ADDIS ABABA UNIVERSITY

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EFFECTS OF RADIUS OF CURVED RAIL ON RAIL WEAR

A Thesis Submitted to the School of Graduate Studies of Addis Ababa institute of Technology in Partial Fulfillment of the Requirement for the Degree of Masters of Science in Railway Mechanical Engineering

by

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DECLARATION

Addis Ababa University

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This is to certify that the thesis prepared Seid Abdulhakim entitled: Effects of Radius of Curved Rail on Rail Wear and submitted in fulfillment of the requirements for the degree of masters of Science (Railway Mechanical Engineering) compiles with the regulations of the university and meets the accepted standards with respect to originality and quality.

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Abstract

One of the most sensible issues in the railway industry is the impact on infrastructure of train operations and the damage on vehicles provoked by the track conditions. These issues have a significant impact on the life cycle costs of the railway networks. The study of vehicle-track interaction is important in reducing the operation and maintenance costs, by increasing the life cycle of both vehicles and tracks, and increasing the speed, safety and comfort indexes of the railway systems.

This thesis focused on the analysis of effects of radius of curved rail on rail wear. Multi-Body simulation software SIMPACK is employed to create a one car railway vehicle model with two bogies with an axle load of 10 ton. The rail material which is applied on the study is UIC 60 standard carbon steel. To verify the simulation results, the SIMPACK model is run on varies track radius (sharp curves) range from 100 m to 800 m under dry condition. Parameters like track radius, wheel rail contact patch area, contact pressure, vehicle speed, wheelset yaw angle are investigated to analyse there effects on rail wear.

The dynamics in the vehicle to the rail on canted curve was analyzed using a quasi-static approach which at instant time the vehicle is loaded vertically by its weight and laterally by centrifugal force. This lateral force is balanced by using centripetal force and its direction may be inwards and outwards depending on velocity and weight of the vehicle. It is applied on outer rails while the vehicle is moving above equilibrium speed and it is applied at the inner rail while the vehicle is moving below equilibrium speed.

The increase in radius of the rail has exponentially decrease the wear conditions at constant velocity on curved track at both rail head. The outer rail side is damaged severely when the vehicle is moving above equilibrium speed and the inner rail side is damaged severely when the vehicle is running below equilibrium speed. Wheel rail contact patch area increases with decreasing of radius of rail at the same time the wear volume also increases. Contact patch area and wear rate have direct relationship. Contact pressure and track radius have direct relation. As track radius increases contact pressure decreases due to decrease of contact patch area. Wheelset yaw angle has inverse relation with radius and direct relation with that of contact patch area and wear of rail.

All above parameters affecting wear property of rail should have to be controlled to minimize damages on rolling stock and rail. Curved railway tracks requires more attention than that of straight track due to the introduction of centrifugal force which makes the vehicle tends to move laterally during curving motion. Severity of wear can be reduced using equilibrium speed of the vehicle and following vehicle speed restriction on curves is important otherwise leads to excessive wear on the contacting surfaces. Interchanging of inner rail and outer rail can be considered as a means to lengthen the service life of the rail.

Generally knowing the effects of these parameters on overall system helps for understanding the mechanism, minimize damages on rail and can be added for practical rail way curved track design and speed restriction on curves.

Table of Contents

List of figure

List of table

Nomenclature

UIC	Union International des Chemins (International Union of Railways)
φ	Cant angle
D	Degree of a curve
2l	Track gauge
ht,	Super elevation height
Veq	Equilibrium velocity
ay	Lateral acceleration
az	Normal acceleration
R	Radius of a curved track
Y	Lateral wheelset displacement
ro	Nominal wheel radius
rl,	Left wheel radius
rr,	Right wheel radius
R1	Right rail radius
R2	Left rail radius
A	Contact area
λ	Wheel conicity
θ	Angle of attack
ξy,	lateral slippage
ξsp,	Spin slippage
ξ,	Longitudinal slippage
αv,	Rotational velocity
∑MB	Total moment force about right rail
∑MA	Total moment force about left rail

υ_η	Lateral creepage
υ_ξ	Longitudinal creepage
ω_z,	Spin creepage
v_ξ	Sliding velocity in longitudinal direction
v_η	Sliding velocity in lateral direction
Ø	Roll angle of the wheelset
F_ξ	Creep force in longitudinal direction
F_η	Creep force in lateral direction
M_3	The spin moment
M	Vehicle total mass
Ts_x	The energy expended by creep on contact patch
ω	Angular velocity of the wheelset
t	Time
WR	Wear number

