



Addis Ababa University
Addis Ababa Institute of Technology
African Railway Centre of Excellence

**INVESTIGATING THE EFFECT OF SLEEPER SPACING AND
MATERIAL ON THE MECHANICS OF WHEEL RAIL DRY
CONTACT**

BY

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Degree of Masters of Science in Railway Engineering (Rolling Stock)

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Declaration

I, Joan Tondo do declare that this thesis research is my work except that which has been referenced. It has never been submitted for any prior academic award or qualification.

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Approval

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Dedication

To God who has given me good life and health, to my mother for all your love and prayers, to my mother-in-law for the love and for caring for your granddaughter all this time, to my colleagues who have shared their knowledge and words of encouragement, to all of my relatives who have made this journey possible. To the love of my life Davis for always loving, trusting and supporting me and believing in me, to my daughter Stephanie for the patience and those innocent little words of encouragement.

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Abstract

In this study, the effect of sleeper spacing, and sleeper materials on wheel rail contact mechanics was investigated. The selected sleeper materials were; concrete, timber (F34) grade, structural steel, and the selected values of sleeper spacing were; 500 mm, 600 mm (the current Addis Ababa light rail transit spacing) and 700 mm to analyse the mechanics (strain, stress, force and pressure, deformation and sliding distance). Wheel rail contact at both straight and curved track taking train operating speed of 40km/hr was modelled using Finite element (FEM). First, a 3-dimension model consisting of a wheel set on railway track supported by sleepers was developed in solid works environment and then imported to Ansys workbench.

By taking 600 mm spacing as a benchmark, using 500 mm spacing in the curved track with the three material models results in higher values of contact pressure and stress in addition to the higher contact forces on the right side. However, lower values of equivalent elastic strain, total deformation and sliding distance in addition to lower values of contact forces on the left side are observed. Additionally, using 700 mm spacing results in lower contact stresses, equivalent elastic strain and contact pressure in addition to higher values of total deformation and sliding distance. Furthermore, the contact forces are higher on the left side and lower on the right side with the 700 mm spacing. Additionally, for the case of straight track and the three materials, the use of 500 mm yields higher values of contact stress, elastic strain, contact forces and pressure but lower values of total deformation and sliding distance. However, with 700 mm spacing, the reverse was true.

By taking concrete sleeper material as a bench mark, using timber (F34) sleeper material lower contact stress, strain, pressure and force but larger deformation and sliding distance and the reverse is true when the structural steel sleeper material is used. However, it is important to note that the effect of the sleeper material on contact mechanics is relatively small. According to the study, it was concluded that the most suitable sleeper spacing and material were 600 mm and concrete respectively. The results of the study are useful in improving wheel rail interaction by reducing on wear volume of wheels and track hence increasing their service life, and vibrations, squeal noise, derailment, and cracks among others by ensuring lower levels of contact stress, strain, pressure, forces, total deformation and sliding distance.

Key-words: Wheel-rail interface, FEM, Sleeper spacing, Sleeper Modulus, Global track stiffness.

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Nomenclature

AALRT-Addis Ababa Light Rail Transit

BCs- Boundary Conditions

FEM: Finite Element Methods

W/r- wheel-rail

MoE-Modulus of elasticity

CoF- Coefficient of friction

CHAPTER ONE: INTRODUCTION

1.1 Background

Railway transport is believed to have started in Corinth, Greece way back in the 6th century BC. It has gone through various developments until now; from the ancient man/animal-hauled railways, horse-powered funiculars, wagon ways, steam locomotives to the diesel and electric locomotives and trains, and the train speeds have improved progressively from low speeds to high speeds with some of the high speed trains being French TGV (April 2007) and the Japanese maglev trains like the MLX01 (December 2003) and the L0 series maglev (21 April 2015). These trains have been tested to run at 574.8 km/hr, 581 km/hr and 603 km/hr respectively [1], [2].

The growth and advancement of railway transport has placed it in a better competitive position with road, air and water transport because of noteworthy advantages; lesser energy consumption, environmental preservation, higher security, large hauling capacity, and comfort. These and other advantages have caused the world's inclination towards railway transportation to escalate. Nevertheless, in railway transportation, the issues of wear, derailment, corrugation, noise, vibration, rolling contact fatigue etc. have continuously been the cause of the high maintenance budgets as a result of damaged rolling stock components especially wheels and track structures. These issues are believed to originate from the wheel rail interaction and for this reason, several researchers have done and are still doing studies on the factors that directly or indirectly influence the wheel rail interaction so as to obtain better solutions to the above issues with the aim of achieving a reliable, efficient and cost effective railway transportation sector [3]–[9].

It is noticeable that the track has direct effect on the wheel rail interaction but since the track components (sleepers, fastening system, ballast, sub ballast and foundation) generally affect and influence its performance [10]–[12], we cannot deny the fact that the track components too have an effect on the wheel rail contact. Moreover, most of the researchers have studied track stiffness from a global point of view which implies that track stiffness is the summation of the stiffness of each track component [13]. This can be a reason to conclude that the material, mechanical and structural properties of the components of the track have an influence on the wheel rail interaction.

The sleepers which are part of the superstructure in addition to the rails and the fastening system play very vital roles which include; dissemination of wheel loads from the trains down to the

substructure of the track (ballast, sub-ballast, embankment fill, subsoil) and finally to the ground, maintaining rails to gauge and inclination, transmission of forces, insulate rails electrically, and provide a base for rail seats and fastenings [14]. Since sleepers play very essential roles in the railway system, it is important to study how certain parameters affect their performance down to the wheel rail interaction. Moreover, researchers have done work on effect of sleeper modulus and spacing on wheel rail interaction though a lot of work has been done on their effect on vibration [15], track performance [16], their own performance and less on the wheel rail interaction [17]. Also Lei Xu and colleague conducted a numerical study to determine how various finite element types of sleepers affect wheel rail interaction [18] while others like Roman et al decided to investigate how the design and shape of the sleepers affects train track interaction [19].

This study focused on two parameters; sleeper material and spacing. Different values of sleeper spacing and sleeper material were used to determine how they affect wheel rail interaction in terms of contact forces, stresses, pressure, strain, total deformation and sliding distance. First a model comprising a wheel set, track and two sleepers was developed in solid works environment and imported to Ansys workbench in IGES format and simulations were done under dry conditions in both straight and curved track for the different materials; timber (F34), concrete and steel the different sleeper spacing: 500 mm, 600 mm and 700 mm to obtain the results of wheel rail contact stresses, pressure, stain, forces, deformation. As stated earlier the results from the study can be an input in maintenance and railway system improvement for railway operators in the process of revamping or maintaining their system like Uganda Railway Corporation and others.

1.2 Problem Statement

The fact that railway transportation has several advantages does not rule out the presence of technical problems. It is noted that most of these problems arise due to the poor wheel rail interaction. Moreover, the small areas where the wheels make contact with the rails are believed to govern the rail vehicles' performance. This highly-stressed interface is the foundation of almost all research areas of interest in vehicle-track interaction from dynamic performance of the vehicle to maintenance of the infrastructure. Therefore, a good understanding of the contact phenomenon between wheel and rail is fundamental to designing an enhanced railway system of transportation. The problems arising from uncontrolled wheel-rail interaction include; wheel rail wear which if not controlled largely leads to the reduced service life of wheels and rails and big maintenance budgets, noise and vibration which greatly

impact the ride quality, high risk of derailment leading to compromise of the safety of passengers and the railway infrastructure. These problems are basically caused by the uncontrolled forces and stresses induced in the wheel rail contact by several parameters. It is important to know how each and every component of the track or rolling stock affects the wheel rail contact and on this note, the effect of sleeper spacing and materials (MoE) on deformation, contact stresses, strains, pressure and forces.

1.3 Research Question

- i. How does sleeper spacing and materials affect deformation, contact stresses, strain, pressure, and sliding distance in straight track?
- ii. How does sleeper spacing and materials affect deformation, contact stresses, strain, pressure, forces and sliding distance in curved track?
- iii. What are the most suitable sleeper spacing and material for good wheel rail contact?

1.4 Objectives

To investigate the effects of sleeper spacing and materials (MoE) on wheel-rail interaction in particular the induced contact stresses, pressure, strain, forces, deformation and sliding distance for both straight and curved sections of track.

1.4.1 Specific Objectives

- To perform a numerical analysis while varying sleeper spacing and material (MoE) to investigate their effects on contact stresses, pressure, strain, and forces, total deformation and sliding distance for straight track
- To perform a numerical analysis while varying sleeper spacing and material (MoE) to investigate their effects on contact stresses, pressure, strain, forces, deformation and sliding distance for curved track.
- To suggest the most suitable sleeper spacing and material for good wheel rail contact.

1.5 Significance of the Study

So many researchers have done research on the effects of the vehicle and track parameters on wheel rail interaction. Moreover, most of the work has been done on the vehicle parameters and the track and its components in other aspects like vibration, track stiffness, noise, etc. However less research still exists on investigating the contribution of sleeper material (MoE) and spacing on the wheel rail contact stresses, strain, pressure, forces, total deformation and sliding distance. In this way, the results from this study can be an input in making improvements in wheel rail interaction and its related effects on wear, vibrations, squeal noise,

derailment, corrugation, and cracks among others made while basing on a bigger picture of combined parameters and not only on one parameter. A profound and broad understanding of the mechanism of vehicle–track interaction is needed in order to implement realistic tactics to lessen the notorious wheel–rail interaction, to obtain best integrated designs of up-to-date rolling stock and track structures, and ultimately to guarantee safe, smooth, and efficient train operations. Also the results from this study can be used as input for railway organisations that are in the process of refurbishing and maintaining their railway system like Uganda Railway Corporation and others.

1.6 Scope and limitations of the study

This study was performed to investigate the influence of sleeper spacing, and material (MoE) on wheel rail contact in terms of deformation contact stresses, pressure, forces and strain for both straight and curved sections of the track. A static structural analysis was performed using Ansys to investigate the effects of sleeper material and spacing on contact stress, pressure, force, strain, deformation and sliding distance. This study also looked at various literatures on the stated parameters and how they influence wheel –rail interaction related issues. The model used in the study comprised one wheel set, a 1320 mm long section of straight and curved track, and two sleepers. However, the substructure of the track (ballast, subballast and soil) was not considered in the study and the sleepers were assumed to be a fixed support. This implies that a very high stiffness of the subgrade was assumed but it is not always the case. This assumption certainly resulted in higher contact stresses than would be if a less stiff subgrade was considered because the subgrade stiffness has a big effect on the track stiffness.

1.7 Organisation of the thesis

The thesis comprises of five basic chapters; with chapter one comprising of the background of the study, objectives, scope and limitations, chapter two comprising of reviewed work from journal articles and thesis work, chapter three comprising of the methodology used to obtain the study objectives with general procedures used to during the FEA, chapter four consists of the results and discussion and finally chapter five in which conclusions, recommendations and future scope of study are presented. There is also a section for references.

CHAPTER TWO: LITERATURE REVIEW

2.0 Introduction

The demand for railway transport has been on the rise at present typically due to its major advantages which include; lesser energy intake, ability to transport a large number of passengers and very heavy goods at a time, higher security and environment conservancy. The issue of the rail and wheel interaction is one of the most essential aspects in the railway system, and has always been taken into thought for many years by many researchers.

2.1 Wheel rail interaction

Many factors influence the interaction between wheel and rail: the mechanical properties of track and vehicle components, the conditions in the wheel-rail interface, track alignment, curves, axle load and train speed. A lot of research has been done on factors that influence wheel rail interaction;

Tamás et al [20] analysed the contact pressure distribution, the contact area and the sliding procedure in a railway wheel-rail contact by using finite element method and discovered that principal shear stress reaches its maximum under the wheel tread which is conformable with the Hertz theory and it was observed that the contact pressure distribution is about symmetric. They also concluded that the sticking zone in the wheel rail contact region is larger in the wheels' stationary position and changes to full sliding with the wheel displacement. Their main aim was to understand the wheel rail contact phenomena under stationary, sticking and sliding scenarios by applying a frictionless contact for stationary scenario and sliding contact for the sliding scenario.

Buddhe analysed the influence of the wheel material on the contact stresses by comparing UIC-60 and IRS-T12 rail-wheels using analytical method and FEM- Ansys. The results showed that contact stresses are indeed influenced by the wheel material because the stresses from the contact with IRS-T12 rail-wheel were less than UIC-60 rail-wheel and he made his conclusions in terms of material properties of the wheels [21]. Srivastava et al [22] and Ayana [1] studied the influence of wheel and rail profile topology on distribution of contact zones, contact stress and contact pressure by using the analytical formulation based on Timoshenko's approach and Hertzian contact theory respectively and FEM. The results showed that the stresses decrease with increase in profile radii and contact area decreases with increase in wheel taper. Aleksander et al [5] also did the same study but used the quasi- Hertz method in addition to FEM to define the contact zones and stresses for different wheel set angles of attack.

Haile [23] in his thesis, studied the wheel-rail contact stresses for a straight track, transition track and curved track using static structural analysis in Ansys. The results showed that the contact stresses were highest in the curved track, followed by the transition track and the lowest were observed in the straight track. This implies that the track alignment has an effect on the wheel rail contact stresses. He further studied the how the coefficient of friction influences contact stresses. It was observed from the results that the contact stresses are highest with high coefficient of friction.

Mutswatiwa et al. [24] and his colleagues investigated the influence of variation in the railway track modulus on wheel rail interaction using elasto-plastic explicit dynamic Finite Elements (FE). The variation in railway track modulus was expressed by assigning different moduli of elasticity to the sleepers. It was observed that high discrepancy of the track modulus can cause increase in the contact forces, stresses and contact area.

Sowndarya et al. [25] also used finite element analysis i.e ANSYS to perform the contact analysis by considering the loading and boundary conditions of the rail and wheel contact for a stress analysis. In their study, a 3-D finite element model comprising a wheel and rail rolling contact was developed on the most critical section of rail track to analyse the peak contact pressure using the phenomena of surface contact analysis module. Static Structural analysis and transient structural analysis were both carried out and results obtained.

Mohammad et al. [26] modelled and analysed a railway system containing wheel, rail, axle, and pads using ANSYS software. They used elastic-plastic materials and mapped meshing dependent upon the current international railway systems which enabled high precision of the results. The simulation results acquired were in agreement with the actual life experiences and their study further contributed to the demonstration of important steps for obtaining more representative 3D solutions to the related problems.

The dynamic behaviour of a rail vehicle running through a curve is majorly affected by wheel-rail contact forces, which are influenced by the adherence, the slipping and wear [27], [28]. Wang et al. [29] notes that changes in alignment and geometric size in the curved track can exacerbate the wheel-rail interaction, increase vibration, and disturb the running safety and comfort. On the small-radius curved track, this phenomenon would be more serious. Also matsumuto et al. [30] illustrated the effect of curve radius on the derailment coefficient and concluded that the coefficients increase with a corresponding increase in curvature as well as the friction of the inside wheel/rail contact surface, which is influenced by wayside lubricant

application. Additionally, Martínez-Casas et al. [31] emphasises that the train–track interaction is more notorious when the rail vehicle is passing through a curve and hence it is important to study the factors that affect the curving phenomena.

From the research already done, it is noted that factors like speed, curve radius, turnouts, track stiffness with little on the contribution of sleeper MoE and sleeper spacing on wheel rail contact has been done. Therefore this study will increase and provide deeper understanding of effects of sleeper spacing and materials on contact stresses, pressure, forces and strain and. The research on sleeper spacing so far is basically on its effect on phenomena that occur in wheel rail interaction like vibration, noise, wear but how it affects the forces, stresses, contact pressure has not been given much attention and yet it is very important information for revamping and maintenance of a railway network.

In curves, the outer rail is longer than the inner rail. In a curve, as the wheelset moves laterally towards the outside, the rolling radii of the outer and inner wheels will increase and decrease respectively because of the conical shape of the wheels. The outer wheel with a larger circumference is forced to move further in relation to the inner wheel for common rotational speed. If the wheel set travels amply far laterally to the outside of the curve, the difference in the rolling radius will be sufficient to cover up for the variance in rail lengths causing the wheel set to acquire "pure rolling" [32]–[34].

However, the wheel set is not always in a perfectly radial position in a curve but it is sometimes in an under-radial position. Wheel set yaw relative to the track as shown in figure 2.1 is called "angle of attack" (ψ).

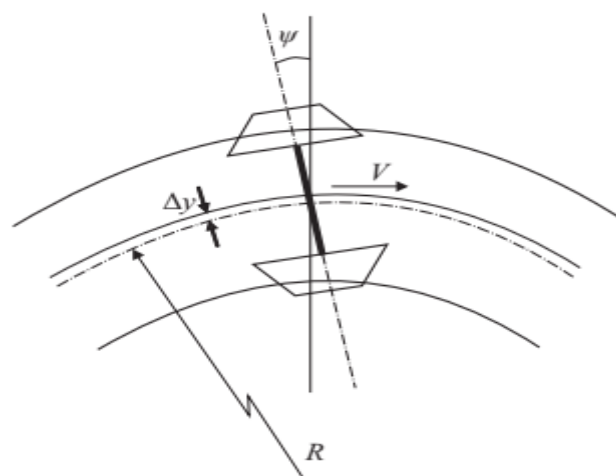


Figure 2. 1; curving of a wheel set [32]

When the wheels' contact surface moves relative to the rail, sliding motions termed as creepage transpire. “Creepage is defined as the quotient of sliding velocity (v) and vehicle speed (V)” and comprises; longitudinal, lateral and spin creep.

$$\text{Longitudinal creep} = u_x = \frac{v_x}{V}$$

$$\text{Lateral creep} = u_y = \frac{v_y}{V}$$

$$\text{Spin creep} = \varphi = \frac{w}{V}$$

These creepages result in creep forces (friction forces) on the wheel and rail. The yaw moment should be greater than the repelling forces of the primary suspension in order for the wheelset to steer radially in a curve. This can be achieved by creating a more 'soft' primary suspension although it gives a poor stability on curved track at higher speeds [32].

2.1.1 Contact theories

From the start, the wheel-rail contact has been of crucial interest to railway engineers which is evidenced from the vast research fields which involve wheel-rail contact. Nevertheless, the most fundamental step in detailed investigation of mechanisms and phenomena emanating from wheel rail interaction is to identify the contact patch shape and size and the stress distribution in it. The study of rolling-sliding contact, is said to have begun in 1926 with the Carter's study on a two-dimensional rolling contact problem which was expounded to a 3D rolling contact based on the sliding contact theories by Johnson about thirty years later. Kalker suggested a theory for solving the problem numerically although it is limited by elastic half-space assumption [3]. The Hertz contact theory published by Heinrich Hertz in 1882, is the most famously applied contact theory in solving many engineering problems [35]. The contact theories are explained in detail below;

a. Carter's theory

The first published rolling contact theory is by Carter in 1926. At about the same time Fromm published his thesis on creep analysis of rolling elastic discs where he proposes a similar theory. Carter's paper was on running quality of electric locomotives and its goal is to relate creepage and creep force of the driving locomotive wheels while putting emphasis on tractive and braking forces of the driving wheels of the locomotive. His work was based on earlier studies by Reynolds on belt drives.

b. Kalker's linear theory

During the period of 1957-1972, Kalker developed the Linear Theory which was based on the work originated by De Pater in 1956. The linear theory uses the well-known Kalker coefficients C_{11} , C_{22} , C_{23} and C_{33} , to determine the relation between creepages, creep forces and moment. Then, the creep forces and spin moment can be expressed in the matrix form as follows [36];

$$\begin{bmatrix} F_x \\ F_y \\ M_z \end{bmatrix} = -G_{ab} \begin{bmatrix} C_{11} & 0 & 0 \\ 0 & C_{22} & \sqrt{ab}C_{23} \\ 0 & -\sqrt{ab}C_{23} & abC_{33} \end{bmatrix} \begin{bmatrix} \xi_x \\ \xi_y \\ \varphi \end{bmatrix} \quad (2.1)$$

c. Hertz contact theory

This was developed by a twenty three year old graduate student called Hertz in 1880 during his Christmas vacation. Hertz's model of contact stress is based on the following assumptions:

- The materials in contact are homogeneous and the yield stress is not exceeded
- The effect of surface roughness is negligible
- Contact stress is caused by the load which is normal to the contact tangent plane which effectively means that there are no tangential forces acting between the solids
- The contact area is very small compared with the dimensions of the contacting solids

Matin [37] introduced the coordinate system during contact and Hertz's contact in figures shown below:

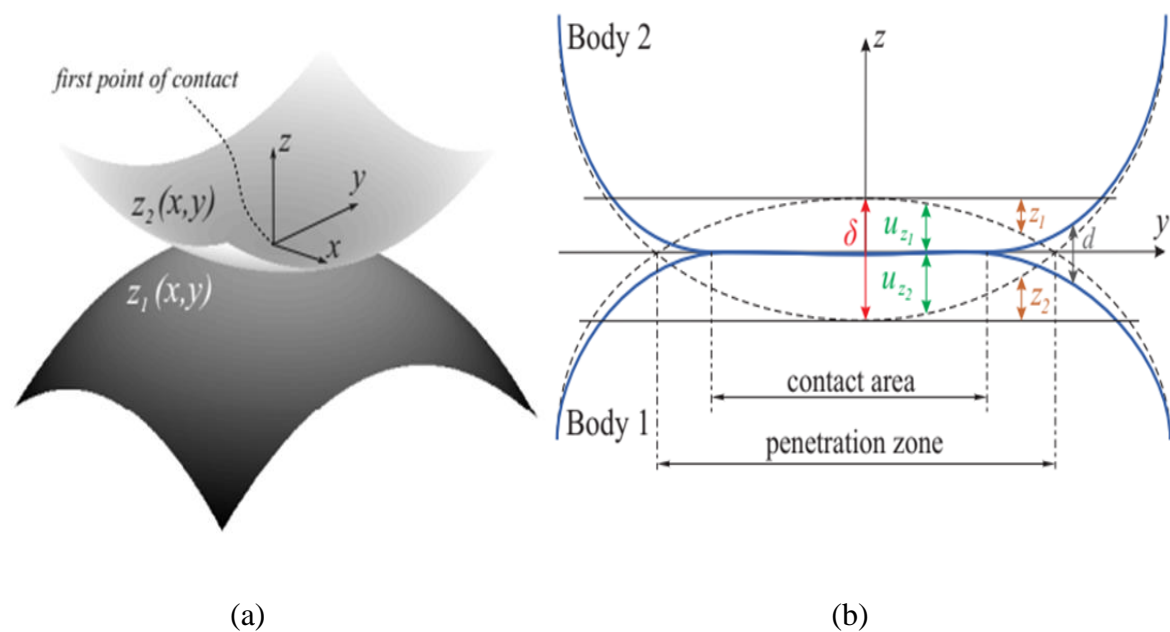


Figure 2. 2; (a) defined coordinate system and (b) contact terminology [37]

While basing on the assumptions above, Hertz suggested the solution for the determination of elliptical contact area and pressure distribution between two bodies in contact. As shown in figure 11 (b), the surfaces of the bodies in contact before loading are represented by $z_1(x, y)$ and $z_2(x, y)$ for the first and the second body, respectively. The distance between each contacting surface is given by:

$$Z_1 = A_1x^2 + B_1y^2 \quad (2. 2)$$

$$Z_2 = A_2x^2 + B_2y^2 \quad (2. 3)$$

Where: x and y are the coordinates of the point inside the contact area in the XY - plane.

