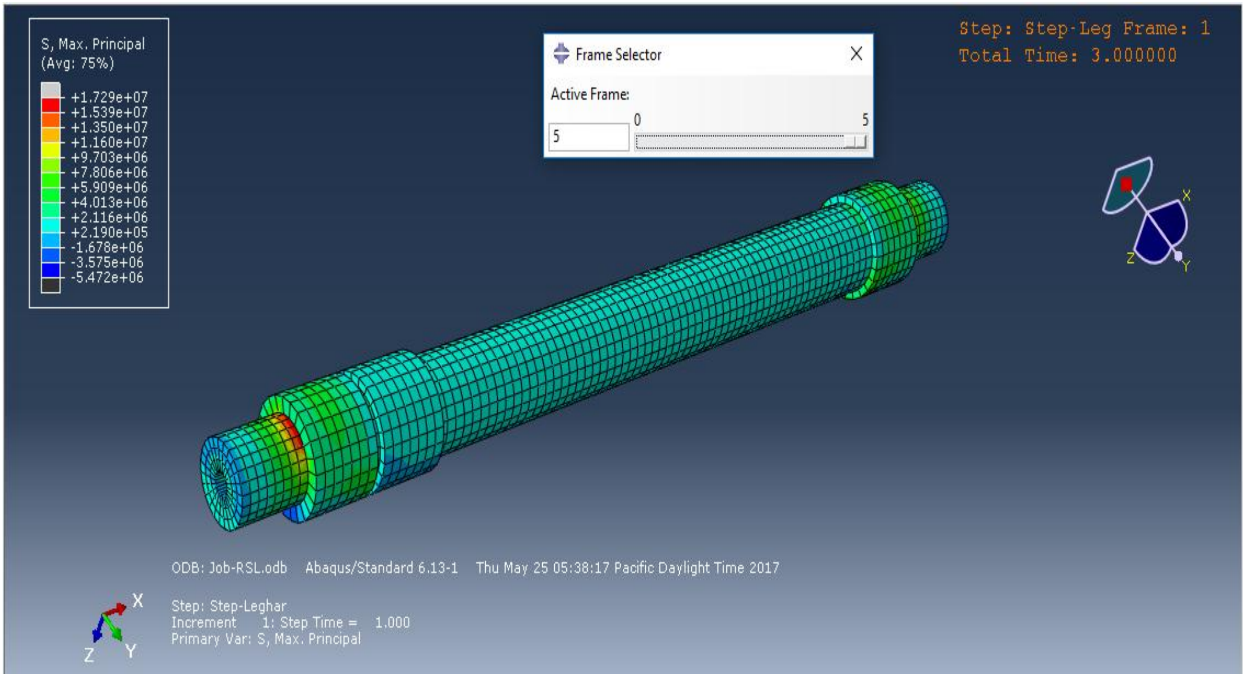


Fig 5. 7: Maximum Principal Stress

Here on the figure above shows where and in how much amount the Maximum Principal stress develops on the axle around the stepped region. The maximum principal stress on this region is 1.729×10^7.When the stress on the axle increase and increase, the fatigue life of the axle gets decrease and decrease. So they have inversely proportional relationship. The reason why the maximum principal stress develops around this area is because it is found in the middle of the region where the reaction forces of the wheels and the applied force applied and it is the end of the small diameter to the bigger diameter of stepped axle. At this region, it is where the Maximum Principal Stress is getting high and at final stage of the axle life it's where the crack initiates.