CHAPTER 1

1. INTRODUCTION

1.1 Background of the study

Development of railways was decisively influenced by the industrial revolution, the introduction of steam and the extensive exploitation of coal and iron mines. The first railway lines began operating in most European countries around 1830 and most railway networks attained maximum density at the beginning of the 20th century. The adoption of electric traction, in the early 20th century, permitted further development of the railways, while the development of signaling and automatic train gave railways their present form, in the 1950s [1]. Today, rail transport plays a very important role in the transportation network, moving goods, heavy goods in particular, and people all over the world. To effectively solve the problem of passenger transportation and that of freight goods, the government of Ethiopia also decided to build a railway infrastructure in the city of Addis Ababa and in different parts of the country. To satisfy such demands, it needs to have a railway transportation system with standard safety, comfort to customers, and reliable to user's etc. Such demands need the advanced design and analysis of functional components, which have a direct or an indirect impact on the customers and users demand. The railway train running along the track is one of the most complex dynamic system in engineering, it has many degree of freedom, the interaction between wheel and rail involves both complex geometry of wheel trade and rail head and forces generated by relative motion in the contact area [2]. Undesirable effects would be observed when a train is travelling on a curved railway track. These negative effects include passengers, discomfort, risk of derailment, wear of rails and wheel flanges, noise and so on [3]. Inadequate guidance on curves results in higher lateral forces between wheel and rail, rapid wear of wheels and rails and the possibility of derailment. In recent years there has been continuous increase of axle loads, tonnage, train speed, and train length which has increased both the productivity in the rail sector and the risk of rail breaks and derailments. These problems have increased the maintenance and replacement costs. If undetected, these problems can cause derailment causing huge loss of revenue, disruption of service, resulting damage of assets, and loss of lives [4]. New technologies and better safety standards are constantly being introduced associated with curve radius, speed and elevations but there will always some risk associated with accidents and early degradation of the components.

The track response for dynamic loading has different aspects of which one is rail tribology. Wear is significant problem for railway companies [5]. Wear of railroad rolling stock and rails costs millions of dollars each year in all rail systems throughout the world. Excessive levels of noise are generated at the rail/wheel interface in conjunction with wear, which is unacceptable in an environmentally responsible rail network [6]. Friction plays a major role in rail wheel interaction, particularly in adhesion, braking, wear and rolling contact fatigue damage, formation of wheel burns, steering and hunting of locomotive and wagon bogies, and wheel climb/ mount leading to derailment. The energy loss due to friction of a wheel on a rail can reach about 24 % of full energy consumption on the traction, which is basically consumed on the overcoming of friction forces between wheels and rails especially in curves [7]. For the heavy loaded frictional contact of wheels and rails, the variable values of friction forces, vibrations, noise and various kinds of destruction are characteristic. Therefore the control of friction between a wheel and a rail is necessary. Damage accumulation because of wear, fatigue and plastic deformation significantly reduces the service life of the railway wheels and rails. Studies on rail wear and lubrication for maintaining reliability and safety of rolling stock and rail track can reduce the operational risks and the costs by prevention of rail failures, rail breaks and derailments. This thesis focuses on analysis of the effect of radius of curved track on wear of rail under dry condition. By doing this research on the title effects of curve radius on wear of rail has many advantages for the global level and as well for the country ‘Ethiopia’ which is constructing railway infrastructure.

1.2 Objective of the thesis

1.2.1 General objective

- Analyze the effect of the radius of curved rail on rail wear.

1.2.2 Specific objectives

- To determine the effect of varies curve radius of rail on wear of rail.
- To determine the effect of varies curve radius of rail on contact patch area and rail wear.
- To determine the effect of track radius on wheelset yaw angle and wear of rail.

1.3 Significance of the study

The railway transport system is one of the most crucial transport systems in the world with very high speed, safety and durability. Now a day, there is a high demand of railway transportation systems in the world including in our country Ethiopia for a long distance transport of passengers and goods. To satisfy such demands, it needs to have a railway transportation system with standard safety, comfort to customers, and reliable to user's etc. Such demands need the advanced design and analysis of functional components, which have a direct or an indirect impact on the customers and users demand. The movement of the wheelsets along the track is characterized by a complex contact where there are relative motions on the longitudinal and lateral directions and also relative rotations of the wheels over the rails. Especially in curved tracks these motions generate large tangential creep forces and moments at the wheel-rail interface, which are ultimately responsible for increasing wear and tractive effort loss. Wear is the major cause of material wastage and loss of mechanical performance and any reduction in wear can result in considerable savings. Friction is a principal cause of wear and energy dissipation. Considerable savings can be made by improved friction control. It is estimated that one third of the world's energy resources in present use is needed to overcome friction in one form or another [8]. This research has a great impact on future analysis and applications of wheel/rail contact in general and in particular in the Ethiopian context. Especially, it will contributes a lot for future work, like in designing of new rail track which are going to be constructed in the future.

1.4 Scope of the study

The objective of this research is to analyse the influence of rail curvature on wear of rail. The analysis involves the effect of change of vehicle speed, contact patch area, wheelset yaw angle, pressure on contact patch and change of radius of track curvature. Discussion is also made on the effects of these all factors mentioned above.

1.5 Limitation of the thesis

In this thesis the analysis of wear of rail is based on centrifugal loading only neglecting other forces such as wind. So all calculation from the point of axles loads and forces generated due curving motion of the vehicles.

1.6 Research methodology

To analyse the influence of rail curving on wear of rail, there are procedures that should be followed:

- Mathematical modeling of
  - kinematic and dynamic mathematical model
  - Wear model
  - Vehicle model developed using Simpack software
- Simulation of wheel rail contact using SIMPACK software

1.7 Organization of the thesis

This thesis is organized in to six chapters. In the first chapter, background and justification of this thesis work and the objectives to be achieved are discussed. In chapter two, a review of literature relevant to this thesis work. Chapter three is about formulating analytical relations which deal with the interaction between vehicle and curved rail by using kinematic and dynamic analysis. In chapter four, materials, methods and conditions that applied for result analysis and interpretations of the study are presented. In chapter five results of the analysis are summarized and discussions are made based on using deferent ranges of a radius of rail track, velocity of a vehicle, wheel rail contact patch area, yaw angle, wheel rail contact pressure, using outputs from SIMPACK software simulation and graphs showing parametric relation are plotted using Microsoft office Excel. Finally, chapter six gives conclusion and recommendations achieved from this thesis work and propose future work in this field of study.