The distance between the un-deformed surfaces (before loading) is called separation;

$$z = |z_1| + |z_2|$$

$$z(x, y) = Ax^2 + By^2 \quad (2. 4)$$

Assuming each body has two principal radii; with one in the y - z plane (R_1^r, R_2^r) and another in the x - z plane (R_1^t, R_2^t) as shown in the figure below;

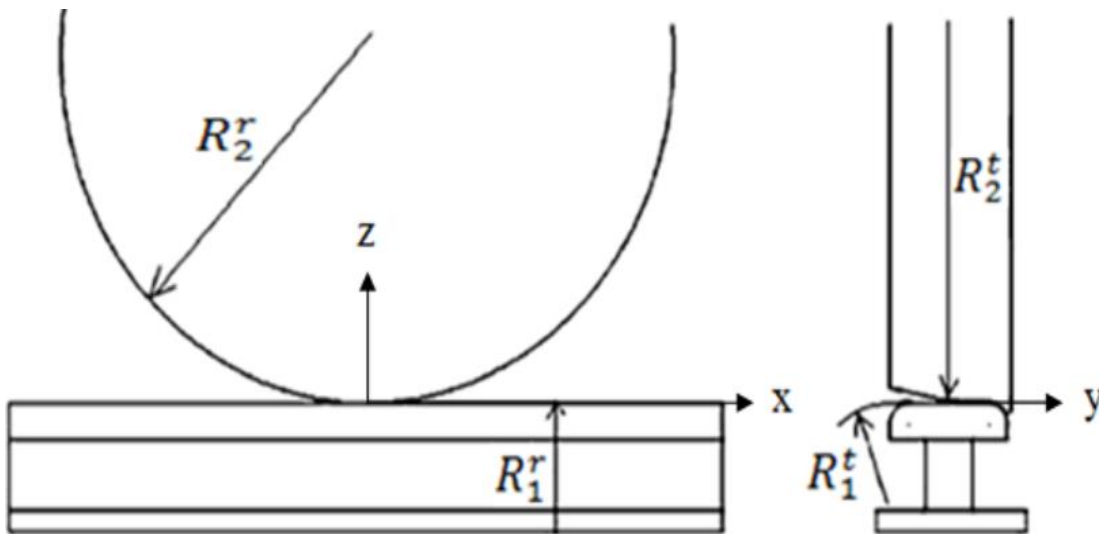


Figure 2. 3; Wheel-rail outline displaying different principal radii of curvature

Where,

R_1^r - Rolling radius of the rail

R_2^r - Rolling radius of wheel

R_1^t - Transverse radius of curvature of rail

R_2^t - Transverse radius of curvature of the wheel

A and B are constants depending on the magnitude of the principal curvatures of the contact surfaces and on the angle, ‘ θ ’ between the planes of the principal curvatures of the two surfaces and ψ the angle between the normal planes of the radii of curvatures i.e. x_1 and x_2 axis are inclined to each other by an angle ψ , hence the constants A and B are determined from the following equations;

$$(A + B) = \frac{1}{2} \left[\frac{1}{R_1^r} + \frac{1}{R_1^t} + \frac{1}{R_2^r} + \frac{1}{R_2^t} \right] \quad (2.5)$$

$$(B - A) = \frac{1}{2} \left[\left[\frac{1}{R_1^r} - \frac{1}{R_1^t} \right]^2 + \left[\frac{1}{R_2^r} - \frac{1}{R_2^t} \right]^2 + 2 \left[\frac{1}{R_1^r} - \frac{1}{R_1^t} \right] \left[\frac{1}{R_2^r} - \frac{1}{R_2^t} \right] \cos 2\psi \right]^{\frac{1}{2}} \quad (2.6)$$

$$\cos \theta = \frac{|B-A|}{A+B} \quad (2.7)$$

The normally loaded bodies are shown in Figure 2.2 (b). The centres of the bodies have to be constrained together by the distance, d , known as an approach. The elastic deformation of the contact surfaces of the bodies, $u_{z_1}(x, y)$ and $u_{z_2}(x, y)$ may be added up to form, $u_z = |u_{z_1}| + |u_{z_2}|$. The distance between the deformed surfaces is expressed by the equation below [1], [38];

$$d(x, y) = z(x, y) - \delta + u_z(x, y) \quad (2.8)$$

Where δ is the total deflection at the centre of the contact.

2.1.2 Wheel-rail contact zones

During the rolling and/or sliding of the wheels on the rails there are three possible contact zones as shown in figure 2.12 below [1].

- Region A: Wheel tread-rail head –usually occurs when the railway vehicle is treading a straight track or one with a very high radius curve and is characterised by low contact stresses and pressure.
- Region B: Wheel flange–rail gauge corner – majorly occurs in curved tracks and is characterised by a smaller contact area, higher contact stresses and pressure with higher wear rates compared to region A. When extreme wear and material flow happen, two-point contacts may develop, where tread and flange contact seem to occur.
- Region C: Contact is not likely to take place here and, in case it does, it results in high contact stresses, leading to unwanted wear features causing inappropriate steering of the wheelset.

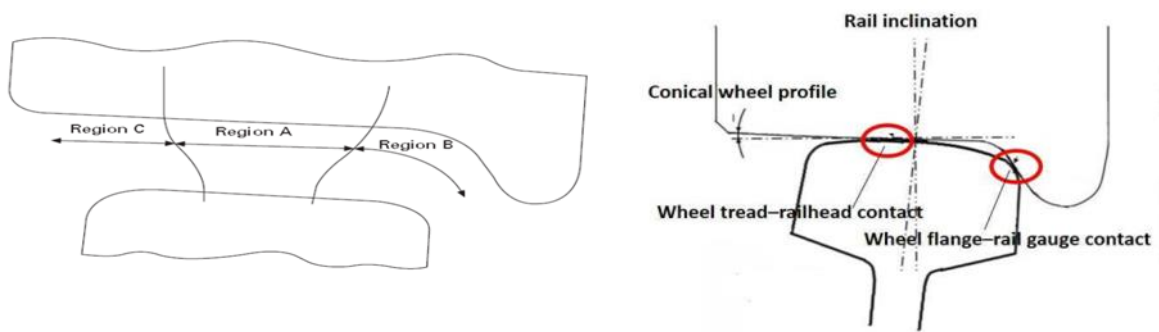


Figure 2. 4; wheel rail contact zones [1]

Furthermore, the contact region can be divided into stick and slip zones. During absence of slip in the contact region, the whole contact region sticks with no tangential force transmission hence resulting into free rolling motion. Slip takes place at the trailing edge and spreads throughout the contact region with increase in adhesion force. As the sticking zone decrease, the slip zone increases causing a rolling- sliding contact until pure sliding occurs during which adhesion force equals the frictional force between the wheel and rail under sliding.

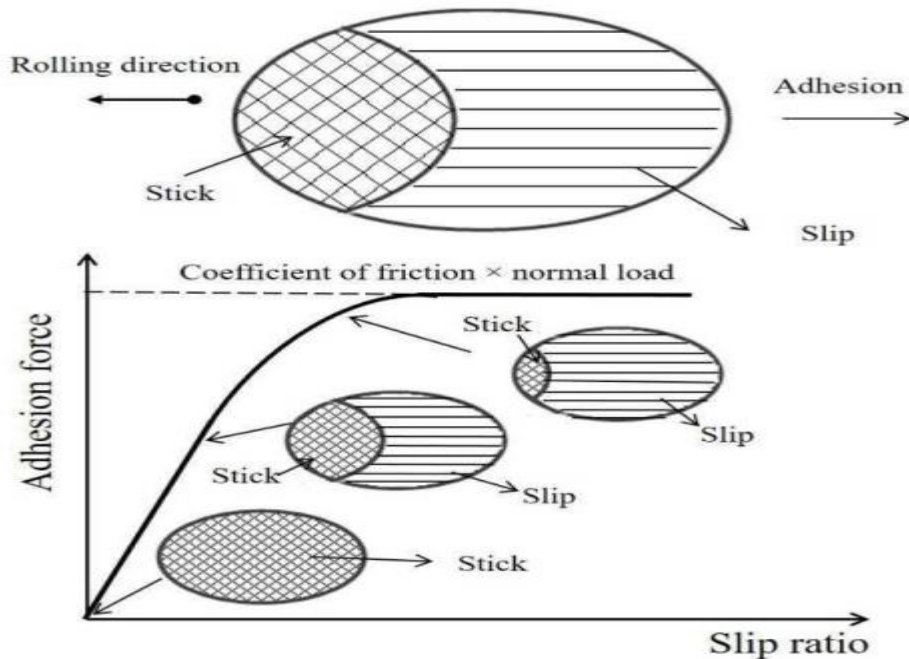


Figure 2. 5; stick and slip zones in the wheel rail contact.

2.4 Research on sleeper spacing and material

David et al. [10] in their study on the influence of variation of track level and support system stiffness on track performance and vehicle track interaction, the track support system stiffness was quantified by the track support system modulus k , which combines the effects of all the

resilient elements below the rail (rail pads, ballast, sub ballast, sleepers, the foundation and possibly under sleeper or pads and mats). They concentrated on ballast stiffness and also did not consider variation in sleeper spacing.

Tzanakakis also in his study of effects of track stiffness on track performance looks at global track stiffness as a combination of the stiffness of different layers and components of the truck. He described the effect of global stiffness in a table 2.1. He also illustrated in figure 2.18 the deterioration process due to track stiffness variations and clearly it is mainly due to increase in wheel/rail interaction forces [13].

Table 2. 1: Influence of global stiffness [13]

Case	Influence / Result	Explanation
Track of low global stiffness	<p>Large displacements and large bending moments of the rail.</p> <ul style="list-style-type: none"> • Dynamic train-track interaction forces will be relatively low. • Influence on long-term fatigue conditions. 	<p>Large rail displacements: more sleepers will be involved sharing the load leading to lower forces on sleepers.</p>
Track of high global stiffness	<p>Small displacement and small bending moments of the rail.</p> <ul style="list-style-type: none"> • Dynamic train-track interaction forces will be relatively higher. 	<p>Small rail displacements have the effect that fewer sleepers will share the load increasing forces for each sleeper.</p>

It is also mentioned track that stiffness is directly affected by sleeper spacing hence causing changes in wheel/rail contact forces [13]. These contact forces when too much, cause degradation of the interfaces in contact as shown in the figure below;

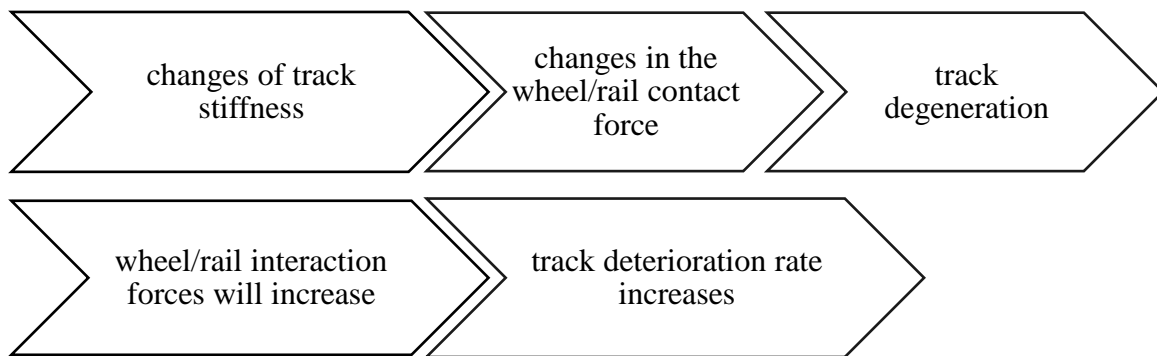


Figure 2. 6: The deterioration process due to variation in track stiffness [13]

Wu and Thompson analysed the effect of sleeper random spacing and ballast stiffness on rail vibration and discovered that the pinned-pinned resonance phenomenon could be suppressed due to random sleeper spacing, but randomly varying ballast stiffness had no significant effect on the average noise generated by the track. The consistency of sleeper spacing is an elementary qualitative feature of a typical railway track. It affects the consistency of viscoelastic supports, coefficients, and additional mass of sleepers with rotational inertia [19].

Zou et al. [39] compared the effects of vehicle speed and load on the vibration response of rails under varying sleeper spacing, and the results showed that the sleeper spacing directly affects the first-order Pinned-pinned vibration of the rail, and the effects of the speed and the load on the vibration response are different. In the low-speed section, the effect of the sleeper spacing on the vibration reaction is smaller, and the bigger spacing can be suitably selected to decrease the quantity of sleepers to save costs. However, in the high-speed section, the effect is greater, and the speed and load must be expansively well-thought-out to select the ideal sleeper spacing.

Abe et al. [39] investigated the impact of sleeper spacing on railway track vibration and noise. Through numerical experiment, it was confirmed that sleeper spacing variation causes a low anticipated amount of pinned-pinned resonance amplitude. The amplitude of rail deflection excitation as a result of stationary harmonic loading is affected by sleeper spacing for the pinned-pinned resonance. Furthermore, they recommended that incredibly low amplitude can be achieved by employment of the optimal sleeper arrangement. They also noted that the sleeper spacing variation can increase the sound pressure level at lower frequencies optimal sleeper arrangement can cause suppression of the noise relevant to the pinned-pinned resonance no matter the train speed. Therefore, it was concluded that the optimal sleeper distribution can contribute to the reduction of vibration and noise caused by the pinned-pinned resonance.

Jabbar et al. [40] in their study of Sensitivity analysis of track parameters on train-track dynamic interaction investigated the factors affecting track responses among which is sleeper spacing. They noted that an increase in the sleeper spacing raises the sleepers' stiffness causing them to give a greater reaction resulting in displacement of the rail. A sensitivity analysis was done by distancing the sleepers and evaluating the effects in a track-train interaction modelling and the results showed that an increasing the sleeper spacing from 55cm to 65cm would cause a 12% increment in the reaction forces of the supports, 18% increment in the displacement of mid span of the rail, 12% increment in the displacement of rail on sleeper and displacement of the sleeper.

Shokrieh et al. [41] did a study on the effect of young's modulus on the performance of sleepers using the Winkler's foundation method. For their case study, it was shown that when the variance between the Young's modulus of the beam and that foundation is noticeable, the stiffer material can be considered as a rigid material, and in this case, the change in the Young's modulus of the rigid material has little effects on the response of the sleeper.

Anderson et al. [42] explained that variations in support condition and differences in ballast stiffness influence the risk of damage on track and train. Assessment of the track stiffness can be used as a factor to boost the maintenance by recognizing local weaknesses along the track and detect hanging sleepers. The rates of degradation of track components and track settlement are affected by the severity of the stiffness variation. When the track geometry starts to deteriorate, the disparities of the train/track contact forces increase, and this increases the track and wheel degradation rate.

2.5 Components/parts used in the study.

The study will focus on the wheel set, railway track and the sleepers. These are defined and explained in detail below;

2.5.1 Wheel set

The wheel set is attached to the railway bogie. Bogie is a structure beneath a train to which axles and hence wheels are connected through bearings. A wheel set consists of two wheels linked by one axle [43]. It serves the following purposes;

- It provides the suitable distance between the vehicle and track.
- It is a guide to the motion of vehicles on the tracks.

- It provides means of transferring axle loads, traction and braking forces to the rails to accelerate and decelerate the vehicle etc.

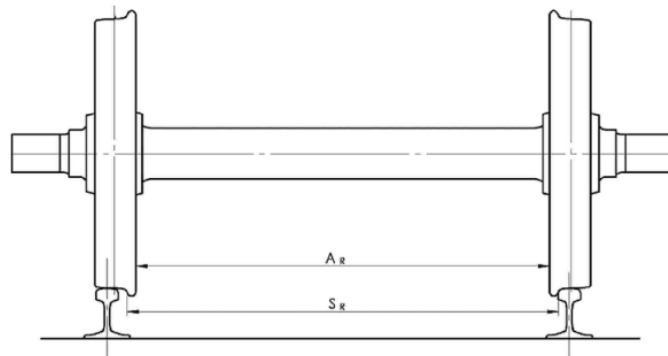


Figure 2. 7; wheelset [43]

2.5.2 Rails

The rails offer the running surface for the wheels and are placed on spaced sleepers to guide the rolling stock. The wheel axle loads are transmitted by the rails to the cross-ties. The modern rail also acts as a conductor on electrified lines in addition to transmitting signals. There are various rail profiles in use but for this study the most commonly used profile; flat-bottom rail, also called Vignole rail was used. It is divided into three parts; rail head- the top surface that contacts the wheel tire, rail web- the middle part that supports the rail head, like columns, rail foot- the bottom part that distributes the load from the web to the underlying superstructure components [43], [44]. The rail profile is shown in the figure below;

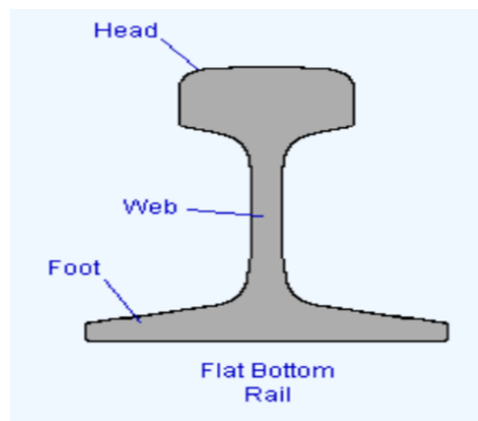


Figure 2. 8; Flat bottom rail parts [43].

2.5.3 Sleepers

These are one of the most significant components of the railway track. They are a part of the track superstructure in addition to the rails and the fasteners. The roles of the sleepers include; transmitting and distributing train loads from the rail foot to ballast bed, to maintain the rail

gauge with the help of the rail fastening system, to maintain rail inclination, and to restrain longitudinal, lateral and vertical movements of the rails [45]. The loads to be transferred by the sleepers can also be referred to as rail seat loads and are vertical, longitudinal and lateral in nature. The vertical loads expose the sleeper to a bending moment which is inclined to condition of the ballast under the sleepers. However, the ability of the sleeper to tolerate both longitudinal and lateral forces is determined by sleeper's size, shape, surface geometry, weight and spacing [46]. The sleepers should also have an adequate level of strength and elasticity since they are exposed to high loads from the trains [17]. The sleeper materials that have been used so far around the world include; timber/ wood, steel, pre-stressed concrete and composite materials.

The timber sleepers have been replaced by concrete sleepers in most parts of the world though the composite sleepers have also started to increase as a replacement for concrete sleepers, this implies that scientists continue to improve the available sleeper materials until now. The timber sleepers due to their desirable mechanical properties are still being used in critical sections of the track like S&C, turnouts and others. It has been noted that the steel sleeper material has not been used so much in most parts of the world [47]–[49]. All of these sleeper materials have strengths and weaknesses, some work best in particular operating conditions, and since they operate in an open environment, they should have material properties that enable them to withstand UV light, temperature fluctuations, and resist insect attack not forgetting the ability to prevent conduction of electricity between the rails [47].

In general, the sleepers are manufactured as mono-block except the concrete sleepers which can be produced as twin blocks. The twin-block sleeper consists of two blocks with one block tied under each rail and are connected by a steel rod. The questions of whether or not one is better in terms of cost, maintenance and installation is still uncertain. The twin-block sleepers have been used in France on both high speed and low speed lines but the mono-block sleepers are the most common around the world even on the AALRT [50].



a) Mono-block concrete sleeper



b) twin-block concrete sleeper

Figure 2. 9: Types of concrete sleepers [44]

2.6 Gap in the literature review

It is evident from the research above that research on sleeper spacing has mostly been done on its effect on vibration and noise induced in the truck and the ground, and on track deterioration. This study will focus on the influence of the sleeper spacing and sleeper material on wheel/ rail interaction in terms of stresses, contact forces, contact pressures provoked due to variation of these parameters under dry contact conditions.

CHAPTER THREE: MATERIALS AND METHODOLOGY

3.1 Data collection

The data collected for the study was both primary and secondary data; the primary data which included the track conditions, optimum vehicle speed, currently used sleeper material and spacing was collected from AALRT. The secondary data included; vehicle weight specifications and axle load, sleeper materials and material properties of other sleeper types (timber (F34) and steel) that are not currently used at AALRT, the mechanical properties of the wheel, rail and axle material, other sleeper spacing (500 mm and 700 mm) and related research was collected from journals, conference proceedings, and text books.

3.1.1 General AALRT track conditions

Table 3. 1; AA LRT track conditions

Parameter	Value
Track gauge	1435 mm
Rail cant	1/40
Minimum radius of horizontal curve	50 m
Maximum super elevation of curve	120 mm
Type of rail	50 kg/m
Sleeper spacing	600 mm

3.1.2 General AALRT vehicle specifications

Material properties and data are used based on Ethiopia Railway Corporation, Addis Ababa Light rail Transit (LRT). The optimum vehicle operation speed is 40 Km/h, axial load $\leq 17t$ and axle number is four. The details concerning the vehicle weight are shown in the table 3.2 below;

Table 3. 2; Vehicle weight specifications [1]

Loads	Car body weight (t)	Passenger weight (t)	Total weight (t)
Overload capacity	48.76	15.24	64
Axle load	$\leq 17 t$		

Note: Take 60kg as average weight of each passenger

Overloading conditions have been considered in the study because it is the most common scenario due to the fact that the population is high in Addis Ababa hence the trains are filled to overcapacity especially during peak hours in the morning as people go to work and in the evening when returning from work.

3.2 Materials and sleeper spacing

This section outlines the mechanical properties of the parts (rails, wheels, axle and sleepers) used for the study. The table below includes the sleeper material currently used at AALRT (concrete) and the other materials for which comparison is to be made (timber (F34) and structural steel).

Table 3. 3; Material Properties [14], [51], [52]

Materials and material Property	UIC50	ER7	Concrete	Timber (F34) grade	Structural steel	EA1- Axle steel grade
Material density (kg/m ³)	7800	7850	2400	1200	7850	7850
Poisson ratio	0.3	0.3	0.2	0.25	0.3	0.3
Young's modulus (Gpa)	207	210	30.2	21.5	200	210
Tensile yield strength (Mpa)	540	540	2.85	13	250	320
Compressive yield strength (Mpa)	540	540	52	18	250	320
Ultimate tensile strength (Mpa)	780	820			460	550

The currently used sleeper spacing is 600 mm. the other values of comparison (500 mm and 700 mm) have been selected based on a study conducted by Ayantu [53].

3.3 Force distribution on a wheel set.

There is a difference between forces experienced by a wheelset while on the straight and curved track. The difference is that only the vertical force acts on the wheel set on a straight track whereas both lateral and vertical forces act on a wheel set treading a curved track.

3.3.1 Forces on straight track path

Only the wheel load acts on the wheels that are running on a straight track and for an applied load on a wheel-rail interface, normal contact forces develop on the contact patch depending on the total vertical load applied and the contact angle formed as a result of the lateral displacement, y of the wheel set during motion. For this study, equal loads are assumed on both wheels.

$$\text{Load /force on each wheel of the wheel set} = \left(\frac{1}{8}m_v + \frac{1}{4}m_b + m_w \right) \times g \quad (3.1)$$

Where; m_v = the total mass of the vehicle

m_w = the mass of the wheel which according to UIC, S1002 railway standard is 412kg.

m_b = The mass of the bogie

g = the acceleration due to gravity

The value of the normal load F_N is 83000N.

Since the wheel set is rotating on the track, the angular velocity (ω) should be known and it is calculated using the equation below;

$$\omega = \frac{V}{r} \quad (3.2)$$

Where; V = vehicle speed which in this case is the maximum vehicle speed of 40 km/h

R = the radius of the wheel which is 460 mm for this study

By substituting values of speed and wheel radius in m/s and metres respectively, the angular velocity is 24.15rad/s

3.3.2 Contact forces on curved track

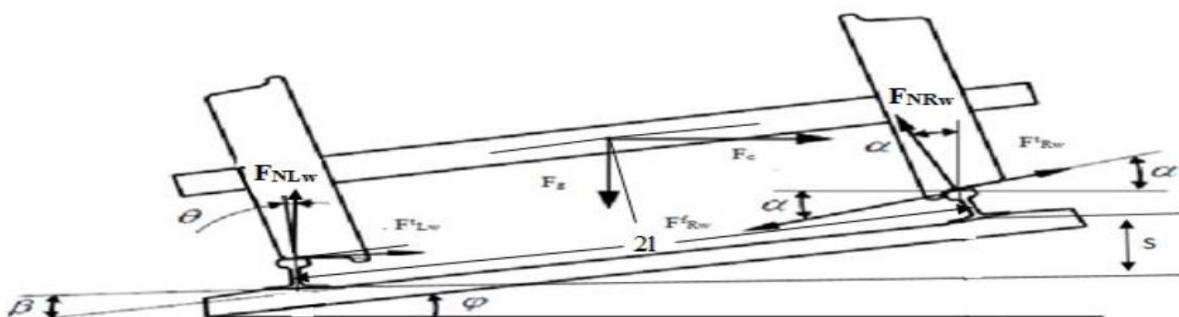


Figure 3. 1; contact forces on curve track [23]

The figure 3.1 is a representation of the forces acting on the wheel set when a vehicle is negotiating a curve and it is shown that there is a flange contact on the right side.

F_{NLW} , F_{NRW} represent the normal contact forces acting on the left and right wheels, respectively.

F_{LW}^t , F_{RW}^t are the lateral contact forces acting on the treads of the wheels.

F_{RW}^f is the lateral flange force on the right wheel, and,

F_g and F_c are the gravitational and the centrifugal forces on the wheel set. The angle, ϕ represents the track cant angle, β is the rail inclination angle and θ and α , are the angles associated to the sets of forces acting on the left and right wheels, respectively.

The vehicle is subjected to a constant radial acceleration when moving in a curve;

$$\text{Radial acceleration, } a_c = \frac{V^2}{R} \quad (3.3)$$

Where; V is vehicle speed which is 40km/hr equivalent to 11.11m/s

R is the curve radius in meters and m is the maximum axle load

The radial acceleration yields a centrifugal force in a radial direction from the centre.

$$\begin{aligned} \text{Centrifugal force, } F_c &= (m \times a_c) \quad (3.4) \\ &= \frac{(16000 \times 11.11^2)}{50} = 39498N \end{aligned}$$

To neutralise the effect of centrifugal force, a super elevation is applied to the outer rail of the curve with respect to the inner rail.