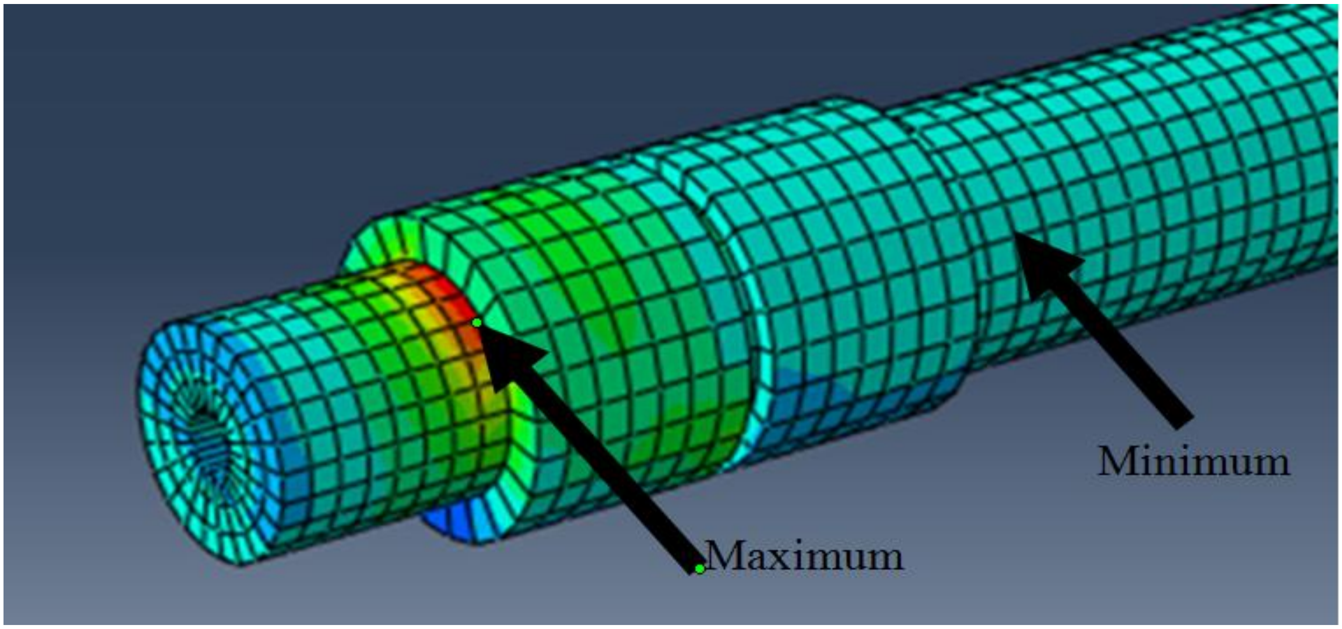


Fig 5. 8: Maximum and Minimum Principal Stress Points developed on the Axle

Table 5. 4: Maximum and Minimum Result of Principal Stress

Colors	**Maximum Principal Stress (N/m^2)**	
	1.729×10^7	Maximum
	1.539×10^7	
	1.350×10^7	
	1.160×10^7	
	9.703×10^6	
	7.806×10^6	
	5.909×10^6	
	4.013×10^6	
	2.116×10^6	
	2.190×10^5	
	-1.678×10^6	
	-3.575×10^6	
	-5.472×10^6	Minimum

Table 5. 5: Result of the worst case (on Leghar Stations)

Station	Maximum principal Stress (N/m^2)	Pressure (N/m^2)	Reaction Forces (N)	Maximum Von – Misses Stress (N/m^2)	N – Number of cycles or Fatigue life
Leghar	1.72×10^7	1.160×10^7	5.12×10^9	1.84×10^7	3.7×10^7

At Leghar station which is on the worst loading condition the result of the Maximum Principal Stress and the von misses stress are 1.72×10^7 N/m^2 and 1.84×10^7 N/m^2 respectively. The result of the pressure is 1.160×10^7 N/m^2. The reaction force is 5.12×10^9 N. The fatigue life is as calculated above is 3.7×10^7 cycles. When the result compares with the standard axle life, the AALRT has less fatigue life. Some references used to compare the result.

Rostislav Fajkoš, Bohumír Strnadel and **Radim Zima** on their study on Fatigue Limit of Induction Hardened Railway Axles has been clearly demonstrated that the final fracture instability can be, in the case of **Giga-cycle** fatigue. [4]

C.Klinger, D.Bettge, R.Haccker, T.Heckel, D.Gohlke,D.Klingbei, When they study on Failure Analysis on A Broken Railway Axle, A specific axle operated with a very high cycles in service $3*10^6$ km~10^9 cycles. [12]

Chapter Six

6. Conclusion and Recommendation

6.1. Conclusion

The main purpose of the project is finding the fatigue life of the axle on its worst condition or case. According to the data collected and by using approaches and software the life of the axle found. It is 3.7×10^7 cycles. For most materials this kind of life is infinite. But for axle, it is applicable or operational. A failsafe design for the axle is not acceptable, because failure on the axle causes derailment which results such a massive damage for property and a loss of many lives. Practically it is not possible to know the life of the axle from AALRT due to the lack of transparency that hinders the comparison with the calculated axle fatigue life on this project. But it is tried to compare the axle fatigue life result with the standard axle life. When comparing with the standard axle life it's found that the life of the AALRT axle life has less fatigue life. According to the standard axle life the cycle of the fatigue life reaches more than Giga cycles. The result validates and proven. In general when the amount of passengers loaded on the train increase, the life of the axle gets decrease. Therefore AALRT should load passengers according to its design load limit to make the life of the axle longer. Some ideas can be propose for the solution for this problem. If the train add additional trailer the concentration of load on one trailer will be less and disperse. Another solution is that using better fatigue resistance material for the axle would make longer the life of the axle.

6.2. Recommendation

Due to the lack of transparency, it is not possible to find the fatigue life of the axle of AALRT. If the fatigue life that calculated in this study compared to the fatigue life of the AALRT axle better result would be concluded. It preferred to compare and contrast the fatigue life with calculated one. The other thing that should be recommended is that AALRT should have fatigue test laboratory to check the fatigue life and the crack initiation. This helps to detect the risk and give the proper service to axle and other train parts before too late.

