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By

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Optimization of Bolted Rail Joint using FEM of NRNE

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Optimization of Bolted Rail Joint using FEM of NRNE

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Optimization of Bolted Rail Joint using FEM of NRNE

ABSTRACT

Proper functioning of bolted joint is essential for the safe operation of the railway continuity of the rail and for breaking system. The conventional bolted joint resembles structure butt joint consisting of two pieces of rails connected together through two joints bars on either side of their web .The main objective of this study is analyzing and optimize the stresses on bolted joint due to the vertical, lateral and longitudinal force to predict and minimize stress caused. This studies use the hertz contact theory to perform the analysis for three geometric of bolt. The three dimensional model has been developed on modeling package of CATIAV5R16. In CATIAV5R16, different components of wheel/rail assembly i.e. wheel, rail, joint, bars, nuts, bolts are created separately then all components are assembled, and create a complete model of wheel/rail assembly. Assembly model has created in assemble workbench of CATIA after individual component of joint had created on part work bench. After the assembly is accomplished on CATIA, it was imported in to the ANSYSR16 to analyze the stress caused by vertical, lateral load and longitudinal load. ANSYS solves governing differential equations by breaking the problem into small elements. The wheel structure discretized using CONTA174 element. CONTA 170 three dimensional elements are used to discretize rail. During the analysis of wheel/rail contact in ANSYS software, the parameters have been used axle load, angular velocity and gravitational acceleration. From ANSYS software simulation static and dynamic result when M32 geometric, the rail joint highly exposed to the equivalent, normal, shear stresses than the rest. For this reason, the service life of the rail joint of M32 is small as compared to the remaining rail joint location. When M34 bolt geometric; the stress on the rail joint is small in comparison to M30 and M32 bolt geometric. So M34 bolt geometric will be better to use by national railway than M32 bolt geometric.

Key Words: Stress, bolted Joint, Modeling, Finite Element Method.

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NOMENCLATURE

NRNE: National Railway Network of Ethiopia

RJ: Rail joint

CWR: continuous welded rail

IRJ: Insulated Rail Joint

FEM: Finite Element Method

a: Minor semi axes of the contact ellipse

b: Major semi axes of the contact ellipse

F: Vertical load

m& n :Hertz coefficients

Kw: Constants that depend on the material properties of railway wheel

E_r : Poisson‟s ratio wheel material

E_w : Young‟s modulus of the railway wheel material

r : Young‟s modulus of rail material

R_{1wr} : Principal rolling radii of the wheel

R_{1r} : Principal rolling radii of rail

R_{2w}: Principal transverese radii of curvature of wheel

R_{2r}: Principal transverese radii of curvature of radii

K : Stiffness matrix of the system

U: Nodal displacement vector

Γ_1: Boundary with zero displacement,

Γ_{2}: Boundary where measured displacements are given

Γ_3: Boundary with unknown contact forces F_c

Γ4: Boundary where there are known applied force

U_1: Displacements on constrained boundary Γ1

U_2: Known and measured displacements on free boundary Γ2

U_3: Unknown displacements on contact boundary Γ_3

U_4: Unknown displacements on boundary Γ_4

σ : Stress vector,

D: Elastic stiffness matrix.$\in^{el}$:
Strain that cause stress E_x: Young"s
modulus in the xdirection σ_x:
Stress in x direction σ_y: Stress in
y direction σ_z: Stress in z direction

σ_{xc}^{f} : Normal stress failure in x direction

σ_{yc}^{f}: Normal stress failure in y direction

σ_{zyc}^{f}: Normal stress failure in z direction

σ_{xy}: Shear failure in xy direction

σ_{xy}: Shear failure in yz direction

σ_{xy}: Shear failure in xz direction

CHAPTER 1: Introduction

1.1-Background

One of the essential components of the railway is the rail joints which are primarily bolted joints that connect pieces of rails together and also the rail joint is typically considered to be one of the weakest locations in the track superstructure. Rail joints are used to connect two adjacent rails. Typically, a pair of joint bars is placed on each side of the two adjacent rail ends, fixed with at least four or six bolts. Two main categories are used to classify rail joints: bolted joints and insulated joints (IJs). Bolted joints can be further categorized into compromise joints and standard joints. Compromise joints are used when two rails with dissimilar sections need to be connected, while standard joints are installed with two similar rail sections. Standard joints are mostly seen in bolted-joint rail (BJR) track, however, they can be observed in continuously welded rail (CWR) track as well, which are used as temporary joints to connect long CWR strings before welding. Rails are manufactured in relatively short lengths of 60 metres to 120 metres of different sizes, and they are required to be attached to one another, end to end, during installation. Rail joints are points at which these short length segments of rails are structurally connected. They are joined by either welding or mechanical connections. The mechanical connectors consist primarily of fishplates and mechanical fasteners (bolts and nuts) that mount the fishplates to the web sides of the rail. Hence, a mechanical rail joint is a typical example of bolted joints. Rail joints can be categorized as conductive rail joints and insulated block joints (IBJs) depending on their operation. While the conductive joints allow electricity to pass through it to the adjoining rails, the insulated block joint electrically isolates the connecting rails. Figure shows examples of the rail joints.

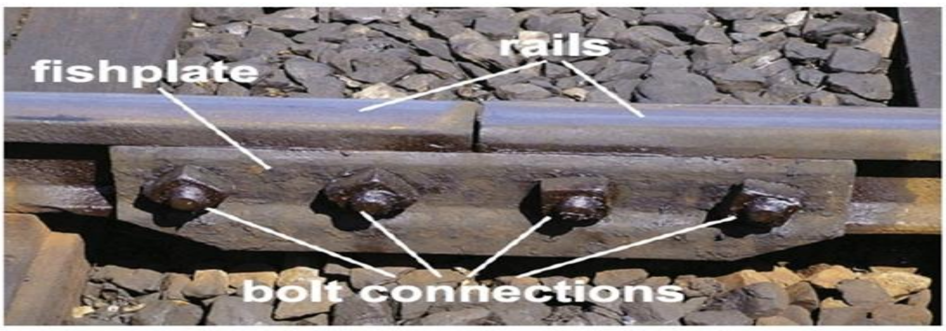


Figure 1.1: composant of bolted rail joint [2].

Wear is one of the main damage mechanisms for the rail track, especially in the joint of two rails head. The joint parts of the rail have a lower vertical bending stiffness than the normal rail track. As a result, large deflections in the joint region are generated due to the presence axle load. The large deflections can accelerate track deterioration. This in turn yields larger wheel load caused by the dynamics of the passing wheels and damaged tracks. The lower vertical bending stiffness, coupled with the rising tonnage rates and axle loads experienced in the last decade years from numerous industries means sections of track that feature joint bars suffer a very short service life. To reduce such maintenance costs and other indirect cost it is better to analyze and understand the wheel/rail contact stress distribution, deflection and wear type on the joint part of the rail head.

1.1.1-Railway History of Ethio-Djibouti

Le Chemin de fer Djibouti–Ethiopien (Ethio-Djibouti Railway Transport), commonly known as the CDE, is a single-track railway line connecting Addis Ababa with the port of Djibouti by 1917 by which date the service operated along a line of 784 kilometers.

As the Ethio-Djibouti Railway deteriorated through lack of maintenance, Ethiopia lost railroad access to the sea. The existing metre-gauge railway had been originally built by the French between 1894 and 1917; it had all the deficiencies of a colonial-era railway, with steep gradients and tight curves. Since China was financing the construction of a standard gauge railway network in East Africa, Ethiopia and Djibouti chose to abandon the metre-gauge railway and build a new standard gauge link.

In 2011, the Ethiopian Railway Corporation awarded contracts to two Chinese state-owned companies for the construction of a new standard gauge railway from Addis Ababa to the Djibouti border.

In 2012, Djibouti selected the China Civil Engineering Construction Corporation to complete the final 100 km to the port of Djibouti

The completed Ethiopian section was formally inaugurated on 5 October 2016 in Addis Ababe. The two primary contractors have formed a consortium to operate the railway for the first 3–5 years, while local personnel are trained. On 10 January 2017, the 100 km section of Djibouti side was inaugurated in a ceremony held in the new Nagad railway station by Djibouti

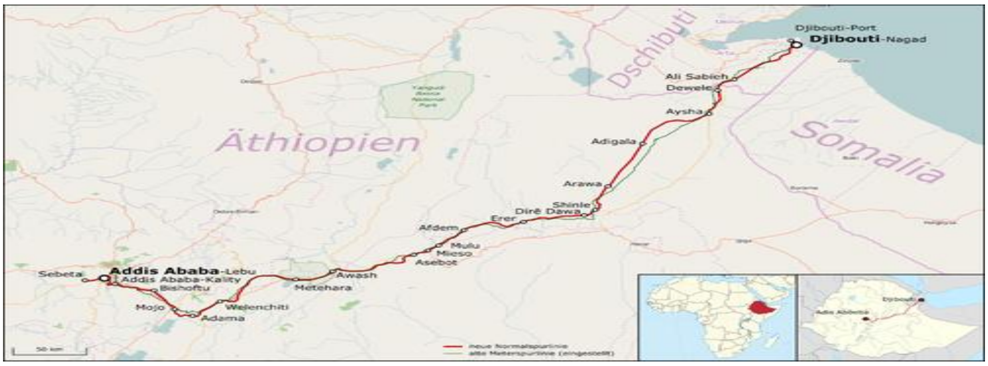


Figure 1.2: Ethio-Djibouti Railway[33].

Railway Track

The railway constituting conventional rail system may be divided into subsystems of either structural areas or operational areas. The structural parts consist of infrastructure, energy, control, command and signaling, management of traffic operation, and rolling stock. Whereas the operational concerns of railway subsystem consist of maintenance and telematics applications for passenger and freight services.

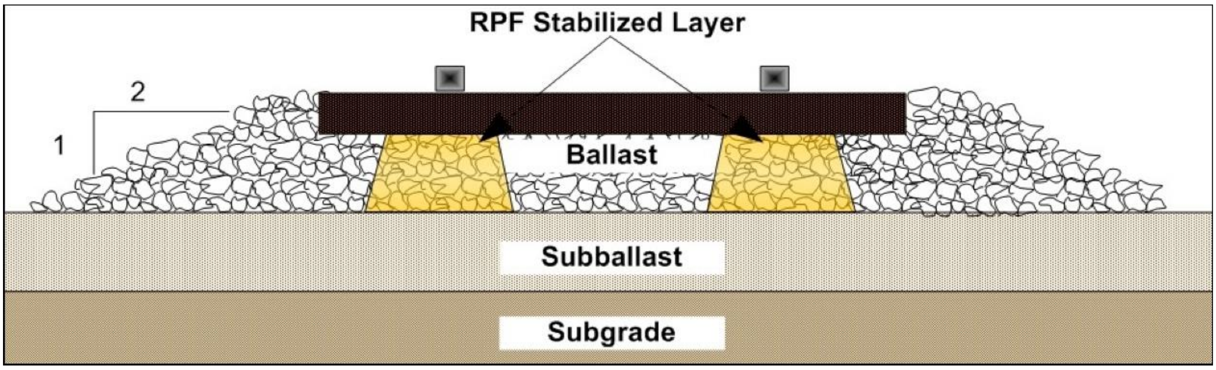


Figure 1.3: Railway track structure[33].

In addition showed that infrastructure subsystem of the railway consist of tracks, points, engineering structures (bridges, tunnels, etc.), associated station infrastructure (platforms, zones of access, etc.), safety and protective equipment.

Track geometry, consisting of several parameters, is a significant factor influencing the ride quality and derailment risks. It describes the position that each rail, or the track centerline, occupies in space. By projecting the track geometry into various planes, track geometry can be specified. Track geometry parameters can be grouped according to the plane they reside in. The main parameters defining the track geometry are gauge (track plane), profile (longitudinal vertical plane), alignment (horizontal plane), cross level or super elevation (transverse vertical plane).

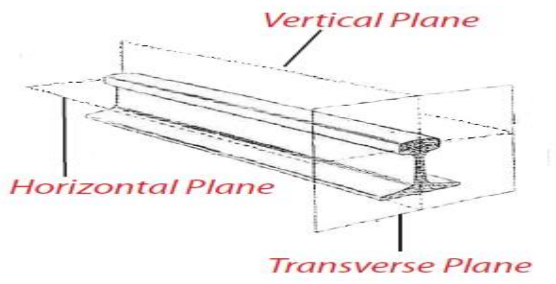


Figure 1.4: Track geometry[33].

Rail track is a fundamental part of railway infrastructure and its components, for ballasted track, can be classified into two main categories: superstructure and substructure. The most obvious parts of the track as the rails, rail pads, concrete sleepers, and fastening systems are referred to as the superstructure while the substructure is associated with a geotechnical system consisting of ballast, sub-ballast and subgrade (formation).

1.1.2-Joining Rails

Rails are produced in fixed lengths and need to be joined end-to-end to make a continuous surface on which trains may run. The traditional method of joining the rails is to bolt them together using metal fishplates or joint bars, producing *jointed track*. For more modern usage, particularly where higher speeds are required, the lengths of rail may be welded together to form continuous welded rail (CWR) and bolted joint. There are essentially two different type of a rail joint

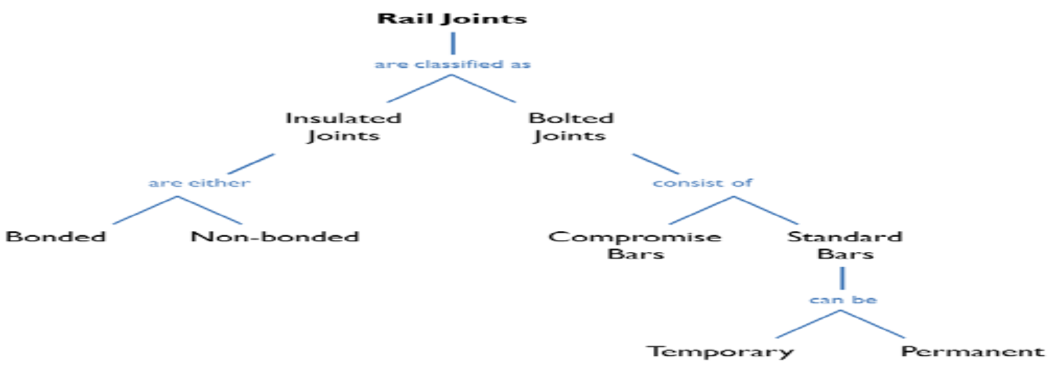


Figure 1.5: Different type of rail joint[2].

The rail joints create discontinuities in the running surface of the rails. These discontinuities produce different type of static and dynamic stresses on the running surface .These stresses on the rail head produce weakness on the joint and rail end. They cause different type of failures at the rail joint compare to the normal rail. Based on other country experience, the following failures mode is at joint: - delamination of end part of rail and joint bar, broken joint bar and looseness of the bolt, fatigue wear of rail end due to repeated loading of a surface., fretting damage on the joint parts and on rail due to the relative oscillatory movement of small amplitude that may occur between two contact surfaces subjected cyclic load.

❖ Bolted joint

The purposes of the rail joint (made up of two joint bars or more commonly called angle bars) are to hold the two ends of the rail in place and act as a bridge or girder between the rail ends. The joint bars prevent lateral or vertical movement of the rail ends and permit the longitudinal movement of the rails for expanding or contracting.

The joint is considered to be the weakest part of the track structure and should be eliminated wherever possible. Joint bars are matched to the appropriate rail section.

Each rail section has a designated drilling pattern (spacing of holes from the end of the rail as well as dimension above the base) that must be matched by the joint bars.

Although many sections utilize the same hole spacing and are even close with regard to web height, it is essential that the right bars are used so that fishing angles and radii are matched. Failure to do so will result in an inadequately supported joint and will promote rail defects such as head and web separations and bolt hole breaks. There are three basic types of rail joints

- ✓ Standard
- ✓ Compromise
- ✓ Insulated

Figure 1.6: Three basic types of rail joints[29].

1.1.2.1-Standard Joints

Standard joint bars connect two rails of the same weight and section. They are typically 24" in length with4-bolt holes for the smaller rail sections or36" in length with 6-bolt holes for the larger rail sections. Alternate holes are elliptical in punching to accommodate the oval necked track bolt. Temporary joint sin CWR require the use of the 36. Bars in order to permit drilling of only the two outside holes and to comply with the FRA

Track Safety Standards requirement of maintaining a minimum of two bolts in each end of any joint in CWR.

Figure 1.7: Standard Joints[30].

1.1.2.2-Compromise Joints

Compromise bars connect two rails of different weights or sections together. They are constructed such that the bars align the running surface and gage sides of different rails sections. There are two kinds of compromise joints:

- ✓ Directional (Right or Left hand) compromise bars are used where a difference in the width of the head between two sections requires the offsetting of the rail to align the gage side of the rail.
- ✓ Non-directional (Gage or Field Side) are used where the difference between sections is only in the heights of the head or where the difference in width of rail head is not more than 1/8" at the gage point. Gauge point is the spot on the gauge side of the rail exactly 5/8" below the top of the rail.

To determine a left or right hand compromise joint:

- ✓ Stand between the rails at the taller rail section.
- ✓ Face the lower rail section.
- ✓ The joint on your right is a "right hand"
- ✓ The joint on your left is a "left hand".

Figure 1.7: Compromise Joints type

1.1.2.3-Insulated Joints

Insulated joints are used in tracks having track circuits. They prevent the electrical current from flowing between the ends of two adjoining rails, thereby creating a track circuit section. Insulated joints use an insulated end post between rail ends to prevent the rail ends from shorting out.

There are three types of insulated joints:

- ✓ Continuous
- ✓ Non-continuous
- ✓ Bonded

Continuous insulated joints are called continuous because they continuously support the rail base. No metal contact exists between the joint bars and the rails. Insulated fiber bushings and washer plates are used to isolate the bolts from the bars. The joint bars are shaped to fit over the base of the rail. This type of insulated joint requires a special tie plate called an "abrasion plates" to properly support the joint. Non-continuous insulated rail joints are called non-continuous because these joints don't continuously support the rail base. A special insulating tie plate is required on the center tie of a supported, non-continuous insulated joint.

Metal washer plates are placed on the outside of the joint bar to prevent the bolts from damaging the bar.

There are two common kinds of non-continuous insulated joints:

- ✓ Glass fiber.
- ✓ Polyurethane encapsulated bar.

The glass fiber insulated rail joint (See the bar to the right in Figure 3-8) replaces the joint bar with a reinforced glass filament bar. Metal washer plates are placed on the outside of the joint bar to prevent the bolts from damaging the bar .The polyurethane encapsulated insulated bar is a steel joint bar completely encapsulated in polyurethane over the entire joint bar surface. The Poly joint uses insulating bushings to insure that track bolts do not short out the track.

Bonded insulated rail joints (commonly called plugs or slugs) are made up of two pieces of rail, which utilize an epoxy resin to glue the insulated bars t=o the rail sections. They are bolted together using bushings to isolate the bar from the rail steel itself. The bolts maintain the alignment of the bars and rail until the epoxy cures. The bars are typically of a heavier section (Section)to provide extra support for the epoxy. These units can be purchased in a variety of made up lengths. The completed assembly is then Thermite welded into the track structure. His is the preferred type of insulated joint to use in continuous welded rail (CWR).

Figure 1.8: Insulated Joints type[29].

1.1.3-Type of Rail joint

1.1.3.1-Classification According to Position of Rail joint

Two types come under this category

1.1.3.1.1. Square joints

The square joints are mostly used on straight portions and this is one of the most common types of the joint which is provided on the railway tracks or the permanent way. The joints in one rail are exactly opposite to the joints in the other rail it means the joint on the other rail section being laid parallel at the gauge distance is going to be just opposite side to this one and that is how at every regular distance we are going to have a joint on both the cases, this rail as well as this rail and this is why they are termed as because this is making a sort of a square condition.

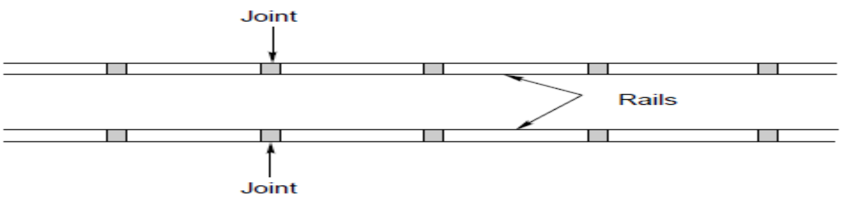


Figure 1.9: Square joints[29

1.1.3.1.2. Staggered joint

The staggered joint is mostly provided on curved tracks because they hinder the centrifugal force that pushes the track outward. In the case of curved tracks is a larger distance to be travelled on the outer curve as compared to the inner curve. The rail length being provided, the rail length is a constant because of the difference in the distance being travelled on the two tracks the joint will be coming at different location.