The equilibrium of forces can be written as below;

For vertical forces;

$$\begin{aligned} \sum F_y &= F_{NRW} \cos \alpha + F_{NLW} \cos \theta + F_{RW}^t \sin \alpha - F_{RW}^f \sin \alpha - F_{LW}^t \sin \theta - F_g = 0 \\ F_g &= F_{NRW} \cos \alpha + F_{NLW} \cos \theta + F_{RW}^t \sin \alpha - F_{RW}^f \sin \alpha - F_{LW}^t \sin \theta \quad (3.5) \end{aligned}$$

For lateral forces;

$$\begin{aligned} \sum F_x &= -F_{NRW} \sin \alpha + F_{NLW} \sin \theta + F_{RW}^t \cos \alpha - F_{RW}^f \cos \alpha + F_{LW}^t \cos \theta + F_c = 0 \\ F_c &= F_{NRW} \sin \alpha - F_{NLW} \sin \theta - F_{RW}^t \cos \alpha + F_{RW}^f \cos \alpha - F_{LW}^t \cos \theta \quad (3.3) \end{aligned}$$

Gravitational force, $F_g = w = mg = 16000 \times 9.81 = 156960N$

Where; w , is axle weight and m , is maximum axle load

F_{Lw}^t , F_{Rw}^t , and $F_{Rw}^f = \mu F_N$; μ being equal to 0.4 and F_N the normal load on the wheel

Rail cant, $\beta = \sin^{-1}(1/40) = 1.433^\circ$

$$\sin \varphi = \frac{S}{2l} = \frac{120mm}{1505mm} = 0.079734$$

Therefore, $\varphi = 4.573^\circ$

$$\alpha = (\varphi + \beta) = 6.006^\circ$$

$$\theta = (\varphi - \beta) = 3.14^\circ$$

Where; S is maximum super elevation,

$2l$; is the lateral distance between the points of contact of the wheels with the rails

By solving the equations simultaneously, the vertical forces on the right and left wheel are 201843.28 and 121000N respectively and the lateral forces are 25062N and 24930N respectively.

3.3.3 The effect of contact temperatures.

Temperature variations have an effect on some mechanical properties of materials and on the stress strain behaviour too. According to Liu et al, they discovered that the temperature effect on rail steels is noticeable when the range of temperature is 100° C and above [54]. Additionally, these temperatures usually occur in freight and passenger trains that cover long distances and this is not the case for this model since it is a very short distance. Also, from AALRT, it is known that the range of temperatures from start to stop of the train is usually 20° C to 50° C on average respectively and for this matter, the temperature effects are negligible in this particular study.

3.4 Modelling using Finite Element Method

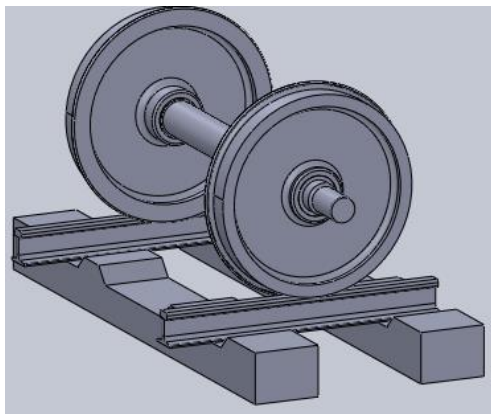
Finite Element Analysis is a numerical method of deconstructing a complex system into elements. The software executes equations on which the behaviour of these elements depends and solves them hence creating a broad explanation of how the system performs. These results are illustrated in tabular or graphical forms. This nature of analysis is usually applied in the design and optimization of systems that are far too difficult to analyse by hand.

The FEA software package used for this particular study is Ansys 2020 which is an analysis tool for structural analysis, including linear, non-linear, static and dynamic studies. The contact

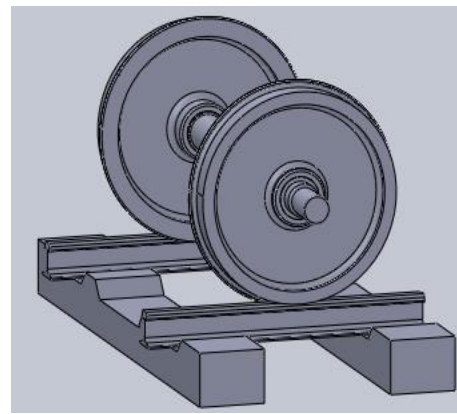
mechanics including contact stresses, pressure, and forces, strain, deformation and sliding distance will be determined in this study.

3.4.1 General procedure for FEM.

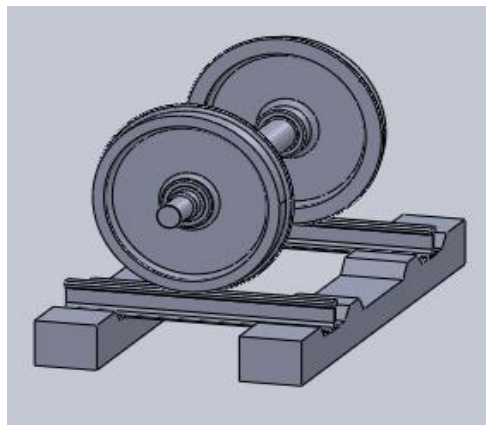
- 1) An assembly of the wheel set on a track and sleepers is developed in solid works. First the parts are developed independently (wheel, axle, rail and sleeper) and then assembled as shown below with the different sleeper spacing under study. The sleeper spacing used in this study was 500 mm, 600 mm and 700 mm.



a) 500 mm



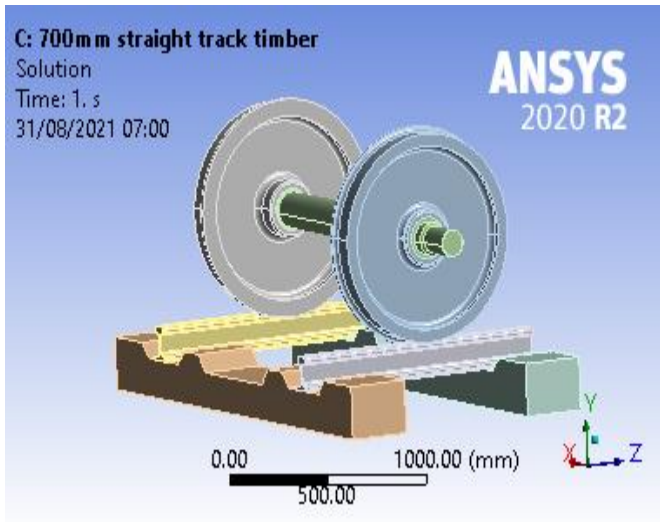
b) 600 mm



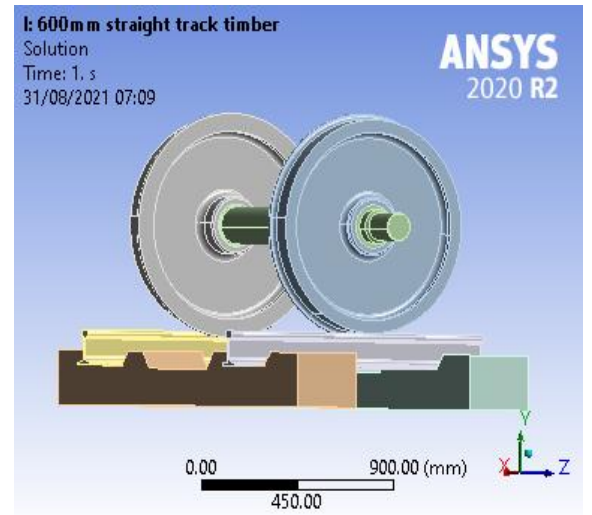
c) 700 mm

Figure 3. 2; Solid works models

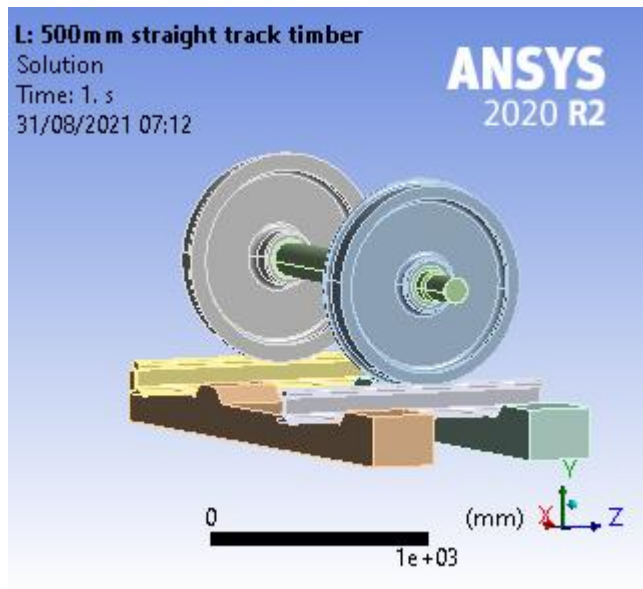
- 2) The solid works assemblies with various sleeper spacing, were imported to Ansys in IGES format for numerical analysis as shown below;



a) 700 mm



b) 600 mm



c) 500 mm

Figure 3. 3; IGES formats for the different models used

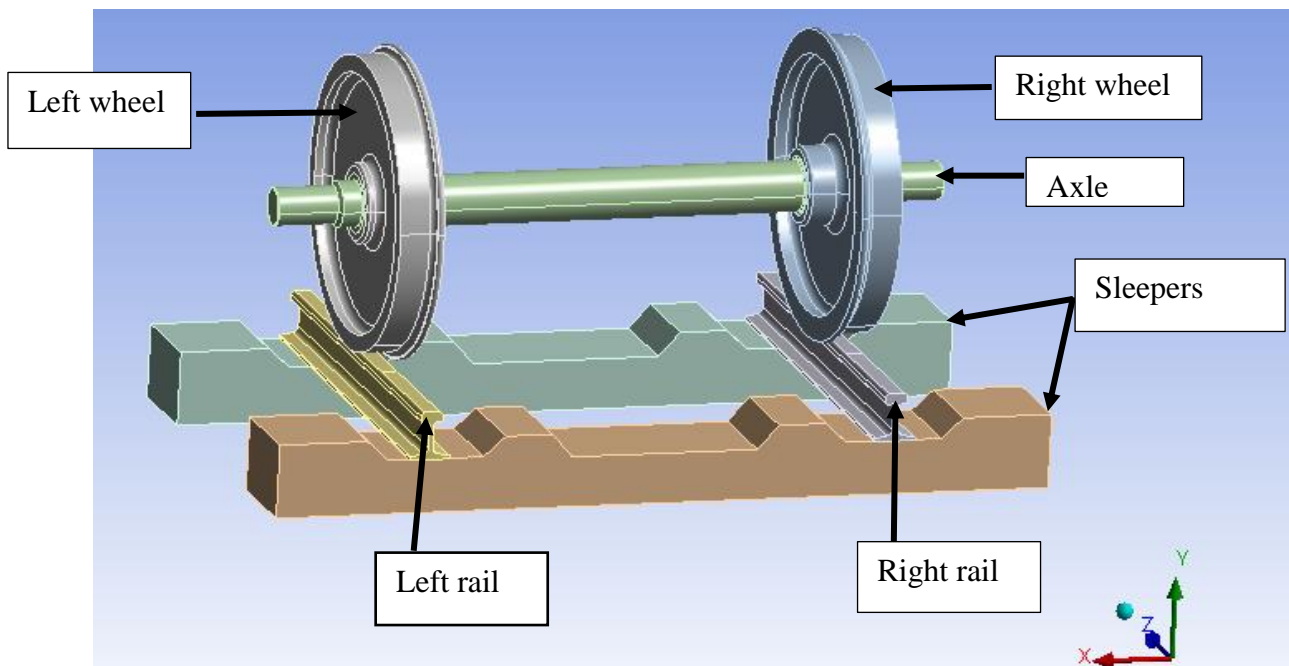


Figure 3. 4: Three dimensional model of the wheel rail system used for the study

- 3) Materials with their respective properties were selected from the materials library considering linear elasticity for all the bodies in the model. The material properties are shown in table 3.3 above.
- 4) Contacts were modelled. Basically two contact types are applied in this study; 1- frictional contact between the rails and the wheels with a CoF of 0.4 to represent the dry contact conditions and it's the contact of interest for this study. Augmented Lagrange and penalty formulations are frequently used in contact analysis however, Augmented Lagrange generates minimum amount of penetration compared to the penalty method and for this reason, augmented Lagrange has been chosen as the contact formulation method. When the vehicle is moving in a curve, the wheel flange sometimes has contact with the rail gauge corner [23]. 2- Bonded contact between the sleepers and rails, wheels and axle with pin ball radius set to auto detection value. The contacts are shown below;

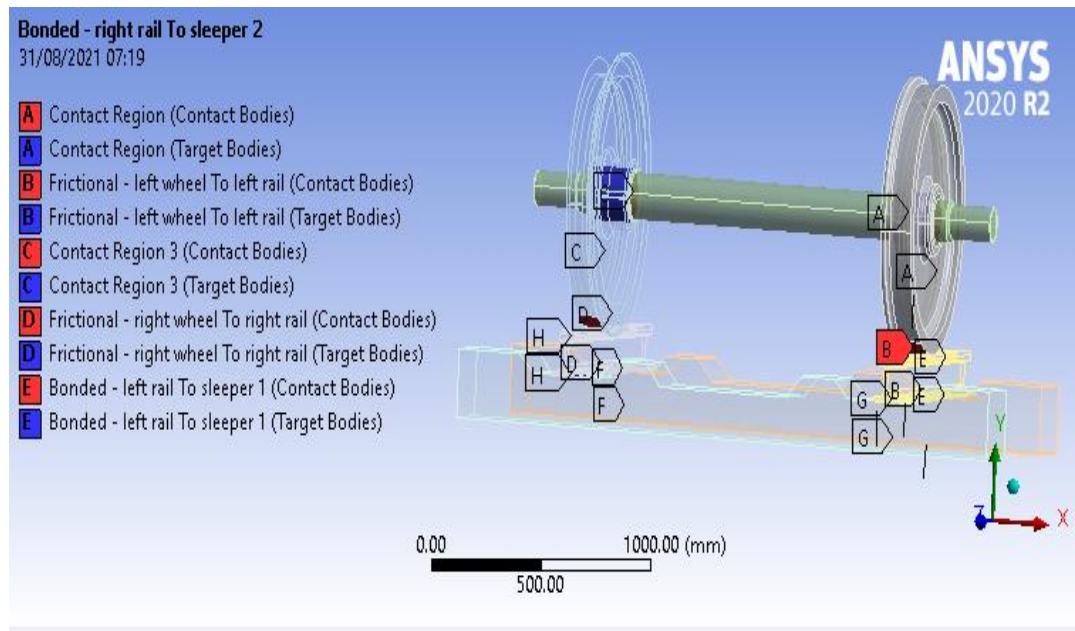


Figure 3. 5: Contacts modelled for the study

5) Meshing. This is the discretization of the model into elements and nodes. It is important to note that the higher the density of the mesh, the more accurate the results. Nevertheless, this comes with higher simulation time and resources. Therefore, for the mesh used for the model included fine mesh for the whole model, a sizing using sphere of influence of radius 50 mm with element size of 1mm was used in the wheel rail contact in order to get better accuracy since this is the point of interest for this study, patch conforming method was further applied to achieve uniform element shape for both wheels and rails which is tetrahedron shape. This was done to enable achievement of better results while limiting the simulation time. The element type used for all the bodies in the model under investigation is SOLID187, CONTA174 for contact surface and TARGE170 for target surface. The statistics of the meshes for straight and curved track are shown in the table below;

Table 3. 4; Mesh statistics for all models used.

Straight tack	700 mm	600 mm	500 mm
Nodes	366172	367143	367281
Elements	221537	222278	222287
Curved track	700 mm	600 mm	500 mm
Nodes	388687	367479	392199
Elements	235485	222291	237713

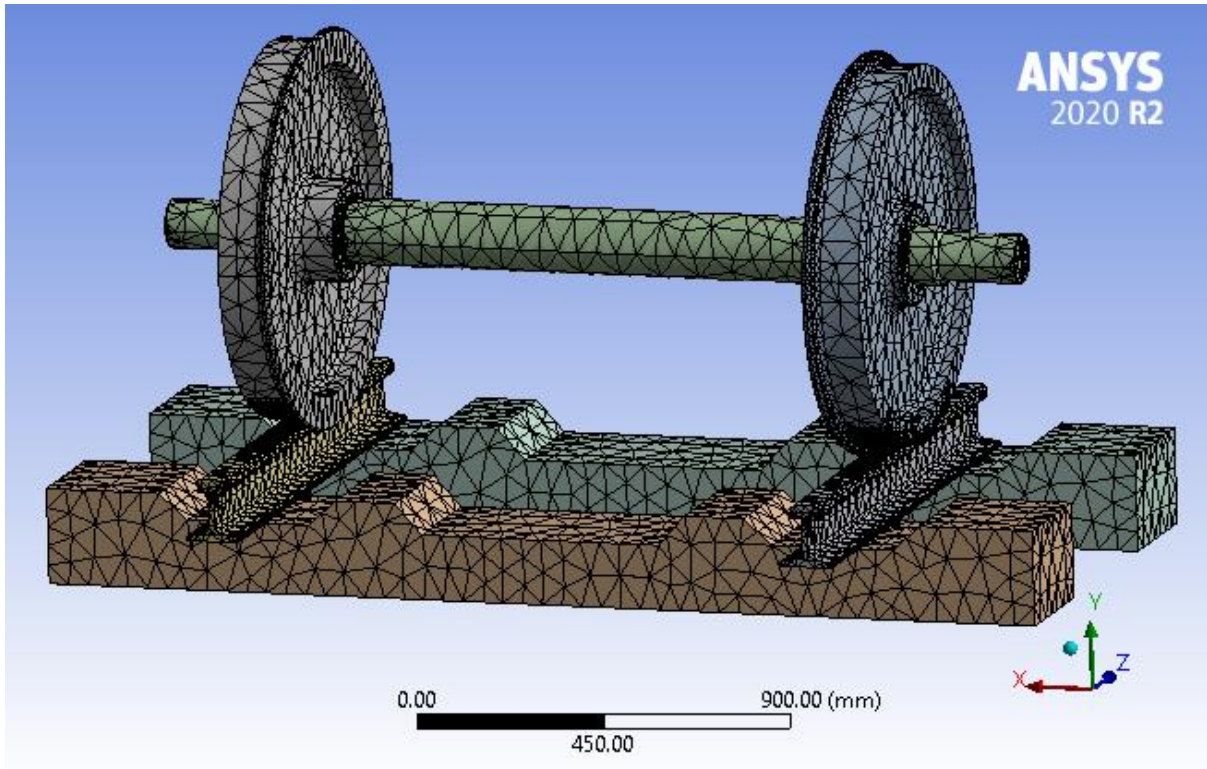
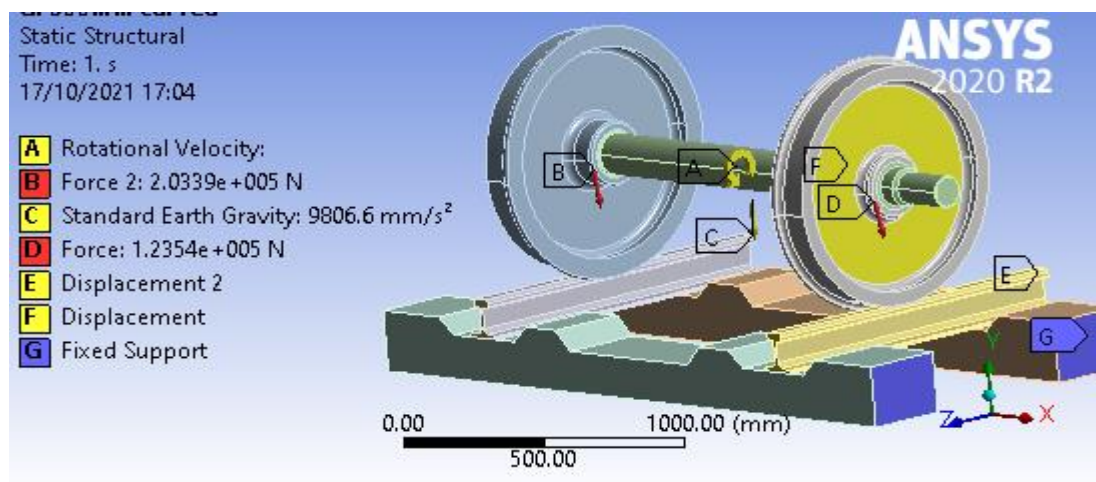
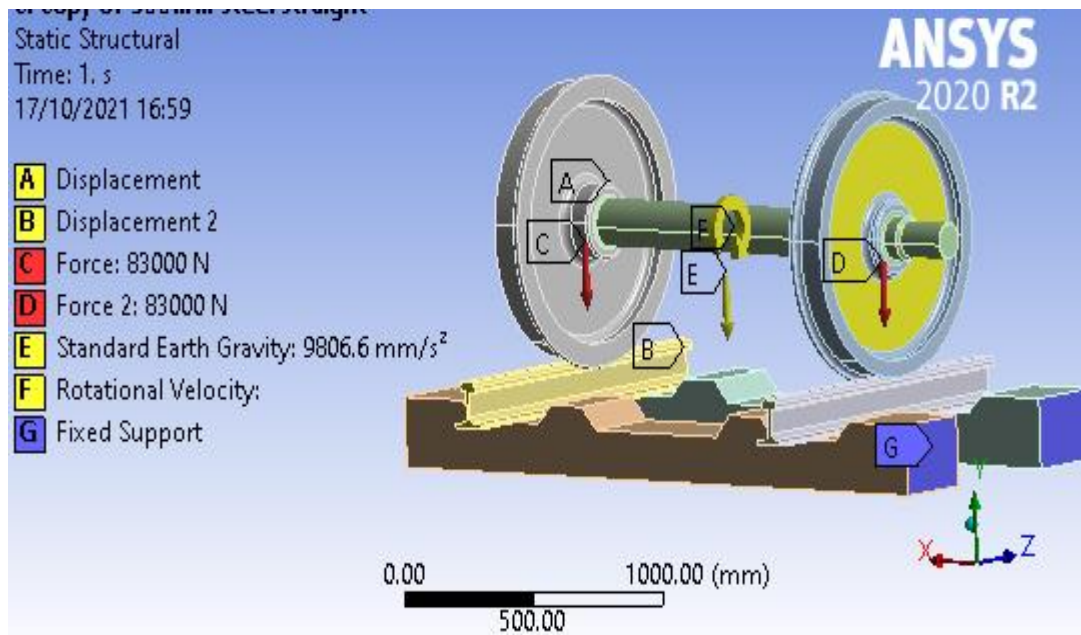


Figure 3. 6: Mesh of the model

- 6) The boundary conditions were applied with only vertical force applied on the face of the wheel hub in the straight track model and both lateral and vertical force applied in the curved track model. A rotational velocity, displacement, and standard earth gravity were also applied on the model. Fixed support was applied on the sleepers at the ends and the bottom and a displacement was applied on the rails to allow motion in only the longitudinal direction, in such a way, the deflection of the rail can be seen since it is very much affected by the sleepers. The boundary conditions are shown below;



(a)



(b)

Figure 3. 7; Boundary Conditions used for simulation for (a) curved track and (b) straight track.

c) Total deformation

Considering concrete material as the bench mark since it is currently used, using structural steel sleeper materials instead yields a slight increase in contact stress and pressure. And a 24.3% decrease in the total deformation.