6.3. Future works

In this thesis project the life of the axle is clearly calculated and solved. The main reason why it's concluded that the axle has decreased life span is by comparing the fatigue life with the standard axle life. This is because of the train is operating beyond its design load limit. According to minimizing the axle load if further future work can be studied around these area, the problem will have better and alternative solutions.

- Adding extra trailers to the operating train car would minimize the concentrated load on one place
- Using better fatigue resistance materials for the axle makes longer the life span of the axle. So if further researches can be studied around axle material, better solutions will be found.

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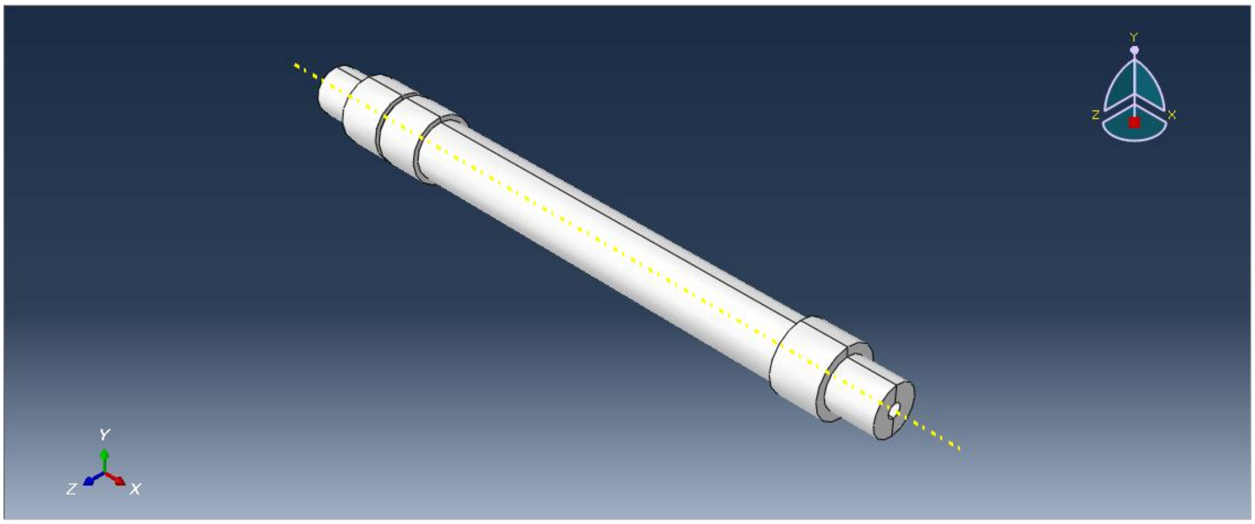
Appendix

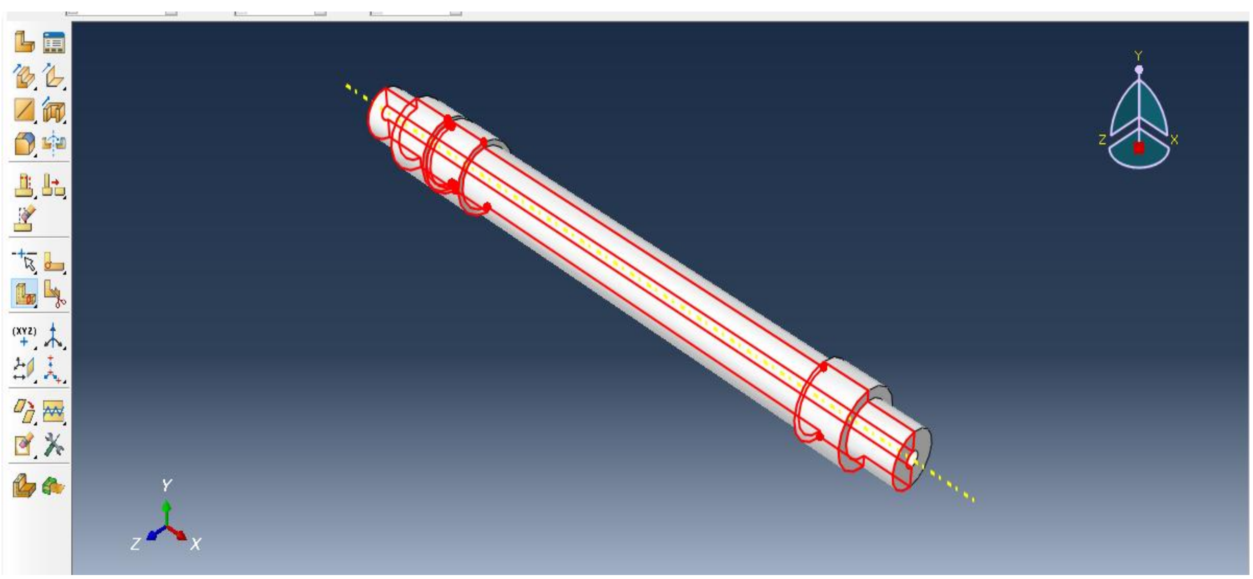
Steps to perform the stress analysis on Abacus software