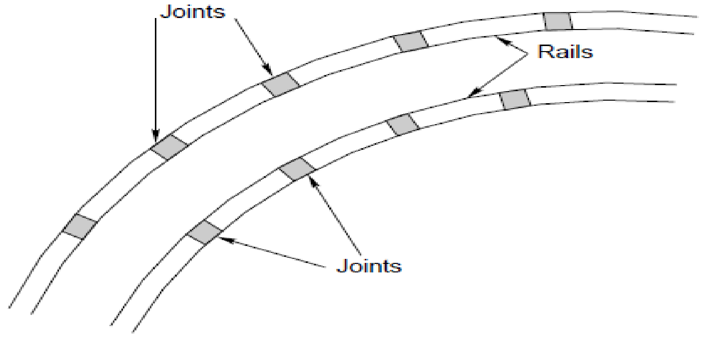


Figure 1.10: Staggered joint[29].

1.1.3.2-Classification According to Position of Sleepers

Three types of rail joints come under this category.

1.1.3.2.1. Supported joint

In this type of joint, the ends of the rails are supported direct lyon the sleeper. It was expected that supporting the joint would reduce the wear and tear of the rails, as there would be no cantilever action. In practice, however, the support tends to slightly raise the height of the rail ends. As such, the run on a supported joint is normally hard. There is also wear and tear of the sleeper supporting the joint and its maintenance presents quite a problem. The duplex sleeper is an example of a supported joint (Fig. 15).

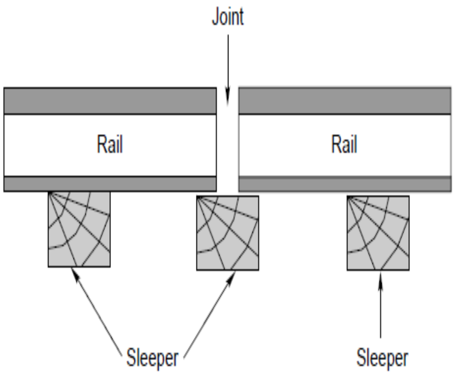


Figure 1.11: Supported rail joint[2].

1.1.3.2.2. Suspended joint

In this type of joint, the ends of the rails are suspended between two sleepers and some portion of the rail is cantilevered at the joint. As a result of cantilever action, the packing under the sleepers of the joint becomes loose particularly due to the hammering action of the moving train loads. Suspended joints are the most common type of joints adopted on railway systems worldwide (Fig. 16).

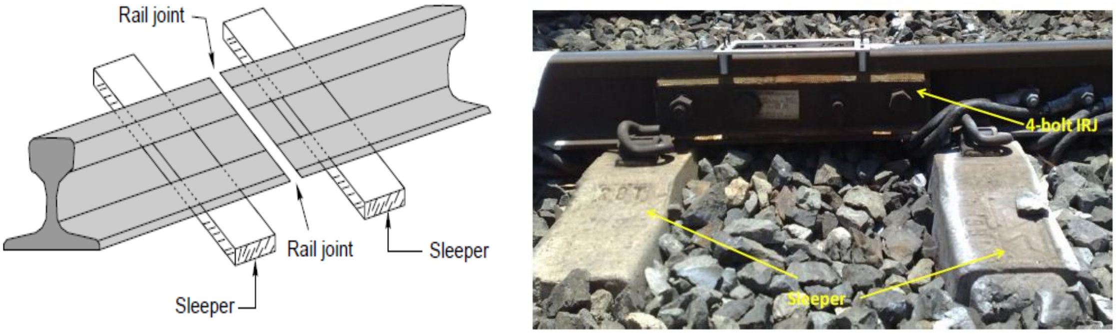
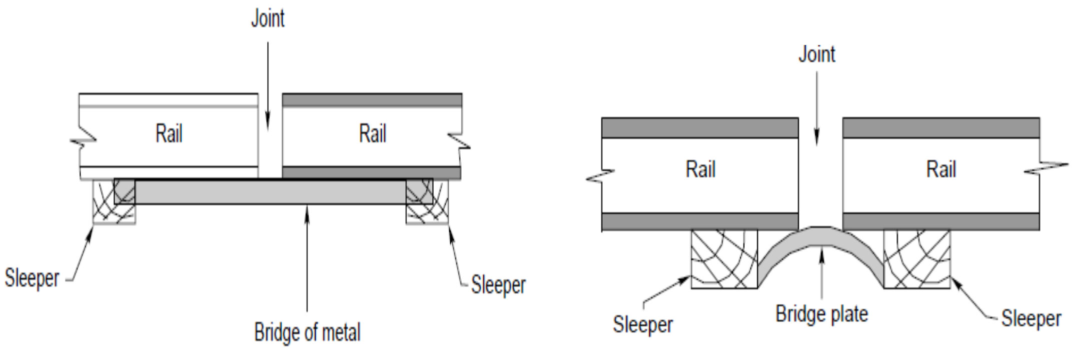


Figure 1.12: Suspended joint[29].

1.1.3.2.3. Bridge joints

The bridge joint is similar to the suspended joint except that the two sleepers on either side of a bridge joint are connected by means of a metal flat [Fig. 3(a)] or a corrugated plate known as a bridge plate [Fig. 3(b)].

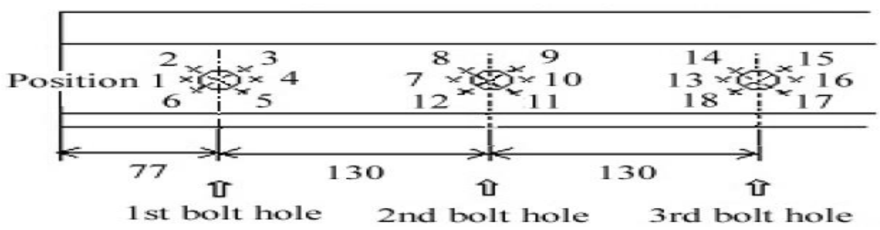


3. Figure 1.13 (a): Bridge joint with metal flat 3. Figure 1.13 :(b) Bridge joint with bridge plate [29].

1. 2-Statement of problem

Due to improving the vehicle dynamics performance to restrict the vibration at a desirable level, to increase ride comfort and to minimize the maintenance cost. The rails accommodate the wheel loads and distribute these loads over the sleepers or supports. Lateral forces from the wheel sets and longitudinal forces due to traction and braking of the train are also transmitted to the sleepers and further down into the track bed. The rail joints create discontinuities in the running surface of the rail. These discontinuities produce different type of stresses on the running surface due to vertical, lateral and longitudinal load. These stresses on the rail head produce the deflection on the joint that causes to vary on the rail and rail joint profile. A rail joint is the weakest link in the track. There is a break in the continuity of the rail in horizontal as well as in vertical plane at this location because of the expansion gap and Imperfection in the rail heads at joint. As the result, the profile of rail joint component change from its normal profile it originates the discomfort and safety problem to the passenger and accident on the vehicle by causing wear on joint part. And also cracking at rail-end bolt holes is a major safety concern being the attributable cause of a number of derailments. It is also a cause of premature rail replacement, imposition of rail speed restrictions and a significant factor in rail inspection and maintenance costs. Due to above reason this paper try to address the following questions:

1- At which position, the size of bolt have minimum stress cause by vertical, lateral and longitudinal force



2-What types of stresses are inducing on the bolted joint?

1.3-Objectives of study

1.3.1 -General objectives of study

is optimization of bolted rail joint using FEM of NRNE.

1.3.2 -Specific objectives

- ✓ Modeling different size of bolt
- ✓ Analysis the stress on bolted joint due to vertical, lateral and longitudinal
- ✓ Analysis of the maximum deflection on the bolted joint
- ✓ Analyze the vertical, lateral and longitudinal forces on the rail joints
- ✓ Compare the stress of the different size

1.4. Methodology

A three dimensional Finite element model is developed for both of bolted rail joint at Catia. The important feature is the 3D modeling of the contact surface between the connected plates and the bolt and the plates.

The model is import in Ansy validated by determine the results with a closed form solution for a both problem of determining the stress around the bolted rail joint And identified the stress at different geometry of bolt.

1.5-Scope and limitation of the study

This research scope analyze the stress due to vertical, longitudinal and lateral load at different geometry of both, its effect on the bolted rail joint at when the joint is on the sleeper location of the standard bolted rail joint by using 3D finite element method. It helps to demonstrate how stresses contribute to the wear and deterioration of the rail track with different dimension of bolt. And optimization of the design of structural both leads to increase in their load carrying capacity, minimizing weight and cost.

Static and dynamic analysis of a bolted rail joint structure is carried out. The dynamic analysis includes transient analysis of the bolted rail joint structures. This study is neither to reduce stress nor to succeed the bolted rail joint, rather to show stress distribution, and their contributions to wear standard rail joint.

1.6. Organization of the Paper

The body of this study is divided into five main chapters. The first chapter discusses background, objectives and methodology of the study. In addition, the details of the rail joint type and rail joint used for analyze. The second chapter covers the review of some of the journal articles, conference papers and publications which were referred to during the study, wheel /rail mechanic and rail joint stress. Model and analysis of stress on bolted rail joint is discussed in the third chapter. Modeling contact at rail joint, stress model using hertzian theory, rail support and wheel rail simulation presented. The results obtained from the analysis of the rail bolted joint and discussions based on these results are included in the fourth chapter. Finally, the fifth chapter cover conclusions drawn based on the results of the analysis, recommendations and future work.

Chapter Two: Literature review

2.1-Introduction

Review of literatures reveals that different previous work which helps for the guidance of this work. These previous related works may be journals, conference papers, design works and books related to this paper. Selection of appropriate conditions, approaches and methodologies with these journals and other materials will strengthen for the successful accomplishment of the paper.

2.2-Wheel/rail Contact Mechanics

In studies of railway, wheel-rail rolling contact is one of the main issues. The contact forces developed in the contact are the most important external inputs to the vehicles and the track, and are also the direct cause of the damage of wheels and rails like wear, corrugation, fatigue and fracture. In recent years, with the continuous increase of the running speed and the axle load, the influence of the contact forces on the wheel-rail damage and the track deterioration has received more and more attention. The stress field created by the contact stresses was first introduced by Heinrick Hertz in 2000[1]. Assessment of contact stresses at the wheel–rail interface is one of the most important aspects of railway research, considering the many phenomena involved. For this reason, many scientists have approached the problem mainly by means of theoretical or numerical solutions based on the Hertz's theory, which can be considered the basic starting point for all subsequent research. However, there are limiting conditions for the applications of the Hertz contact theory:

I. The contact between elastic bodies should be frictionless.

ii. The significant dimensions of the contact area should be much smaller than the dimensions and the radii of curvature of the bodies in contact.

iii. The contact between elastic bodies should be described by second-order polynomials.

Due the above reason hertz's theory applied only for the straight surface and smooth it is not applied on the roughness and joint part of the rail head. Other method need to analysis the rough surface and join part of the rail head. Researcher also used different mechanism to deal the interaction between the wheel and rail.

[2]: Studied are the joints are those points which are the points of weaknesses and which causes number of wears or tears or the damages. We will start from these particular points. And also the rail section on the track gets eroded, or wearied out, or damaged or they fail out.

And also at studies will devised as rail joints, the welding of rails, advantages of welded rails, short welded rails, and long welded rails. Now coming to these rails and the joints of these one we have of course discussed about number of factors which are the problem, which are caused at any of the rail joint.

[30] Went a step further in analyzing the effects of the discontinuity under rolling conditions by modeling a whole joint with an explicit integration scheme to account for the high frequency dynamics.

Rail, wheel, joint bars and isolating material were represented in 3D while the rail, isolating material and joint bars were modeled as one component that means no contact was defined between them. The study reaffirmed the significant influence of the discontinuity on the wheel/rail contact. Pang and Dhanasekar showed that not only the pressure distribution, but also the contact patch size and the maximum contact pressure differed from Hertzian solution.

[3] Studied the stress around fish bolt hole by using static loading tests and field tests then compare to stresses were calculated by using FEM. Tensile stress field in the vertical direction occurred when joint bolts were fastened and the maximum stresses were generated at lateral positions of holes. Maximum stress amplitudes were observed at 45-degree position to longitudinal axis of rail under a vertical load. Based on results, they established a method to evaluate the stresses at the edges of fish bolt holes when fish bolts were fastened and trains passed.

[4]: Described the results of computer modeling of the effects of epoxy debonding on the stresses and strains in a bonded insulated joint subjected to longitudinal force. The identified measurable changes in the joint's strain distribution that correlate with the extent of debonding, to serve as the basis of a non-destructive monitoring and evaluation technique model show that, under thermal tensile loads, strains at the center of the outer surface of the joint bar tend to increase as debonding begins near the end post. The strain at this point tends to stabilize after the debonding reaches the innermost bolt hole. Strain at a point between the outermost and middle bolt holes starts off relatively stable, but increases after debonding passes the innermost bolt hole. Strains in the event of a pull-apart depend on the friction parameters chosen for the ruptured epoxy. In all cases, results suggest that any increase in debonding causes an increase in the elastic relative displacement of the rail ends under load.

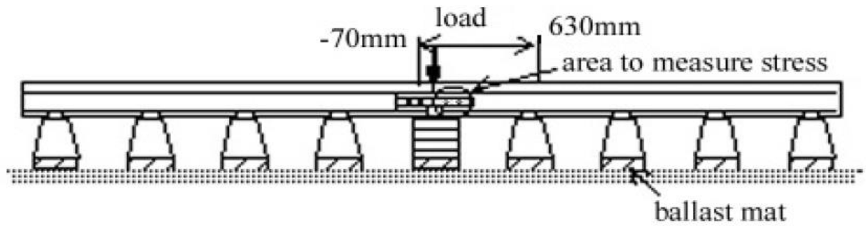
[5]:studied the fatigue life of rail joint bar by using three dimensional finite element model of rail joint bar and dynamic load is applied to estimate the fatigue life of the joint bar. Different component of the rail joint bars are being created separately and assemble in Autodesk Inventor. The model consists of assembly of the rail, joint bars, bolts, nuts, washers, and wheel. A three -dimensional finite element analysis of rail joint bars is carried out in ANSYS after importing from Autodesk Inventor. The static and dynamic loads are being applied to estimate fatigue life and endurance strength at the section. The material properties of the rail and wheel are assumed same. The material properties of the wheel and rail are considered bilinear kinematic hardening in ANSYS.

[6].A finite element analysis is conducted to study dynamic elastic–plastic stress. The ANSYS implicit code and LS-DYNA explicit code are coupled to simulate the process of the wheel contacting or impacting the rail joint. Contact elements are used to simulate the interactions between wheel and rails, between rails and joint bars, between joint bars and bolts and between bolts and rails. The effects of train speed, axle load and the height difference of rail joint on the contact forces, stresses and strains at rail head are investigated.

Bolted joints are used to join two rails in jointed rail territory. In continuous welded rail (CWR) territory, bolted joints are used to temporarily join rails before they are welded. Compromise bars are used to join two rails with differing sections. Insulated joints are further categorized as bonded or non-bonded joints. Bonded insulated joints are glued. Non bonded insulated joints are basically bolted joints having some type of electrical insulating properties. While these three types of rail joints perform differently and have different operational objectives, they share many similar design features. For example, they all use bolts and bars and create a discontinuity in the running surface of the rail. Bolts generally fail due to yielding, and bars usually fail due to fatigue. The discontinuity in the running surface of the rail creates conditions that can accelerate track degradation around the joint. At a minimum, the gap at the rail ends within the rail joint is a source of impact loading from passing wheels.

2.3-Rail Joint Stresses

Rail joints are essential component in track circuitry that joining the two rails end to end and make a continuous surface on which trains may run. Previously rails were bolted together by using two joint bars, one on each side of the web with four or six bolts through the rail track as a geometric requirement.



These joint bars have a lower vertical bending stiffness than the straight rail track. As a result; large deflections in the joint region are generated while wheels pass through. The large deflections can accelerate track deterioration. Due to this parts need more research, many researcher are interesting to do their research around this area to solve the problem related to discontinuous of the rail joint to improve the performance, the service life ,maintenance cost and to increase the comfort of the passenger. This part deals with previous work is related to the rail joint.

[8] In addition to bending, thermal, and residual stresses, rail joints are also subjected to dynamic loads due to these discontinuities. A lower difference in elastic modulus can reduce the interface stress magnitude in the rail ends; however, it could adversely result in higher stress in the insulation material, which may lead to its earlier failure. Electrical failure often results when the loose joint experiences contact between metal surfaces on the rails and joint bars or bolts – a result of fretting, deterioration or wear in the insulator, relative component movement, and related processes. According to FRA"s accident data, a total of 242 accidents related to joint failures occurred from 2000 to 2009. Most joint bar failures occurred due to cracks initiated from bolt holes or at the bottom or top edges of the joint bars. The number of accidents caused by joint bars was relatively consistent until 2007. The sharp decrease observed in 2008 and 2009 appears to be due to lower traffic and the overall downward industry trend in accidents.

[9]: This studied an engineering analysis of different designs and failure modes of the IRJ and a 3D finite element model for analyzing the stresses experienced by three different joint bar sizes (30mm, 34mm, and 40mm) of width. From the results it was a small reduction in the stresses encountered by the rail when joined with a pair of joint bars of increased moment of

inertia considering the thickness range considered. It suggests that increasing bending stiffness by increasing the thickness of the joint bar is not a good way to reduce stresses and displacements of the rail joint. An important way of increasing bending stiffness of the bar is to increase its height. The increase of stiffness of the bar by increasing height is more dominant compared to that of thickness.

[10]: used 3D finite element analysis to study the effect of the discontinuity of the rail ends and the presence of lower modulus insulation material at the gap to the variations of stresses in the bolted rail joint (IRJ) is presented. It is shown that the maximum stress occurs in the subsurface of the railhead when the wheel contact occurs far away from the rail end and migrates to the railhead surface as the wheel approaches the rail end; under this condition, the interface between the rail ends and the insulation material has suffered significantly increased levels of stress concentration. The ratio of the elastic modulus of the railhead and insulation material is found to alter the levels of stress concentration. Numerical result indicates that a higher elastic modulus insulating material can reduce the stress concentration in the railhead but will generate higher stresses in the insulation material, leading to earlier failure of the insulation material.

[11]: Studied the fatigue life of rail joint bar by using three dimensional finite element model of rail joint bar and dynamic load is applied to estimate the fatigue life of the joint bar. Different components of the rail joint bars are being created separately and assemble in Autodesk Inventor. The model consists of assembly of the rail, joint bars, bolts, nuts, washers, and wheel. A three -dimensional finite element analysis of rail joint bars is carried out in ANSYS after importing from Autodesk Inventor. The static loads are being applied to estimate fatigue life and endurance strength at the section. The material properties of the rail and wheel are assumed same. The material properties of the wheel and rail are considered bilinear kinematic hardening in ANSYS.

[12]: studied the influence of design parameters on static response of the railway track structure by implementing Finite Element Method (FEM). He modeled and simulated different railway track and compared separately between different by using three categories of finite element analysis (eigenvalue analysis and general static analysis).

Anne [13]: Studied possible changes in insulated rail joint design in order to improve the performance of the insulated joint. The finite element program ABAQUS is used to model the supported butt joint. In this model, the rail, joint bars, epoxy, and ties surrounding the joint are modeled using solid elements. The remaining ties are modeled as an elastic foundation.

The rail is subjected to a tensile load, as well as a vertical wheel load that is applied to the rail using Hertz contact theory. Parametric studies are performed by varying the tie width, joint bar length, and joint bar dimensions. Two different wheel load locations are also investigated: centered about the end post, and halfway between the tie under the end post and the tie just to the left of the end post.

[14] Determined the vertical displacement of the rail and the shear and peel stresses in the epoxy of the joint considering both a static wheel load and a tensile load. The influences of design parameters of IRJs such as size of sleepers and thickness and length of joint bars on stresses in rail and epoxy were presented Ding.

[15] Studied the looseness of the bolts at the joints and concluded that the simulation results revealed that sufficient bolt tension was necessary to prevent the loss of structural integrity of the joint due to bending load. The loss of bolt tightness increased the length of separation of the contact surface that affected the behavior of the joint.

[16] This paper discusses the effects that track parameters such as foundation configuration and track surface condition have on joint bar stresses. Joint bars were notched using the Electro-discharge machining (EDM) technique to initiate cracks and were tested under controlled conditions at the Facility for Accelerated Service Testing so that crack growth rates under simulated mainline heavy haul freight operations could be evaluated. The limited test data showed no significant difference between concrete and wood tie track in terms of joint bar stresses. However, joints installed in curved track showed a considerable increase in stress state and required more maintenance than joints installed in tangent track. In addition, the data showed that standard joint bars experience higher stresses than insulated rail joints. Similarly, increasing the number of joint bar bolts and the magnitude of the bolt torque can have positive effects on joint performance. Limited thermal force data showed that insulated joints, once welded in place, behave like the surrounding rail and can develop considerably higher thermal stresses than standard rail joints. After cracks initiated from EDM notches at the bottom of the joint bars on tangent track, two joint bars out of eight broke within 35 million gross tons of traffic.