b) Concrete sleeper material

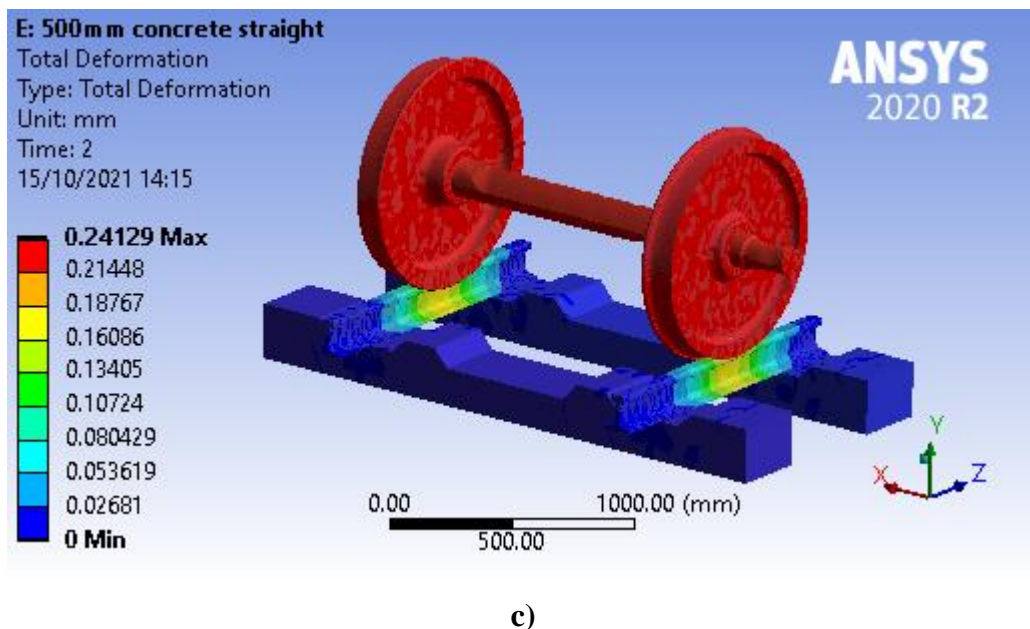
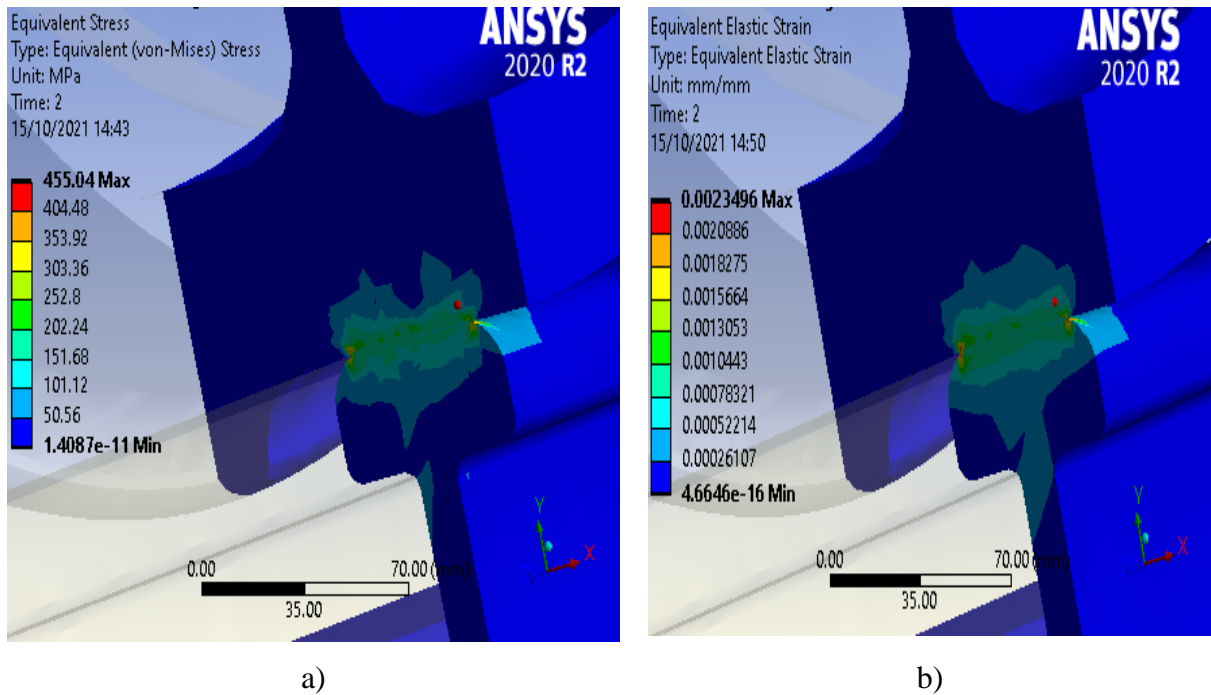
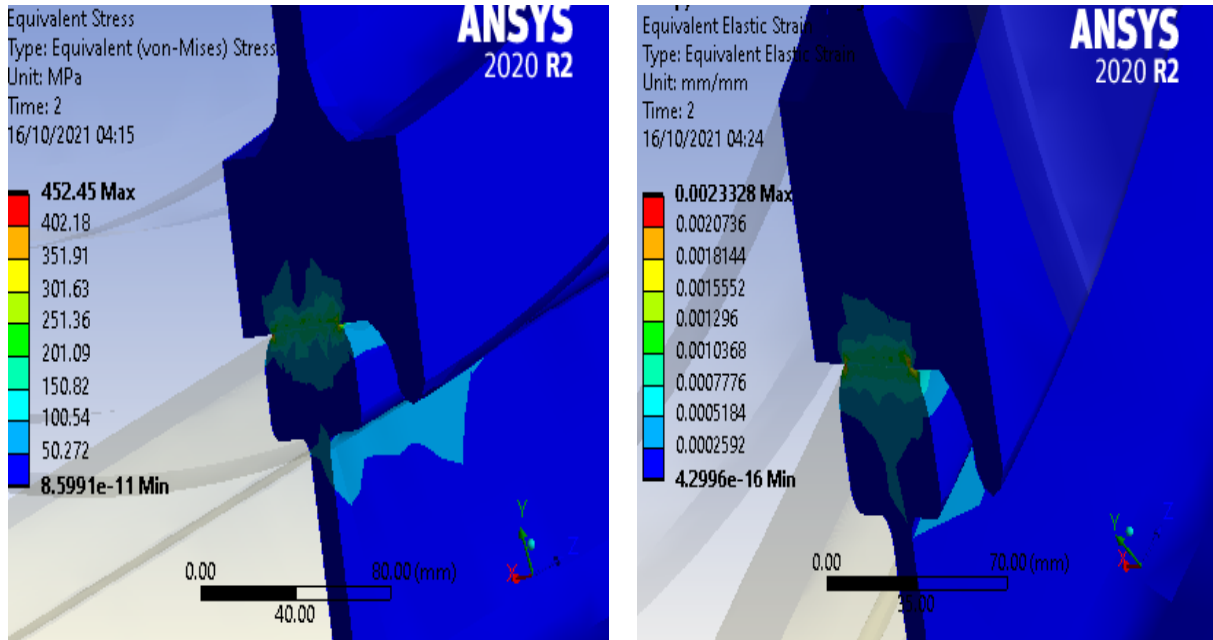


Figure 4. 2; Concrete- 500mm, straight track- a) Von mises stress, b) elastic strain, c) Total deformation

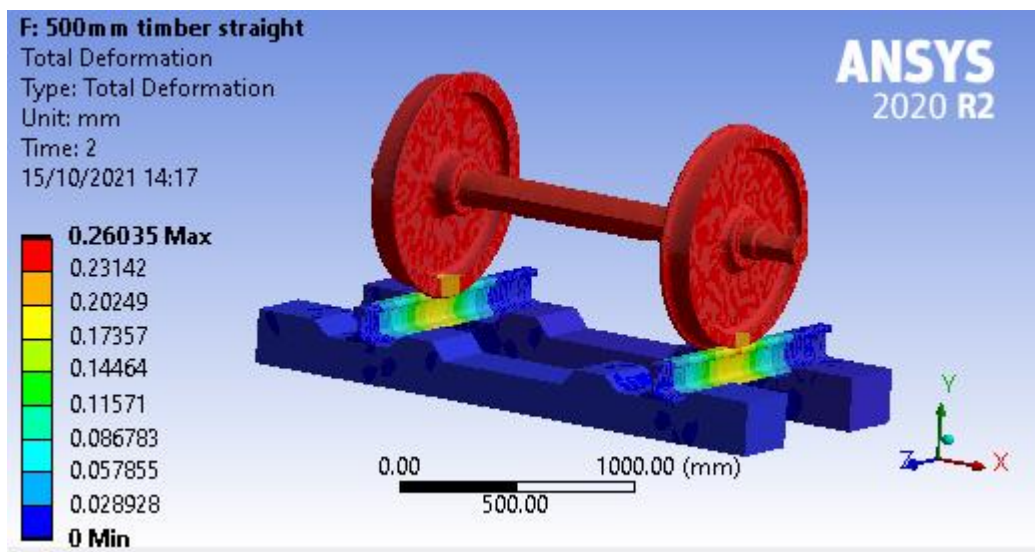
Using timber (F34) instead of concrete sleeper material yields a slight decrease in contact pressure and stress but a slight increase in the total deformation.

c) Timber (F34) sleeper material



a)

b)

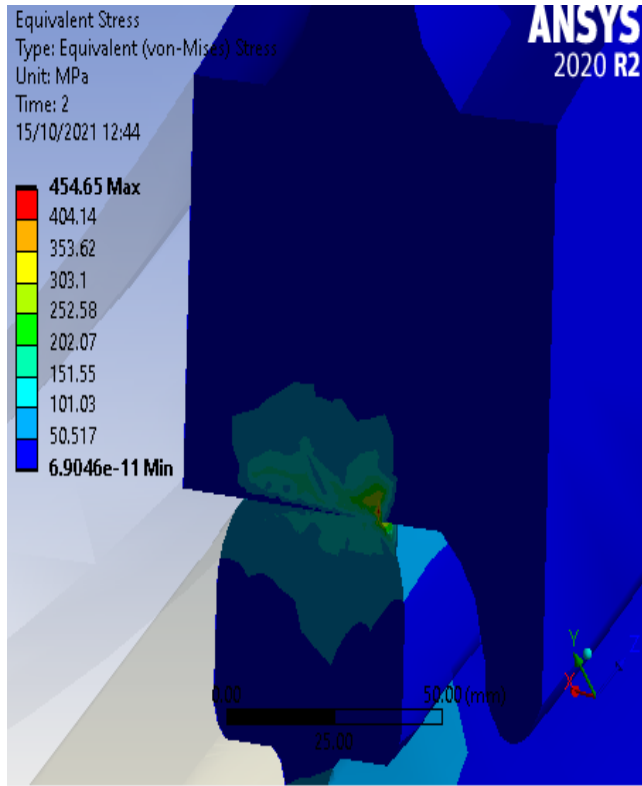


c)

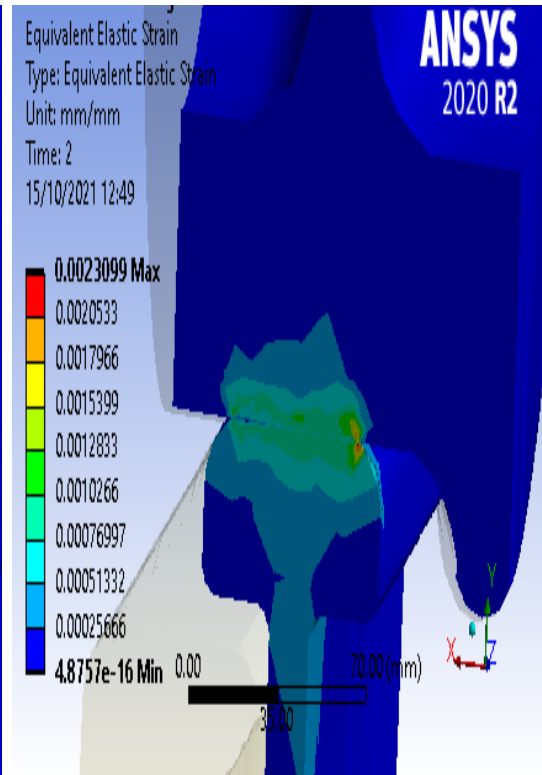
Figure 4. 3; timber (F34)-500 mm, straight track- Von mises stress, b) elastic strain, c) Total deformation

4.1.2 600 mm sleeper spacing on straight track

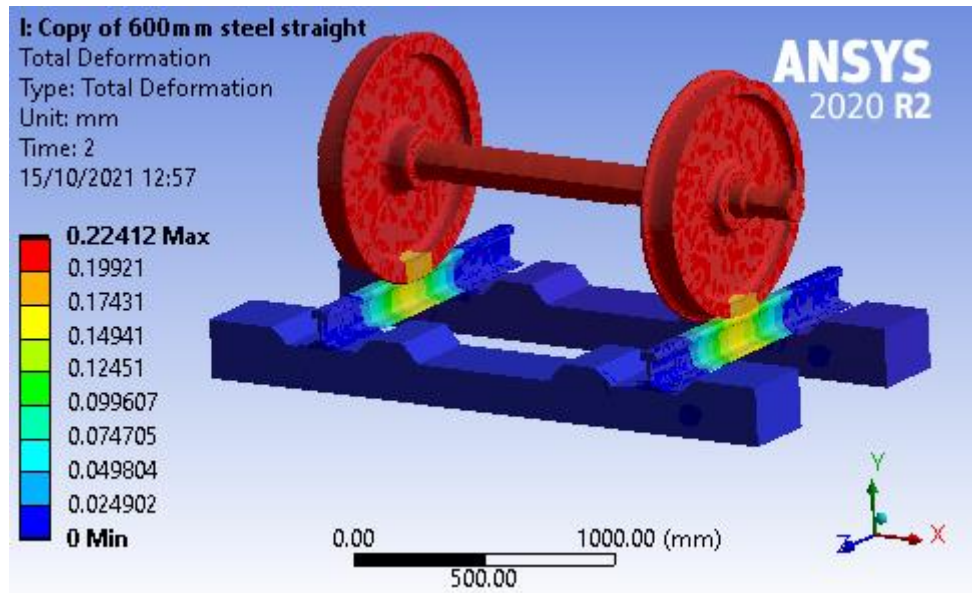
a) Structural steel sleeper material



a)



b)



c)

Figure 4. 4; Steel-600 mm, straight track- a) Von mises stress, b) equivalent elastic strain, c) total deformation

The results show that replacing concrete with steel sleeper materials (with a higher MoE) yields an increase in contact stress and pressure but a 23.1% decrease in total deformation.

b) Concrete sleeper material

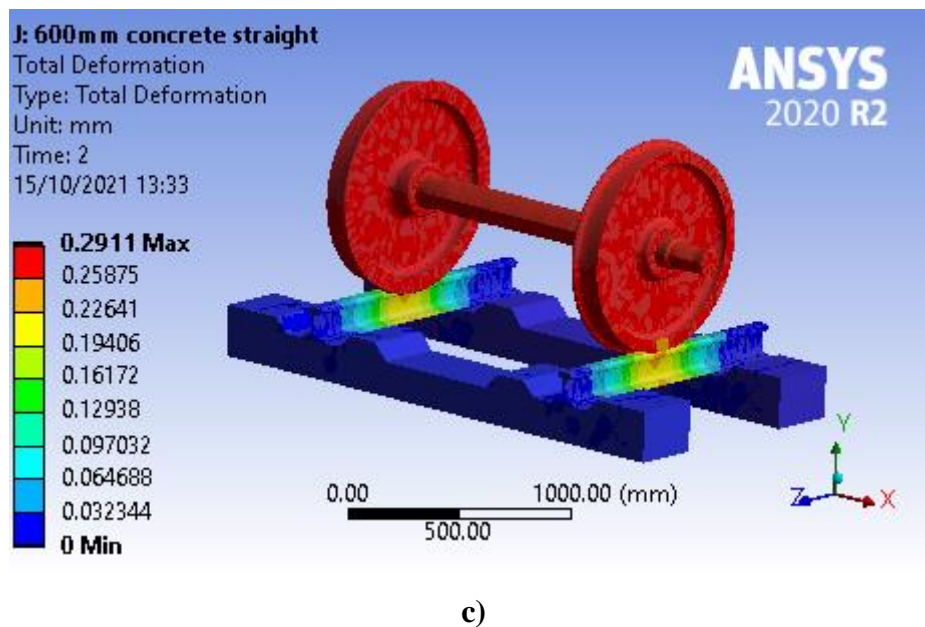
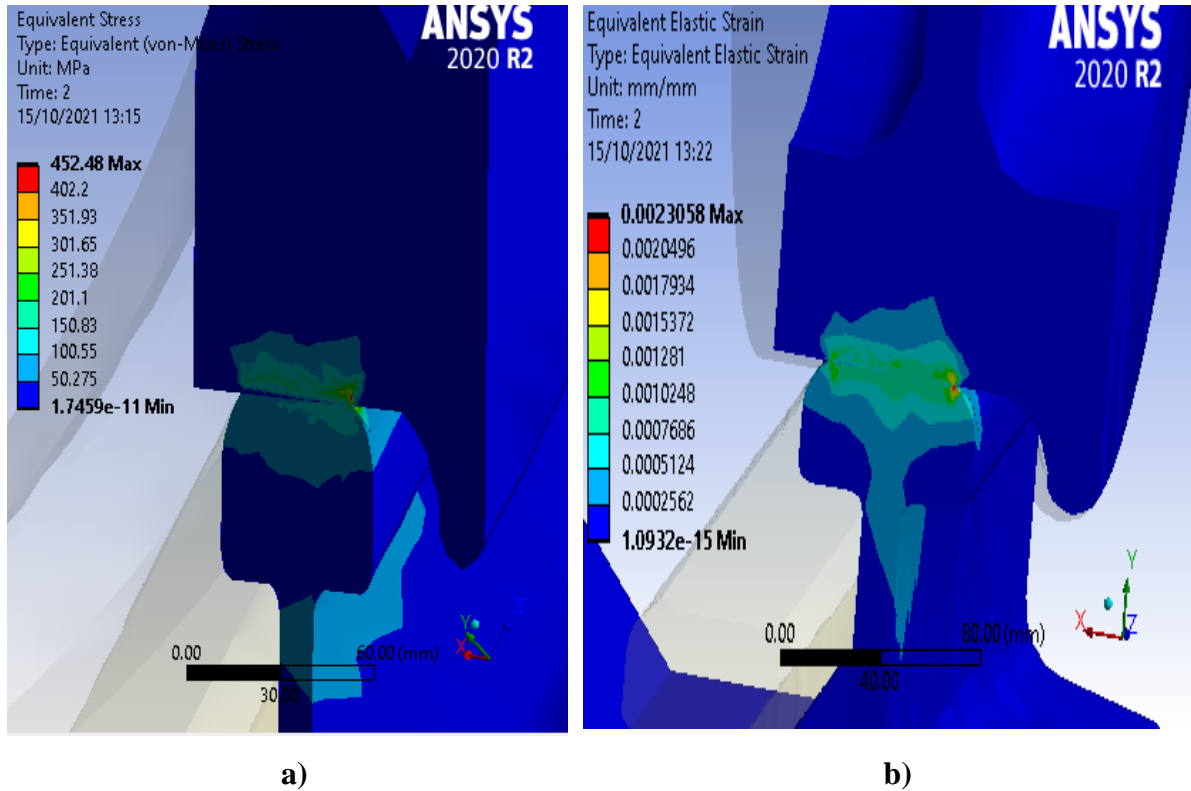
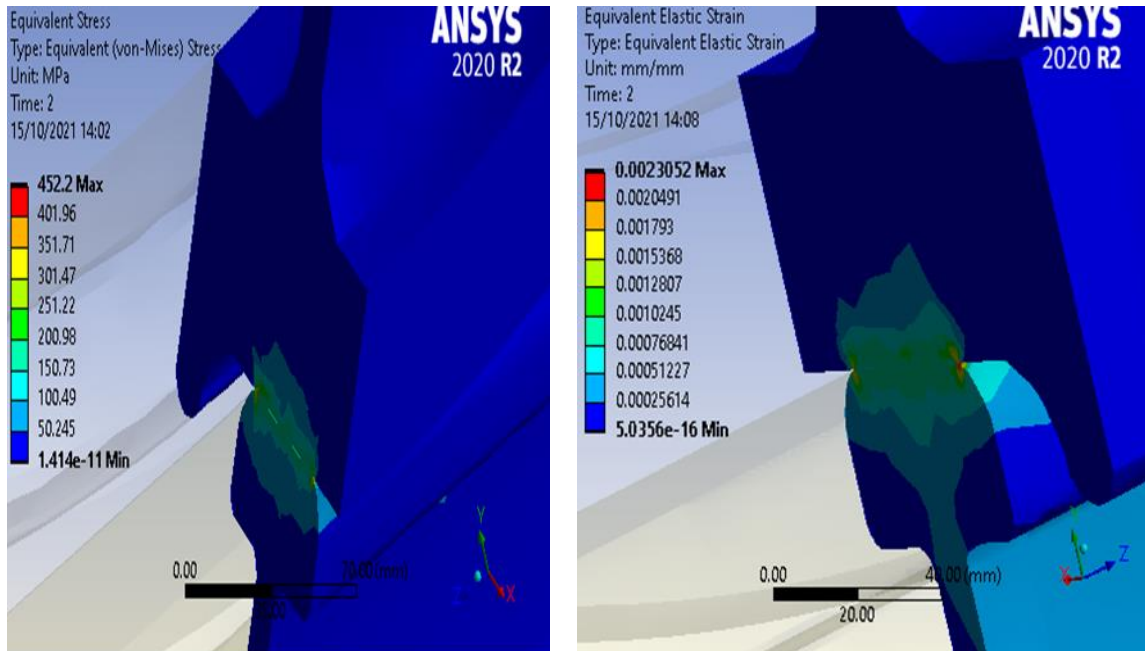


Figure 4. 5; Concrete-600 mm, straight track: a) Von mises stress, b) equivalent elastic strain, c) total deformation

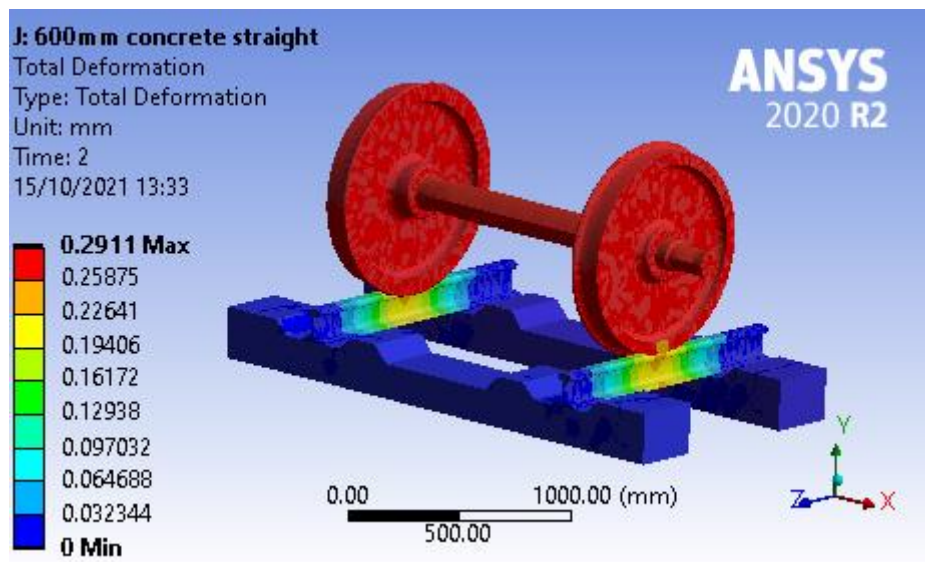
By replacing the concrete sleeper with timber (F34) sleeper materials (with a lower MoE), results in a decrease in the contact stress and pressure but a 7.5% increase in the total deformation.

c) Timber (F34) sleeper material



a)

b)

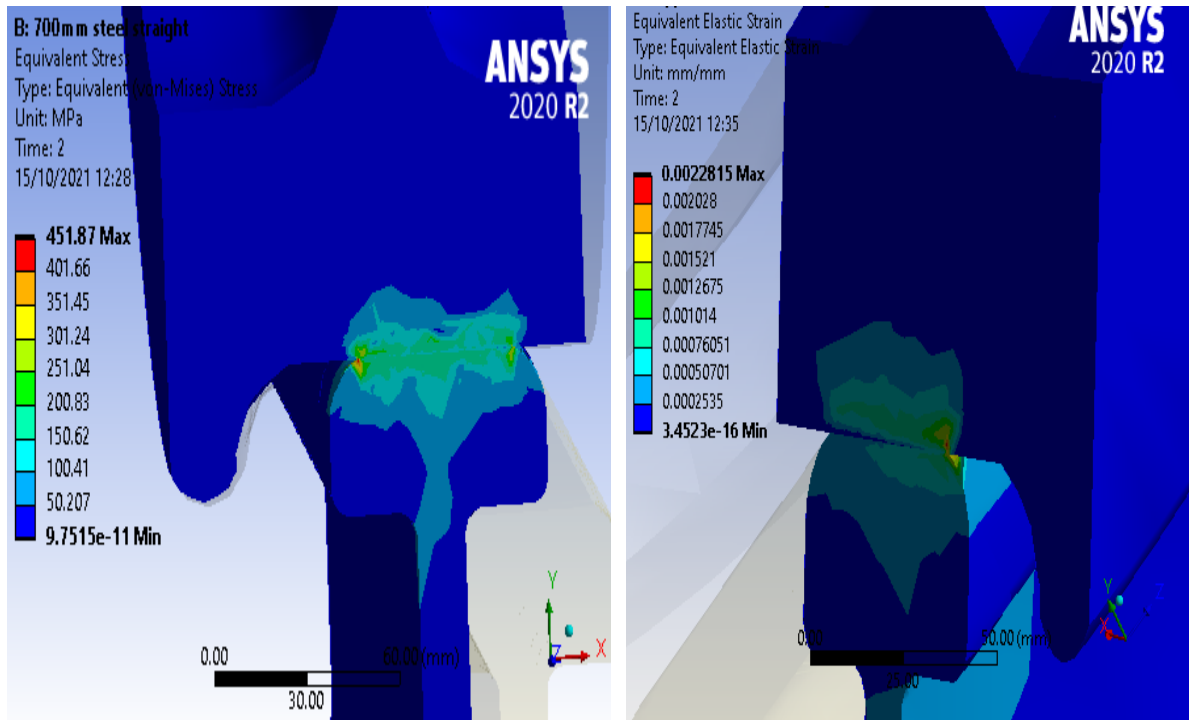


c)