1. Modeling

- Modeling the Axle on catia v5 software and import in Abacus software by changing it in the form of IGS format

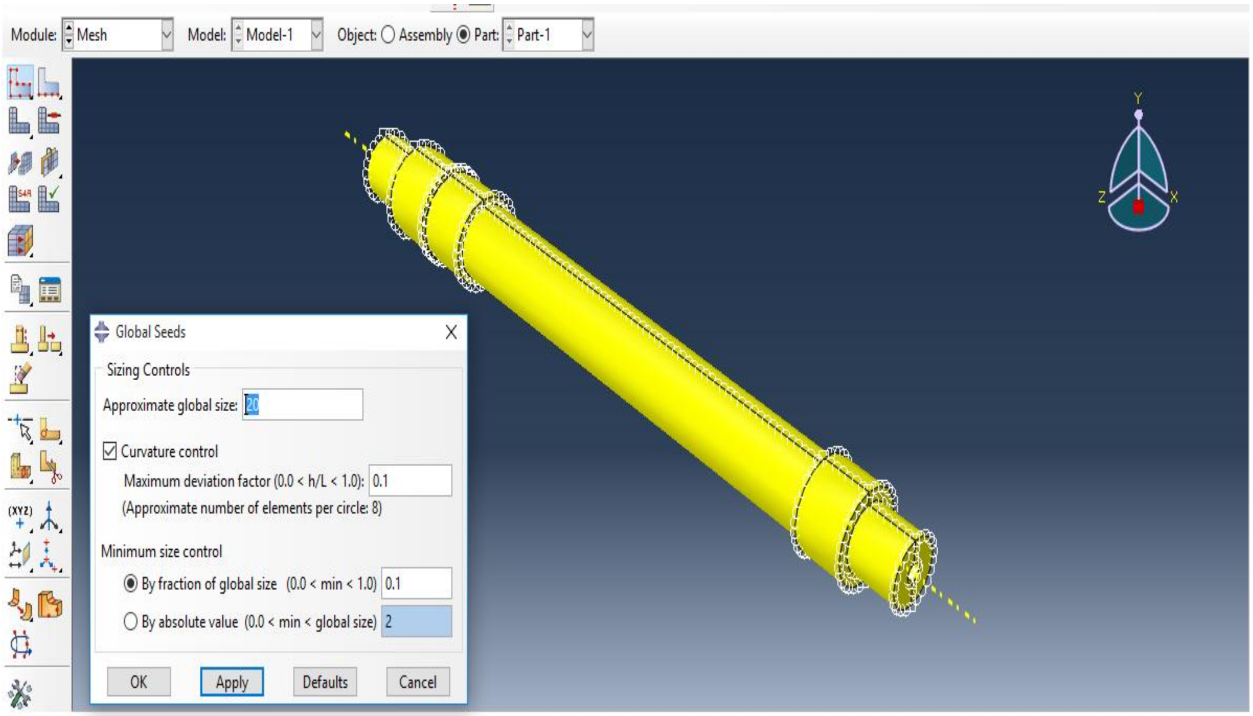


2. Partitioning



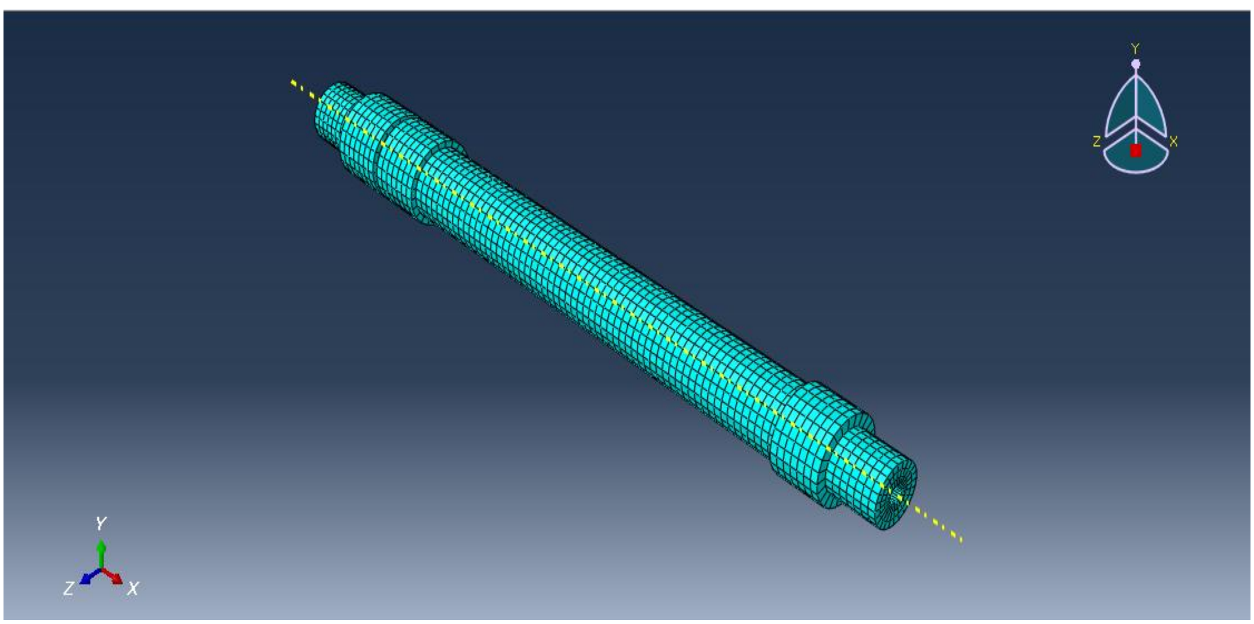
Partitioning on the behalf of the axle helps to apply the load