[17] studied the effects of epoxy deboning on the stress and strain are in a bonded insulated joint subjected to longitudinal force. Finite element method is used to model the different type of deboning feature. They studied the effect of the deboning by using ABAQUS software from their result; it shows under thermal tensile loads, strains at the center of the outer surface of the joint bar tend to increase as deboning begins near the end post. The strain at this point

tends to stabilize after the deboning reaches the innermost bolt hole. Strain at a point between the outermost and middle bolt holes starts relatively stable, but increases after deboning passes the innermost bolt hole. From the standard joint model it was concluded that: the results from the shell element model matched the expected behavior of straight rail and a standard joint reasonably well, based on comparisons with prior numerical simulations and analytical beam on elastic foundation theory. As aforementioned rail joint is the weakest link in the track the components should be optimized by strong materials as well as parameter variation and to increase the bolt connection strength of the joint to have the best stress values.

[22] Investigate the effects of axle load and train speed at rail joint. A three-dimensional finite element analysis of a rail/wheel contact is conducted on the rail joint section of track and dynamic load is applied to develop an estimate respective stresses at the section. In Autodesk Inventor, different components of wheel/rail assembly i.e. wheel, rail, joint, bars, nuts, bolts, and washers are created separately then all components are assembled, and create a complete model of wheel/rail assembly. The finite element program ANSYS is used to model the contact analysis. This ANSYS is used to simulate the loading and boundary conditions of the rail and wheel contact for a stress analysis.

[32] Studies use the hertz contact theory to perform the analysis for three various position of bolted rail joint on sleeper. This study covers only the stress and failure caused by the wheel load. The 3D model has done on modeling package of CATIAV5R16. Assembly model has created in assemble workbench of CATIA after individual component of joint had created on part work bench. The rail joint position at to sleeper. The stress distribution when the rail joint between sleepers is high in comparison to rail joint approach to sleeper and rail joint on sleeper. The stress distribution is when the joint on sleeper is small in comparison to the remaining.

2.4-Failure of bolted rail joint

Rail joints are often considered to be one of the weakest spots in track superstructure, mainly because of the discontinuities of both geometric and mechanical properties that arise as a result of the nature of rail gap, height mismatch and dip angle, as well as a lower bending stiffness at the rail joint compared with regular track. Those discontinuities can lead to high-impact loads when the wheels pass over the rail joints, which may cause or exacerbate defects, including rail-end batter, loosened bolts, deteriorated support condition, and excessive deflection, eventually leading to possible failure modes such as bolt hole cracks, head–web separation, bent or broken bolts, and cracked or broken joint bars. Bolt hole cracks and head–web separations are two of the most common defects that can develop in the rail joint area (Figure 2, *a* and *b*). These defects can cause rail breaks or loss of rail running surface. However, the initial cracks are difficult to detect because they are covered by joint bars, thus limiting or eliminating the effectiveness of inspection methods that rely on either human vision or machine vision.

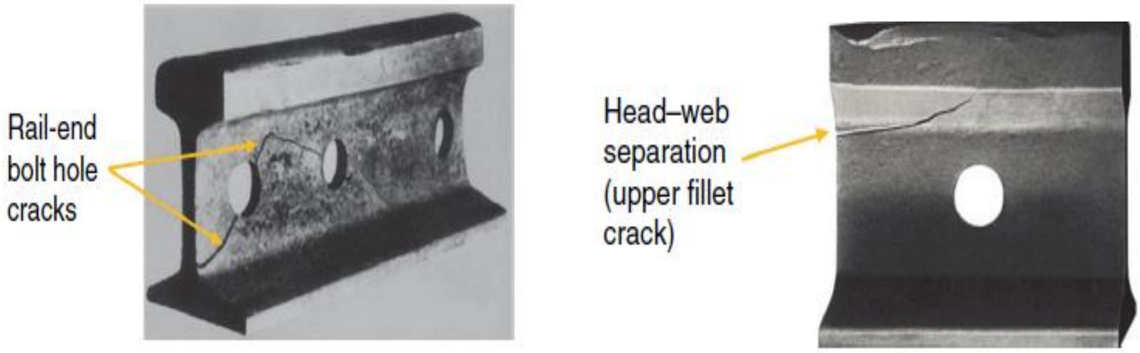


Figure 2.Typical (a) rail-end bolt hole crack, (b) head–web separation

[23] Have analyzed different designs of bolted joints, which are externally loaded. The mount stiffness of bolt and abutment predicted in this study was higher than the bolt load. A load application factor was introduced in VDI 2230 and a new fraction was determined, which was much lower than the existing one. In practical design, there was no effect of the location of the external load on the application factor, but it was affected by the layout of the bolt joint. The load factor Φ based on the mounting stiffness's of the bolt and abutment is only valid when the external load is applied directly at the bolt head.

[24Studied the fatigue life of rail joint bar by using three dimensional finite element model of rail joint bar and dynamic load is applied to estimate the fatigue life of the joint bar. Different components of the rail joint bars are being created separately and assemble in Autodesk

Inventor. The model consists of assembly of the rail, joint bars, bolts, nuts, washers, and wheel. A three -dimensional finite element analysis of rail joint bars is carried out in ANSYS after importing from Autodesk Inventor. The static and dynamic loads are being applied to estimate fatigue life and endurance strength at the section. The material properties of the rail and wheel are assumed same. The material properties of the wheel and rail are considered bilinear kinematic hardening in ANSYS.

[25] Carried out static loading tests in a laboratory, field tests to measure the stresses around the fish bolt holes, static stress analysis of jointed rails and bending fatigue tests of rails with bolt holes. Based on the study results, they developed a method to evaluate the fatigue life of jointed rails.

[7] Carried out an analysis of stresses developed in insulated rail joint. He showed that due to degradation and failure of insulated joints there may be chances of failure of signal which will cause delay in train and also there is a chance of accident. There are two types of failure occur in IRJ: electrical and mechanical failure. Before failing electrically it leads to mechanical failure, which occur due to different stresses develop in various component of IRJs. He listed joint stresses and failure. These are bending stress, thermal stress, residual stress, shear stress, von misses stress, stresses due to static and dynamic load, fatigue failure, broken bolt, poxy layer debonding from rail, joint bar or both, joint bar failure, delamination of end post, crushed end post and metal flow.

The main purpose of this research is to analyze the stress due to longitudinal and lateral load and its effect on the bolted rail joint at three different geometric of bolt on the standard bolted rail joint by using 3D finite element method. It helps to demonstrate how stresses contribute to the wear and deterioration of the rail track.

Chapter Three: Analytical Methods and condition

For the analysis, all standard data for the rail profile are gathered from China standard in accordance to theTB10082-2005 (50kg/m)). All track cases are considered and the analysis is done by using track parameter of TB10082-2005. This is because the Nation Railway Network Ethiopia (NRNE) track is constructed by using this china standard. For analyzing the stress analysis, it is very useful to use FEM simulations based on the adopted analytical model for defining fatigue load sources at surfaces which are in contact with the joint. This study will use the FEA software ANSYS14 version to carry out stress analysis and the mechanical stresses of a railway bolted rail joint during wheel running conditions will be determined.

3.1-Materiel properties of bolted rail joint of NRNE

The railway tracks are mostly steel material in accordance to the EN 13674-1:2011 (E). Steel is the most common and widely used metallic material in modern society. Steel contains 50% iron and one or more alloying element. These elements generally include carbon, manganese, silicon, chromium, phosphorus, sulpher etc. Each element has a specific role in the steel making process or in achieving particular properties E.g. Strength, hardness and quality. Most of the materials used in this specific project are based on Nation Railway Network Ethiopia (NRNE).

In this study, the material of rail used in NRNE is U71Mn, which is equivalent to R260Mn. Rail material rail property in accordance to EN 13674-1:2011 (E)

Rail Stresses

Bending Stress – Bending of the rail that occurs from vertical wheel loading and lateral wheel loading. Vertical wheel loading normally results from loading between the tie supports, and causes tensile longitudinal stresses in the rail base area and head/web fillet area. Lateral wheel loading applies tensile longitudinal stresses in the rail web area and head/web area of the rail field side.

3.1.1-propertie of rail

Rail is the most important and most expensive element of the track structure. It is the point of contact with the vehicle wheel, the structure beam supporting the vehicle load and one local where noise is generated. The rails are the long steel members whose purpose is to support and guide the wheels of the train.

Rails are the most fundamentally important members of railways and have the following roles.

- ✓ They directly support the wheel load with large carriages.
- ✓ They impart safe and smooth travel surface to carriages and provide safe passage and guidance.
- ✓ They disperse the wheel load to rail support and facilitate rail track maintenance control. The rail configuration and material are determined in order to fulfill these roles.

The steel rails must be strong enough to carry the large load of a train at acceptable deflections. Deflections can be minimized with the correct cross spacing. The closer the cross ties are to one another, the smaller the internal moment becomes in the rail. The smaller the moment, the smaller the deflection becomes.

Rails are grouped according to their standards, strength, grade, quality and length. The rail steel qualities can be distinguished in to two categories.

- ✓ Normal steel quality, with an ultimate tensile strength of 700-900 MPa
- ✓ Hard steel quality used mainly on curves, and crossings etc. with an ultimate tensile strength of 900-1200 MPa.

  Rail profile has been the object of continuous improvement since the beginning of railways. The cross-sections of gauge rails have been standardized by the UIC.

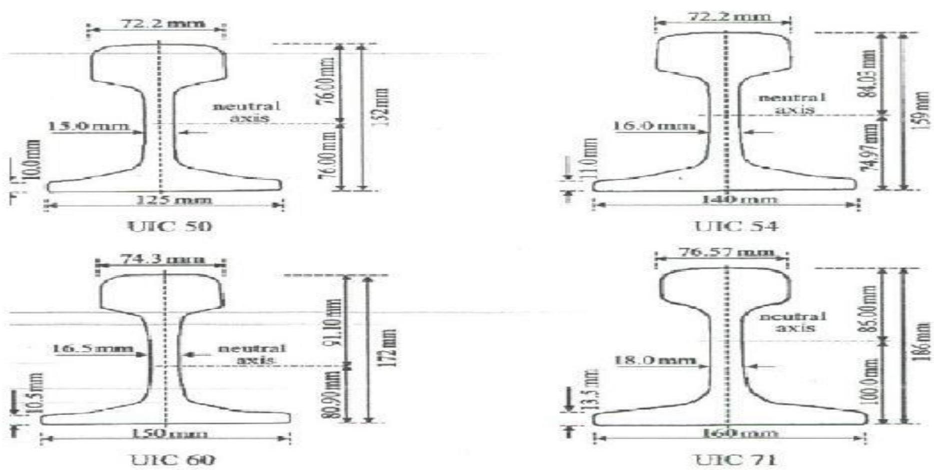


Figure 3.1: Rail profiles UIC 50 (50 E1), UIC 54 (54 E1), UIC 60 (60 E1) and UIC 71 (71 E1)

Table 3.1- Material selection of rail

Item	1	2	3	4	5	6	7
Chemical element	C	Si	Cr	Mn	P	S	Al
Composition	0.65-0.76	0.15-0.58	…….	0.70-1.2	≤0.04	<0.025	< 0.010

3.1.2-propertie of Fish bolts

Bolted joint one of the main functions of the bolt is to maintain an adequate positive clamping force during the service life of the joint in order to prevent leaks, relative movement, wear and fretting**.** Fish bolts are used for connecting fish plates with the rails. Four or six bolts are required for each pair of fish plates. *Bolt* is a device with a head on one end of a shank or body and a thread on the other end. Designed for insertion through holes in assembly parts, it is mated with a tapped nut. Tension is normally induced in the bolt to compress the assembly by rotating the nut. This may also be done by rotation of the bolt head. These bolts are inserted from outside the track and bolted on the inside of the track. Fish bolts have to withstand shear due to heavy transverse stresses, hence they are manufactured of medium or high carbon steel. The length of fish bolt depends on the type of fish plate used. For 44.70kg rail, the fish bolts of 25 mm dia and 127.6 mm length are used. These bolts get loosened due to vibration of moving train and hence these are to be tightened time of time. Too much tightening of bolts is prohibited as it prevents free expansion or contraction of rails due to temperature vibrations.

The mechanical properties are sensitive to the carbon content, which is normally less than 1.0%. For fasteners, the more common steels are generally classified into three groups: low carbon, medium carbon and alloy steel. But Nation railway used low carbon steel materiel

Low Carbon Steels

Low carbon steels generally contain less than 0.25% carbon and cannot be strengthened by heat-treating; strengthening may only be accomplished through cold working. The low carbon material is relatively soft and weak, but has outstanding ductility and toughness; in addition, it is Machinable, weldable and is relatively inexpensive to produce. The most commonly used chemical analyses include AISI 1006, 1008, 1016, 1018, 1021, and 1022.

Table 3.2: Mechanical property low carbon steel for ERC

S.N	Mechanical property	Value
1	Poisons Ratio	0.3
2	Young'sModulus (GPa)	207
3	Tensile strength (MPa)	482
4	Yield strength (Mpa)	302
5	Density (Kg/m)	7800
6	Elongation	25%

3.1.3. Fish plate material

The function of the fishplate is to hold the two rails together both in the horizontal and vertical planes. The fish plates are manufactured from a special type of steel having composition of Carbon, Manganese, Silicon, Sulphur and Phosphorous 0.30 to 0.42%, Not more than 0.8%, Not more than 0.15%, Not more than 0.06 % respectively. The fish plates are so designed that the fishing angles at the top and bottom surface coincide with those of the rail section so as to have a perfect contact with the rail.

Table 3.3- Material selection of joint plate

Item	**1**	**2**	**3**	**4**	**5**	**6**	**7**
Chemical element	**C**	**Si**	**Cr**	**Mn**	**P**	**S**	**Al**
Composition	**0.52~0.60**	**0.50~0.80**	**…….**	**17~0.37**	**≤0.035**	**<0.025**	**< 0.010**

3.2-Dimension of bolted rail joint

3.2.1-Rail dimension

The rail is designed as a beam with the cross section of a UIC 60kg/m standard rail which is presented in Figure. Because the rail is interacting with wheel and Transfers the sleepers; it remains the rail cross section profile with exact dimensions, in order to keep the accuracy of the results. The dimension of rail shall conform to the appropriate profile of Nation Railways

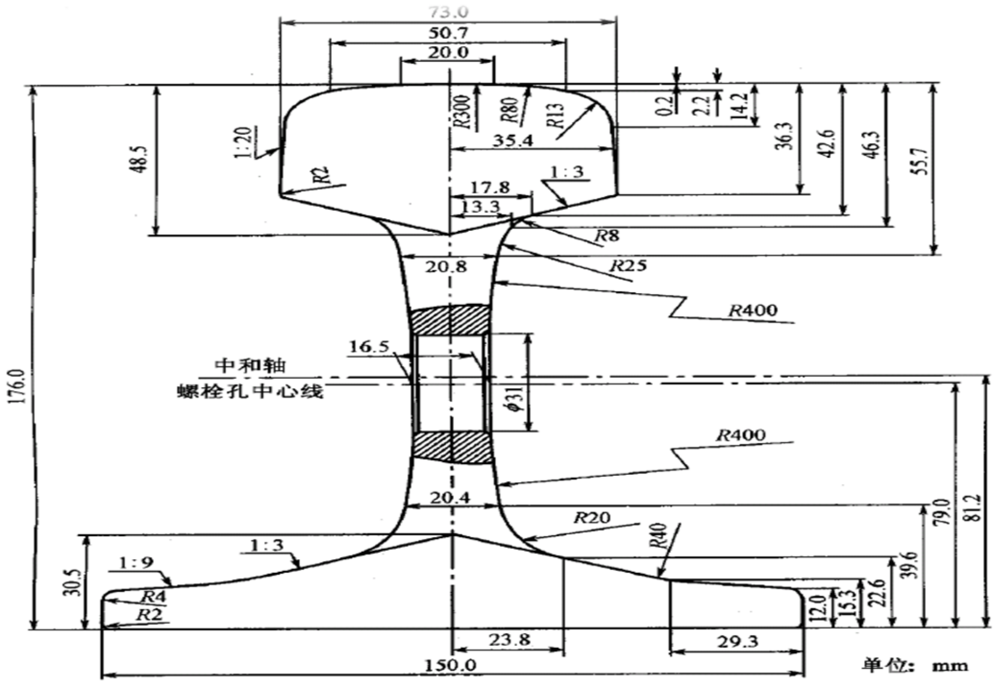


Figure 3.2: Dimension of rail for ERC

Parameters	Dimension in(mm)
The overall height of the rail	172
The height of railhead	59.4
The width of the rail	150
The width of railhead	77.3

Material type	Mild steel
Modulus of elasticity, E	200 Gpa
Poisson's ratio, A	0.29
Thickness	10 mm
Width	50 mm

Table 3.4 Mechanical property and dimension of rail material

3.2.2- Joint plate dimension and Mechanical property

Two joint bars or fishplates and six bolts are used to connect the ends of adjoining rails. This joint assembly is to line-up rail ends horizontally and vertically and to create smooth running surface for rolling contact of wheel and rail.

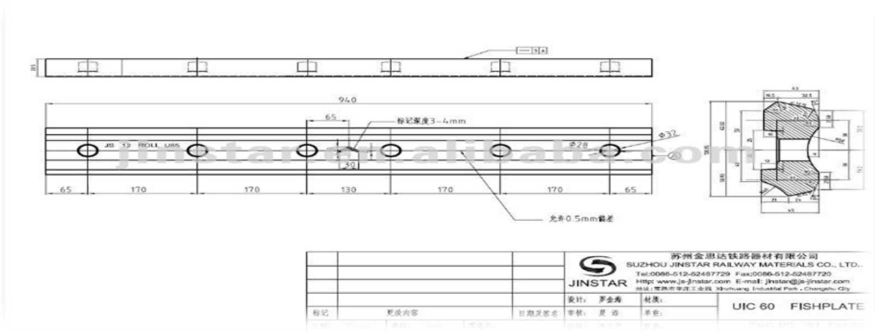


Figure 3.3: Dimension of joint plate for ERC

Standard	Material	Tensile strength Rm	Size(kg)	Yield strength	Elongation of Area Z	Brinell hardness	Cold bend d=3a
TB/T2345-2008	steel 55#	60	≥785 Mpa	≥520 Mpa	≥9%	HBW 227-388	30 degree, no crack

Material type	**Mild steel**
Modulus of elasticity, E	200 Gpa
Poisson's ratio, A	0.29
Thickness	10 mm
Width	50 mm

Table 3.4- Mechanical Property and dimension of the joint Plates

3.2.3-Main Technical Parameters used on the Stress Analysis of bolted rail joint due to lateral and longitudinal forces

This research thesis is used to simulate the Nation railway network of Ethiopia project bolted rail joint stress analysis due to lateral and longitudinal forces. For this simulation, NRNE tramcar is considered .Currently, this NRNE use of bolted rail joint.

Table 3.5-Technical parameters used for the analysis ERC

Technical parameters	**Values**
The speed of the vehicle	120 km/h for design speed
Tram car length	25500mm
Weight of tram car	44.7t
Track gauge	1435mm
Distance between two sleeper pads	0.658m for mainlines and 0.625m for elevated
Rail standard in NRNE	TB10082-2005(60kg/m)
Length of the rail	25m
Height of the rail	172 mm
Top width of the rail	72 mm
Foot width of the rail	150 mm
Rib thickness	16.5mm
Cross sectional area of rail	76.86cm
Young modulus of rail	207 MPa
Density of rail	7800 kg/m
The pads	rubbercushion (polyethylene)
Type of sleepers	Concrète
Wheel diameter	840mm for motor wheel, 800mm for driven wheel
Sleeper top width	220mm
Sleeper bottom width	250mm
Sleeper length	2200mm
Rail pad length	180mm
Rail pad width	150mm
Sleeper height	208mm
Load on each wheel	11.18 ton+3%
Stiffness of ballast	100KN/mm
Stiffness of pads	50 ~ 70 KN/mm (for analysis take 60KN/mm)
Rail pad thickness	7.5mm

3.3-Test rigid of bolted rail joint

3. 3.1- Modeling Contact at bolted Rail Joint

The most common profiles of the wheel and the rail-head are shown in Figure 1. The wheel profile consists of a flange to guide the trains along the rails and a conical tread that contacts rail head, the wheel tread – railhead contact occurs at a very small area, and rail has many curvatures to guide wheel properly. The contact positions of the wheel / rail are different in the different situation. However, this paper uses the contact between the wheels tread and rail head.

The contact area between wheel and rail are very small compared to their dimension.