CHAPTER 2

2. LITERATURE REVIEW

Tribology is a new field of science, most of the knowledge being gained after the Second World War. Tribology is derived from the Greek word ‘tribos’ meaning rubbing or sliding and it focuses on friction, wear and lubrication of interacting surfaces in relative motion. Friction occurs when two bodies are in relative rubbing motion, and rubbing means that the bodies are in actual contact and generates wear and heat. Wear takes place on both rubbing surfaces as a result of relative motion between surfaces [8]. Lubrication on sharp curves has been commonly adopted as a method of reducing friction between the wheel flange and rail gauge face to minimize wear and energy consumption [3,5]. Many friction models show that friction is affected by the rheology of the third body, slip distance and load with the shear stress, of which slip distance exhibits the dominant influence [6]. In curves there can be a large sliding component on the contact patch at the gauge corner of the rail head since the wheel flange is in contact with the gauge corner of the rail. Flanging causes large sliding motion in the contact patch [9]. Sliding increases wear rate in the contact under the poorly lubricated condition that is typical of wheel/rail contact. During sliding wear, an increase of the severity of loading (normal load, sliding velocity, or surface temperature) leads to a sudden change in the wear rate (volume loss per sliding distance). It was found that mild wear dominated at the rail head, but at the rail edge severe wear was clearly occurring. For horizontal curved track radius below 1500 m at any of case of vehicle run with flange contact whereas within elevated track of curve radius of 700 m rather than 1500 m any of the cases with lower conicity wheel/rail combinations run in flange contact. Where there are separate flange and tread contact points, the conditions at the point with the higher contact stress and traction coefficient. At the tighter 700 m radius, the wheels are much closer to full slip and the traction coefficient changes much more markedly. Therefore, curve radius of the track has a strong influence on rail wear. The influence also depends on the vehicles and their behaviors especially speed [7]. Speed can adversely influence the curving performance of the vehicle and, in turn, lead to wear and stresses in rail and wheel. It is because the point of application of the load moves with the running speed [10]. Due to the centrifugal forces at the curves, the outer rail bears substantial amount of forces that wears it out. By applying a super elevation, wear can be decreased. It helps prevent overturning of the vehicle, to overcome the centrifugal force of the vehicle on curves [11]. Degradation on either the high rail or low rail lying in the same curve radius depends on the speed of the vehicle. The higher the speed, the more the degradation will be on the high rail. This is because much of the wheel flange is in contact with the inner surface of the high rail than the inner surface of the low rail due to centrifugal force acting on the vehicle. The lower the speed, the more the degradation will be on the low rail [12]. The wear of wheels and rails, especially the wheel flange wear and the rail side wear on curves, is a long-standing problem of heavy-haul railways. With the rise in train speed and transport capacity, the wheel-rail interaction aggravates inevitably. Severe wheel-rail dynamic interaction will induce serious wear of wheels and rails, especially in the case of curved tracks. Studies shows that the predominant wear is the side wear of outer rails on small radius curves. The outer rail side wear aggravates rapidly with the decrease of curve radius and as the curve radius rises, the rail side wear drops down greatly [13]. On sharp curves, the outer rail profile with severe side wear nearly conforms to the flange shape. The wheel-rail contact under this condition usually leads to an incremental load on rail side. With a continuous expansion in railway traffic volume, the wear rate presents an upward tendency [14]. A vehicle passing through a curved track will experience a lateral force. This wheel-rail lateral force is a key factor, which is mainly responsible for the rail side wear. On curved track, the wheelset is steered by the lateral forces acting on wheels. If the wheel and rail profiles are designed to be able to provide enough rolling radius difference, the creep forces will push the wheelset to adjust the radial position without wheel flange contact. Conversely, small rolling radius difference results in large attack angle of the wheelset, and the wheel flange usually contacts with the rail side. Once the wheelset is steered by the wheel flange, severe rail side wear may happen [13]. Lubrication at wheel flange and rails on sharp curves is considered as an effective solution for reducing wear loss of material from effective cross-section of rail and wheels. Rail administrations around the world have been increasing axle loads and traffic densities in rail networks. The rail wear rate decreases with increase in curve radius for both high and low rails as shown in figure below.