and set boundary conditions on the half part of the axle. It used the three point partitioning way.

3. Assigning Global size

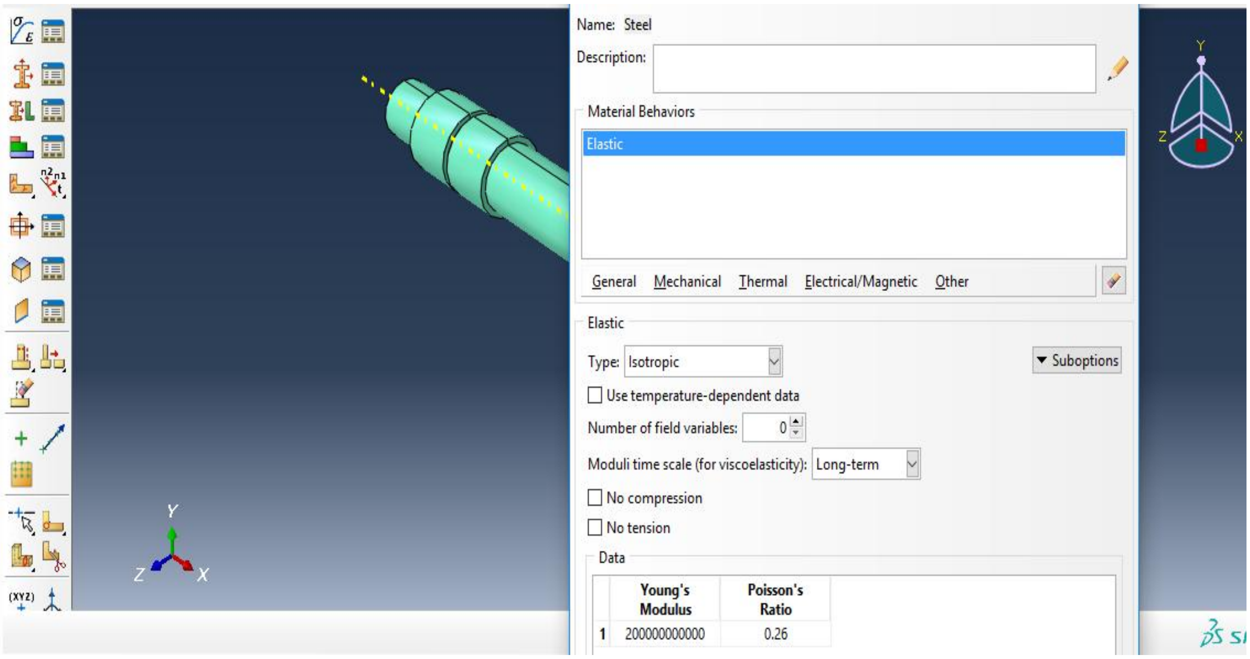


Assigning global size helps to specify the number of meshing elements on the axle. The approximate global size number is 20.