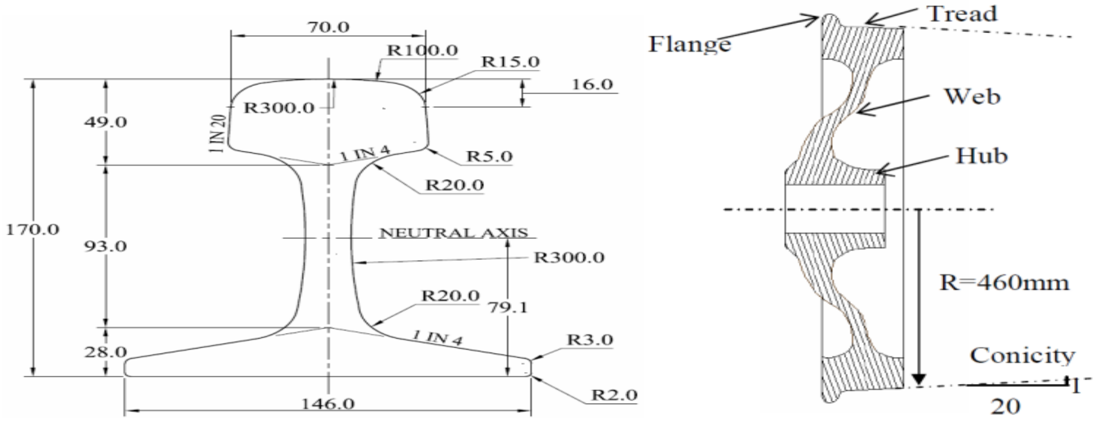


Figure 3.4: Profiles of rail and wheel

Rails are produced in fixed lengths and need to be joined end-to-end to make a continuous surface on which trains may run. Bolted rail joint are designed and installed at site with a view to providing a smooth running surface. Unfortunately the difference in the elastic moduli of the railhead steel and the end post material causes discontinuity However; presence of the joint caused the high stress on the rail, which produces the diverse type of wear at the rail head. This paper concerns on the stress distribution on bolted rail joint due to the lateral and longitudinal loads.

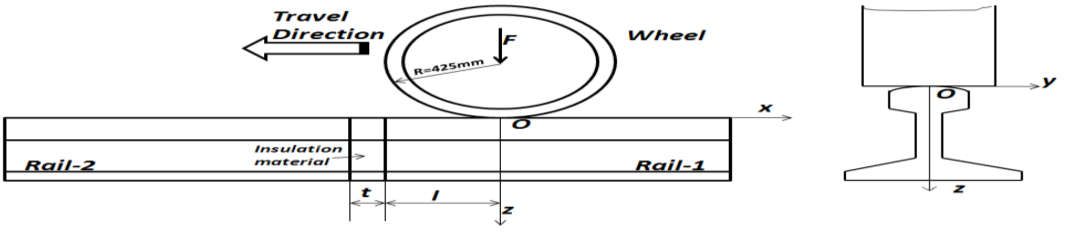


Figure 3.5: Wheel-rail contact configuration

3.3.2- Hertz Contact Patch Theory

As the wheel is located far away from bolted rail joint, the contact can be treated as Hertzian contact problem. Hertz proposed the solution for the determination of contact area and pressure distribution between two bodies in contact. Assume that each body has two principal radii: one in they-z plane (R1x, R2x) and the other in the x-z plane (R1y, R2y). The separation for this case can be written as,

$$Z(x, y) = Ax2 + By2 \text{..} 3.1$$

Where,

$$A = ½(1 /R1y + 1/ R2y) \text{..} 3.2$$

$$B = ½(1/R1x + 1 /R2x) \text{..} 3.3$$

With A and B known as relative longitudinal and lateral curvature respectively.

According to Hertzian contact theory, the contact surface between two curved surfaces, such as a wheel and a rail, can be represented as an ellipse with a major semi-axis a and a minor semi-axis b. The Hertzian patch and pressure distribution is shown in Figure 3.3

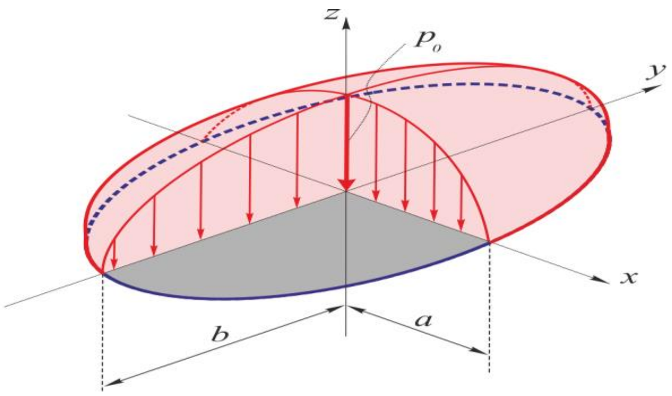


Figure 3.6: Hertzian contact theory

The pressure exerted over this elliptical area is parabolic in two directions and is defined according to the following equation:

$$P(r) = p_o \sqrt{1 - \frac{x^2}{a^2} - \frac{y^2}{b^2}} \text{..} 3.4$$

Where, p0 is the maximum contact pressure at the first point of contact. The integration of the contact pressure over the contact patch must be equal to the total applied force due to static equilibrium. The maximum pressure p0 is therefore related to the prescribed contact force, F through. The value of p0 is given by:

$$p0 = 3F/2\pi ab \text{..} 3.5$$

Where, F is the applied normal force.

Based on the Hertz contact theory, the contact point is very small relative to the overall ldimension of wheel and rail surfaces. This very small contact point has elliptical shape. There is small gap on the pressure distribution due to the end gap on the joint. The magnitudes of a and b also depend on the applied normal force, as well as the profile and materials of the wheel and rail. They are expressed as

.

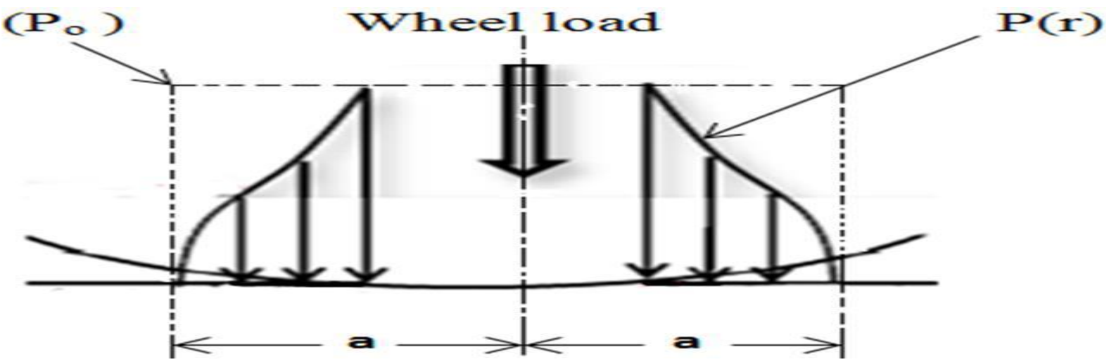


Figure3.7: Elliptical shape

If x=0 and y=0 the point of contact is on the centerline of the rail head the stress is maximum, which is equal to:

$P = p_0$, where p0 =3F/2$p$ab.. 3.6

From the above formula **a** and **b** are semi axes of the contact ellipse whereas X and Y are the required coordinates to specify the point of contacts on the rail surface based on the lateral rail surface parameter. However, based on Hertz contact formula and assumptions, the stress due to wheel/rail contact decreases and becomes zero when it goes far away from the centerline of the rail head. Similarly, the wheel/rail contact stress is inversely proportional to the major and minor axis of the contact ellipse [18].

The contact area determined as follows:

a = m$(3\pi F(\frac{K_w+K_r}{4K_3}))^{1/3}$**.. 3.7**

b = n$(3\pi F(\frac{K_w+K_r}{4K_3}))_{1/3}$ **.. 3.8m**

and **n** are Hertz coefficients and they are given as a function of the angle (0_o -180_o)$\theta = cos-1$ **$(K4/K3)$..3.9**θ is rail curvatureKwandKr are constants that depend on the material properties of railway wheel and rail respectively.

$Kw=1-(vw)\,2/(\pi Ew)$..3.10

$Kr=(1-vr)\,2/(\pi Er)$..3.11

- ✓ vr, and Er are Poisson's ratio and young's modulus of railway rail material
- ✓ vw, and Ew are Poisson's ratio and young's modulus of the railway wheel material respectively.

Whereas $K3$ depends on the geometric properties of the two bodies

$K3=A+B=$ $1/2\,(1/R1w+1/R2w+1/R1r+1/R2r)$**...................................3.12**

K $_4=\frac{1}{2}\left(\left(\frac{1}{R_{1w}}-\frac{1}{R_{2w}}\right)^2+\left(\frac{1}{R_{1r}}+\frac{1}{R_{2r}}\right)^2+2\left(\frac{1}{R_{1w}}-\frac{1}{R_{2w}}\right)\left(\frac{1}{R_{1r}}-\frac{1}{R_{2r}}\right)cos2\varphi\right)^{1/2}$ **=B-**

A…..3.13$R1w$ **$1r$**, are the principal rolling radii of the wheel and the rail respectively

$2w$ and**$R2r$**, are the principal transverse radii of curvature of the wheel and rail respectively

The direction of the axes of the contact ellipse can be determined based on the radii of curvature and the rolling radii for the two bodies in contact [16].

If $1/R1w+1/R1r\geq1/R2w+1/R2r$,the transverse semi axis of the contact ellipse (y direction) is greater than or equal to the longitudinal semi-axis.

If $1/R1w+1/R1r\leq1/R2w+1/R2r$, the transverse semi axis of the contact ellipse (y direction) is less than or equal to the longitudinal semi-axis

3.4. Methods

3.4.1-Finite element theory for Contact body

The general conditions considered during the wheel rail contact simulation are the assumption of the Hertz contact theory .Finite element theory is used to show that relationship among the contact force, applied force, support and free displacement of wheel /rail contact. During the formulations of finite element the contact between the wheel and rail is assumed:

- ✓ Isotropic and homogenous material
- ✓ no friction (Hertz wrote that the surfaces of both bodies had to be completely smooth)
- ✓ both bodies were considered as half-spaces
- ✓ The contact ise lastic

The linear elastic bodies have four boundaries condition as shown in the figure below.

- ✓ Γ1 is the boundary with zero displacement.
- ✓ Γ2 is the boundary where measured displacements.
- ✓ Γ3 is the boundary with unknown contact forces Fc and unknown contact deformation or displacements.
- ✓ Γ4 is the boundary where applied forces Fa and the other are free surface except those mentioned above. The displacements on boundary Γ4 are unknown.

Let's recall the general form of static finite element system, which is

KU=F [5]……………………………………………………………………3.14

K, U and F are stiffness matrix of the system, nodal vector displacement, and nodal vector forces. According to the classification of the boundary, it constructs the finite element equation in the following form:

$$\begin{bmatrix} K_{11} & K_{12} & K_{13} & K_{14} \\ K_{21} & K_{22} & K_{23} & K_{24} \\ K_{31} & K_{32} & K_{33} & K_{34} \\ K_{41} & K_{42} & K_{43} & K_{44} \end{bmatrix} \begin{bmatrix} U_1 \\ U_2 \\ U_3 \\ U_4 \end{bmatrix} = \begin{bmatrix} F_1 \\ F_2 \\ F_3 \\ F_4 \end{bmatrix} \quad \ldots\ldots\ldots\ldots\ldots\ldots 3.15$$

K_{ij} and $F_1$are sub-stiffness matrix and vector of reaction forces on the boundary Γ_1.

F_2 is a vector forces on the boundary Γ_2 with measured displacements, usually there is no force on the measured boundary.

F_c and F_a are vector of unknown reaction or contact forces on the boundary Γ_3 and vector of known applied forces on the boundary Γ_4.

U_1 and U_2 are known displacements on constrained boundary Γ_1 and measured displacement on free boundary Γ_2

U_3 and U_4 are unknown displacements on contact boundary Γ_3 and unknown displacements on boundary Γ_4 with known applied force Fa,the free surface with no applied force and the internal nodes where net force is zero.

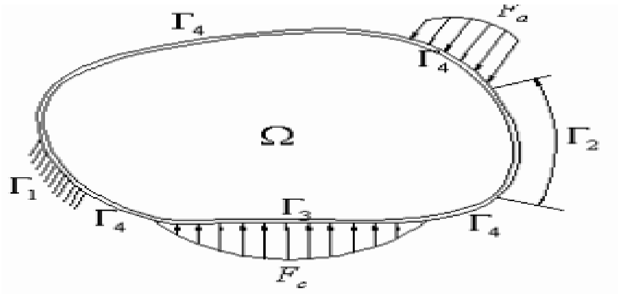


Figure 3.8: Contact model analyze[19].

The stiffness matrix is singular and no unique solution for displacement is possible if the structure is unsupported for the above structure stiffness equation. For this reason all displacement on the boundary Γ_1 are zero, that means $U_1=0$.When apply this condition to the system matrix and vector in equation 3.15 FEA equation becomes: Matrix and vector in equation 3.15 FEA equation becomes:

$$\begin{bmatrix} K_{21} & K_{22} & K_{23} & K_{24} \\ K_{31} & K_{32} & K_{33} & K_{34} \\ K_{41} & K_{42} & K_{43} & K_{44} \end{bmatrix} \begin{bmatrix} U_2 \\ U_3 \\ U_4 \end{bmatrix} = \begin{bmatrix} F_2 \\ F_C \\ F_a \end{bmatrix} \quad \ldots\ldots\ldots\ldots\ldots 3.16$$

To calculate the contact forces F_c at U_2. Multiple 3rd row of the stiffness matrix with displacement matrix, then equation became:

$$\mathbf{K_{42}U_2 + K_{43}U_3 + K_{44}U_4 = Fa} \ldots\ldots\ldots\ldots 3.17$$

$$\mathbf{U_4 = K_{44}^{-1}[Fa - K_{42}U_2 - K_{43}U_3]} \ldots\ldots\ldots\ldots 3.18$$

Multiple the 2nd rows with displacement column, the equation became;

$$\mathbf{K_{32}U_2 + K_{33}U_3 + K_{34}U_4 = F_c} \ldots\ldots\ldots\ldots 3.19$$

$U_3 = K_{33}^{-1}$ [Fc $-K_{32}U_2 - K_{34}U_4$]………………………………..…………………………...3.20

When equation 3.17 substitute in the equation 3.18, U3 became;

$U_3 = K_{33}^{-1}[F_c-(K_{33}- K_{34} K_{44}^{-1} K_{43}) U_2 - K_{34}K_{44}^{-1}F_a + K_{34}K_{44}^{-1}K_{43} U_3]$………………….. 3.21

Therefore the displacement at the contact point is: **$U_3 = K_{33}^{-1}[F_c-(K_{33}- K_{34} K_{44}^{-1} K_{43}) U_2 - K_{34}K_{44}^{-1}F_a + K_{34}K_{44}^{-1}K_{43} U_3]$…………………..3.22**

Generally, the contact between the wheel and rail are considered to determining the failure effect of rail end and rail joint.

Stress -strain relationship of structural analysis

$\{\sigma\}= \{K\} \{\epsilon el\}$………………………………………………………………………………….3.23

ϵel, σ and D are elastic strain vector, stress vector and elastic stiffness matrix.

$\{\sigma\}=\{\sigma x,\sigma y,\sigma z,\sigma xy,\sigma yz,\sigma xz\}$ ……………………………………………………………….3.24

$\{\epsilon el\}= \{\epsilon\} -\{\epsilon\}th$ ………………………………………………………………………………………3.25

$\{\epsilon\}=\{\epsilon\}th+\{D-1\}+\{\sigma\}$……………………………………………………………...3.26

$$\{D-1\}= \begin{bmatrix} 1/E_x & -v_{xy}/E_x & -v_{xy}/E_x & 0 & 0 & 0 \\ -v_{xy}/E_Y & 1/E_Y & -v_{yz}/E_y 0 & 0 & 0 & -v_{zx}/E_z \\ -v_{zy}/E_z & 1/E_z & 0 & 0 & 0 & 0 \\ 0 & 0 & 1/G_{xy} & 0 & 0 & 0 \\ 0 & 0 & 0 & 1/G_{yz} & 0 & 0 \\ 0 & 0 & 0 & 0 & 1/G_{xz} & 0 \end{bmatrix}$$

E_x and $G_{xy}$are young's modulus in the x direction and shear modulus in the xy plane$V_{xy}$and $v_{yx}$are major poison's ratio and minor poison's

Also the $\{D-1\}$ matrix is presumed to be symmetric, so that

$v_{xy}/E_Y=v_{xy}/E_x$…………………………………………………………………………………….3.27

$v_{zx}/E_z=v_{xz}/E_x$…………………………………………………………………………………….3.28

$\nu_{zy}/E_z = \nu_{yz}/E_x$ **3.29**

$\nu_{zx}/E_z = \nu_{xz}/E_x$ **3.30**

$\nu_{zy}/E_z = \nu_{yz}/E_y$ **3.31**

The element integration point strain and stress are:

$\{\epsilon el\} = \{B\}\{U\} - \{\epsilon\}th$, for this case, $\{\epsilon\}th$ is zéro3……………………..……........3.32

$\{\sigma\} = \{D\}\{\epsilon el\}$ 3.33

B and $\{\epsilon\}h$are strain - displacement matrix evaluated at integration point and thermal strain

$\epsilon el$is strain that cause stress

Maximum stress failure criteria

$$\varepsilon_x = \text{Maximum of} \begin{cases} \dfrac{\sigma_{xc}}{\sigma f_{xc}} & , \dfrac{\sigma_{xy}}{\sigma f_{xy}} \\ \dfrac{\sigma_{yc}}{\sigma f_{yc}} & , \dfrac{\sigma_{yz}}{\sigma f_{yz}} \\ \dfrac{\sigma_{zc}}{\sigma f_{zc}} & , \dfrac{\sigma_{zx}}{\sigma f_{zx}} \end{cases}$$

σ_x, σ_y and σ_z are stress in x, y, z direction.

σ^f_{xc} , σ^f_{yc} *and z*are normal stress failure in x, y, z direction.

σxy,σ_{xy} andσ_{xy} are shear failure in xy, yz, xz direction.

3.4.2-Three directional model of joint

The three dimensional models of bolted rail joint were developed by using modeling software Catia and analysis was carried out with use of Ansys software to predict stress distribution among regions of failure of bolted joint.FEM is a numerical method for solving problems of engineering and mathematical physics. This process includes Finite Element Modeling and Finite Element Analysis.FEM or FEA is a computational technique used to obtain approximate solutions of boundary value problems in engineering. And also ANSYS is a general purpose finite element of modeling package for numerical solved problems. Like any finite element software, ANSYS solves governing differential equations by breaking the problem into small elements. When the wheel is contact element and the rail is target element, the wheel and rail will be different. The wheel structure discretized using CONTA174 element. CONTA 170 three dimensional elements are used to discretize rail.

3.4.3- Steps for finite element analysis:

FEA is mainly divided into three following stages:

- ❖ Preprocessing
    - ✓ Creating the model.
    - ✓ Defining the element type
    - ✓ Defining material properties
    - ✓ Meshing
    - ✓ Applying loads
    - ✓ Applying boundary conditions
- ❖ Solution: Assembly of equations and obtaining solution
- ❖ Post processing: Review of results such as deformation plot, stress plot, etc.

3.4.4- Constitutive Models

In order to define the stresses and to optimum the geometry on bolted rail joint because of lateral and longitudinal forces, a structural sketch is prepared using CATIA V5 modeling and analysis software. This model is exported to ANSYS -14 for final result analysis. The purpose of the structural sketch is to define the rail profile and bolted rail joints for stress analysis.

The finite element models of the rail joint bolts are established. According to the national standards, the material of the bolts is steel whose Young's modulus is 207 Gpa, Poisson's ratio is 0.3, yield strength is 7640Mpa and tensile strength is 880Mpa. The geometry dimension of the bolt as follows:

1. M30 the length of bolt bar is 133mm, the nominal diameter with screw thread is 30mm, the length of the thread is 60mm, and the thread pitch is 3.5mm.

2. M32 the length of the bolt bar is 134, the nominal diameter is 32mm, the length of the thread is 64mm, and the pitch is 3.5mm.

4. M34 the length of the bolt bar is 134, the nominal diameter is 34 mm, the length of the thread is 64mm, and the pitch is 4mm.

For this analysis, this research develop the model is used to investigate the stresses developed in standard joint due to lateral and longitudinal forces of NRNE. The profile drawing includes rail profile, standard rail joint.