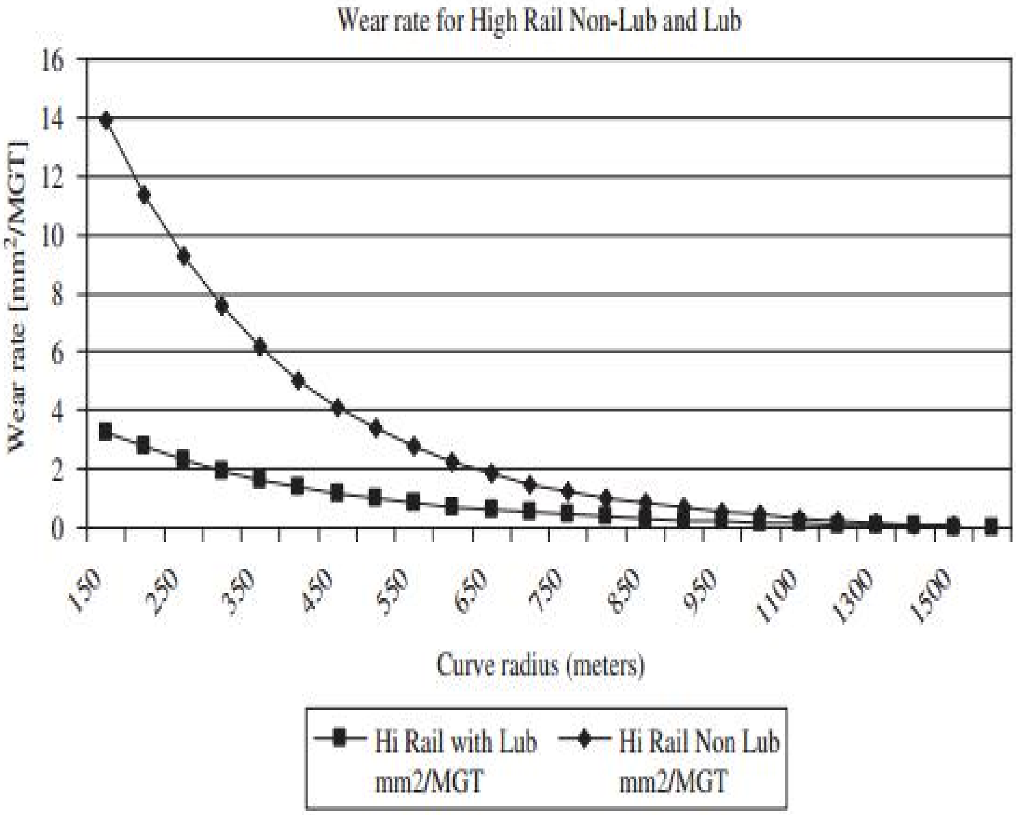


Figure 2.1: Traffic wear rate for high rail non lubricated and lubricated [15].

The wear rate ratio between non-lubricated and lubricated sites decreases for the curves with larger radius [15]. In the mild wear it was observed that the wear process is slow, similar to oxidation. In the sever wear it occurs much faster, similar to adhesion wear, as observed in curves under dry conditions. Mild wear observed at the wheel tread and rail crown. Sever wear is observed at the wheel flange and gauge face [6]. The Stockholm local network studied the lubricated and non-lubricated rails for UIC900A and UIC1100 grade rail steel under various seasons. The study found that the contact situation in terms of pressure and sliding between rail and wheel, strongly influences the wear. The curve radius of the track has influence on wear behavior. It is found that the wear rate increases exponentially for decreasing curve radius. Sharp curves lead to increased track guiding forces on the wheels, leading to increased creep and increased wear. The track side lubrication reduces rail wear significantly. Lubrication benefits as a factor of 9 for sharp curves (300m) and from 600-800m radius curve the lubrication benefit varied from 2 to 4 compared to non- lubricated curves. An increase in temperature of the rail leads to increased rate of wear. A survey of heavy haul railways in the mid 1990s indicated rail lives varying between about 1500 million gross tones (MGT) of traffic in straight track and about 300 MGT in highly curved track. This life also depends on axle load, traffic density, track formation, bogie type and railway track maintenance practices. Rail life has been increased by a factor of two and wheel life by a factor of five using appropriate lubrication. Spoornet (South Africa) has reported that rail life was increased from 27 MGT to up to 350 MGT, depending on curve radius. Canadian pacific (CP) rail indicated that rail life improved by 110% using effective lubrication. Experiments on the Olympic Park loop, NSW (Australia) on a 200m radius curve, indicated that flange wear rate is reduced from 0.36 mm/day (life of 0.2 year) to 0.006 mm/day (life 3.5 years). Eurostar conservatively estimates that effective lubrication saves 1000,000 Euro per year on maintenance and wheel replacement costs [16]. Generally from the above studies it can be concluded that, track curves are observed with high wear rate and the intention of this paper is to see the influence of curve, vehicle speed and lubrication and wear rate in curved rail.

2.1 Wear on wheel and rail interaction

Wear is the principal cause of rail replacement on almost all railroads. Wear tends to be concentrated on the high rail gauge face (i.e., the inner edge of the outer rail in curved track) where contact is made with the wheel flange (Figure 2.2). In straight track and large radius curves, vertical wear of the head is seen. If the rail wears severely, the stress in the rail rises, particularly in the head and, eventually, the rail needs to be replaced. All railroads have different criteria for removing worn rail, but often these criteria imply replacement when about 30 to 50% of the head area has been lost.

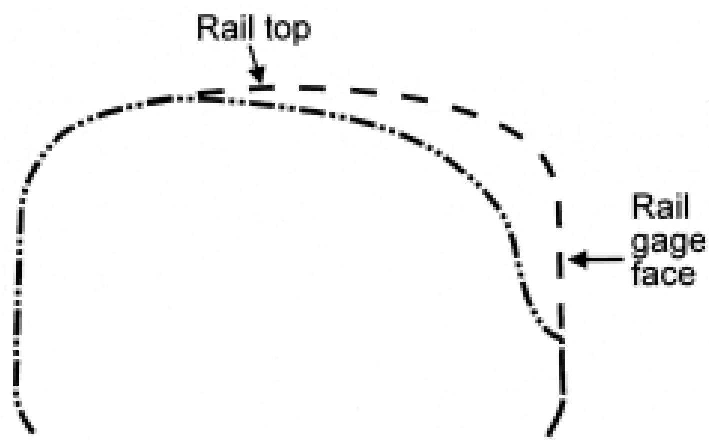


Figure 2 2: wear on the outer (high) rail in curved track [13].