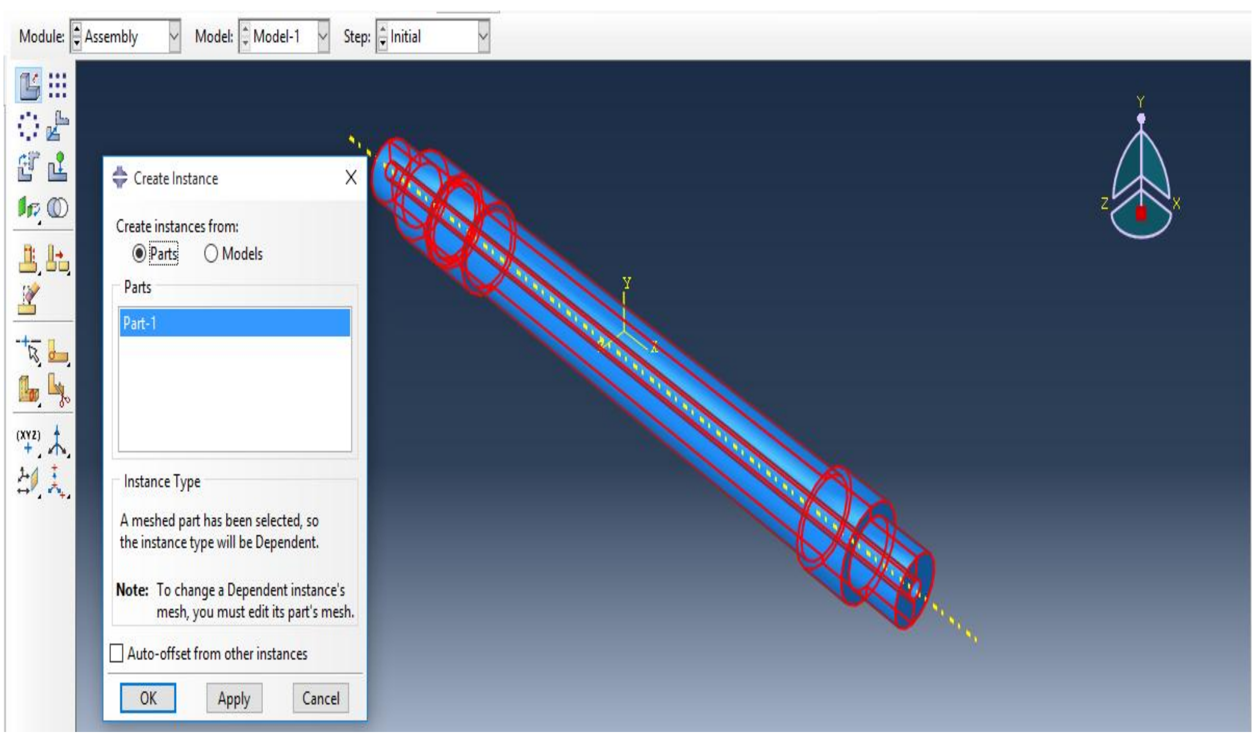
4. Meshing



5. Input Material Property

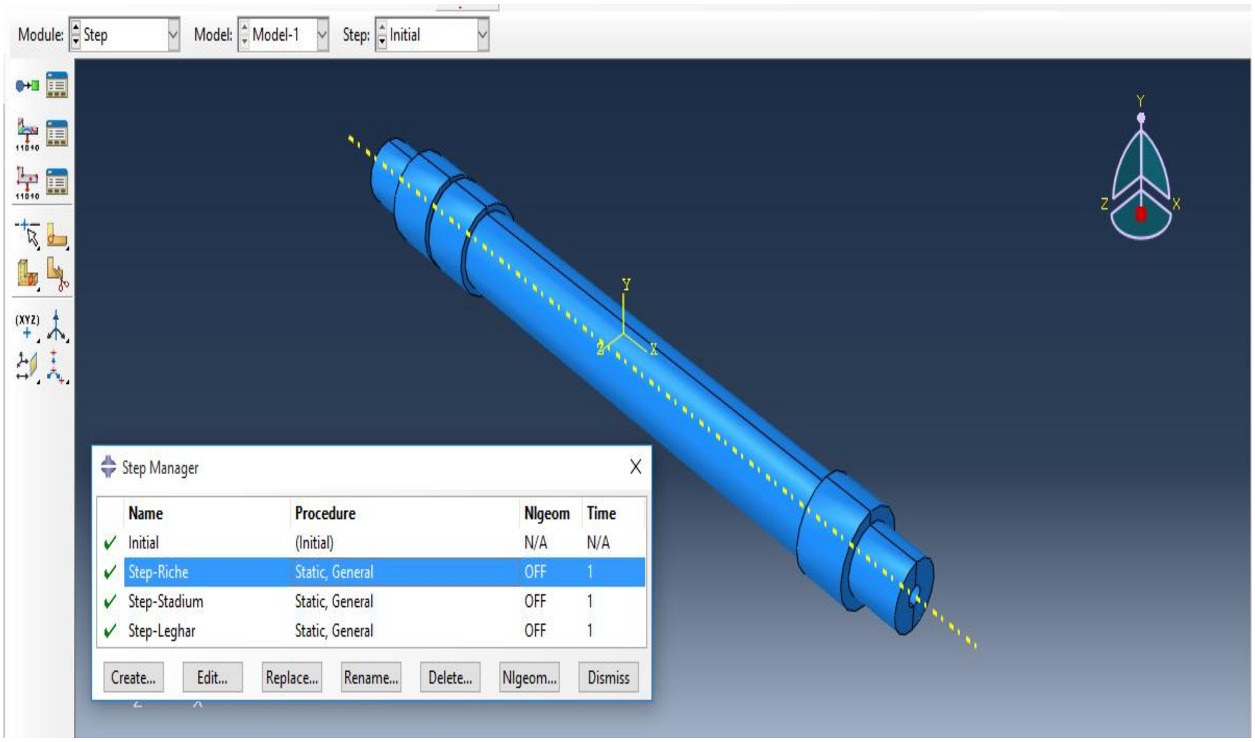