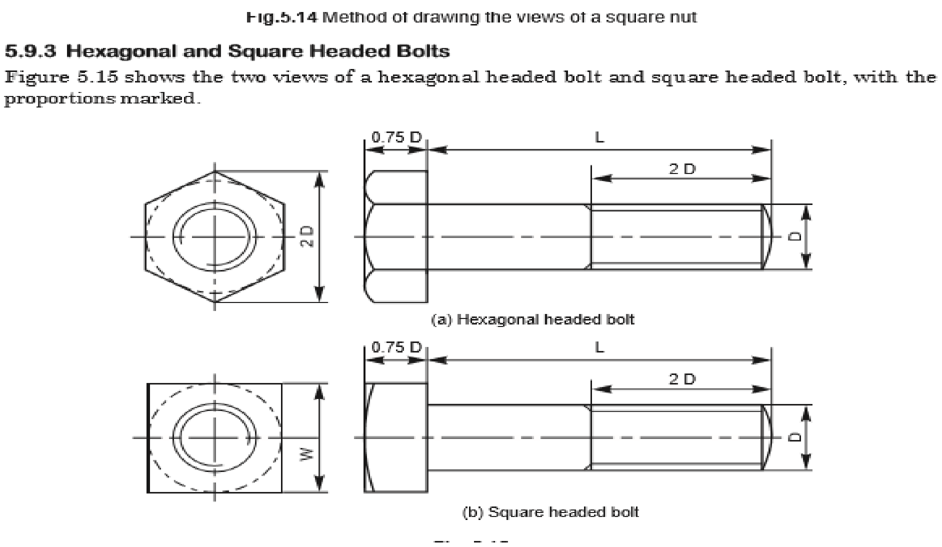
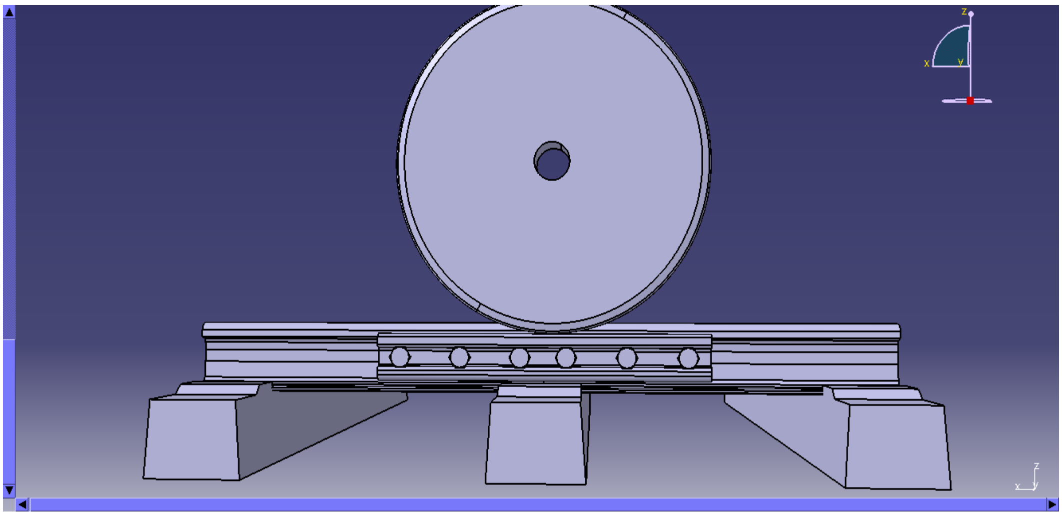


Fig.5.14 Method of drawing the views of a square nut

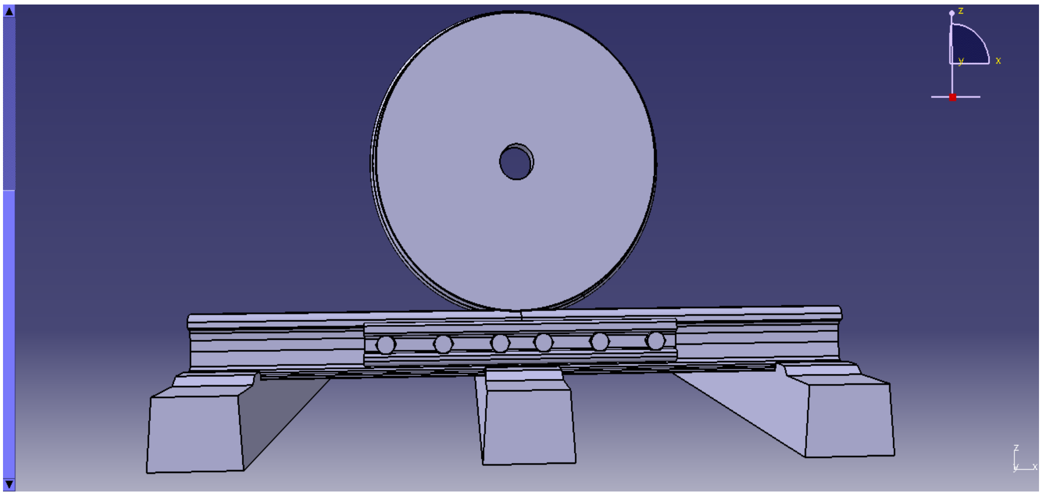
5.9.3 Hexagonal and Square Headed Bolts

Figure 5.15 shows the two views of a hexagonal headed bolt and square headed bolt, with the proportions marked.

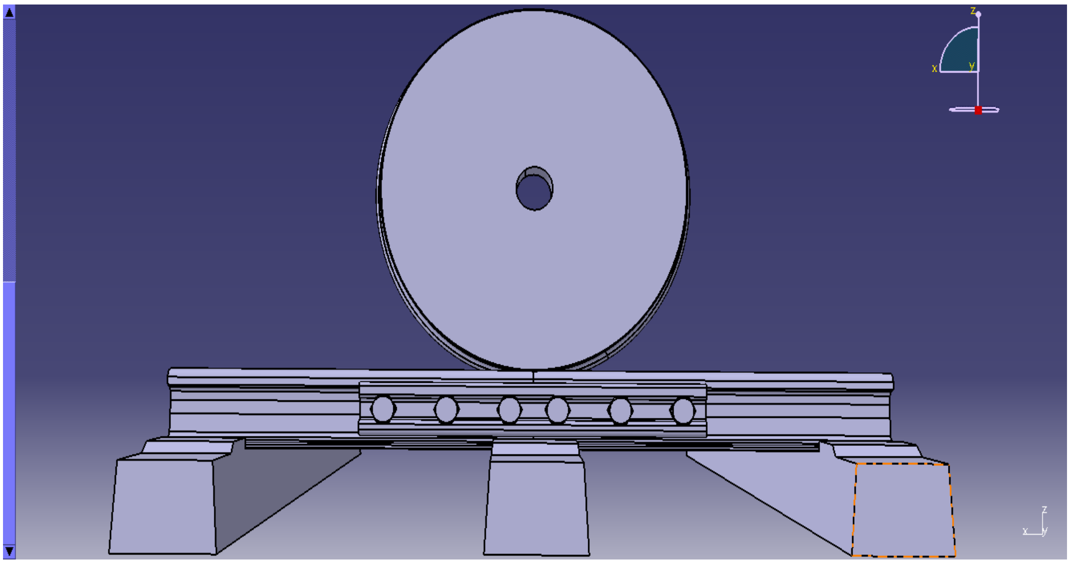
3.44.1-Modeling of optimum geometry of bolt M30



3.44.2-Modeling of optimum geometry of bolt M32



3.44.3-Modeling of optimum geometry of bolt M34



3.5-Load and condition

The forces applied to the track are vertical, lateral (parallel to the ties) and longitudinal (parallel to the rail).these forces are affected by train travel speed. An important point to recognize is that track design involves many force repetition. The rail should provide smooth running surfaces for the train wheel and they should guide the wheel set in the direction of the track. The rails also carry the vertical load of the train and distribute the load over sleepers. Lateral forces from the wheel sets and longitudinal forces due to traction and braking of the train should also be transmitted to the sleepers and further down into the track bed.

3.5.1-vertical force

The main vertical force is the repetitive downward action of the of the wheel load. In addition, this wheel/rail interaction produces a corresponding lift-up force on the tie away from the wheel load point.

The nominal vertical wheel force, also called the static force is equal to the gross weight of the railway car divided by the number of wheels.

The major factor affecting the magnitude of the dynamic vertical force:

- ✓ Nominal wheel load
- ✓ Train speed
- ✓ Wheel diameter
- ✓ Smoothness of the rail and wheel surfaces
- ✓ Track geometric
- ✓ Track modulus of vertical track stiffness

Vertically downward load is sum of 3% allowance, maximum axle load.Carrying capacity of vehicle has calculated by take average of 60kg/ person and the total rate of passenger inside of the tramcar is 118 for NRNE.

- ➢ Tram car weight = 44.7 ton=44000 kg so 1 ton = 1000 kg

$F=ma=mg$=44700kg$*$9.81m /s2=438507 kg

- ➢ The load apply on each wheel = 44.7/8=5.5875ton =5587.5kg
- ➢ Carrying Capacity = $60kg/person*118\ person$ =7080 kg
  7080/8=885 kg
- ➢ Over all capacity =$Tramcarweightoneach\ wheel + CarryingCapacity$ = (5587.5Kg+885Kg)$*$9.81$ms2$/=(6472.5Kg)$*$9.81$ms2$/ =63495.225N=63.495KN
- ➢ Maximum Axle load=11160*9.81=109479.6N

- The total vertical load = maximum Axle load +3% maximum Axle load = (109479.6N+3.004=112763.988 N=112.764 kN
- The load on each wheel =Static force =112.764 /2=56.382kN

3.5.2-Lateral forces

The lateral force applied to the rail head produces a lateral deflection and twist in the rail that mean this force the wheel force transmitted thought friction between the wheel and top the rail and by the wheel flange acting against the inside face of the rail head, particularly on curve. Lateral force causes the rail to bend horizontally and the resultant torque causes a huge twist in the rail as well as the bending of the head and foot of the rail.

Lateral deflection of the rail is resisted by the friction between the rail and the sleeper, the resistance offered by the rubber pad and fastenings, as well as the ballast coming in contact with the rail that means this force is the rail bucking resistance force. The combined effect of lateral forces resulting in the bending and twisting of a rail can be measured by strain gauges. Field trials indicate that the loading wheels of a locomotive may exert a lateral force of up to 2 t on a straight track particularly at high speeds.

The design lateral force wheel force depends upon a number of factors including:

- ✓ Vehicle speed
- ✓ Track geometry
- ✓ Elevation difference between the two rail at same cross section
- ✓ Transverse hunting movement due to the train-track dynamic

As the train speed increases, the lateral force outward on the outside rail of curve increase and simultaneously the lateral force on the inside rail decreases.

The on a railway vehicle are tapered so the diameter decreases from the inside to the outside. This helps center the wheels on straight track and compensates in parts for the greater distance that the outer wheel travel on a curve. Because the wheels are fixed to the axle, both wheels must turn together. Thus wheel slip is requiring to the extent that the circumference of the wheel doesn't compensate for the difference in the inside and outside rail length in a curve.

To achieve the desired alignment, the geometric components of a track are tangent, spirals (transition between straight track and constant radius curve) and constant radius in horizontal and gradient and vertical plane.

Vehicles shall be designed so that under all normal track and operating conditions they do not generate lateral forces which could jeopardize the structural integrity of the rails and track. A vehicle shall not subject the track to lateral forces greater than: [20]

$Y = W/3 + 10$ ……………………………………………………….…………..... 3.33

A vehicle shall be able to negotiate a lateral ramp discontinuity in track alignment, when travelling on a curve at maximum normal operating speed and at maximum cant deficiency, without exceeding a total lateral force level per axle of 71 kN. [20]

The lateral force on the track is:

$Y=WA_d+A_yV_m$ [Mu / (Mu+My)] ½*[Ky+Mu] ½ ……………………………………………….3.34

Where

Y= lateral force per axle (N)

W= static axle load (N)

A_d= maximum normal operating cant deficiency angle (rad)

V_m= maximum normal operating speed (m/s)

M_u= effective lateral unsprung mass per axle (kg)

A_y= 0.0039 rad (angle of lateral ramp discontinuity)

M_y= 170 kg (effective lateral rail mass per wheel)

K_y= $25*10_6$N/m (effective lateral rail stiffness per wheel)

$W=56382$ N,=0.147rad,=120 $km/hr=33.33m/s$,

$M_u=1165kg$, $A_y=0.0039rad$, $M_y=170kg$, $K_y=25*10_6Nm$

$Y=WA_d+A_yV_m$ [Mu / (Mu+My)] ½*[Ky+Mu] ½…………………………….3.35

Y=8893.168 N

Therefore the lateral force by considering the above maximum parameters is 8.893 KN

The lateral force applied on rail depends on the following considerations: - the alignment quality, strength and dynamic characteristics of the rails and track and the speed, mass, suspension and dynamic stability characteristics of the vehicle. Maximum lateral forces are resulting from all speed and curvature combinations.

3.5.3-Longitudinal forces

Due to the tractive effort of the locomotive and its braking force, longitudinal lstresses are developed in the rail. Temperature variations, particularly in welded rails, result in thermal forces, which also lead to the development of stresses. The exact magnitude of longitudinal forces depends on many variable factors.

Source of longitudinal rail force are:

- ✓ Speed
- ✓ Locomotive traction
- ✓ Locomotive and car braking
- ✓ Expansion and contraction of rails from temperature change
- ✓ Track grade
- ✓ Special track that is turnout at grade crossings

Longitudinal defects include vertical split heads, horizontal split heads, head and web separations, split webs, and piped webs. Rail that is disturbed after being in one position for many years is prone to longitudinal defects due to altered stresses and changed loading patterns. In some cases, contour grinding aggravates dormant rail conditions. Rail welds - both factory and thermite process field welds - are also common. Frequently, rail weld defects occur due to inclusions or slag entrapments at the weld interface. In addition, longitudinal weld defects in the mid-web area may result from residual stresses. The longitudinal force shall be applied to the rails and superstructure as a uniformly distributed load over the length of the train in a horizontal plane at the top of rail. The horizontal force component transmitted to the rails and superstructure by an axle shall be concentrated at the rail having direct wheel-flange-to-rail-head contact. Due to vertical or lateral wheel loading bending stress developed. Vertical wheel loading results from loading between tie supports and causes tensile longitudinal stresses in the base and head/web fillet areas. Lateral wheel loading applies tensile longitudinal stresses in the web area and head/web area of the rail field side:

The longitudinal force is composed of two others force which are:

- ❖ The maximum braking force
- ❖ And the traction force

Let us first calculated separately and compare which one is greater than the other.

- ➢ The maximum braking force on the track is:

$Fbmax = \tau \times M \times g$

Where

Fbmax=maximum braking force (N)

τ =friction coefficient

M=mass (Kg)

g= earth gravity (m/s²)

τ =0, 3; M=6472, 5Kg; g=9,81m/s²

Fbmax=0, 3×6472, 5×9, 81

Fbmax=19048,57 N=19049 N

- The traction force on the track is:

$Ft = \mu \times Fn$

Where

Ft= traction force (N)

μ =adhesion coefficient

Fn=normal force (N)

And the adhesion coefficient μ in wet rail is:

$\mu = \frac{13{,}6}{\vartheta + 85}$ where ϑ is the maximum speed of the vehicle

$\mu = \frac{13{,}6}{120+85}$

μ =0,066

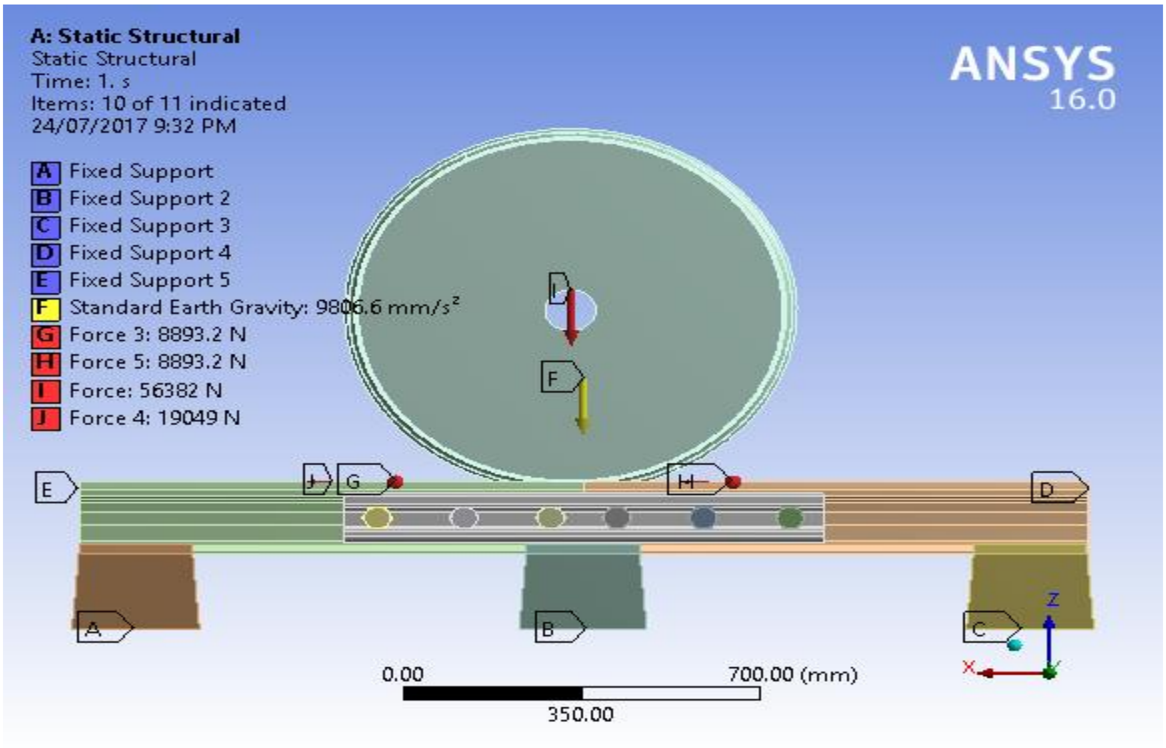
Ft=0,066×56382

Ft=3721, 12 N

Fbmax≫Ft, so the longitudinal force will be equal to the braking force.

3.6. BOUNDARY CONDITIONS

All the material properties and boundary conditions are being applied strictly as per the guidelines made available by National Railways in their manual. The wheel runs at constant speed of 120 km/hr. UIC 60 rail is used for analysis. The diameter of wheel is 840mm. The axle load is 56.832KN. Friction coefficient is 0.3. The material's density is 7800 kg/m3. Material properties of the rail and wheel are assumed to be same. The material properties of the wheel and rail are considered to be bilinear kinematic hardening in ANSYS. Fig. below shows boundary conditions and load conditions on bolted rail joint.



CHAPTER FOUR: RESULT AND DISCUSION

4.1 Result

The analysis is performed by using finite element model consist of static analysis and transient analysis to determine the impact of loads on bolted rail joint. Different bolted rail joint is used to perform finite element analyses, although for this analysis, the materiel of both are same but at different geometric of both to optimize the both of national railway.

4.1.1- Static analysis

A static structural analysis is determine the deflection, stresses, strains, safety factor, and of structures caused by loads that do not induces significant inertia and damping effects.

The load and the structure responses are assumed to vary slowly with respect to time that means steady loading and response condition are assumed.

Loads that are applied on rails and rail joints in a static analysis include:

- ✓ Vertical wheel load (force).
- ✓ Lateral load(by wheel flange)
- ✓ Longitudinal load

As stated earlier, this research focuses on the stresses in the joint. For all finite element models of the joint analyzed, and stresses are compared to determine the influence of bolt geometry on the joint performance.

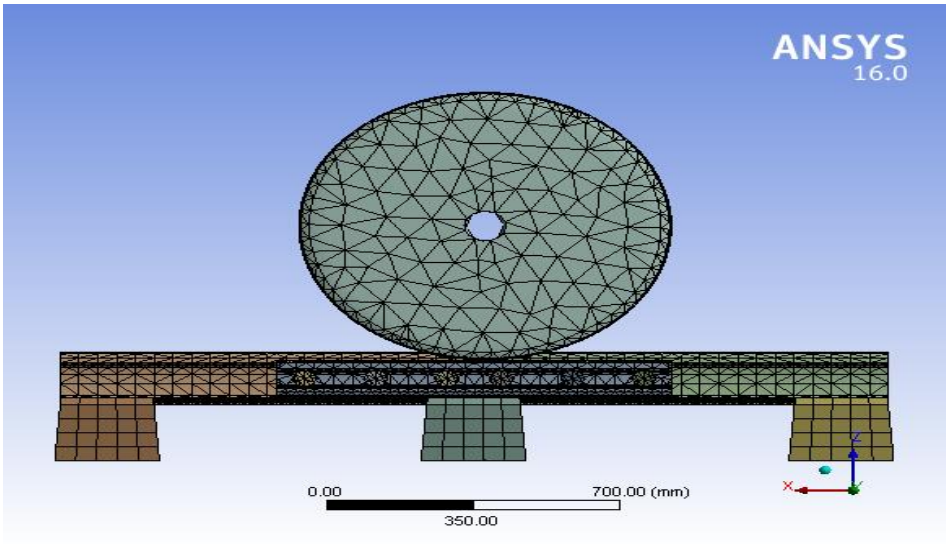


Figure 4.1-Mesh of assembled part on ANSYS workbench

- **Both geometric analyses**
- **Case 1:** When M30 bolt geometry is used.

A. Equivalent (von- mises) stress (MPa)

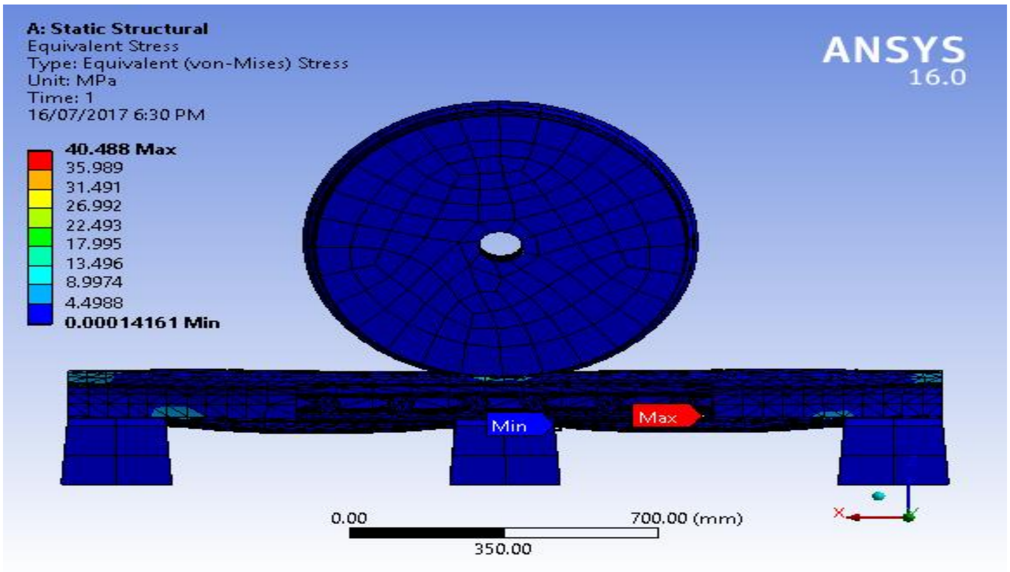


Figure 4.2: equivalent stress on the joint when M30 bolt is used.