Figure 4. 6; Timber-600mm, straight track a) Von mises stress, b) equivalent elastic strain, c) total deformation

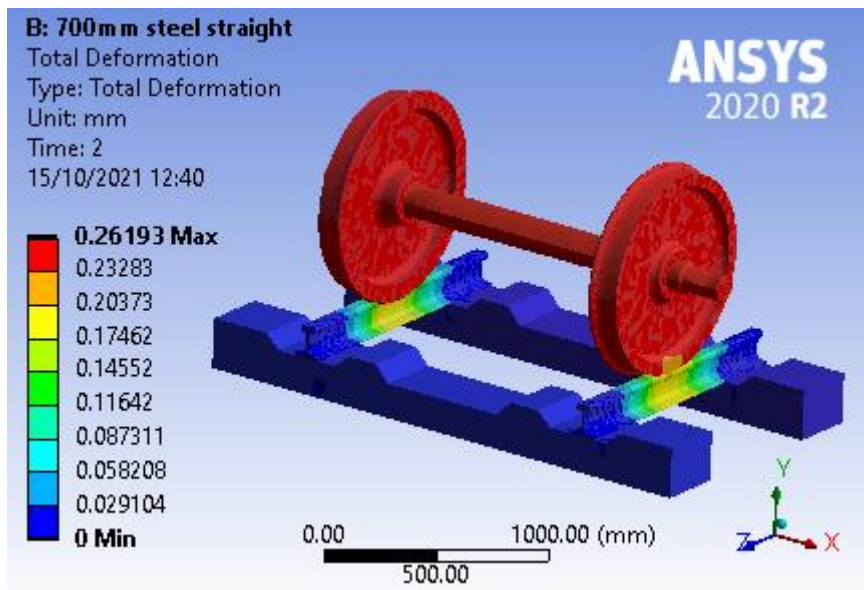
4.1.3 700 mm sleeper spacing, straight track

a) Structural steel sleeper material



a)

b)

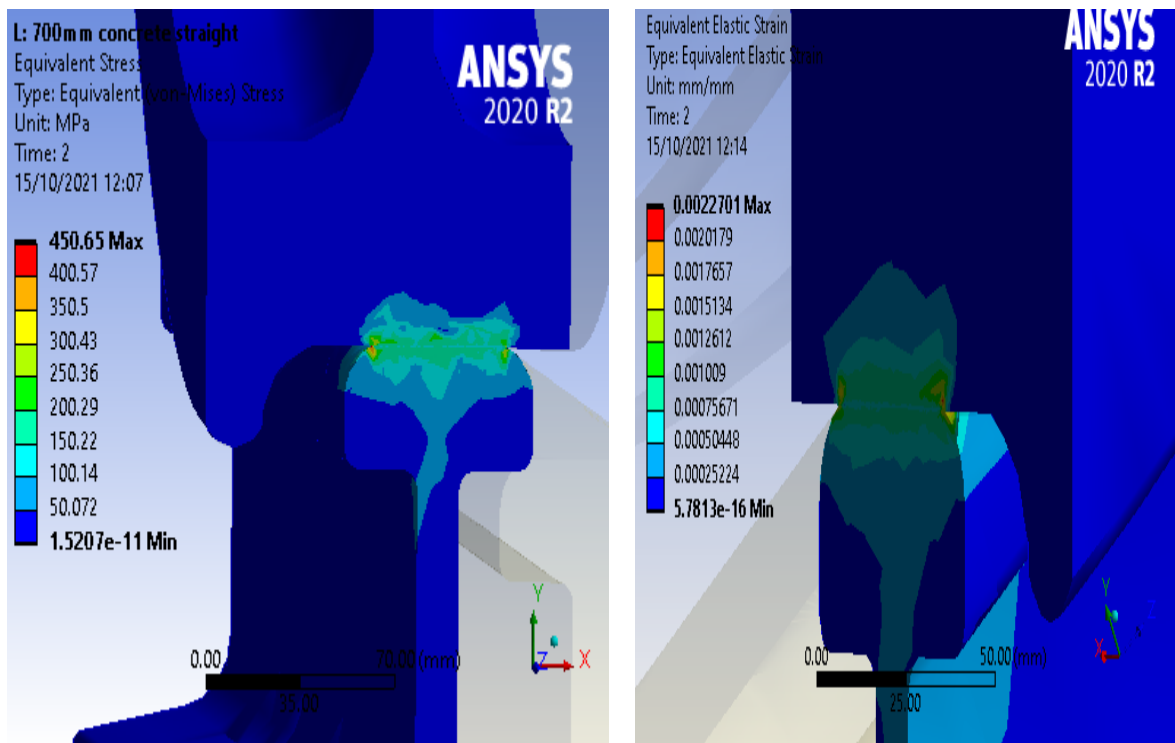


c)

Figure 4. 7; a) Steel-700mm, straight track-a) Von mises stress, b) equivalent elastic strain, c) total deformation

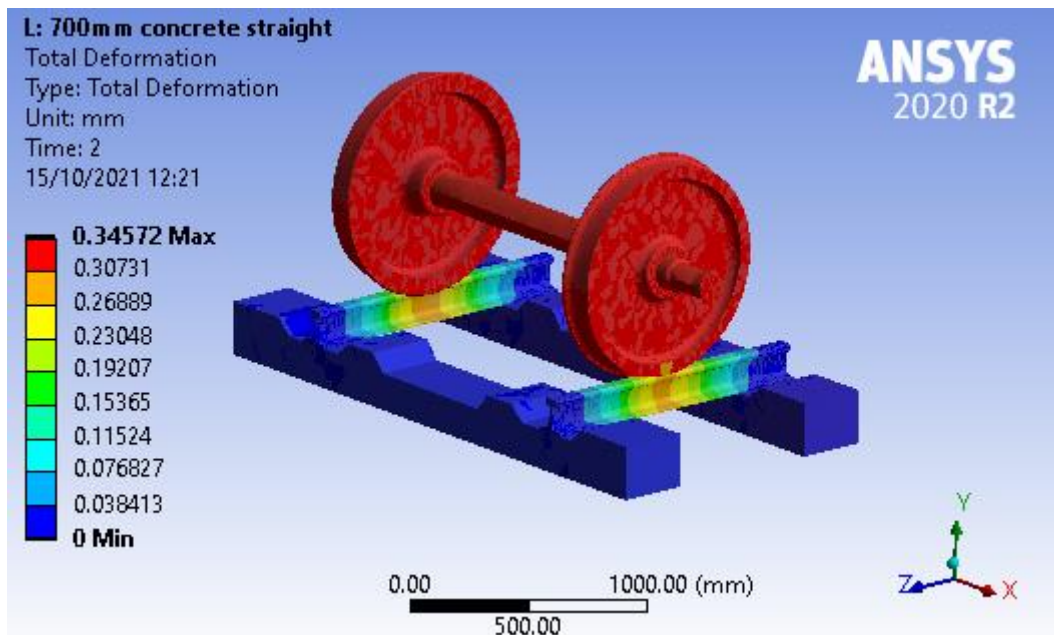
Using structural steel with higher MoE than concrete yields an increase in contact stress, strain but yields a 22.84% decrease in total deformation.

b) Concrete sleeper material



a)

b)



c)

Figure 4. 8; Concrete -700mm, straight track a) Von mises stress, b) equivalent elastic strain, c) total deformation

Using timber (F34) with lower MoE than concrete yields a 6.53% increase in total deformation but results indicate 0.89%, 0.26% and 0.5% decrease in contact stress and elastic strain

d) Timber (F34) sleeper material

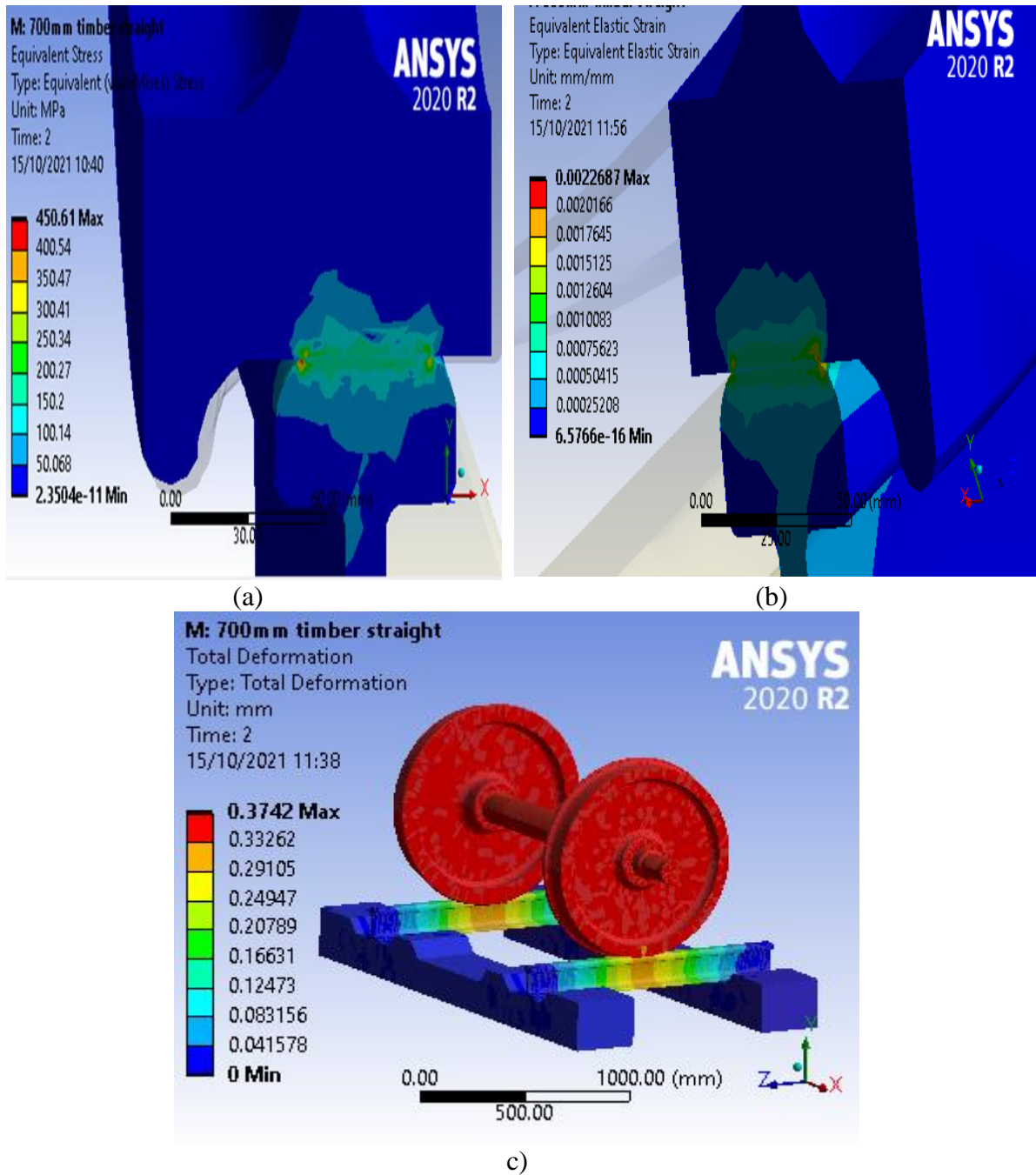


Figure 4. 9; Timber- 700mm, straight track a) Von mises stress, b) equivalent elastic strain, c) total deformation

4.1.4 500 mm sleeper spacing, curved track

a) Structural steel sleeper material

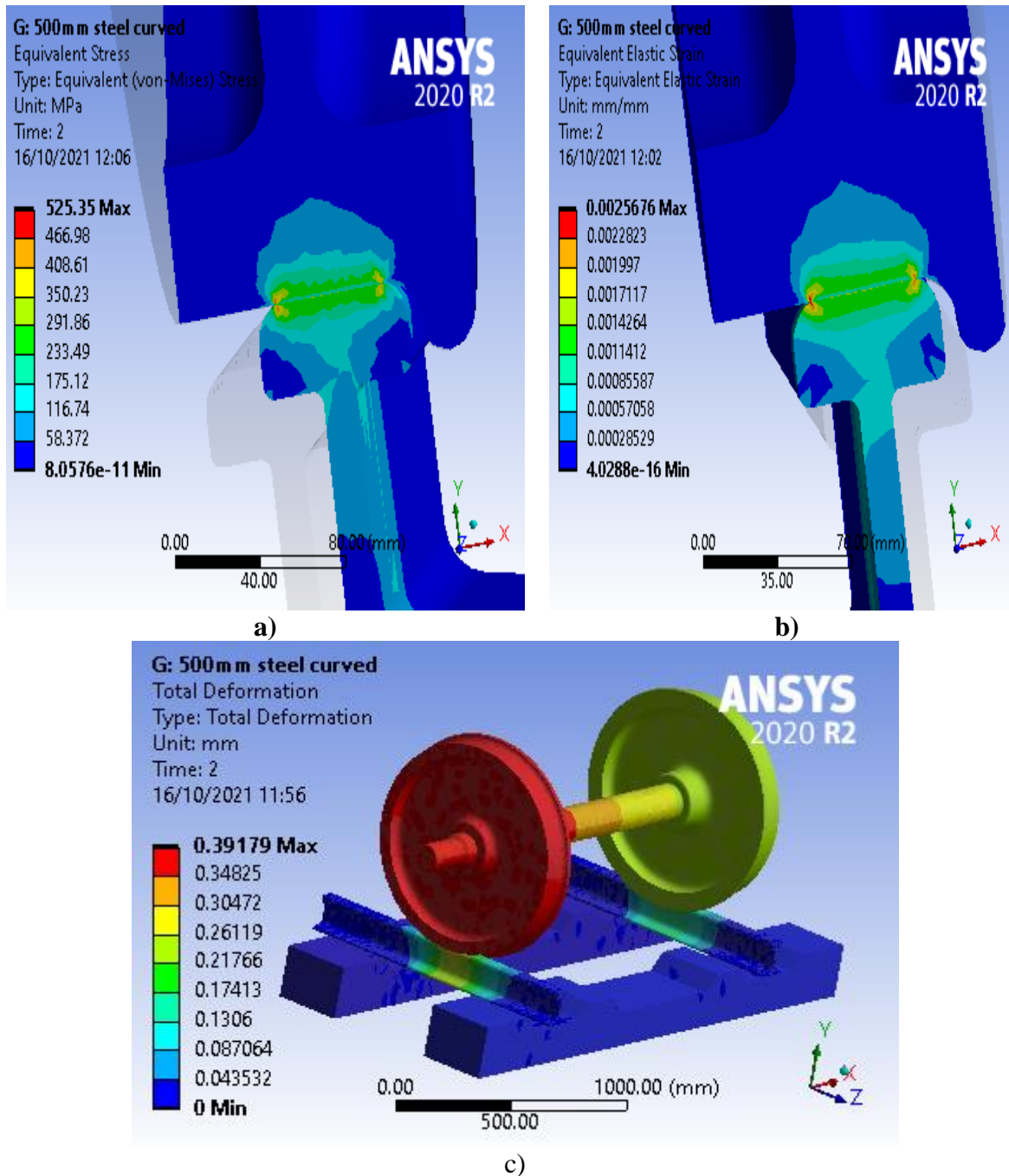


Figure 4. 10; Steel-500 mm, curved track a) Von mises stress, b) equivalent elastic strain,

c) total deformation

Using structural steel sleeper material with higher MoE than concrete yields a 0.07%, 0.17% and 12.58% reduction in contact stress and elastic strain and total deformation respectively. However, there is 0.52% and increase in contact pressure.

b) Concrete sleeper material

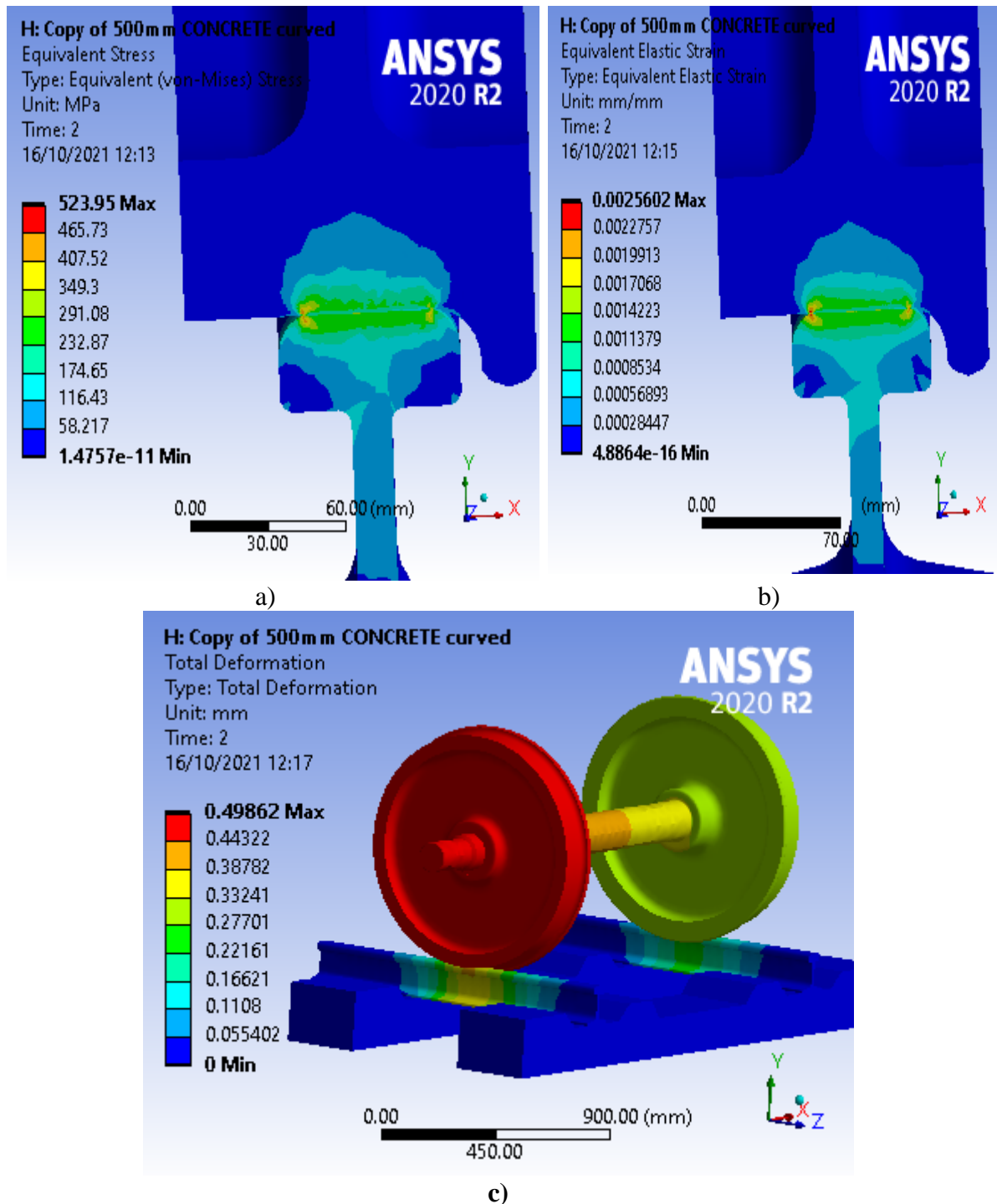


Figure 4. 11; Concrete-500 mm, curved track a) Von mises stress, b) equivalent elastic strain, c) total deformation

Using timber (F34) with lower MoE than concrete yields a 0.11% decrease in contact pressure but results indicate a 1.38%, 1.6% and 5.03% increase in contact stress, pressure and the total deformation respectively.

e) Timber (F34) sleeper material

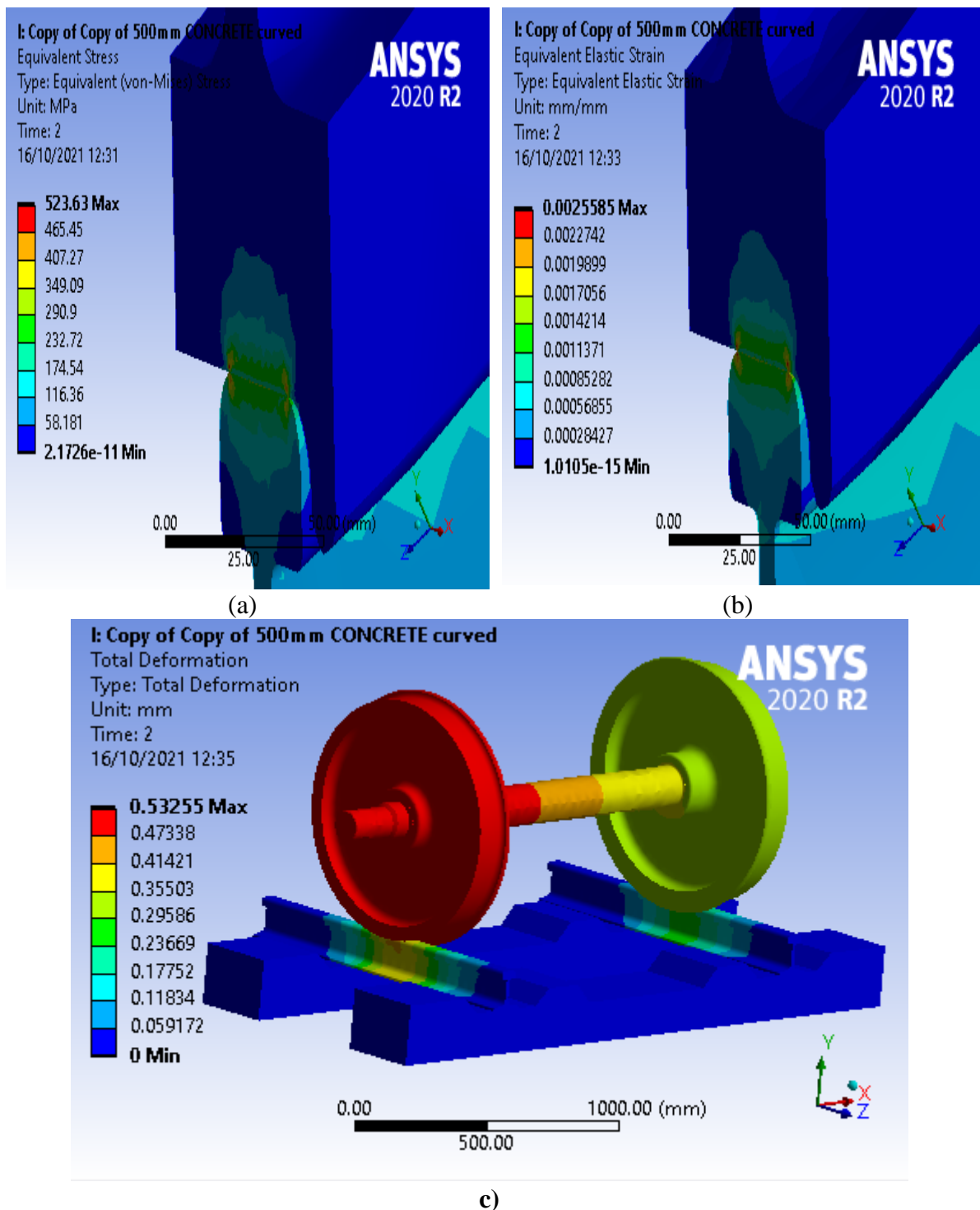


Figure 4. 12; Timber-500 mm, curved track a) Von mises stress, b) equivalent elastic strain, c) total deformation.