6. Assemble the partitioned element

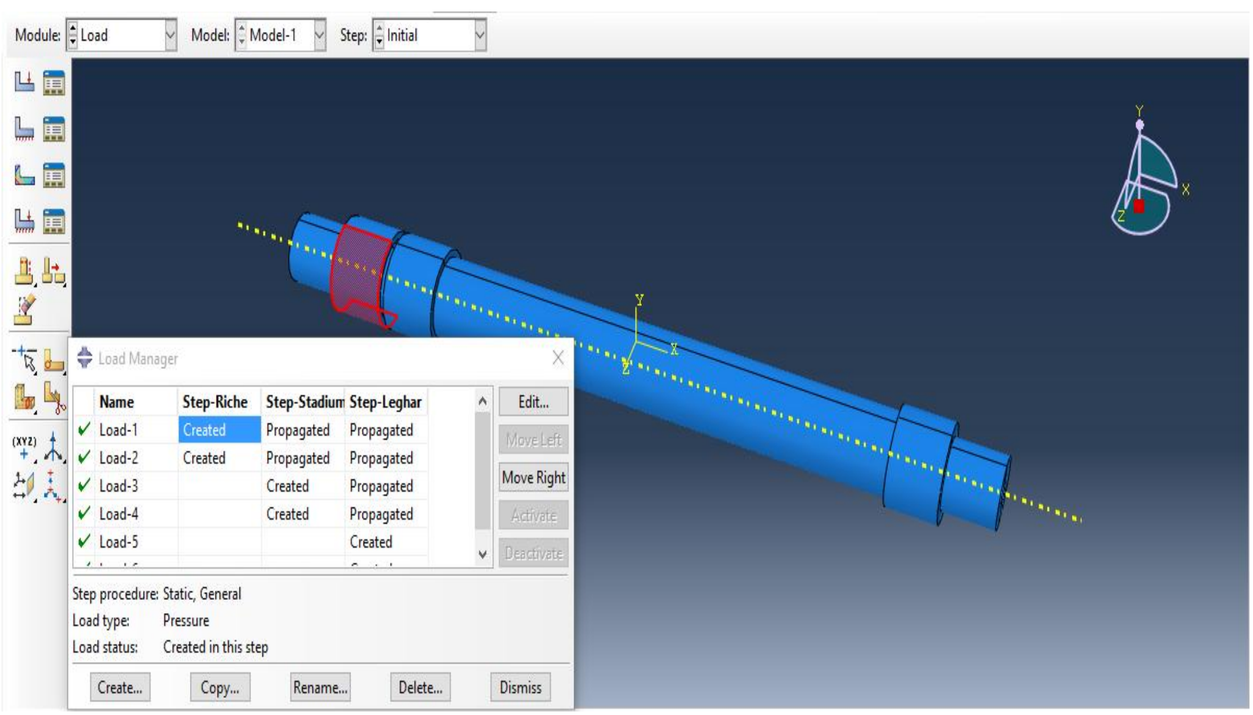


The partitioned axle should assemble to make it as one object on the analysis.