From the ANSYS result when M30 bolt is used the equivalent stress is as shown in figure 4.2: the maximum stress is 40.488Mpa and the minimum is14.161e-5Mpa

B. shear stress

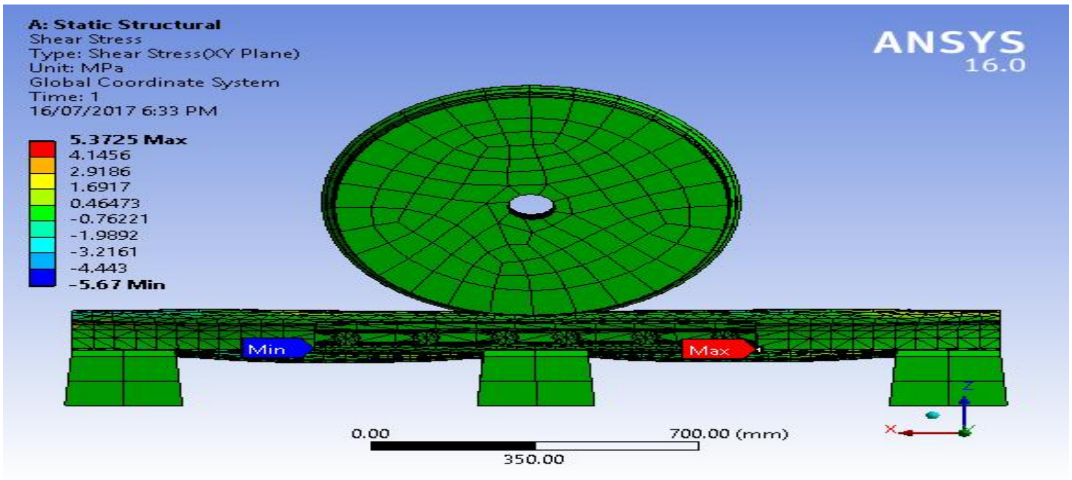


Figure 4.3: shear stress when M30 bolt is used.

And the shear stress is shown in the figure 4.3: the maximum shear stress is 5.3725Mpa and the minimum shear stress is-5.67Mpa.

C. Normal stress (bending stress) (MPa)

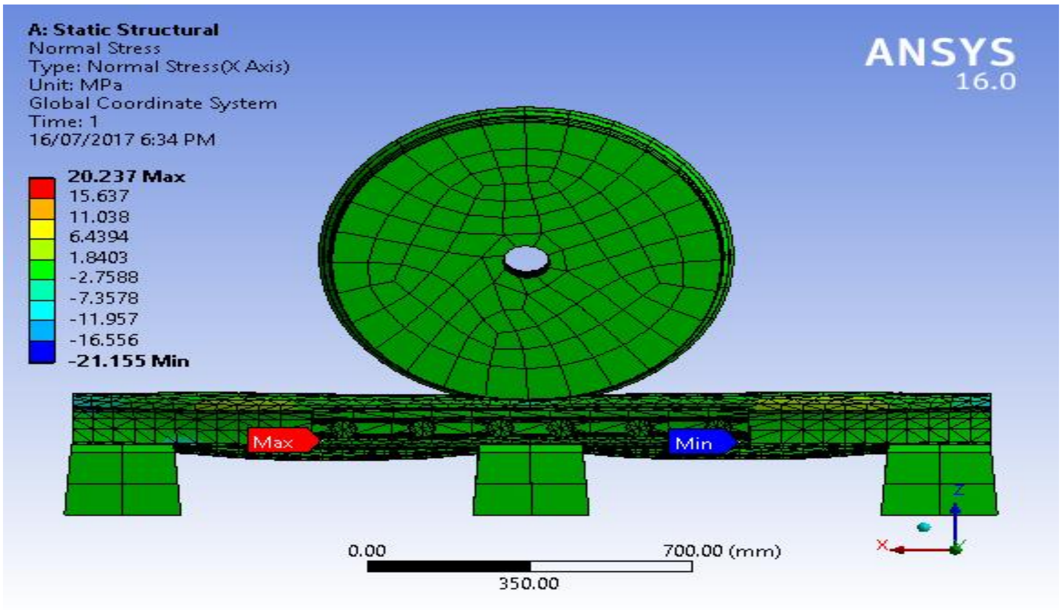


Figure 4.4: Normal stress when M30 bolt is used.

As shown in above figure 4.4: the maximum normal stress is 20.237MPa and the minimum normal stress is -21.155MPa.

D. Factor of safety

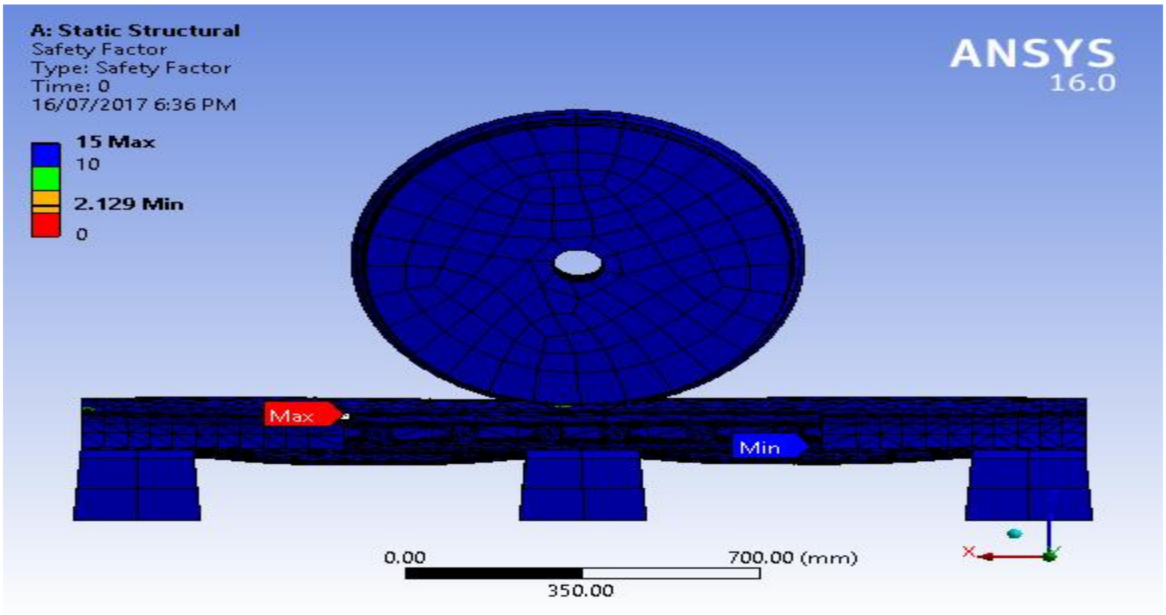


Figure 4.5: factor of safety when M22 bolt is used.

As shown in above figure 4.5, the maximum safety factor is 15 and the minimum safety factor is 2.129

E-Bolt stress

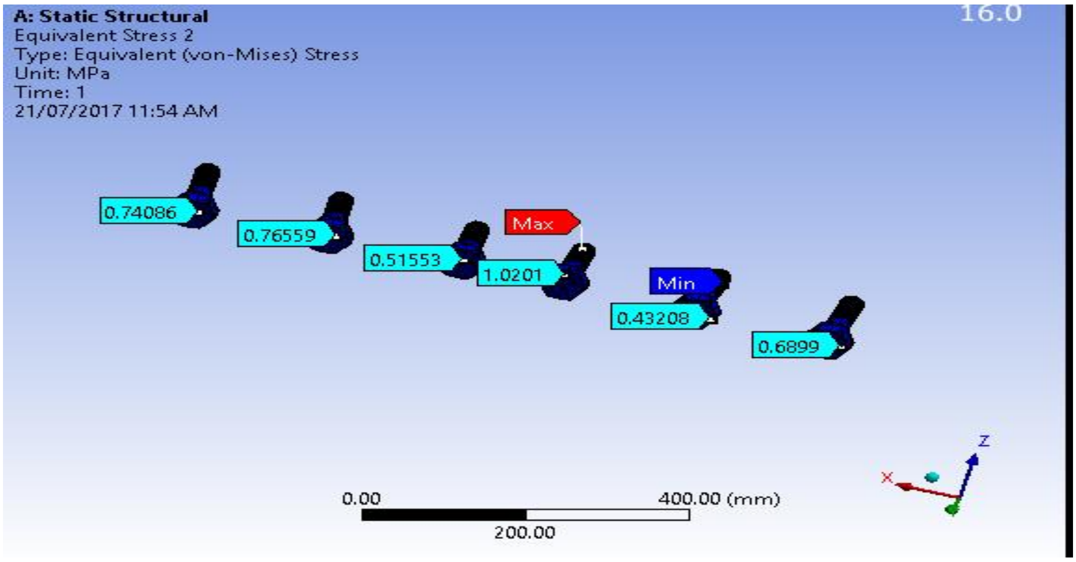


Figure 4.6: bolt stress When M30 bolt is used

The ANSYS result of the equivalent stress 2 is shown in the figure the maximum stress is 1.02Mpa and the minimum is 0.43208Mpa.

Case 2: When M32 bolt geometry is used.

A. Equivalent (von- mises) stress (MPa)

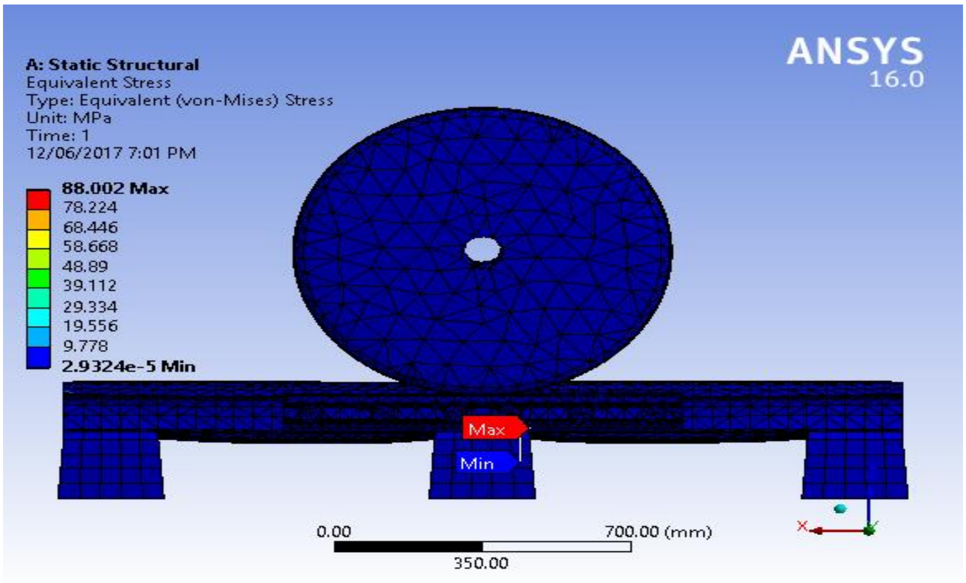


Figure: 4.7 equivalent stresses on the joint when M32 bolt is used.

When M32 bolt is used the ANSYS result of the equivalent stress is shown in the figure 4.7: the maximum stress is 88.002Mpa and the minimum is 29.324 pa.

B. Shear stress (Mpa)

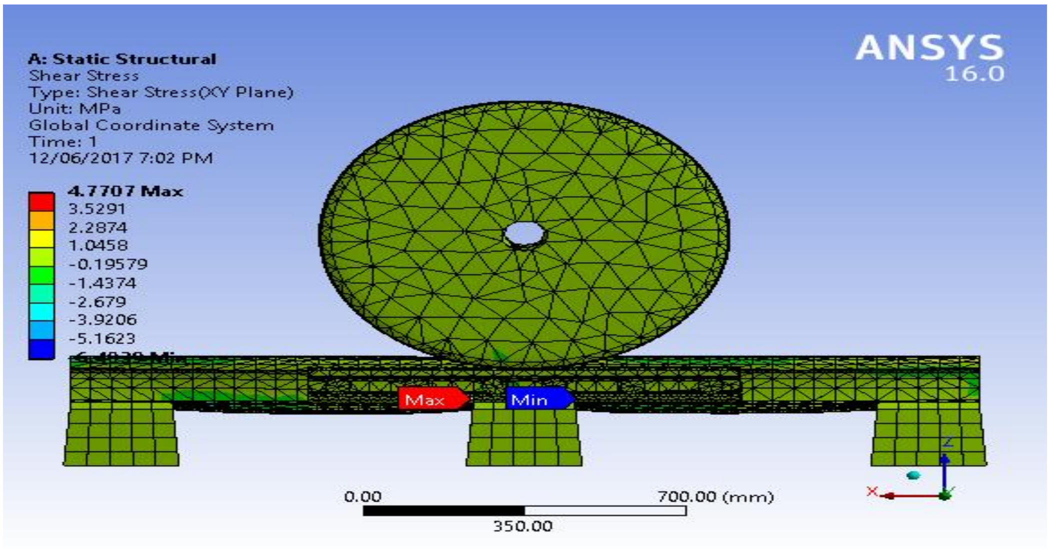


Figure 4.8: shear stress when M32 bolt is used.

And the shear stress is shown in the figure 4.8: the maximum shear stress is4.7707Mpa, and the minimum shear stress is-6.4039Mpa

C. Normal stress (bending stress) (MPa)

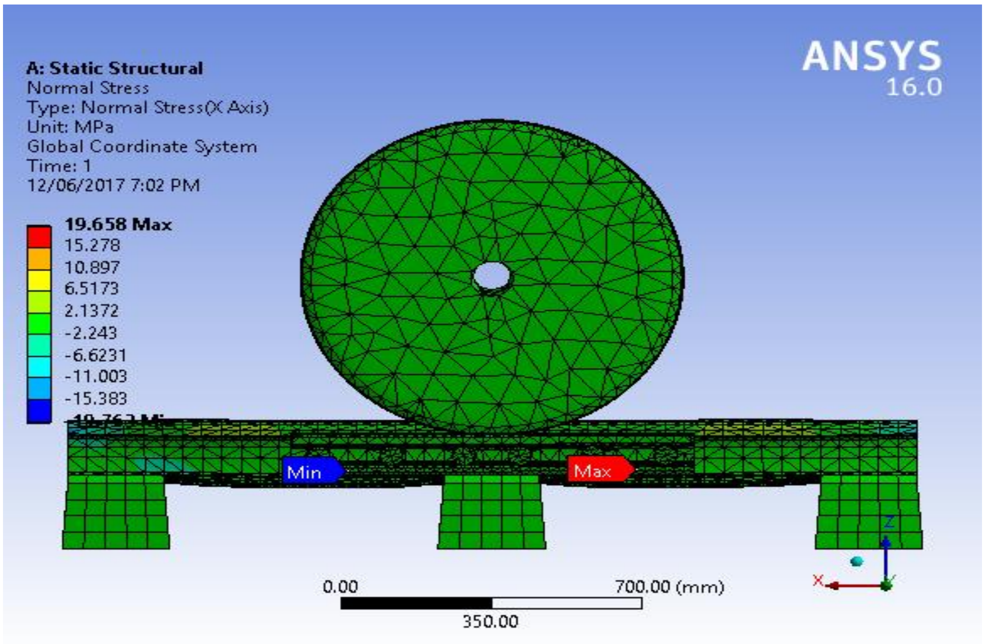


Figure 4.9: normal stress when M32 bolt is used.

As shown in above figure 4.9: the maximum normal stress is19.658MPa and the minimum Normal stress is-19.763MPa.

D. Safety factor

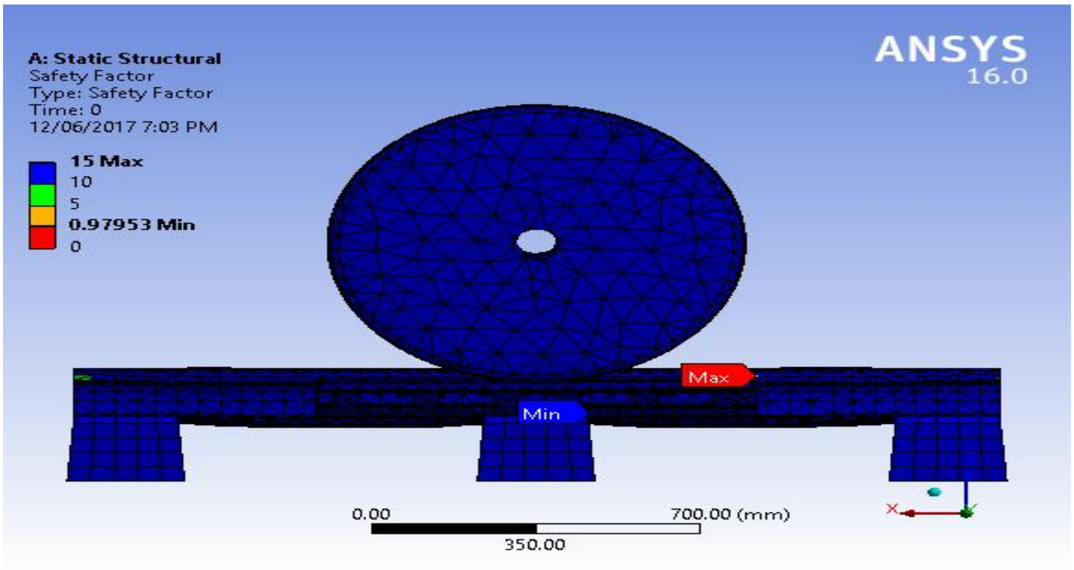


Figure 4.10: safety factor when M32 bolt is used.

As shown in above figure 4.10: the maximum safety factor is 15and the minimum safety factor is 0.97953.

E-Bolt stress used M32 bolts used

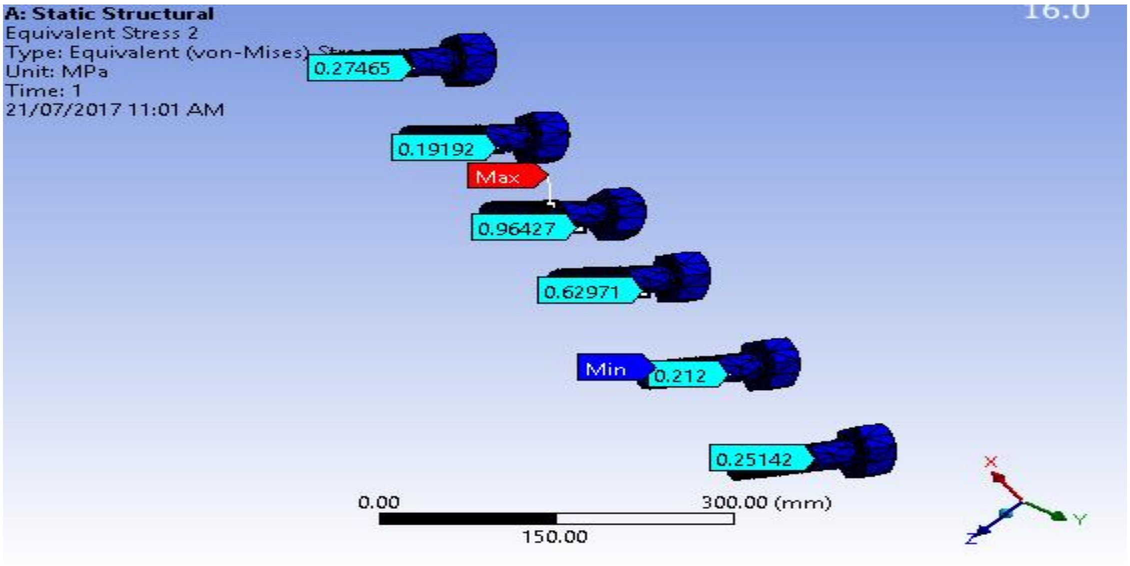


Figure 4.11: bolt stress When M30 bolt is used

The ANSYS result of the equivalent stress 2 is shown in the figure the maximum stress is 0.96471 Mpa and the minimum is 0.212 Mpa.

Case 3: When M34 bolt geometry is used.