4.1.5 600 mm sleeper spacing, curved track

b) Structural steel sleeper material

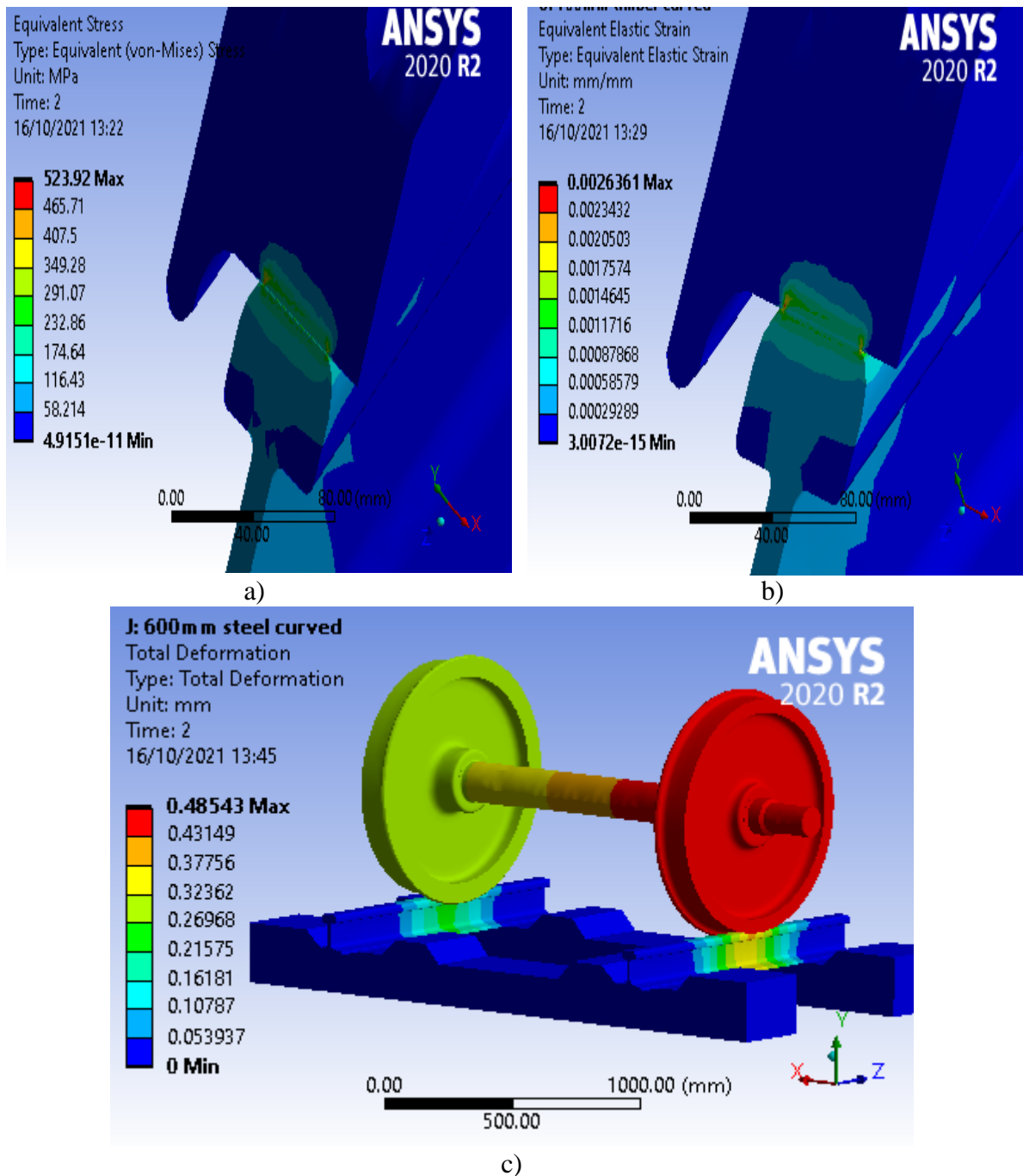


Figure 4. 13; Steel-600 mm, curved track a) Von mises stress, b) equivalent elastic strain,

c) total deformation

By using structural steel instead of concrete sleepers, a percentage reduction of 0.35%, 0.37% and 12% in contact stress and elastic strain and total deformation respectively was achieved. However, a percentage increase of 0.7% was realised in the contact pressure.

b) Concrete sleeper material

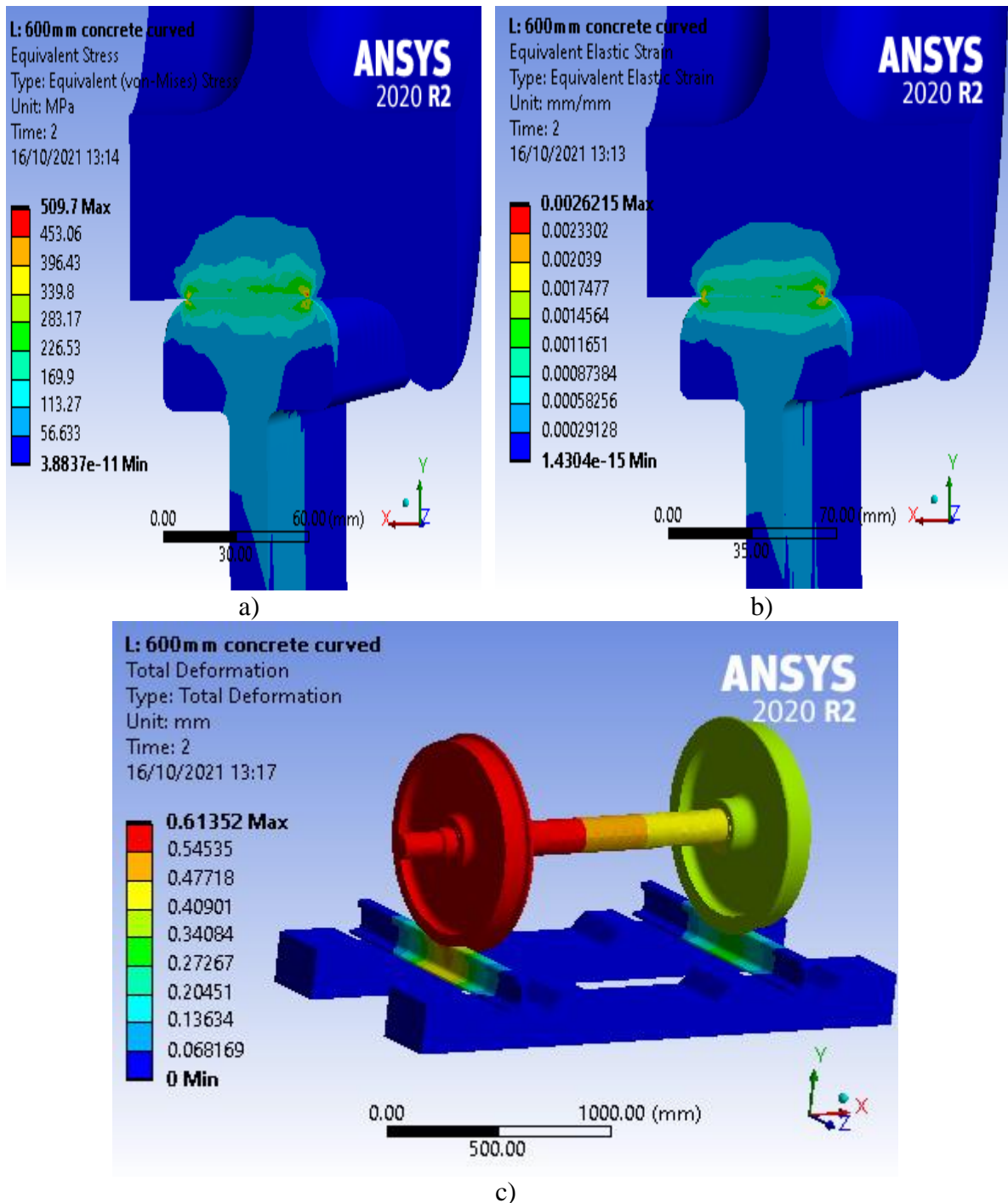


Figure 4. 14; Concrete-600 mm, curved track a) Von mises stress, b) equivalent elastic strain, c) total deformation

Using timber (F34) with a lower MoE than concrete causes a 0.13% reduction in the contact pressure while 4.8% increase in total deformation and 0.3% increase in contact stress and elastic strain is realised.

c) Timber (F34) sleeper material

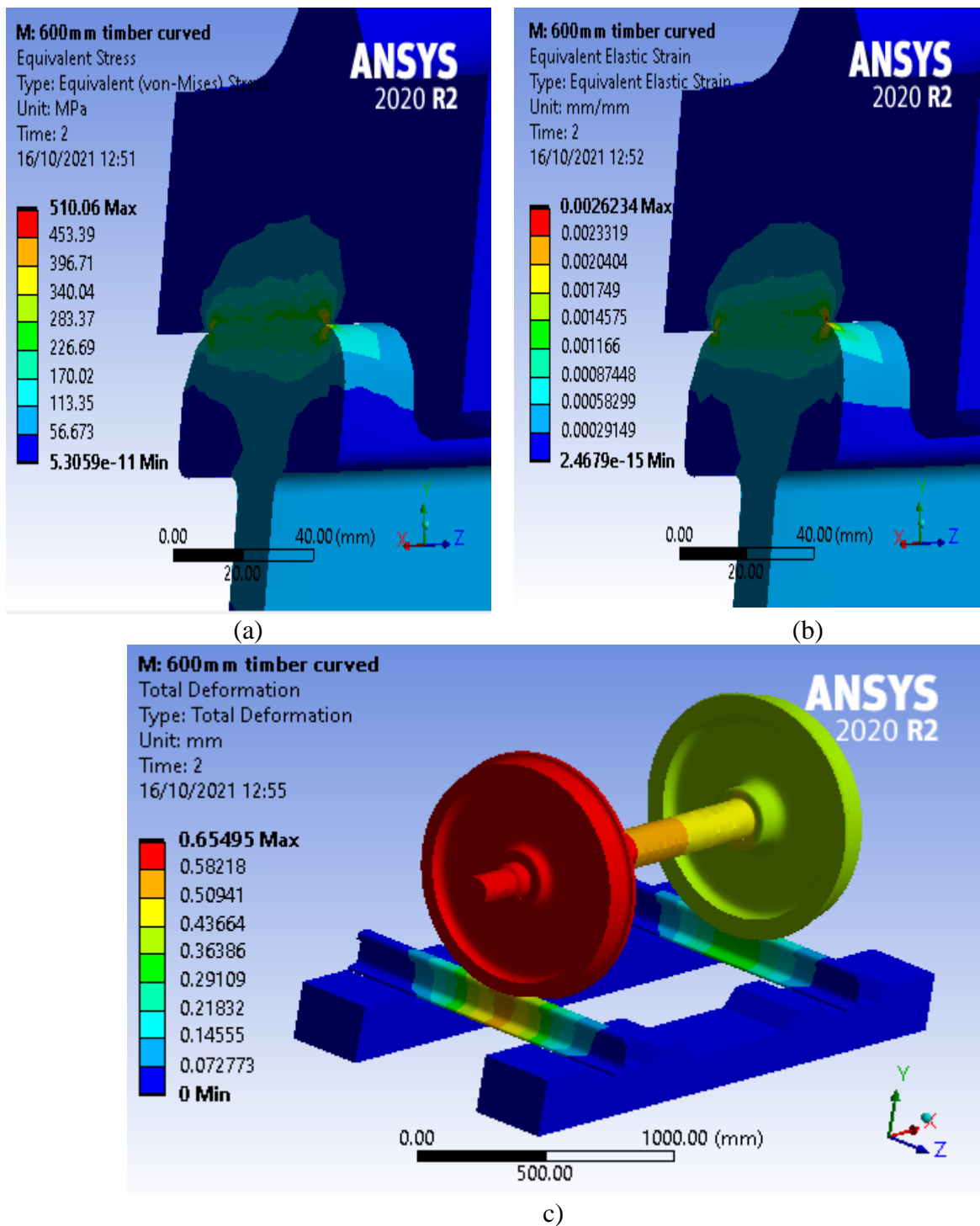


Figure 4. 15; Timber -600 mm, curved track a) Von mises stress, b) equivalent elastic strain, c) total deformation

4.1.6 700 mm sleeper spacing, curved track

a) Structural steel sleeper material

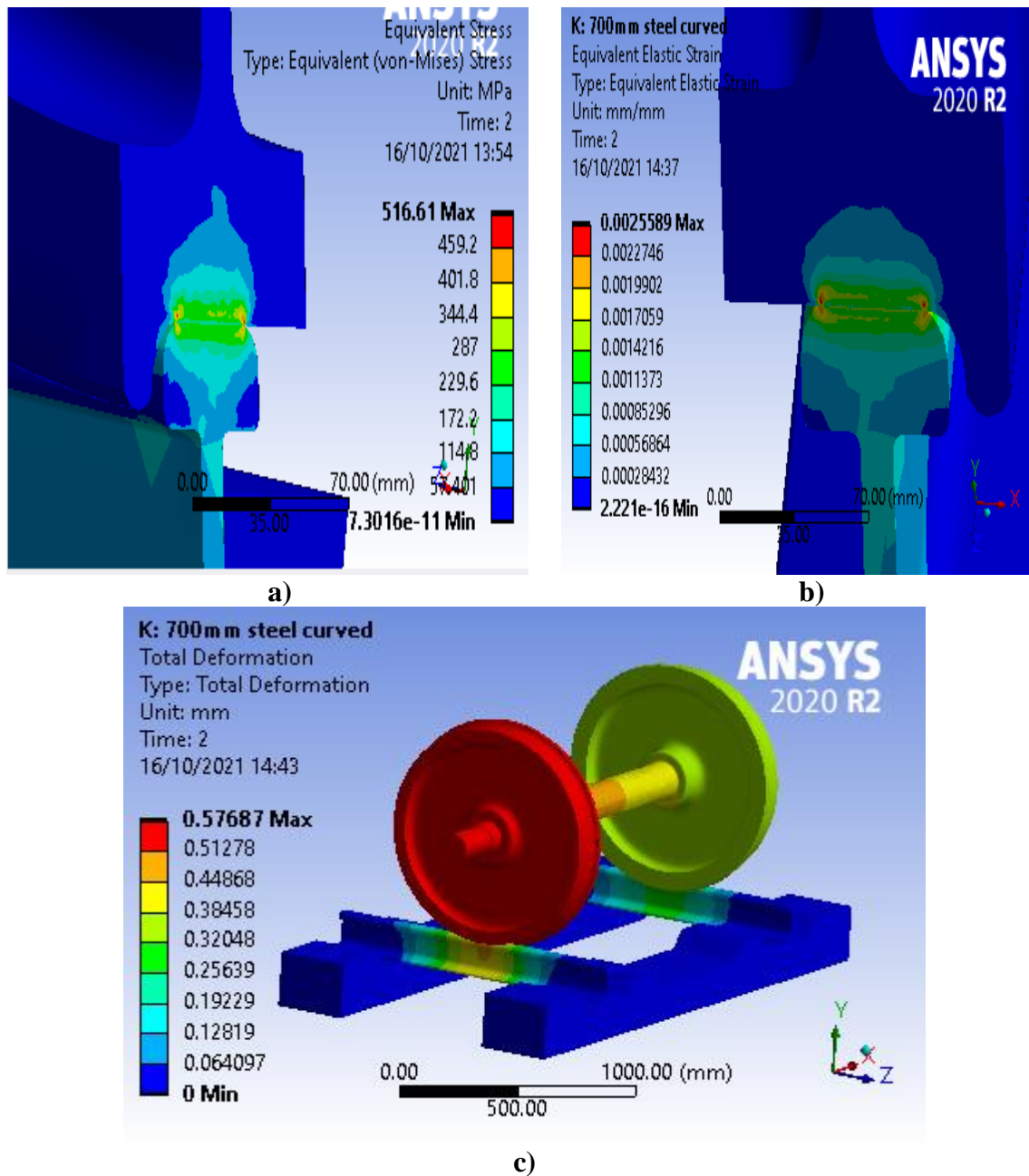


Figure 4. 16; Steel-700 mm, curved track a) Von mises stress, b) equivalent elastic strain, c) total deformation

By using structural steel instead of concrete sleeper materials, a percentage increment of 0.29% and 0.17% was observed in contact stress and pressure respectively but a percentage reduction of 19% in total deformation was achieved.

b) Concrete sleeper material

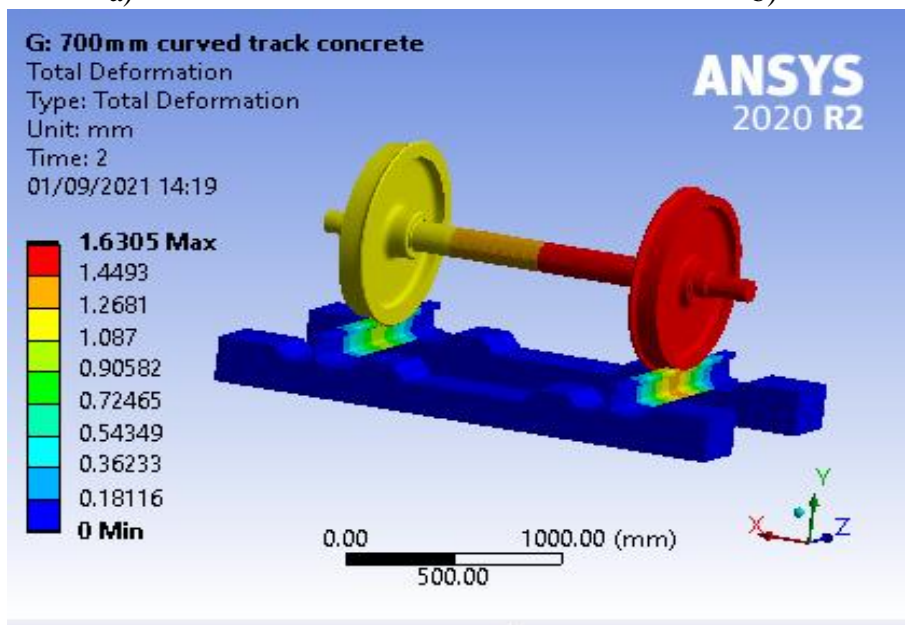
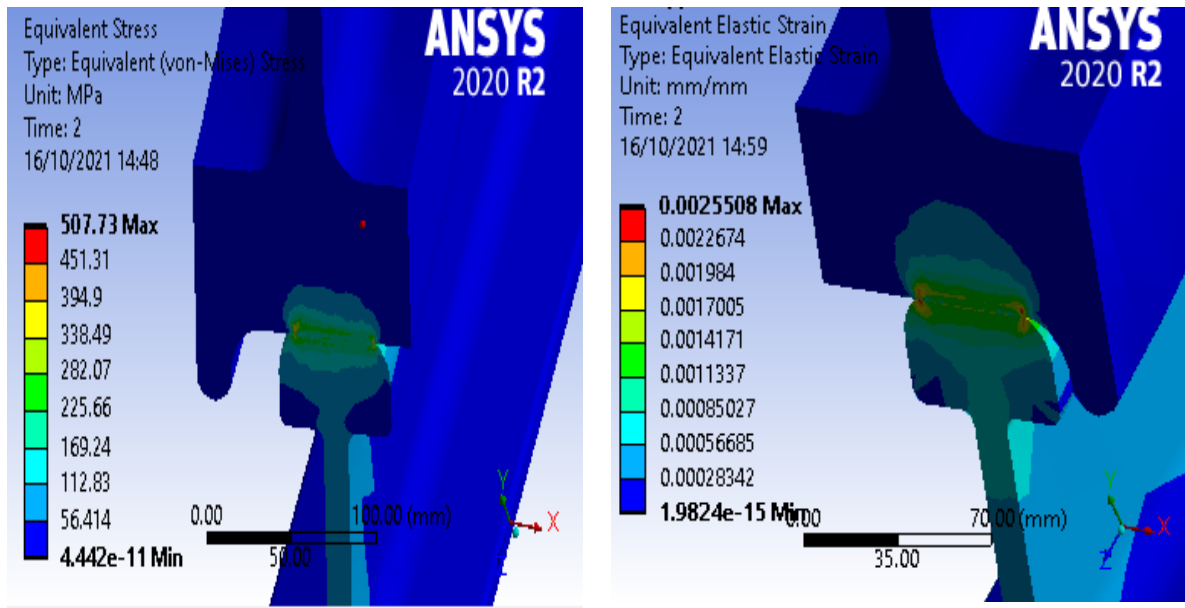


Figure 4. 17; Concrete -700 mm, curved track- a) Von mises stress, b) equivalent elastic strain, c) total deformation

Using timber (F34) with a lower MoE than concrete causes a 0.08% and 0.13% reduction in the contact stress and elastic strain respectively but yields a 0.03% and 5.7% increase in contact pressure and deformation respectively.

c) **Timber (F34) sleeper material**

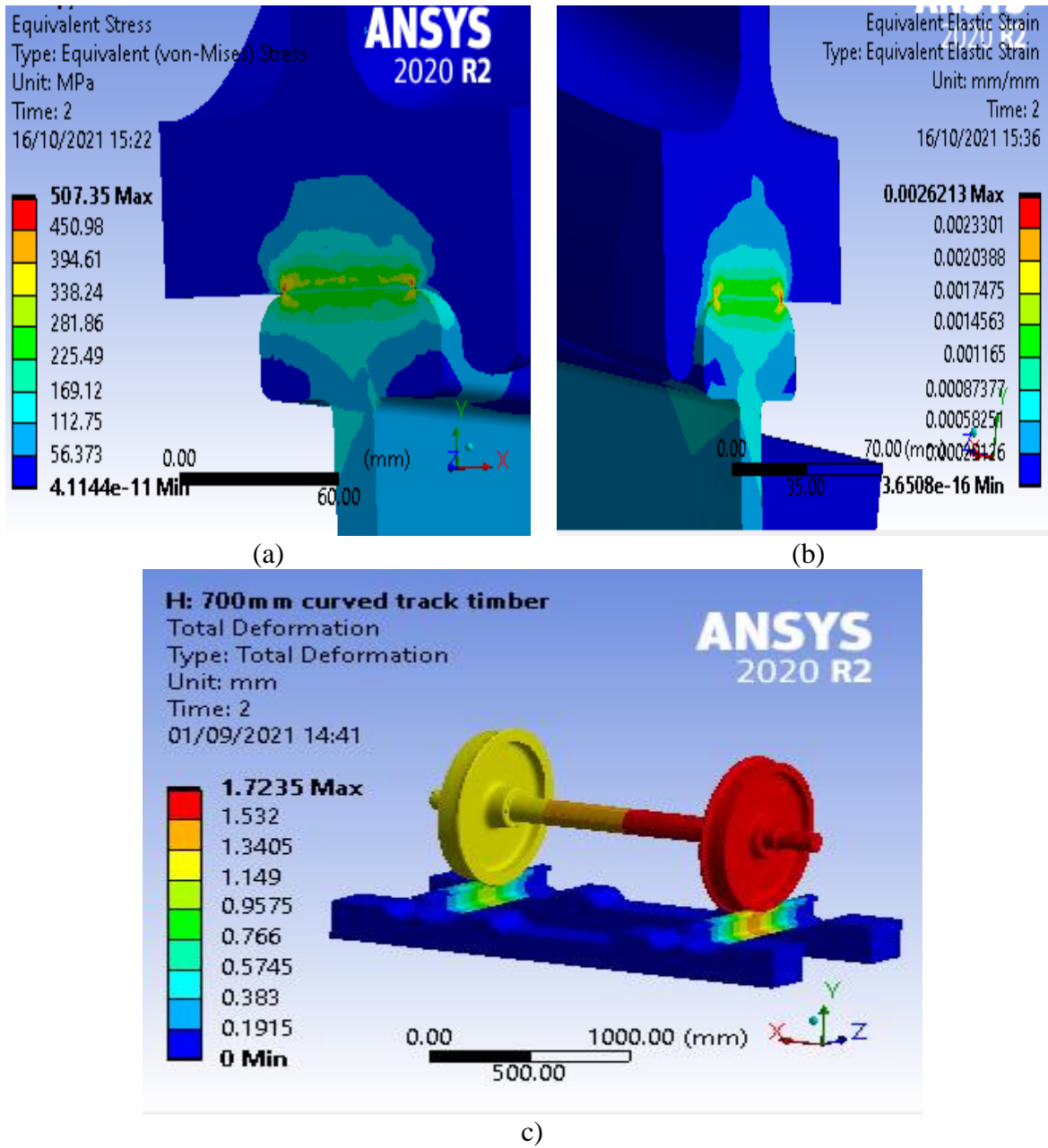


Figure 4. 18; Timber-700 mm, curved track -a) Von mises stress, b) equivalent elastic strain, c) total deformation

4.1.7 Summary of the results for the straight track

Table 4. 1; contact stresses, strain, pressure, and force on a straight track

Sleeper material	Equivalent strain (mm/mm)	Equivalent stress (MPa)	Contact pressure (MPa)	Sliding distance (mm)	Contact force (N)	Deformation (mm)
500 mm						
Timber (f34) grade	0.0023328	452.45	1452.2	0.03021	87818	0.26035
concrete	0.0023496	455.04	1453.5	0.028217	87938	0.24129
Structural steel	0.0023526	455.49	1460.7	0.01817	87995	0.18254
600 mm						
Timber (f34) grade	0.0023052	452.2	1385.8	0.032872	87499	0.31336
concrete	0.0023058	452.48	1392.5	0.030649	87502	0.2911
Structural steel	0.0023099	454.65	1393.1	0.021005	87521	0.22412
700 mm						
Timber (f34) grade	0.0022687	450.61	1366.57	0.039913	87442	0.3742
concrete	0.0022701	450.65	1373.4	0.037035	87464	0.34572
Structural steel	0.0022815	451.87	1378.88	0.023929	87510	0.26193

Using 500 mm instead of 600 mm yields higher values of contact stress, elastic strain, contact forces and pressure respectively for all sleeper materials. It also results in lower values of total deformation and sliding distance respectively for all sleeper materials (moduli of elasticity).