7. Create steps

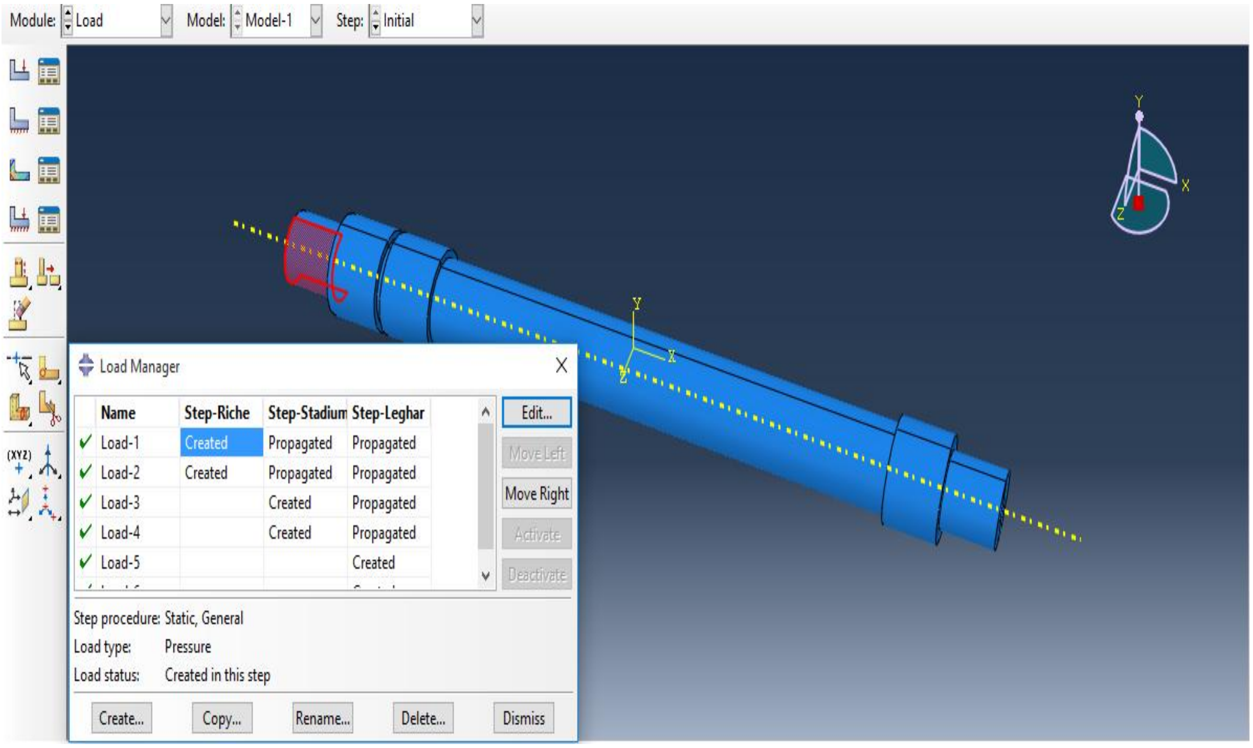


8. Create Boundary conditions



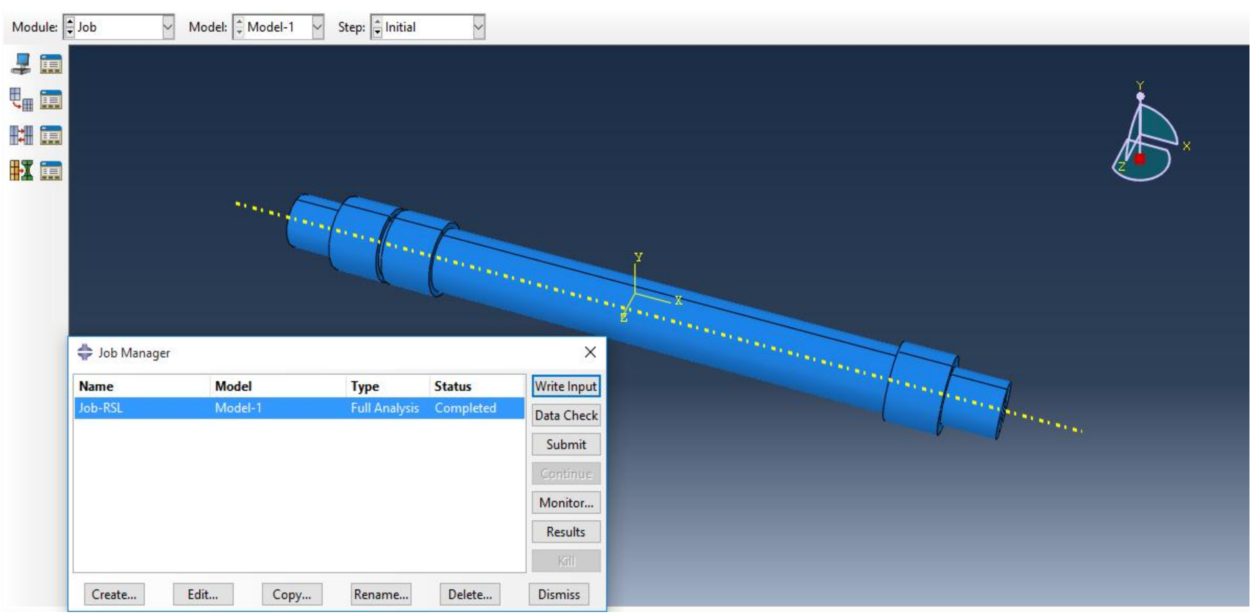
Steps created to apply the loads and boundary conditions

9. Create Loads



Apply the loads on the tip of the axle to analyze the stress.

10. Create Job



Creating job is the final step to submit the inputs and gain the outputs or results.