A. Equivalent (von- mises) stress (MPa)

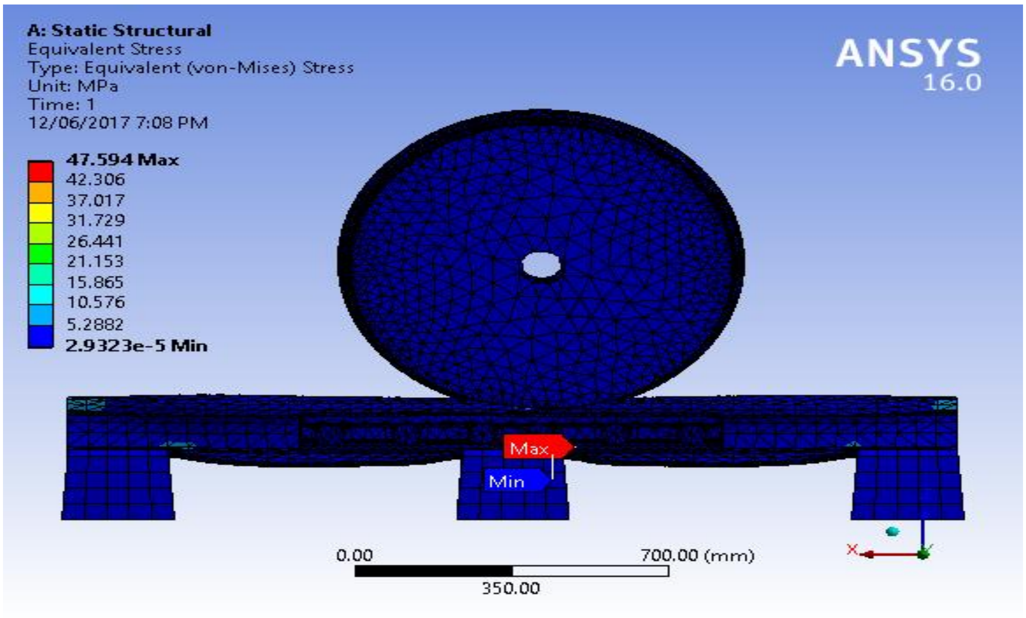


Figure 4.12: Equivalent stress on the joint when M34 bolt is used

When M34 bolt is used the ANSYS result of the stress is shown in the figure 4.12: the maximum equivalent stress is47.594Mpa and the minimum is29.323pa.

B. Shear stress

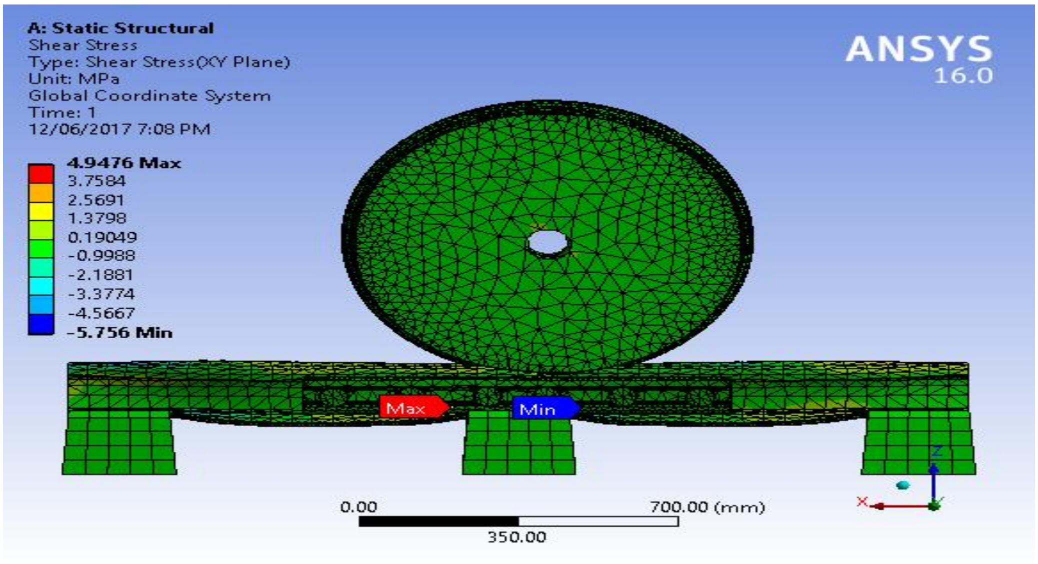


Figure 4.13: shear stress when M34bolt is used.

And the shear stress is shown in the figure 4.13: the maximum shear stress is4.9476Mpa, and the minimum shear stress is-5.756Mpa.

C. Normal stress (bending stress) (MPa)

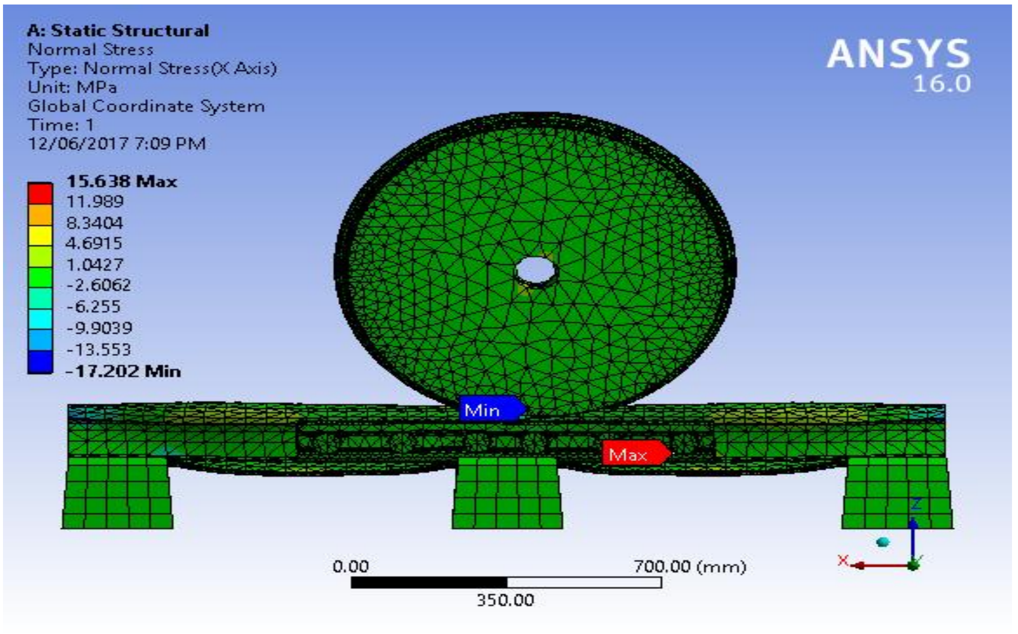


Figure 4.14: Normal stress when M34 bolt is used.

As shown in above figure 4.14: the maximum normal stress is15.638MPa and the minimum Normal stress is compressive stress 17.202Mpa.

D. Safety factor

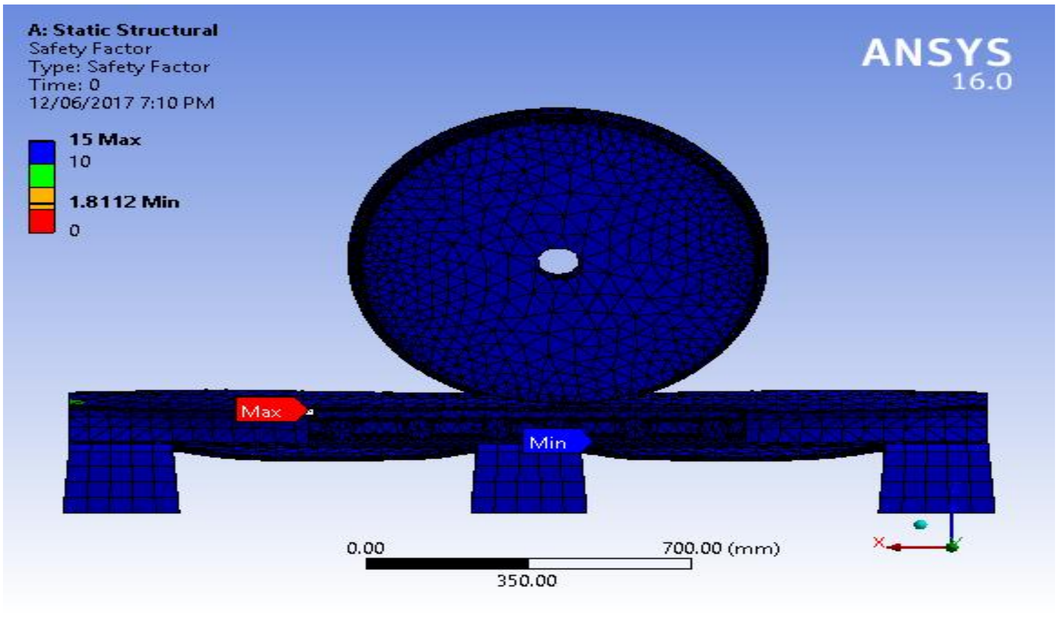


Figure 4.15: safety factor when M34 bolt is used.

As shown in above figure 4.15 the maximum safety factor is15and the minimum safety factor is1.8112.

E-bolt stress when M34 bolt used

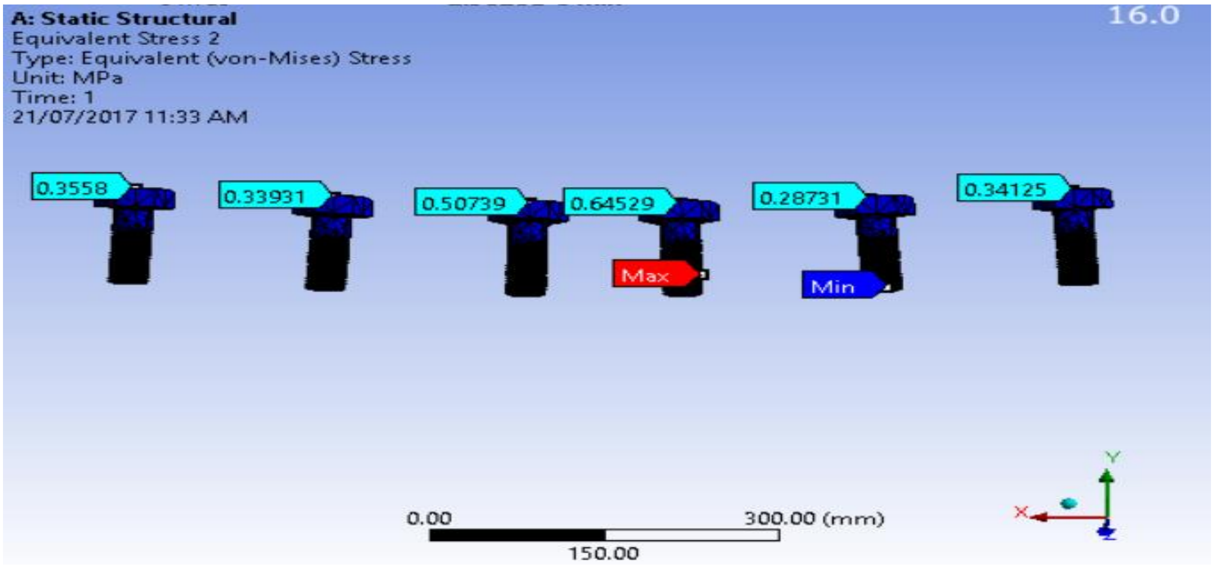


Figure 4.16: bolt stress when M34bolt used

As shown in above figure below: the maximum normal stress is 0.64529 MPa and the minimum Normal stress is compressive stress 0.28731 Mpa.

4.1.2-Transient analysis

Transient dynamic analysis is a technique used to determine the dynamic response of a structure under the action of any general time-dependent loads. You can use this type of analysis to determine the time-varying displacements, strains, stresses, and forces in a structure as it responds to any combination of static, transient, and harmonic loads. The time scale of the load ingissuch that the inertia or damping effects are considered to be important. This paper is used transient analysis to determine the dynamic impact on the rail joint by varying the applied load with very small time. So decelerations of the train is 2m/s2and speed of the train is 120km/h equivalent at 33.33 m/s

Tmax=v/a=33.33/2

Tmax=16.66 s

V=speed of the train

a=deceleration of the train

t max=time maximal

1.**Case 1:**When M30 bolt geometry is used.

A. Transient von mises stress

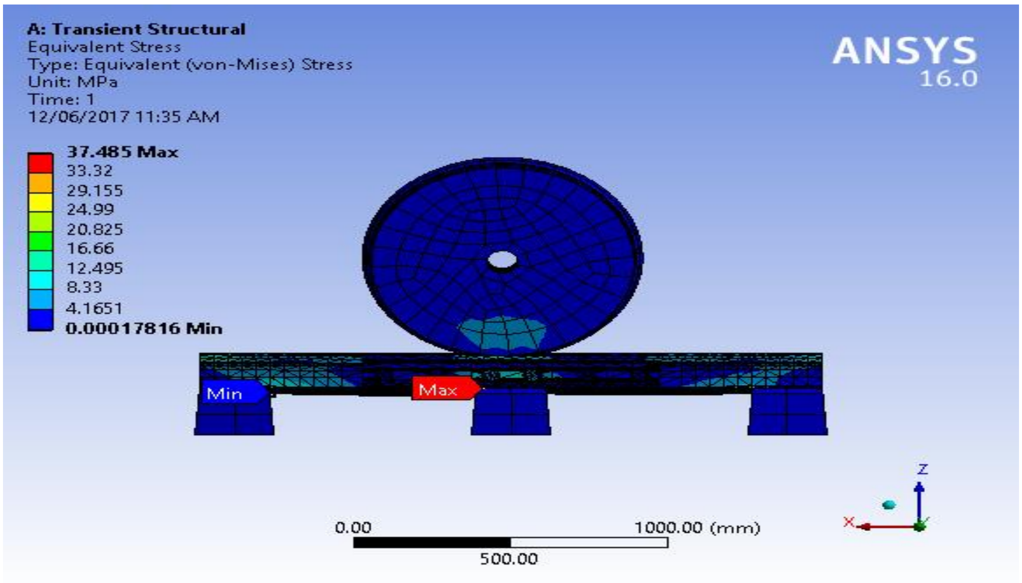


Figure 4.17: Transient von mises stress when M30 bolt geometry.

As shown in above figure, the maximum von-Mises stress is 37.485MPa and the minimum von –mises stress is 17.816e-5MPa

B. Transient shear stress

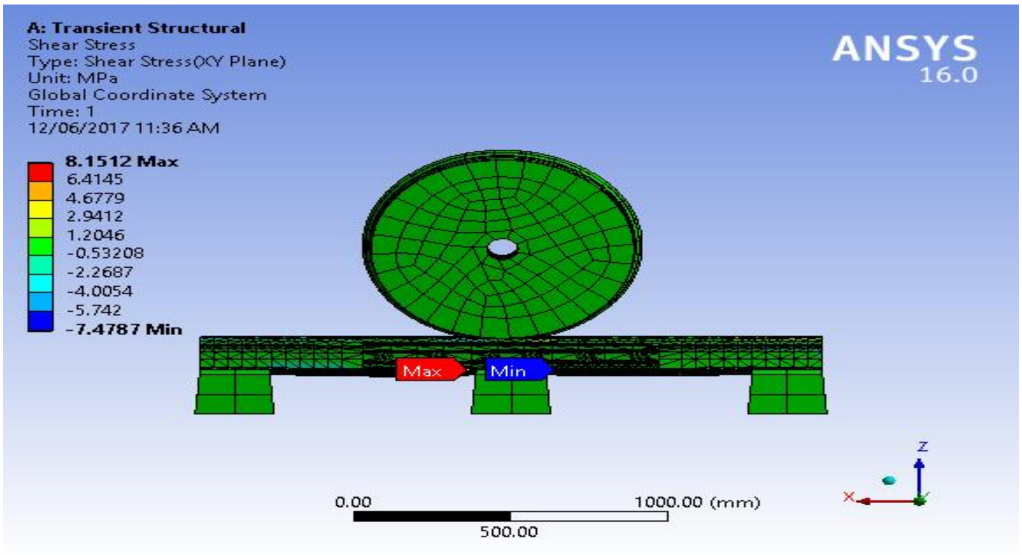


Figure 4.18: Transient shear stress when M30 bolt geometry.

As shown in above figure, the maximum shears stress is8.1512MPa and the minimum shear stress is compressive 7.4787MPa.

C. Transient Normal stress

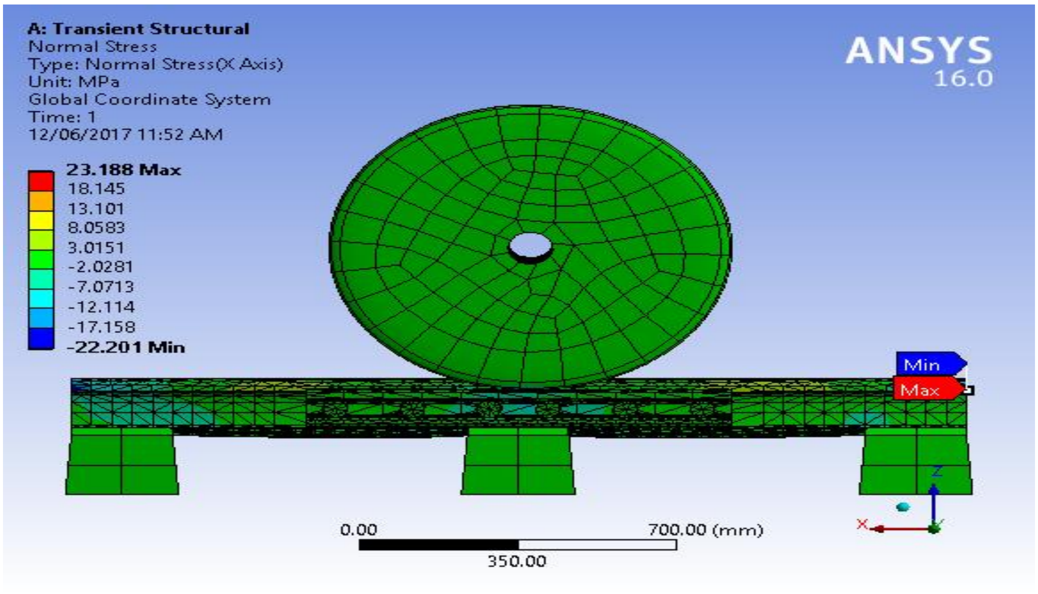


Figure 4.19: Transient Normal stress when M30 bolt geometry.

As shown in above figure, the maximum normal stress is 23.188MPa and the minimum normal stress is compressive 22.201MPa.

2. **Case 2:** When M32 bolt geometry is used.

A. Transient von mises stress

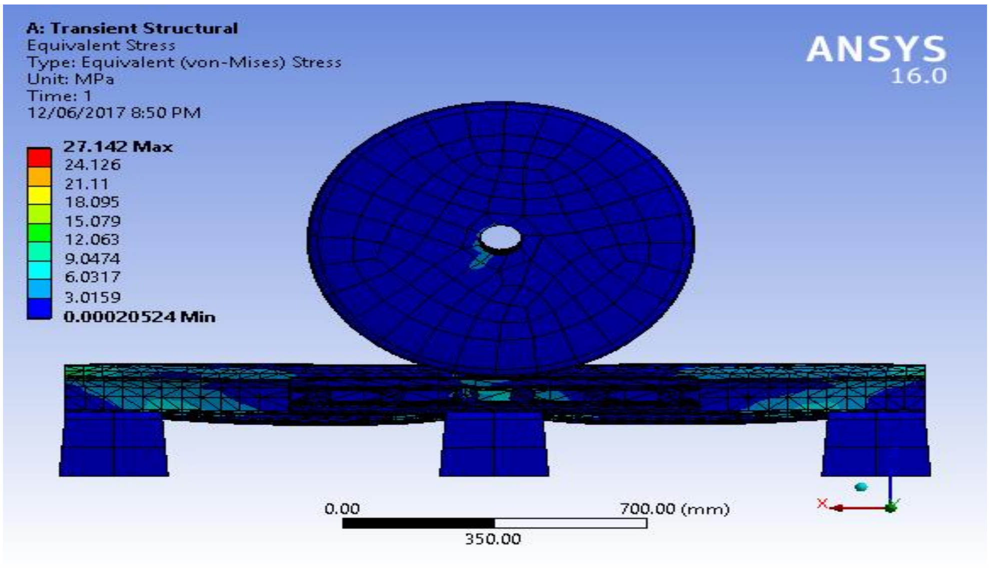


Figure 4.20: Transient von mises stress when M32 bolt geometry.