Wear is a complex phenomenon, to which a number of different damage mechanisms can contribute. Four classical mechanisms that act to produce wear between two bodies can be summarized as follows (Halling, 1989):

1. Adhesion. This can occur when, either in the absence of a lubricant or contaminant film, or when such a film has been disrupted, clean metal surfaces adhere strongly to each other at contacting asperities. Relative tangential motion between the two bodies then shears these junctions, and further motion generates wear particles.
2. Abrasion. This occurs when a relatively soft surface is ploughed, either by a harder surface (as in grinding), or when loose hard particles are introduced between the two bodies.
3. Fatigue. Both adhesion and abrasion require direct contact between the surfaces of bodies in relative sliding motion. In contrast, fatigue wear can occur even when a lubricant layer separates the bodies. In this mechanism, the action of normal and shear stresses produced by a rolling/sliding contact causes cracks to initiate and propagate, leading the surface layers of the material to delaminate.
4. Fretting. This occurs when wear particles are produced as a consequence of low-amplitude vibratory motion between the two bodies.

Adhesion, abrasion, and fatigue can all contribute to wear in wheels and rails. The relative extent to which these different wear mechanisms contribute to wheel and rail wear is not known.

However, examination of wear results from small-scale rolling/sliding dry wear tests can be useful in the investigation of the wear process. These tests typically use a twin-disc (cylinder-on-cylinder) approach, with the discs running with different peripheral speeds to introduce longitudinal creep. The first machines for this type of test were developed in the early part of the 20th century (Amsler, 1922), but more sophisticated machines are now available (Fletcher and Beynon, 2000). Using the twin-disc approach, Bolton and Clayton (1984) found that three types of wear can occur on the head of rails:

1. Type I wear: occurs at low contact stress and creep, and is characterized by large thin wear flakes containing metallic and oxidized wear debris. There is evidence that rolled-out manganese sulfide inclusions contribute to wear, but that rail steel type does not. Wear appears independent of creep once the limiting coefficient of friction is achieved, and is proportional to distance rolled multiplied by the contact stress.

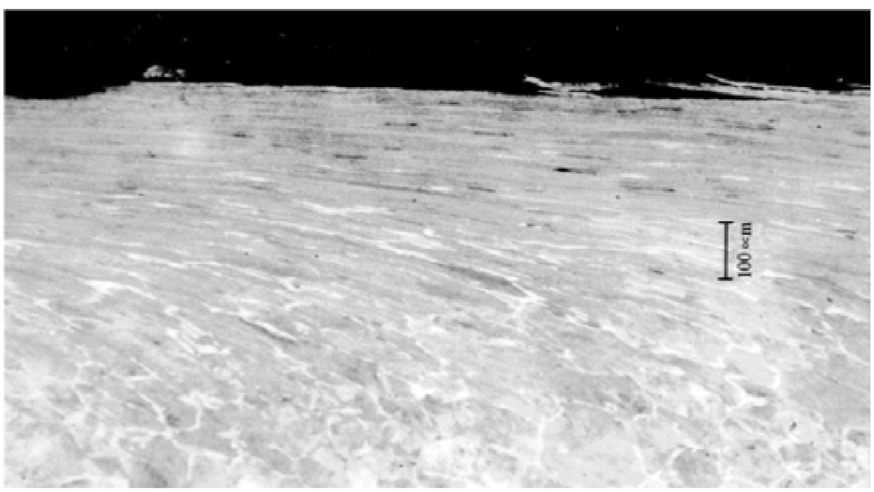


Figure 2.3: Section of rail revealing plastic deformation of the steel near to the rail surface [24].

2. Type II wear: occurs at medium contact stress and creep. Wear flakes are metallic, much smaller, and less regular than those found in Type I wear, and are often compacted. Wear depends both on contact stress and creep.
3. Type III wear: occurs at high contact stress and creep above 0.1. In Type III wear, the surface is much rougher than in Types I and II, with evidence of significant plowing and tearing away of surface particles. Wear particles are very irregular in size, and larger

particles can have visible score marks. Wear rates are at least an order of magnitude higher than Types I and II.

These laboratory results indicate that wear progresses from Type I to Type II to Type III as contact stress and creep increase. Observations of worn rails from curved track in service indicate that Type II wear occurs predominantly from the rail top to the gauge corner, while Type III wear occurs mainly on the gauge face.

Bolton and Clayton (1984) also demonstrated that, for Type II wear, the wear rate (WR, in terms of metal lost per unit contact area per unit distance rolled) depends on the tangential force in the contact (T), the creep, and the area of contact (A):

$$WR=\frac{Ts_x}{A} \quad (2.1)$$

Ts_x is equal to the energy expended by creep, and therefore Equation 2.1 links the wear rate to the energy expended per unit contact area. More generally, for the real wheel-on-rail situation, wear is related to the energy expended in all three types of creep (longitudinal, lateral, and spin) [24].

For the purpose of this project, a simple but widely accepted approach of the wear computation has been adopted [Orvnäs (2011)] [Johnsson, A., Berbyuk, V., Enelund, M. (2012)] [Mousavi, M., Berbyuk, V.(2013)]. It is based on the assumption that the wear present in the rail-wheel contact is linearly related to the energy dissipated in the process.

$$WR= F_\xi . \upsilon_\xi + F_\eta . \upsilon_\eta + M_3\omega_z \quad (2.2)$$

Where F_ξ , F_η are the creep forces in longitudinal and lateral directions and $M_3$is the spin moment. Moreover υ_ξ, υ_yandω_z are corresponding creepages.

$$\upsilon_\xi=\frac{v_\xi}{v} \quad (2.3)$$

$$\upsilon_\eta = \frac{v_\eta}{v} \quad (2.4)$$

Where v_{ξ} and v_{η} are the sliding velocities in the longitudinal and lateral directions, respectively and v is the vehicle speed. Once in equation (2.4), the spin creepage contribution is dismissed, the result is called the wear number [23].