Using 700 mm sleeper spacing instead of 600 mm sleeper spacing yields lower values of contact stress, elastic strain, and contact forces and pressure respectively for all sleeper materials. Also replacing 600 mm with 700 mm sleeper spacing, results in higher values of total deformation and sliding distance for all sleeper materials (moduli of elasticity).

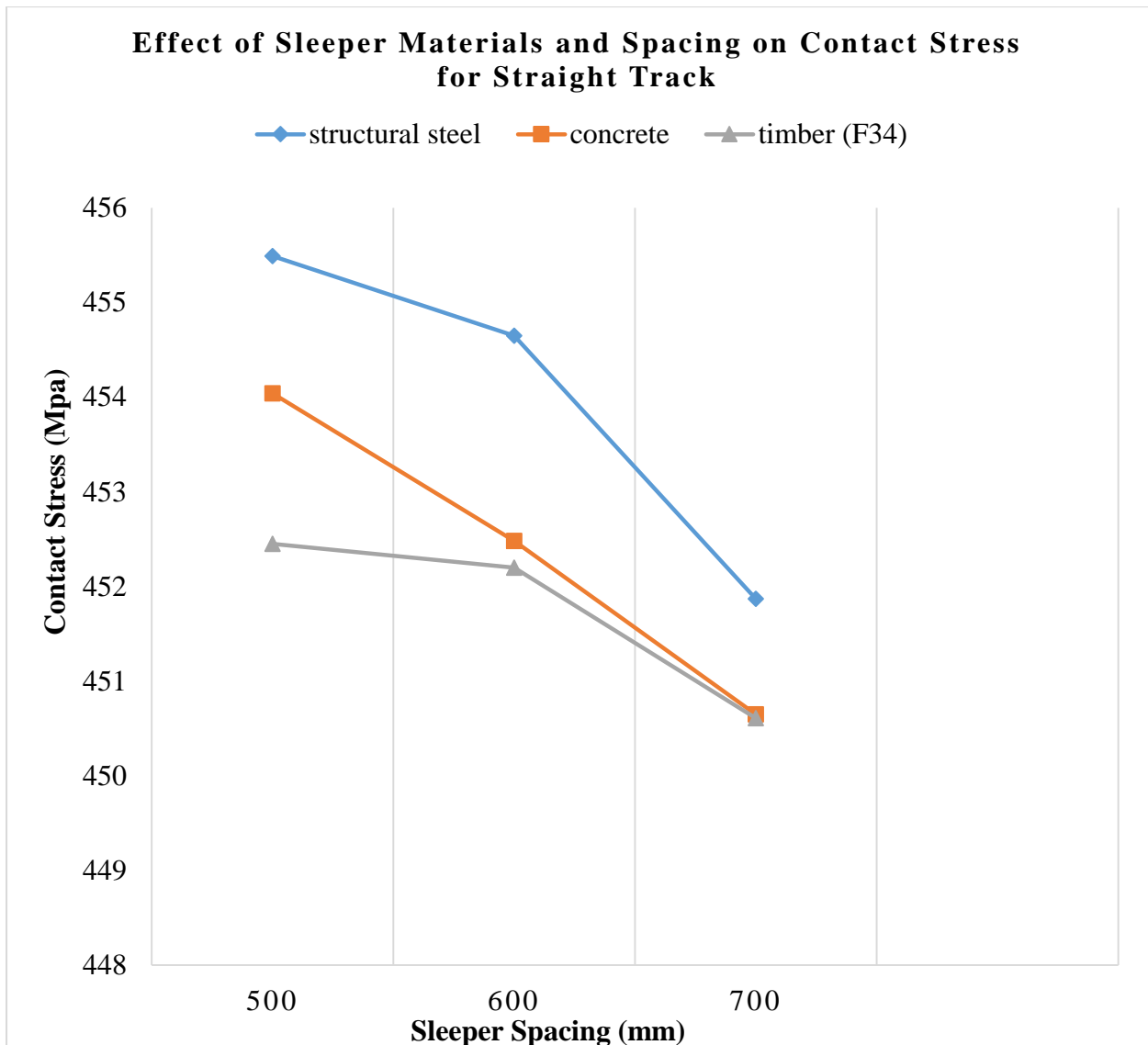


Figure 4. 19: Effect of sleeper material and spacing on contact stress for straight track

From the figure 4.19 above, it is observed that the contact stress generally reduces with increase in sleeper spacing and increases with increase in sleeper MoE. The modulus of elasticity of the sleepers has an influence on the track stiffness; the lower the sleeper MoE, the lower the track stiffness and the reverse is true. Therefore this implies that a track with wooden sleepers will have lowest track stiffness compared to one with concrete and steel mainly because of the *vertical resilient compression of the wood ties*. The contact stress and elastic strain for the straight track is observed to reduce with an increase in sleeper spacing because it yields low global track stiffness which greatly increases the rail deflection allowing more sleepers to share the wheel load which can lead to better load or stress distribution from the track down to the track components by the sleepers [57] and the reverse is true.

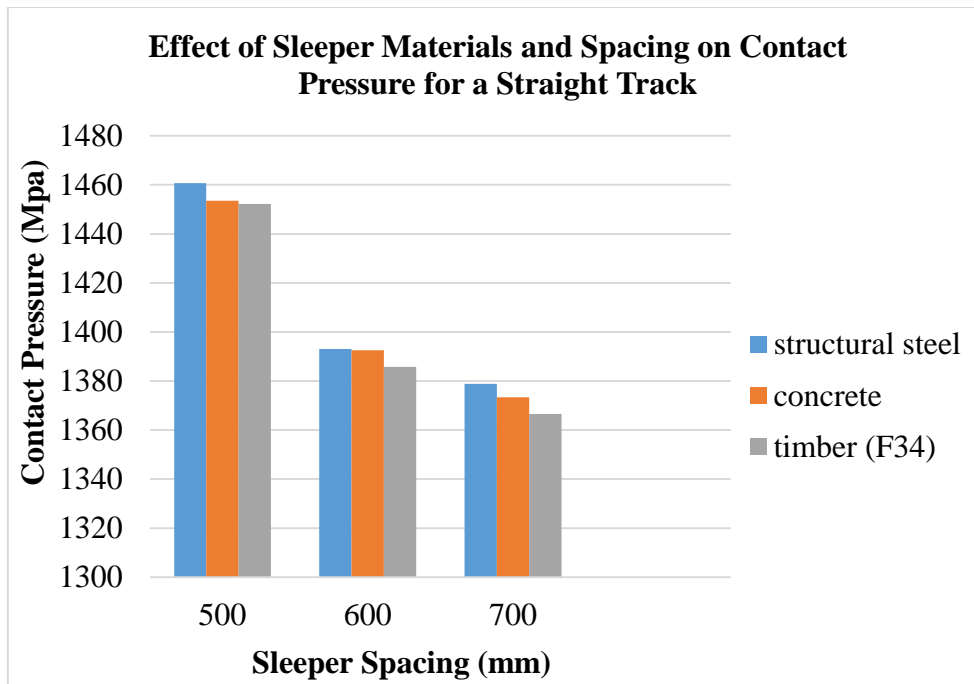


Figure 4. 20: Effect of sleeper material and spacing on contact pressure in straight track

It is observed from the figure 4.20 that the contact pressure increases with increase in sleeper MoE and with reduction in sleeper spacing. That is to say; it is highest with structural steel and lowest with timber (F34) sleeper materials and highest with 500 mm and lowest with 700 mm sleeper spacing. The high contact pressures are a result of high track stiffness which is caused by high sleeper modulus and small sleeper spacing and results in a small wheel rail contact area due to small rail displacements. On the other hand, the low contact pressures are a result of low track stiffness caused by low sleeper modulus and large intervals between sleepers resulting in a larger wheel rail contact area due to large rail displacements [55]. Therefore, since the contact pressure is inversely proportional to the contact area, the results above are logical.

The contact forces as shown in figure 4.21 below are observed to increase with increase in sleeper MoE and decrease with increase in sleeper spacing, that is to say; they are highest with the structural steel and lowest with timber (F34) sleeper materials and highest with 500 mm and lowest with 700 mm sleeper spacing. This is because, both low MoE and large sleeper spacing contribute to a low global track stiffness which results in low wheel rail contact forces. On the contrary, both high MoE and small sleeper spacing contribute to a high global track stiffness which is a reason for high wheel rail contact forces [13].

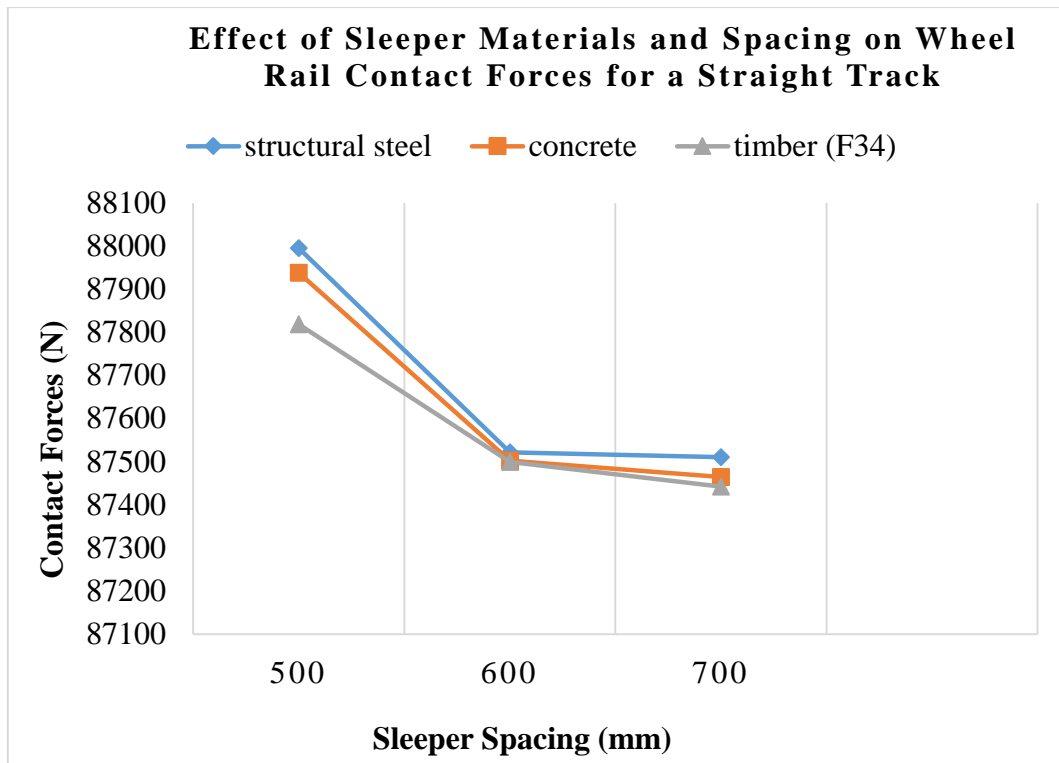


Figure 4. 21: effect of sleeper material and spacing on contact force for a straight track

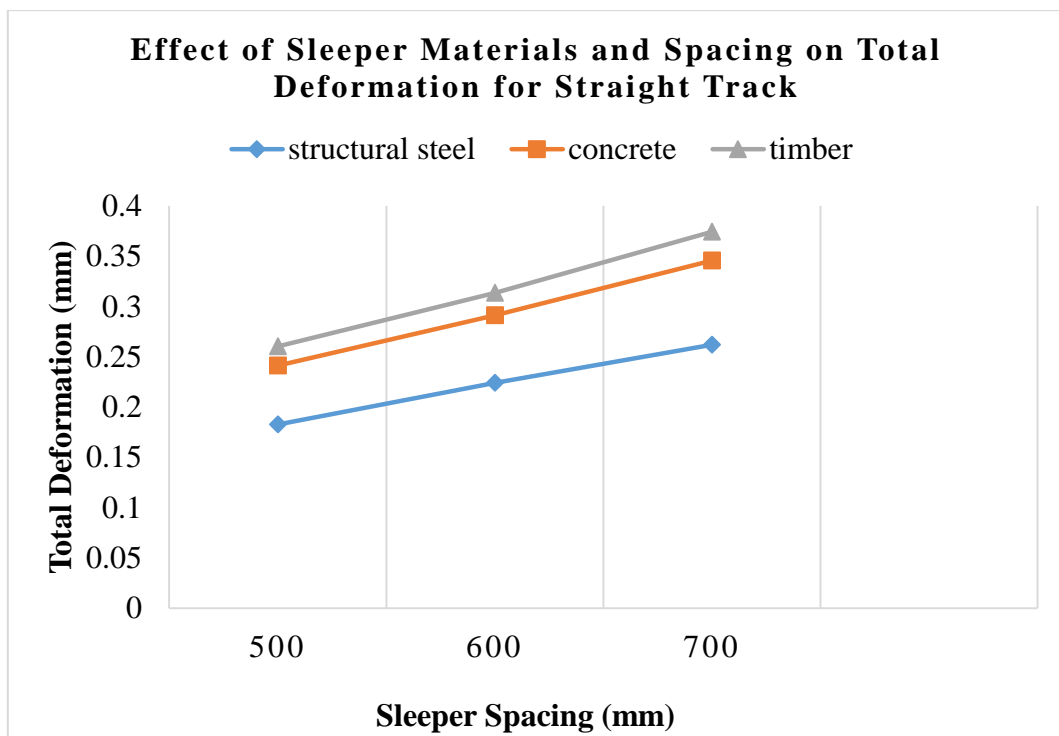


Figure 4. 22: Effect of sleeper material and spacing on total deformation for a straight track

From the figure 4.22 above, the total deformation is observed to increase with a decrease in sleeper MoE and increase with an increase in sleeper spacing i.e. it is highest with the timber (F34) and lowest with structural steel sleeper materials and highest with 700 mm and lowest

with 500 mm sleeper spacing. The low values of total deformation are as a result of high track stiffness which avails enough track resistance to applied loads resulting in reduced track deflection hence reducing deterioration of the track structure and the wheels in contact as discussed by Martin et al. [11]. The high values of total deformation are a result of low global track stiffness which causes large rail deflection leading to deterioration of the track components and rolling stock components [58]. Therefore the results of total deformation and sliding distance are in agreement with Martin et al. [11].

4.1.8 Summary of results for a curved track.

Table 4. 2: contact stress, pressure, strain, and forces

Sleeper material	Equivalent strain (mm/mm)	Equivalent stress (MPa)	Contact pressure (MPa)	Sliding distance (mm)	Deformation (mm)	Contact forces $\times 10^5 \text{N}$	
						left	right
500 mm							
Timber (f34) grade	0.0025585	523.63	1600.5	0.11743	1.1144	1.302	2.0191
concrete	0.0025602	523.95	1602.8	0.10761	1.061	1.2993	2.022
Structural steel	0.0025676	525.35	1611.7	0.067304	0.92755	1.2912	2.0316
600 mm							
Timber (f34) grade	0.0026234	510.06	1556.6	0.12403	1.37	1.3116	2.0067
concrete	0.002615	509.7	1558.7	0.11372	1.3074	1.3082	2.0101
Structural steel	0.0026361	523.92	1560.9	0.075577	1.1506	1.2978	2.0203
700 mm							
Timber (f34) grade	0.0026213	507.35	1531.8	0.13815	1.7235	1.3197	2.0074
concrete	0.0025508	507.73	1539.9	0.12436	1.6305	1.3083	2.0192
Structural steel	0.0025589	516.61	1543.8	0.077469	1.3707	1.3047	2.0274

Using 500 mm sleeper spacing in comparison to 600 mm spacing yields higher values of contact stress, strain and contact pressure in addition to lower values of total deformation and

sliding distance. Furthermore, the values of contact forces on the left hand are lower and those on the right side are higher. Using 700 mm sleeper spacing instead of 600 mm spacing yields lower contact stresses, contact pressure and forces on the right side. However, higher values of total deformation, contact forces on the left side, and sliding distance are noted.

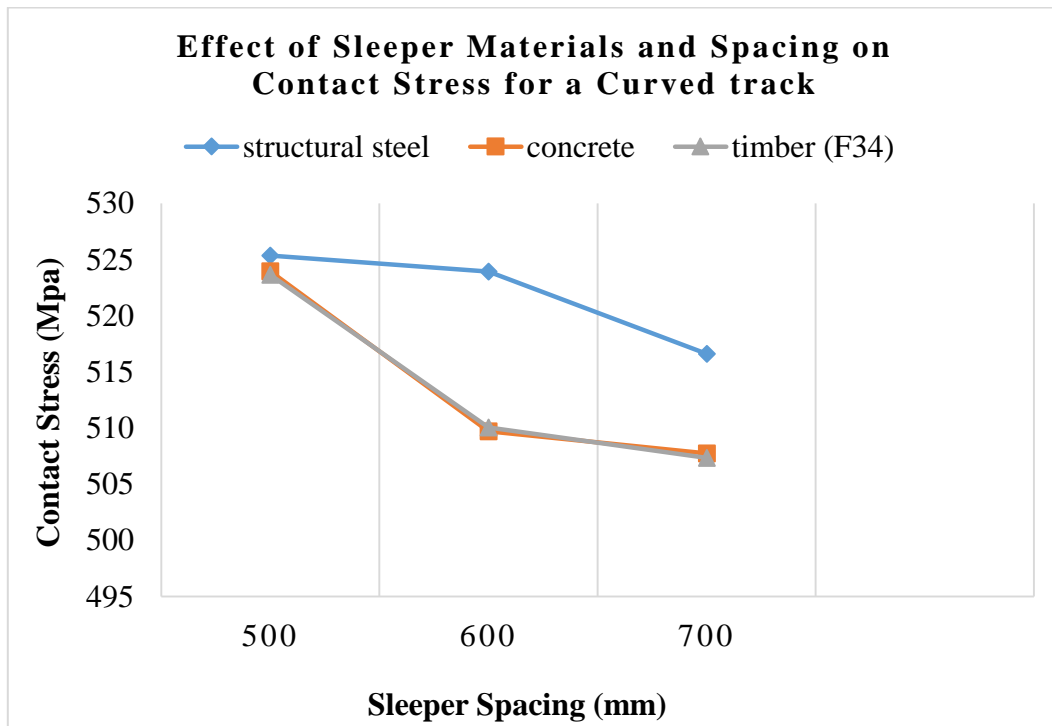


Figure 4. 23: Effect of sleeper material and spacing on contact stress in a curve

From the figure 4.23 above, the contact stresses increase with an increase in sleeper modulus of elasticity and decrease with increase in sleeper spacing. They are observed to be highest with structural steel and lowest with timber (F34) sleeper materials and are highest with 500 mm and lowest with 700 mm sleeper spacing. The low values of contact stress are a result of low track stiffness which is caused by low sleeper modulus and large sleeper spacing. The low track stiffness results in large rail displacements causing many sleepers to share the load or stress from the wheels hence resulting in better distribution of loads from the wheel rail contact to the substructure by the sleepers.

On the contrary, the high values of contact stress are a result of high track stiffness which is caused by high sleeper modulus and small sleeper spacing. The high track stiffness results in small rail displacements causing fewer sleepers to share the load or stress from the wheel rail contact and hence high contact stresses. The contact stresses for both concrete and timber (F34) are almost the same as observed in the figure 4.23. This makes sense considering the conclusion

made by Shokrieh [41] that the sleeper material generally has a small effect on the contact stress because it generally has little effect on the sleeper performance.

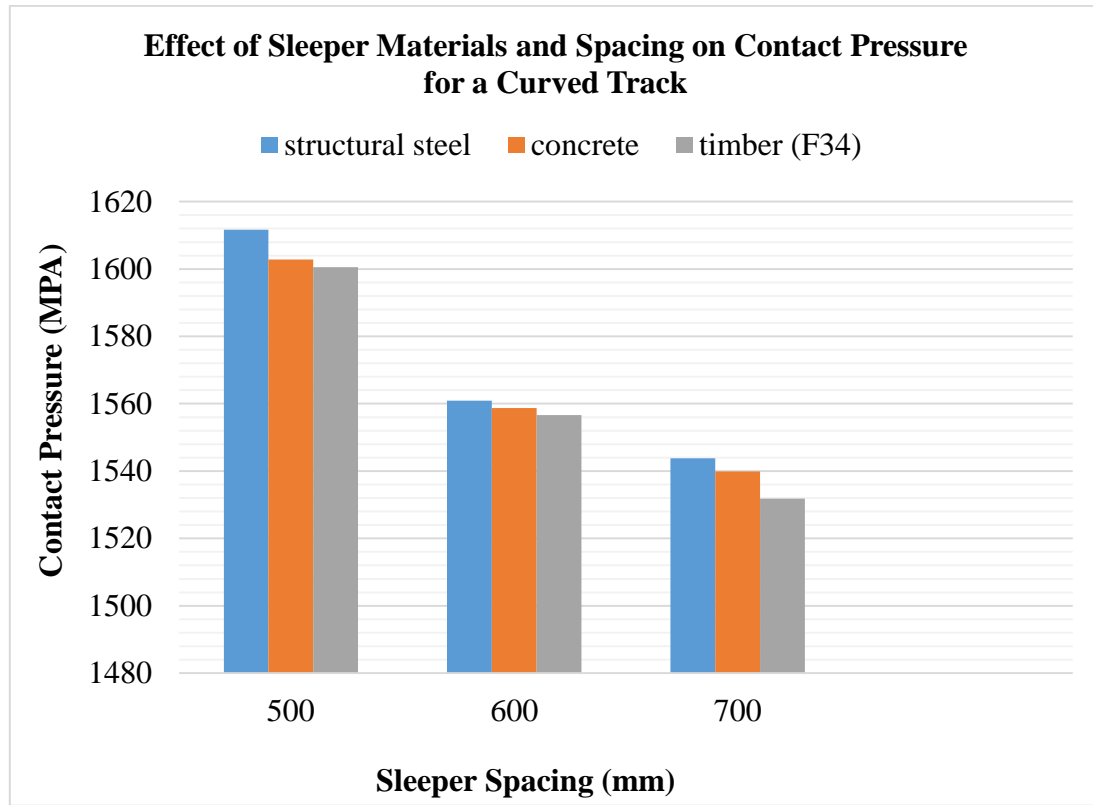


Figure 4. 24: Effect of sleeper material and spacing for a curved track

The contact pressure is observed to generally decrease with increase in sleeper spacing and a decrease in sleeper modulus of elasticity as shown in the figure 4.24. It is seen to be highest with the structural sleeper material which has the highest MoE and generally lowest with the timber (F34) sleeper material which has the lowest MoE. It is also highest with 500 mm sleeper spacing and lowest with 700 mm sleeper spacing. This is because the track stiffness increases with increase in sleeper modulus and with reduction in the sleeper spacing. A high track stiffness causes a small contact area due to small rail deflection while a low track stiffness yields a large contact area due to large rail deflection. Therefore, since the contact pressure is inversely proportional to the contact area, the results above are in agreement with Wehbi [55].