As shown in above figure, the maximum von-Mises stress is 27.142MPa and the minimum von –mises stress is 205.24Pa

B. Transient shear stress

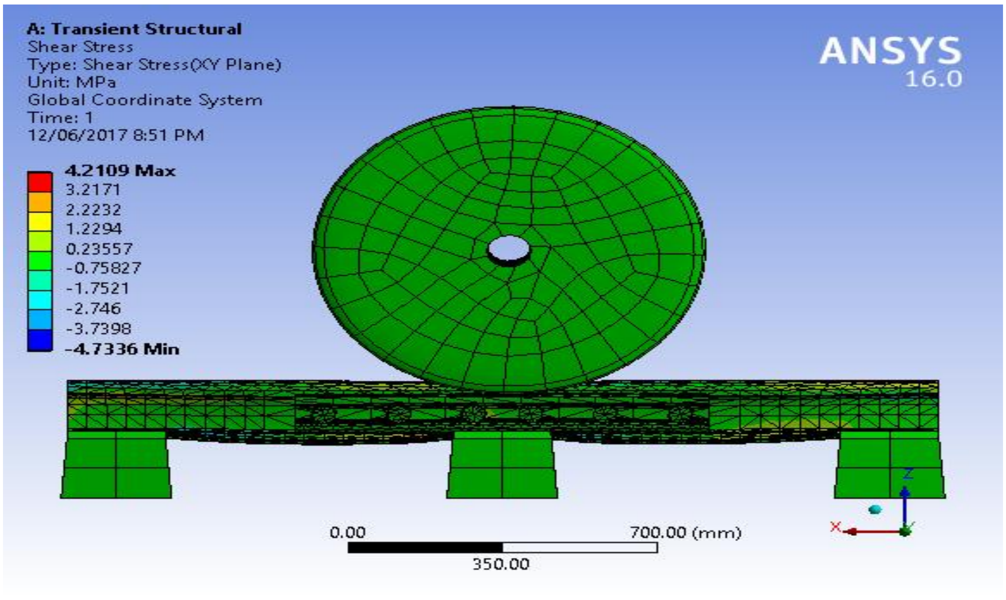


Figure 4.21: Transient shear stress when M32 bolt geometry**.**

As shown in above figure, the maximum shear stress is4.2109MPa and the minimum shear stress is compressive stress 4.7336MPa.

C. Transient Normal stress

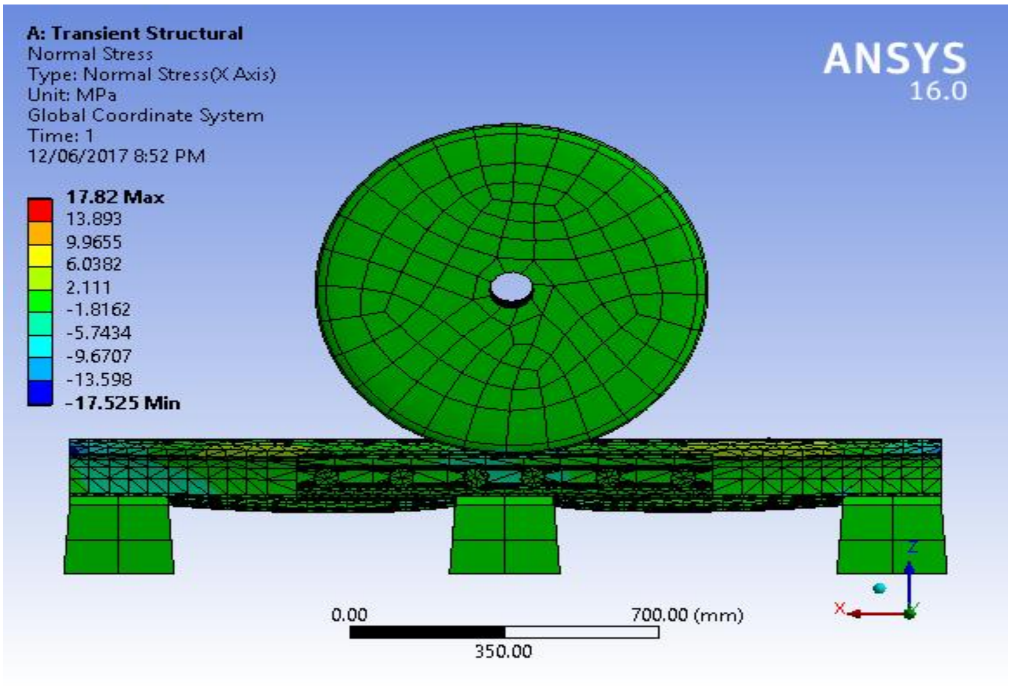


Figure 4.22: Transient Normal stress when M32 bolt geometry.

As shown in above figure, the maximum Normal stress is17.82MPa and the minimum Normal stress is compressive 17.525MPa

3.**Case 3:**When M34 bolt geometry is used.

A. Transient von mises stress

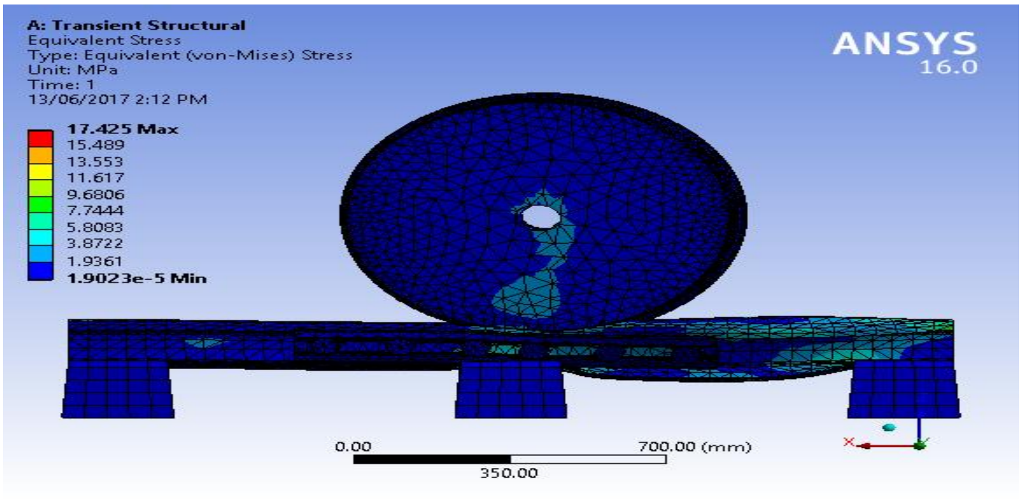


Figure 4.23: Transient von mises stress when M34 bolt geometry.

As shown in above figure, the maximum von-Mises stress is 17.425 MPa and the minimum von –mises stress is 1.9e-5Pa

B. Transient shear stress

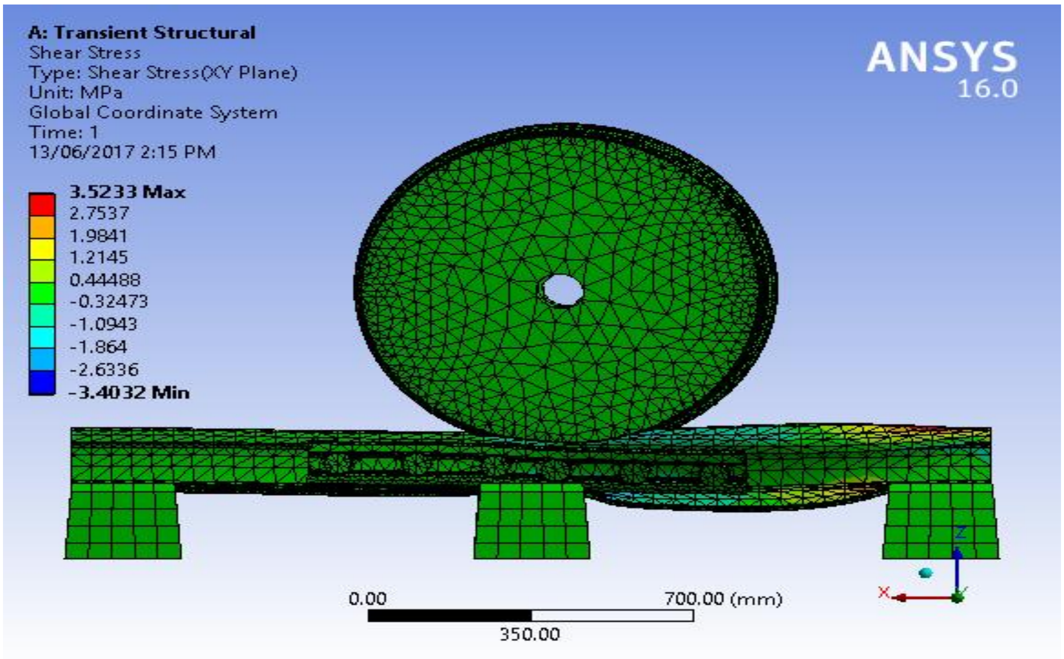


Figure 4.24: Transient shear stress when M34 bolt geometry.

As shown in above figure, the maximum shear stress is 3.5233MPa and the minimum shear stress is compressive stress 3.40336 MPa.

C. Transient normal stress

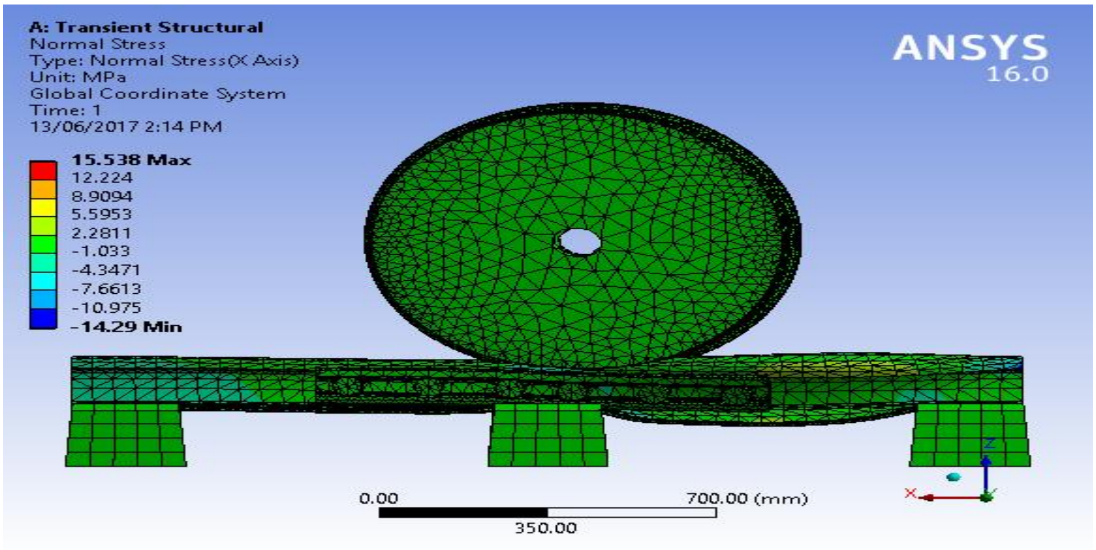


Figure 4.25: Transient normal stress when M34 bolt geometry.

As shown in above figure, the maximum shear normal is 15.538 MPa and the minimum normal stress is compressive 14.29MPa.

4.2 Discussion

This section of the paper specifies the result obtained from the ANSYS software based on hertz contact theory. The above result shows different stress types and failure criteria due to longitudinal and lateral applied load at the bolted rail joint. There are three different bolt geometric at same materiel used on National Railway of rail joints compared each other like this.

Table 4.1: Static and dynamic result summary

Analysis Type	Type of Load			M30	M32	M34
Staticanalysis	Von mises stress(MPa)		Max.	49.523	88.002	47.594
			Min.	29.38e-5	29.32e-5	29.32e-5
	Shear stress(MPa)		Max.	4.9845	4.9407	4.7776
			Min.	-6.0675	-6.4039	-5.756
	Normal stress(MPa)		Max.	14.98	19.658	15.638
			Min.	-150549	-19.763	-17.202
Dynamics analysis	Transient analysis	Transient von mises stress(MPa)	Max	27.142	35.395	17.425
			Min.	205.2e-5	203e-5	190e-5
		Transient shear stress(MPa)	Max	4.21	3.805	3.5233
			Min.	-4.7336	-3.778	-3.4032
		Transient normal stress (MPa)	Max	17.82	16.87	15.538
			Min.	-17.525	-16.414	-14.29

Generally summuzied the table must result two type analysis, three type of stress and safety factor, the above results are included both rail joint and wheel for the reason of the analysis performed using the contact mechanism. For static analysis, As shown from ANSYS result the value of the stress is varying for the three types of bolts. In the M30 bolt as shown in the figure 4.2, figure 4.3, and figure 4.4 the maximum von mises, shear, and normal stress are 49.523MPa, 4.9845MPa and 14.98Mpa. In the M32bolt as shown in the figures 4.6, 4.7, and 4.8 the maximum von mises, and shear stresses are 88.002MPa, 4.7707MPa, and 19.658Mpa. In the M34bolt as shown in the figures 4.10, 4.11, and 4.12 the maximum von mises and shear, normal stresses are 47.594MPa, 4.9476MPa, and 15.638Mpa all are around the bottom of the plate and nearest to the end head of the rail.

For Dynamic analysis, As shown from ANSYS result the value of the stress is varying for the three types of bolts. In the M30 bolt as shown in the figure 4.14, figure 4.15, and figure 4.16the maximum von mises, shear, and normal stress are 27.142MPa, 4.21MPa and 17.82Mpa. In the M32bolt as shown in the figures 4.17, 4.18, and 4.19 the maximum von mises, and shear stresses are 35.395MPa, 3.805MPa, and 16.87Mpa. In the M34bolt as shown in the figures 4.20, 4.21, and 4.22 the maximum von mises and shear, normal stresses are 17.425MPa, 3.5233MPa, and 15.538Mpa.

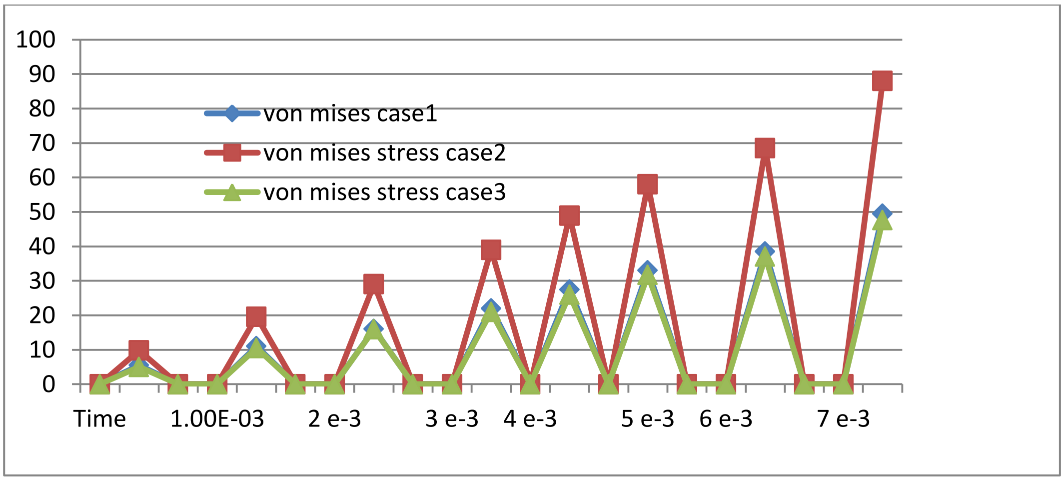


Figure 4.26: Von mises stress versus time graph for static analysis

As shown in above figure, Von-Mises stress verses to time are given for three different geometric of bolted rail joint. The result shows that the von mises stress on rail joint increases with time. The von mises stress increases for three location of rail joint. However, when compare three of them, Geometric M32 stress distribution is highest then the other remaining geometric of bolted rail joint

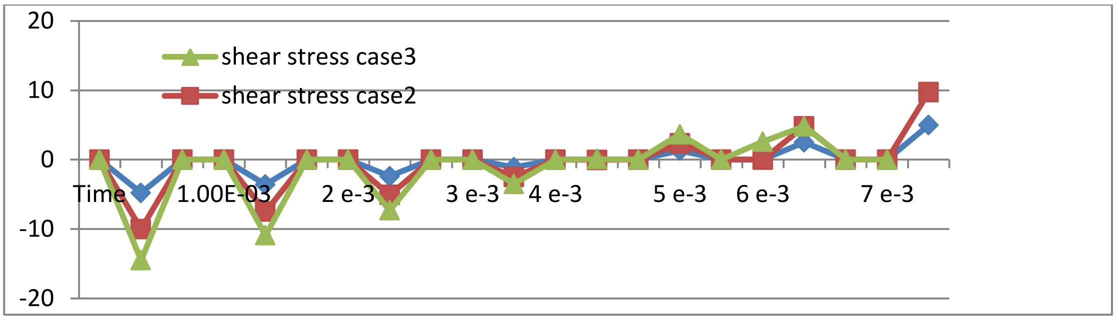


Figure 4.27: shear stress versus time graph for static analysis

The result shows that shear stress on rail joint increases with time. The von mises stress increases for three location of rail joint. However, when compare three of them, Geometric M32 stress distribution is highest then the other remaining geometric of bolted

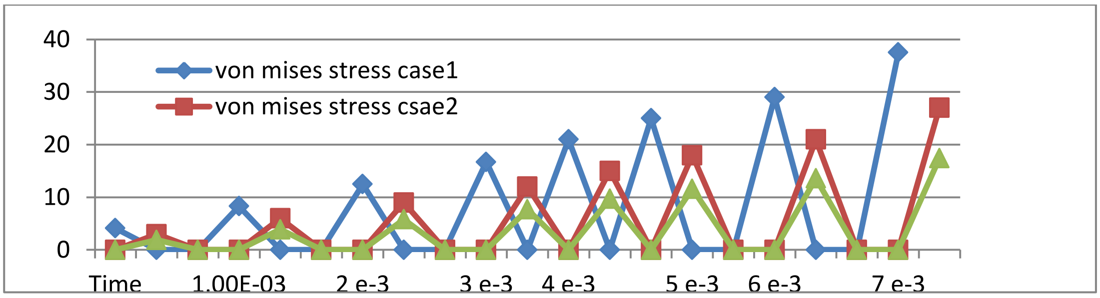


Figure 4.28: Von mises stress versus time graph for dynamic analysis

The result shows that the von mises stress on rail joint increases with time. The von mises stress increases for three location of rail joint. However, when compare three of them, Geometric M30 stress distribution is highest then the other remaining geometric of bolted rail joint.

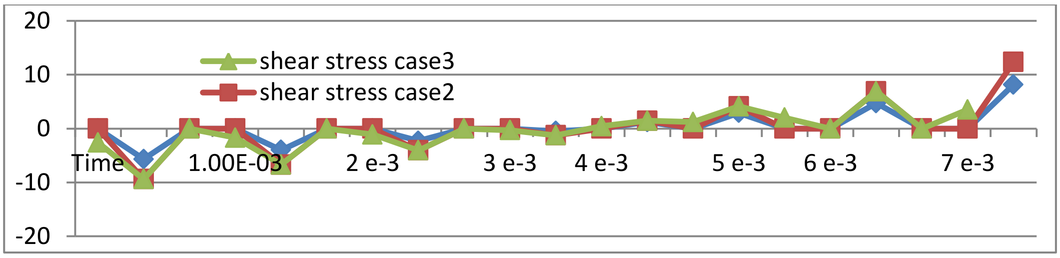


Figure 4.29: shear stress versus time graph for dynamic analysis

The result shows that the von mises stress on rail joint increases with time. The von mises stress increases for three location of rail joint. However, when compare three of them, Geometric M32 stress distribution is highest then the other remaining geometric of bolted rail joint.

Chapter Five: Conclusion, recommendation and future Works

5.1. Conclusion

A three-dimensional finite element model is used to analysis the bolt geometry at bolted rail joint section of track. The finite element program ANSYS is used to model the contact analysis. This ANSYS is used to simulate the loading and boundary conditions of the rail and wheel contact for a stress analysis. From ANSYS software simulation static and dynamic result when M32 bolt geometric, the rail joint highly exposed to the equivalent, normal, shear stresses than the rest. Regarding the stress distribution, when M34 bolt geometric is small in comparison for other bolt geometric. For this reason, the service life of the rail joint of M32 bolt is small as compared to the remaining rail joint location. When M34 bolt geometric; the stress on the rail joint is small in comparison to M30 and M32 bolt geometric. So M34 bolt geometric will be better to use by national railway than M32 bolt geometric.

5.2. Recommendations

Rail joint, one part of rail track, needed more attention than other parts, to reduce the problem related to the element. From this research it was determined that by increasing the bolt diameter, bolt geometricM34 have the minimum stress than other. The geometry of the bolt has seen to have an effect on stress of rail and joint it is better to use geometry M34 for national railway. The bottom of the plate, and the rail head are in high stress these parts should have strengthen to decrease fatigue and wear rate.

5.3. Future works

In this thesis work effects of bolt geometry is observed on the stress of rail joint by finite element method using ANSYS software. Experimental study on the bolt geometry is also necessary to identify the stress more. The joint part of the rail track needs more attention to eliminate problem related to the wheel/ rail contact like fracture of rail and joint bar hole, looseness of bolt, nut, and dislocation and distorted of the rail joints. To minimize the rail and joint wear, the looseness of bolt and nut, and safety problem issues and maintenance cost of the railway track. The following points may be studied further in the rail joint analysis and also see more on following area:

- ✓ Identifying crack initiation and crack probation at the bolts.
- ✓ Fatigue life of bolted rail joint
- ✓ Developed strong material for bolt and nut and rail hole

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