The value of the wear number in the leading outer wheel is the parameter used to quantify the wear objective function in this project and is given by equation (2.5).

$$I'_{W} = \sqrt{\frac{1}{t_{f-t_o}} \int (F_{\xi}.\upsilon_{\xi} + F_{\eta}.\upsilon_{\eta})^2 dt} \quad \ldots\ldots (2.5)$$

According to [Pearce and Sherratt (1991)] this objective function is classified as Table 2.1 illustrates:

Table2.1: Categories for wear number

Wear Number < 100	Low
100 ≤ wear number < 200	Medium
Wear Number ≥ 200	High

CHAPTER 3

3. ANALYTICAL METHOD

3.1RAIL

Rails are longitudinal steel members that are placed on spaced sleepers to guide the rolling stock [17]. Support of traffic load and guidance of vehicles are the two main tasks of the rails. For both tasks the correct contact geometry between wheel and rail is essential [18]. In addition to that rails are used to accommodate and transfer the wheel/axle loads into the supporting sleepers. The most commonly used profile is flat-bottom rail and is divided into three parts:

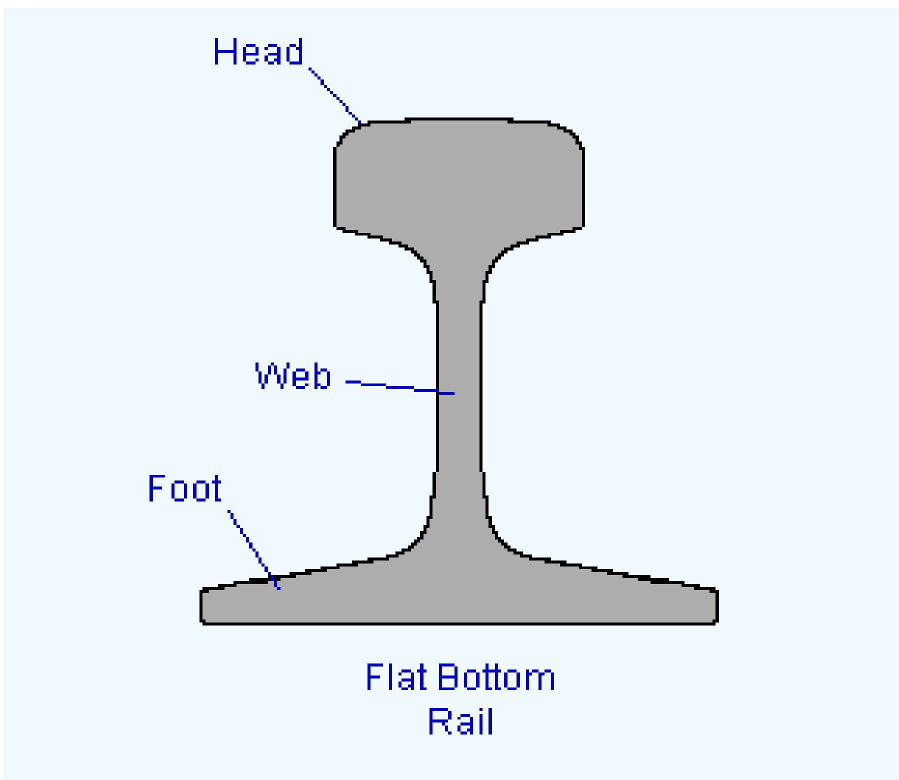


Figure 3.1: Flat bottom rail parts

3.2 WHEELSET

The wheel set is placed attached to the railway bogie. Bogie is a structure underneath a train to which axles and hence wheels are attached through bearings. Bogies are classified according to their configurations in terms of the numbers of axles, the design and structure of the suspension systems. Bogies serve a number of purposes.

- Support of the rail vehicle body.
- Provides stability on both straight and curved track.
- Ensures ride comfort by absorbing vibration and minimizing centrifugal forces when the train runs on curves at high speed.
- Minimizes generation of track irregularities and rail abrasion

A wheel set comprises of two wheels rigidly connected by a common axle and it provides:

- The appropriate distance between the vehicle and the track
- The guide to the motion of vehicles on the tracks
- The means of transmitting axle loads, traction and braking forces to the rails to accelerate and decelerate the vehicle etc.