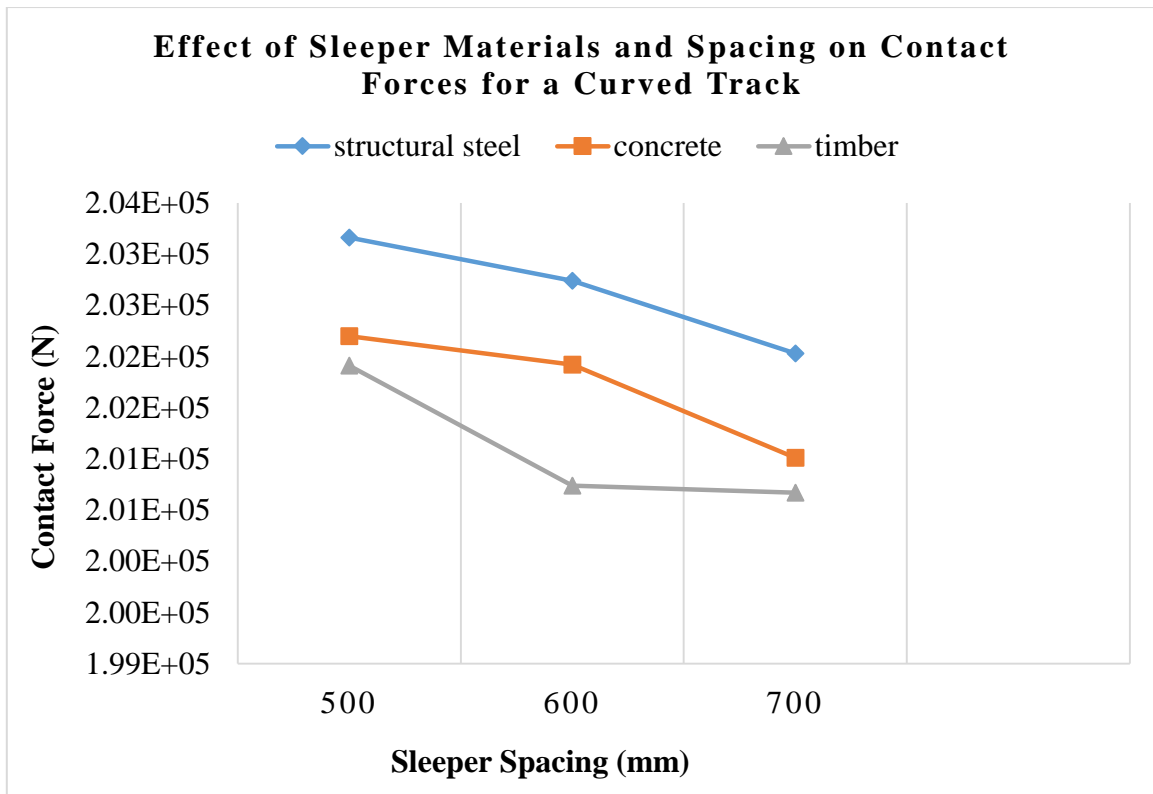


Figure 4. 25; Effect of sleeper material and spacing on contact force for a curved track

The contact forces are observed to increase with increase in sleeper MoE and decrease with increase in sleeper spacing as shown in the figure above.

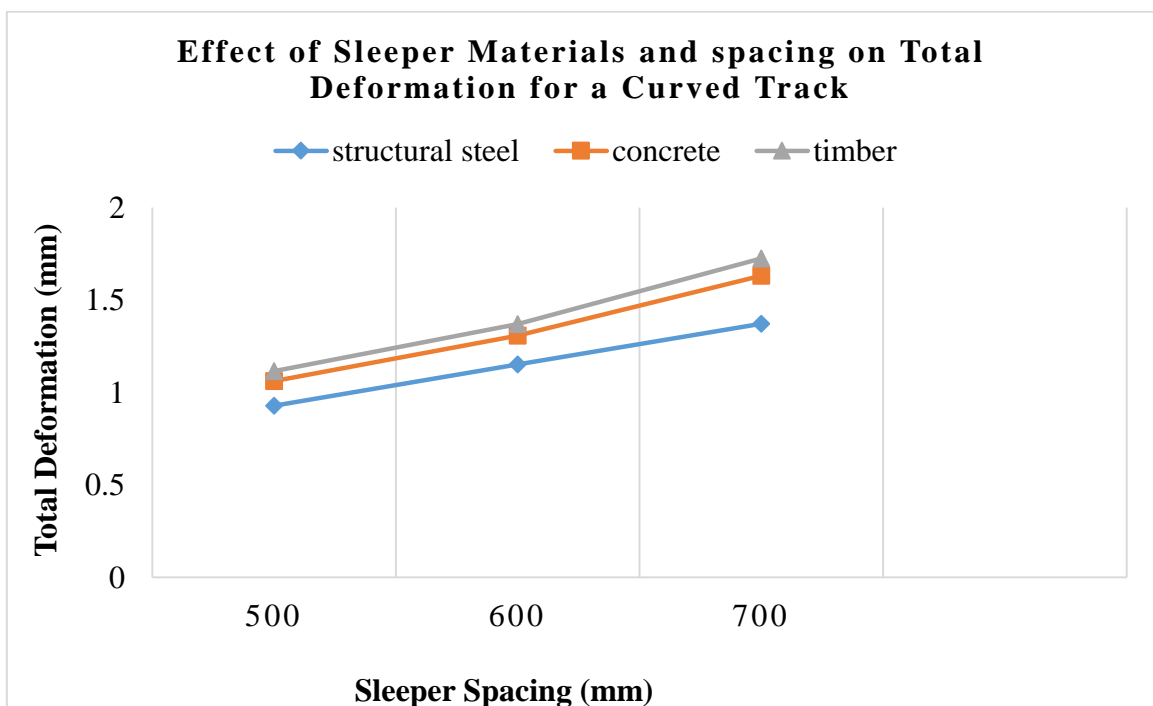


Figure 4. 26: Effect of sleeper material and spacing on total deformation for a curved track

From the figure above, the total deformation is observed to increase with a decrease in sleeper MoE and increase with an increase in sleeper spacing i.e. it is highest with the timber sleeper material that has the lowest MoE and with 700 mm sleeper spacing. The lower values of total deformation and sliding distance achieved when the structural steel sleeper material and 500 mm sleeper spacing are as a result of increased track stiffness which is beneficial because it avails enough track resistance to applied loads resulting in reduced track deflection which reduces deterioration of the track structure and the wheels in contact as discussed by Martin et al. [11] .

4.1.9 Validation of the model

The model was validated using analytical method of Hertzian contact theory. The normal pressure at any point of the contact ellipse of two arbitrarily curved bodies is given by the Hertzian theory;

$$P(x, y) = P_0 \sqrt{1 - \frac{x^2}{a^2} - \frac{y^2}{b^2}},$$

Where; P_0 , is the maximum contact pressure, a , is semi- major axis and b , is semi- minor axis of the contact ellipse.

$$P_0 = \frac{3F}{2\pi ab} \quad (4. 1)$$

In order to determine the maximum pressure, the values of ‘a’ and ‘b’ should be known and the formulae to determine them are as follows according to Timoshenko³²;

$$b = n \left[\frac{3F}{4E^*} \frac{1}{A+B} \right]^{1/3} \quad (4. 2)$$

$$a = m \left[\frac{3F}{4E^*} \frac{1}{A+B} \right]^{1/3} \quad (4. 3)$$

Where;

$$\frac{1}{E^*} = \frac{1-v_1^2}{E_1} + \frac{1-v_2^2}{E_2} \quad (4. 4)$$

E_1, E_2, v_1 , and v_2 are elastic modules and Poisson ratios of the bodies in contact respectively, m and n are non-dimensional coefficients tabulated as a function of the ratio, $g = n/m$ or the angle θ .

$$n = 3 \times 10^{-5}\theta^2 + 0.0045\theta + 0.334 \quad (4.5)$$

$$m = 62.19\theta^{-0.914} \quad (4.6)$$

The table below represents the non-dimensional values of m and n for different values of angle θ ;

Table 4. 3 values of m and n for various values of θ

θ°	30	35	40	45	50	55	60	65	70	75	80	85	90
m	2.731	2.397	2.136	1.926	1.754	1.611	1.486	1.378	1.284	1.202	1.128	1.061	1
n	0.493	0.530	0.567	0.604	0.641	0.678	0.717	0.759	0.802	0.846	0.893	0.944	1

(Source: theory of elasticity by Timoshenko and J. N. Goodier) [1].

The values of the wheel railway parameters used in the study are listed below

Table 4. 4 Parameters and properties of the wheel rail system used

Profile parameters	Value (mm)
Rolling radius of wheel, R_2^r	460
Rolling radius of rail, R_1^r	∞
Radius of curvature of wheel, R_2^t	330
Radius of curvature of rail, R_1^t	300mm
Wheel profile taper, ψ	$1/5$

By substituting the above parameters in equations (2.4) and (2.5), and (2.6);

$$(A + B) = \left[\frac{1}{\infty} + \frac{1}{300} + \frac{1}{460} + \frac{1}{330} \right] = 0.0042688/mm$$

$$(B - A) = \frac{1}{2} \left[\left[\frac{1}{\infty} - \frac{1}{300} \right]^2 + \left[\frac{1}{460} - \frac{1}{330} \right]^2 + 2 \left[\frac{1}{\infty} - \frac{1}{300} \right] \left[\frac{1}{460} - \frac{1}{330} \right] \cos(2 \times 11.3099) \right]^{\frac{1}{2}}$$

$$= 0.0020685/mm$$

$$\cos \theta = \frac{|B - A|}{A + B} = \frac{|0.0020685|}{0.0042688} = 0.48456$$

$$\theta^\circ = 61.02^\circ$$

From table 5, the values of m and n can be found by interpolation as shown below;

$$\theta_1 = 60^\circ, m_1 = 1.486, n_1 = 0.717 \text{ and } \theta_2 = 65^\circ, m_2 = 1.378, n_2 = 0.759$$

Solving for n;

$$\frac{n - n_1}{\theta - \theta_1} = \frac{n_2 - n_1}{\theta_2 - \theta_1} \rightarrow \frac{n - 0.717}{61.02 - 60} = \frac{0.759 - 0.717}{65 - 60}$$

$$n = 0.7256$$

The value of n can also be obtained by substituting the value of θ in equation (4.5)

Solving for m;

$$\frac{m - m_1}{\theta - \theta_1} = \frac{m_2 - m_1}{\theta_2 - \theta_1} \rightarrow \frac{m - 1.486}{61.02 - 60} = \frac{1.378 - 1.486}{65 - 60}$$

$$m = 1.464$$

Also the value of m can be got by using equation (4.6).

From equation (4.4), the value of E^* is 1.1456×10^{11} Pa and therefore, by substituting it in equations (4.2) and (4.3), the value of a, is 7.35 mm and b, is 3.64 mm.

And by substituting the value of a and b, and taking the value of the normal force as 83000N into equation (4.1) the maximum contact pressure is **1473MPa**.

Table 4. 5; Comparison between the analytical contact pressure and the FEA contact pressure

Sleeper spacing	FEA contact pressure (Mpa)	Percentage difference (%)
500		
Timber	1452.2	1.4
Concrete	1453.5	1.32
steel	1460.7	0.84
600		
Timber	1385.8	5.9
Concrete	1392.5	5.5
steel	1393.1	5.4
700		

Timber	1366.57	7.2
Concrete	1373.4	6.8
steel	1378.88	6.4

The percentage differences between the analytical and FEA contact pressure for all sleeper materials and spacing are not so big. The differences could be due to the effect of sleeper spacing and material which are not included in calculating the maximum contact pressure using the Hertzian contact theory.

4.2 Selection of most suitable sleeper spacing and sleeper material

4.2.1 Suitable sleeper spacing

The most suitable sleeper spacing according to this study should most importantly yield generally minimal contact stress, strain, pressure and force, less deformation and sliding distance. 700 mm sleeper spacing generally yields lower contact stress, strain, forces and pressure than 600 mm for the curved track and lower values of all except the contact pressure for the straight track. However, the deformation and sliding distance are higher and yet are some of the causes of deterioration of the track, track components and the wheels majorly in terms of wear. On the other hand, 500 mm sleeper spacing in comparison to 600 mm yields relatively higher contact stress, strain, forces and pressure and also less deformation and sliding distance implying that less deterioration of track, track components and wheels hence increase service life.

It is also evident from the models used for this study that for the same length of track, the smaller the sleeper spacing, the more the number of sleepers needed for the track and more time for maintenance compared to the wider sleeper spacing which may require fewer sleepers. Both of these come with an additional cost with smaller sleeper spacing. With 600 mm sleeper spacing, less number of sleepers will be required compared to 500 mm sleeper spacing. Using 700 mm sleeper spacing may require the least number of sleepers but again it gives the highest total deformation and sliding distance which result in high wear volume because since it is directly proportional to the sliding distance [59]. This and its related effects reduce the efficiency and effectiveness of railway transportation.

Therefore 600 mm sleeper spacing can be the most suitable since the values lie in between 500 mm and 700 mm spacing that is the contact stresses, strain, forces and pressure and total deformation and sliding distance are both not too high or low and also the number of sleepers required is less.

4.2.2 Suitable sleeper material.

The most suitable material according to this study is one which yields relatively minimal contact stresses, strain, pressure and forces in comparison with the rest. From the results above, it can be seen that the timber (F34) sleeper material yields the least contact stresses, strain, pressure and force for the straight track but yields higher contact stress and strain in the curved track. However using it results in the largest total deformation and sliding distance which are not good because they aid in track and track components deterioration hence reducing their service life. Structural steel on the other hand yields higher contact stress, strain, pressure, and forces for the straight track and results in lowest contact stress and strain in the curved track although the smallest deformation and sliding distance are observed with it. Concrete portrays results that are between steel and timber (F34) and therefore can be considered as the most suitable sleeper material.

Therefore, if the choice is to be made by only looking at the results of the study, then the concrete sleeper material is better than the structural steel and timber (F34) sleeper materials. However, since the effect of sleeper material on wheel rail contact mechanics is relatively small, the railway corporation can choose to consider other factors in addition to this for effectiveness and efficiency. For example, the wooden sleepers are easy to handle, have good resilience, have good electrical insulation, and can easily adapt to non-standard situations. Furthermore, their softer and more flexibility properties in comparison with other sleeper materials, enables them to provide better sleeper-ballast interface with larger and more ballast grain contact [48]. It has also been noted that sleeper flexibility is directly proportional to the distribution of wheel load over the sleepers but inversely proportional to the sleeper bending moments [58]. The softness of timber subjects it to a higher flexibility compared to concrete and steel sleeper materials.

However, they have more exaggerated deflection, greater propensity to become centre bound, are very expensive in terms of acquiring the material and maintenance, have a short service life unless increased by coating them with a chemical substance called creosote which has been considered to be environment unfriendly and banned in some places, not forgetting that fungal decay and end splitting also contribute to the failure of timber sleepers. These and other disadvantages may be a reason why timber sleepers have been replaced by the concrete sleepers though they are still being used in some sections of the tracks such as over susceptible older structures like bridges and at locations where impact loads may be present such as at S&C and at transitions.

The steel sleepers have not been used widely in the world but have been used in some few parts due to their high strength to weight ratio, they are sturdier than timber and less expensive than concrete but it has high susceptibility to corrosion mainly at the rail seats where there is a risk of fatigue, leading to higher maintenance costs, their installation can be problematic due to the difficulty in creating homogeneous and complete contact with the ballast within the \cap -shaped profile of a steel sleeper [50]. They are noisy and require special fastenings for electrical insulation. These disadvantages are a main reason for the unpopular use of steel sleepers in most railway systems all over the world.

Concrete sleepers have a high gauge retaining capacity aided by their extra weight compared to steel and wooden sleepers, longer service life due to the fact that they are not affected by changes in temperature, they are water resistant, sun resistant and corrosion resistant in comparison to wooden and steel sleepers, they can withstand fire hazards better, since they need less maintenance, they result in lower maintenance cost and increase on availability of the track, they also provide good insulation and therefore can be used in circuited tracks. Compared to timber sleeper material, they are environmental friendly because they don't require special chemical treatment like creosote which is harmful to the environment, the materials for manufacturing the concrete sleepers is readily available too. However, despite the many advantages, concrete sleepers introduces a much harder interface between the sleeper and ballast, potentially raising the contact stresses on the ballast grains and causing greater ballast damage [48], [49] not forgetting the difficult in handling due to the heavy weight compared to other sleeper materials.

CHAPTER FIVE: CONCLUSIONS, RECOMMENDATIONS AND FUTURE WORK

5.1 Conclusion

In this study, investigation of the effect of sleeper spacing and material on contact stresses, strain, pressure, forces, deformation and sliding distance was made. It involved performing a static structural analysis of model comprising a single wheelset on a track supported by sleepers in Ansys workbench. From this analysis, the contact stresses, pressure, forces, strain, total deformation and sliding distance were obtained. It can be observed from the results that sleeper material and spacing have an effect on wheel rail contact mechanics. Some of the major conclusions while making concrete sleeper material and 600 mm sleeper spacing as a benchmark in the study include the following;

- The sleeper spacing and sleeper material both have an influence on the wheel rail contact mechanics in both straight and curved track.
- The most suitable sleeper spacing and material is 600 mm and concrete sleeper material.
- The values of contact stress, pressure, force, elastic strain, total deformation and sliding distance are generally higher in the curved track than in straight track section.
- The contact stress, pressure, force and elastic strain generally increase with a decrease in sleeper spacing and with an increase in the sleeper MoE for both straight and curved track.
- The effect of sleeper material on wheel rail contact mechanics is generally relatively small for both straight and curved track.
- The total deformation and sliding distance values increase with an increase in sleeper spacing and with a decrease in sleeper MoE for both straight and curved track.
- The contact pressure in the curved track section also increases with a decrease in sleeper spacing and with an increase in sleeper MoE.
- The trend of the contact stress in the curved track tends to deviate from that of the straight track since it appears to be highest with the 600 mm spacing.

- The contact forces on the right side increase with a decrease in sleeper spacing and with an increase in sleeper MoE and the reverse is true for the contact forces on the left side for a curved track.

5.2 Recommendations

- The use of 600 mm sleeper spacing and concrete sleeper materials can be maintained since they portray a better contact and result in less wheels and track deterioration in terms of wear due to not too much or too low contact force and sliding distance according to the study.
- Since the effects of sleeper material on wheel rail contact are relatively low, it's important to consider more than these factors in making decisions on which sleeper material (modulus of elasticity) to use.

5.3 Future scope

- To study the influence of variation of sleeper spacing as a cause of track irregularity on the wheel rail contact.
- To perform a dynamic analysis while varying sleeper material and spacing in a straight and curved track with the aim of studying their effect on wheel rail interaction.
- To study the effect of sleeper spacing and material on wheel rail interaction under wet, dry and lubricated conditions.
- To study the effect of sleeper spacing and material on wheel rail wear.

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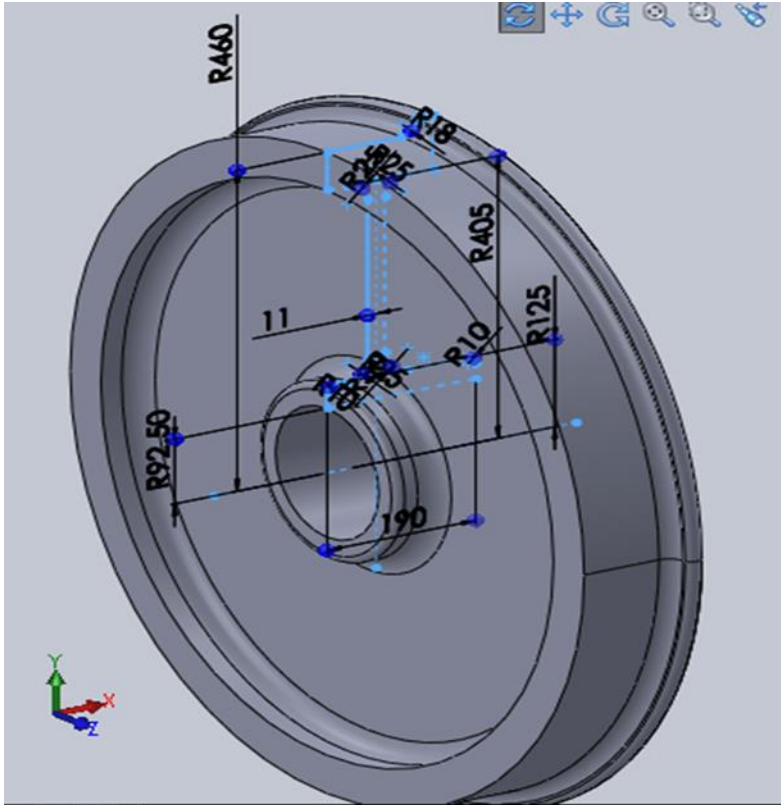
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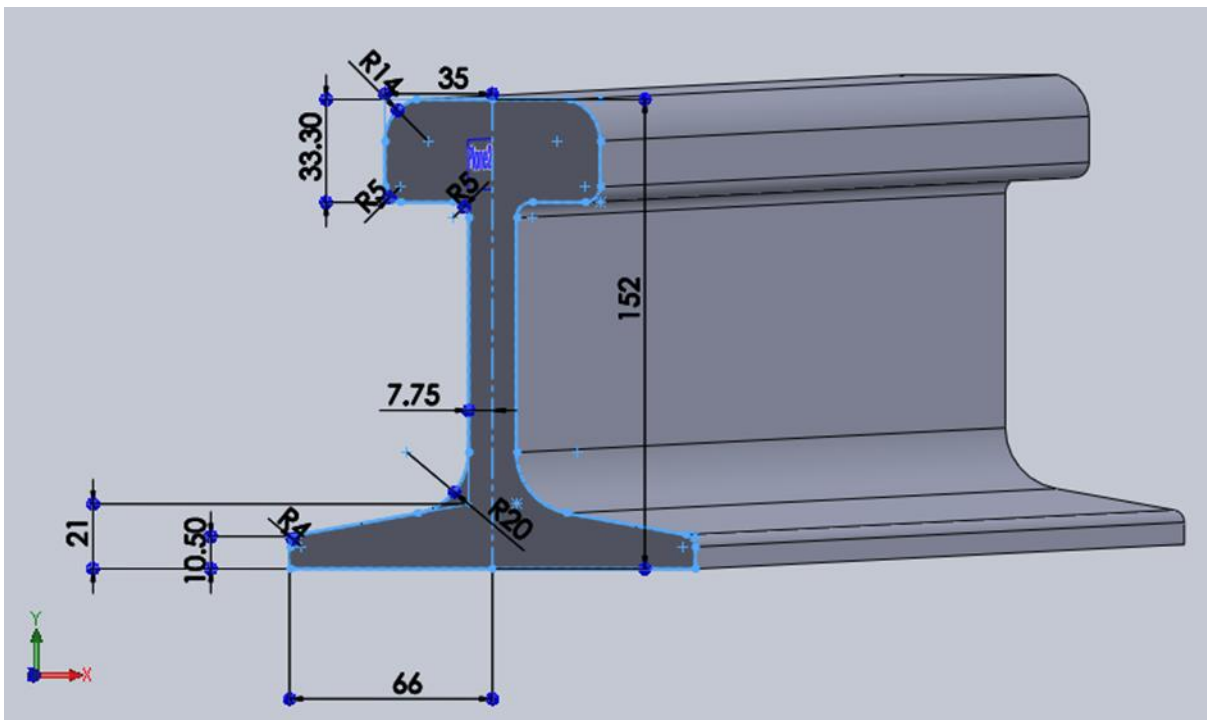
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Appendix

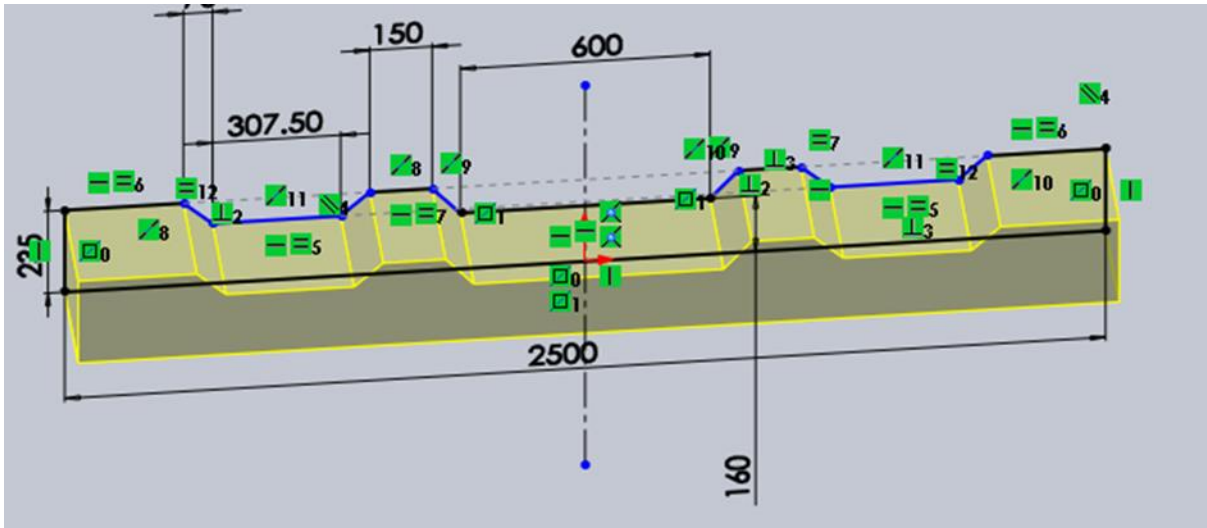
a) Wheel



b) Rail



c) Sleeper



d) Axle