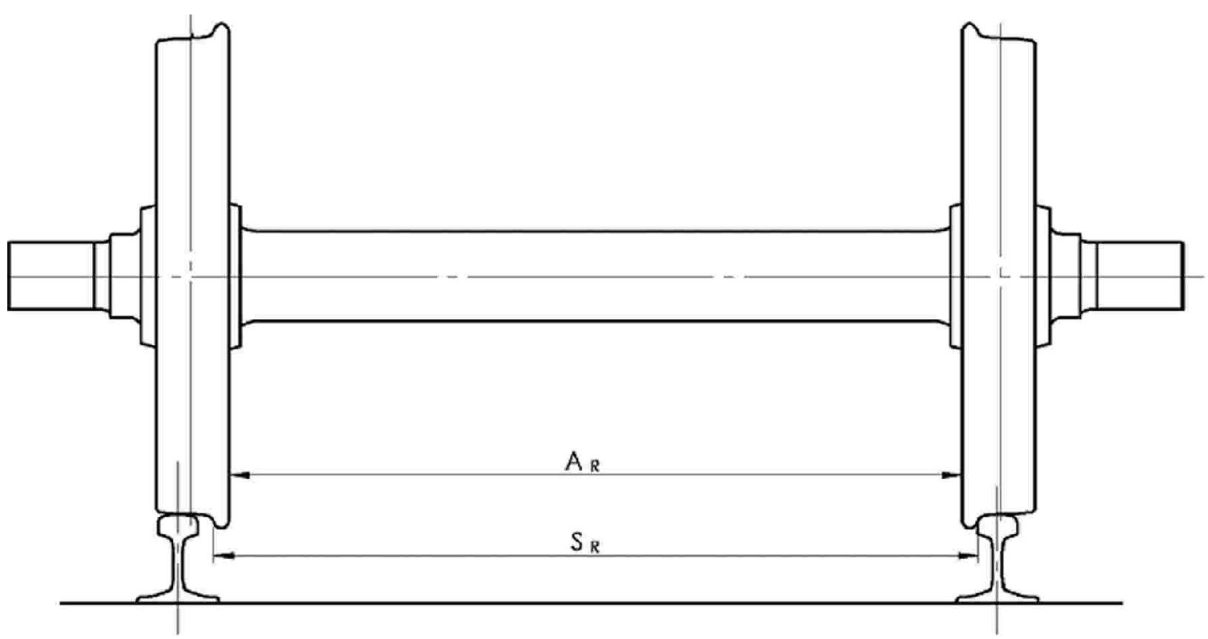


Figure 3.2: A Wheelset

Generally the railway wheel set has 6 degrees of freedom broadly classified as translational and rotational degrees of freedom. The translational degrees of freedom comprise three components that is translation along:

- X-axis
- Y-axis
- Z-axis

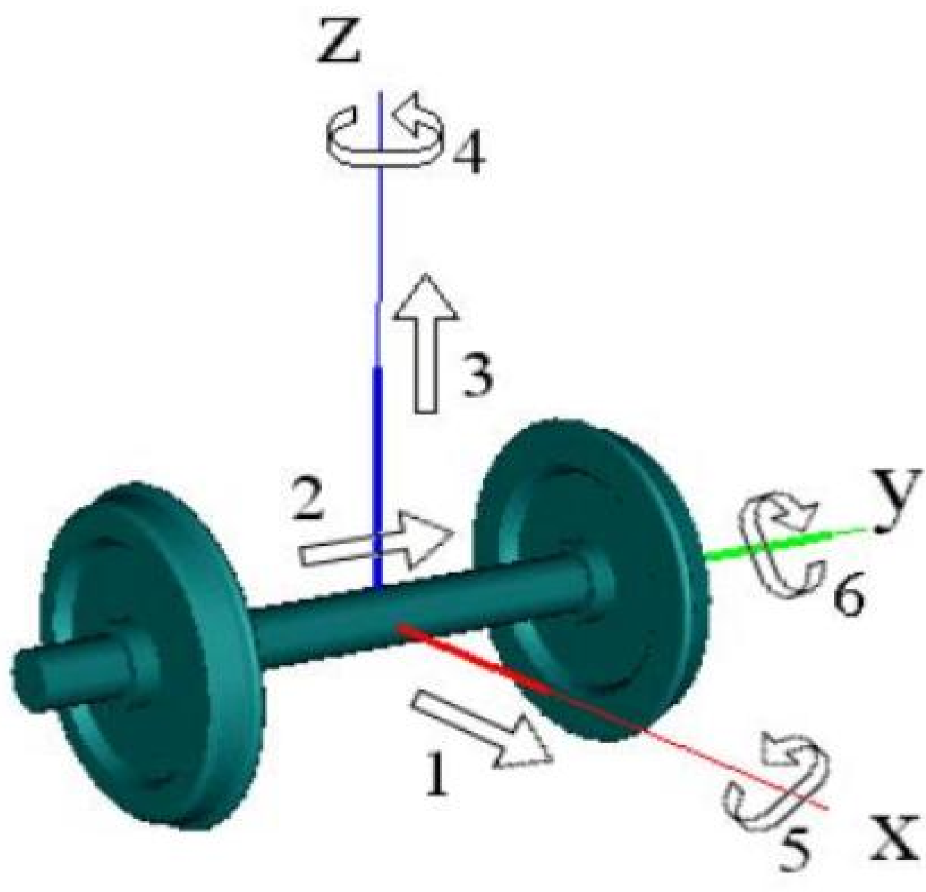


Figure 3.3: Wheelset degrees of freedom

Similarly the rotational degrees of freedom consist of three components that is rotation about:

- X-axis
- Y-axis
- Z-axis

In conventional railway vehicles the wheels, assembled in a wheelset, are not free to rotate independently. Hence, their treads are coned in order to allow them to negotiate curves without slipping. However, it is recommended to reduce the wheel/rail conicity to allow wheel sets to remain stable up to much higher mileages, and to minimize contact between the flange root of the wheel and the gauge corner of the rail [19].

3.3 Rail-Wheel Interaction

The dynamic behavior of railway vehicle is greatly affected by the rail-wheel dynamic interactions. This interaction (wheel/rail) mainly depends on wheel/rail contact geometry. The changes in contacting geometry of rail/wheel depends on different parameters like the variation of wheel and rail profile, track gauge, track radius, rail inclinations, railhead surface irregularities, and flexibility of rail support. The main parameters influencing the wheel rail contact geometry are the profiles of wheels and rails, rail inclination and track gauge [20]. In this paper some of the parameters listed above affecting the contact geometry are assumed to be constant. Analysis of the wheel/rail contact geometry by considering all affecting parameters will make it complex. Therefore for better understanding and analysis of the condition under consideration, the grouping of interrelated parameters will result meaningful final outputs. Considering the other parameters constant and giving attentions to the analysis of wheel rail wear on curved track is the main work of this paper. The wheel/rail contact profile is characterized by using the equivalent conicity, contact angle, lateral movement of wheel set, the wheel/rail material properties (elasticity), axle load, vehicle speed and yaw angle.

Any load from the train is assumed to be transmitted to the track through wheel and the rail head to the ballast without affecting the environment. However, the large portions of these forces are distributed /dissipated in the wheel railhead contact surfaces. Therefore, from this general idea it is possible to conclude that the main failure of railway transport system is on these interacting interfaces. From the data observed in the developed countries using the railway transport system at a large extent, these failures are extremely a problem of developed world. This failure causes huge cost as a railway wheel rail maintenance and renewal. As stated above this is due to high stresses at the wheel rail interfaces, which causes failure like wear, fatigue and a combination of the two. Understanding of interaction between these surfaces indicates and predicts the way how to treat them and how to prevent them from failures which is a better guide to select the appropriate materials, geometry, design, orientation etc. That is why this paper mainly concerns about the analysis of wheel rail interaction on curves to determine wear property of the interacting bodies. The figure below shows the general wheel rail interactions.

Figure 3.4: Wheel rail interaction [14].

3.4 Curved Rail Properties

The main elements of rail way industry are track structure and rolling stock. A track consisting of rails, fasteners, sleepers, ballast plus underlying sub grades [21]. Sleepers hold the rails to correct the gauges and transmit loads, fasteners are used for the purpose of fastening against sleepers and the ballast is a part of foundations. Among these, rail is a central discussion point in this paper. It is the main bearing element of track, is made of longitudinal steel members that help to accommodate wheel loads and distribute these loads over the sleepers or supports. It withstands large dynamic loads in vertical, longitudinal and lateral direction. Strong and wear proof rails are required for safe railroad traffic. The rails are the most important element in railway study. Their function is to guide the wheel-sets, to support the loads and to damp the vibration generated during motion. All rails are inner tilted with a cone angle (mostly 1/20 or 1/40), used for stability during motion, to maintain contact of wheel-set to the center of the track and adjust velocity within the same angular velocity of wheel-set axle. Track geometry has a great influence on the tribology of rails by changing the vehicle load and force transmission. Railroad track geometry is intrinsically three-dimensional. For practical purposes the vertical and horizontal components of track geometry are usually treated separately. In this thesis only horizontal circular curves which use a uniform radius for the entire distance between adjacent tangent sections are used.

A circular curve used on railway system is identified with the following parameters:

- Direction of curve may be considered as clockwise or anti-clockwise depending on the motion of the trains through the curved rails. And in this thesis outer rail is at the right side of the driver and the inner rail is at the left side considering clockwise movement of the vehicle over the rail.
- Radius, R: The radius, R is the radius of the circle at the center line of the track.
- Degree D:The degree, D also used to describe the curve by curvature in place of radius. A D-degree curve turns the forward direction by n degrees over some agreed upon distance. The usual distance in US road work is 100 ft (30.5 m) of arc as seen in fig below.

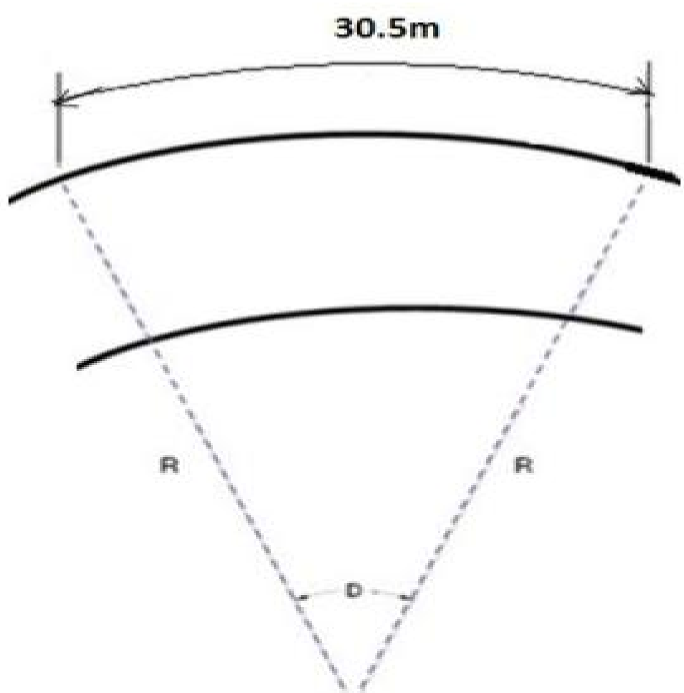


Figure 3.5: Horizontal track curve

The circumference of the curve, 2*π*R, for the angle at the center of the curve is 360°. Therefore, the angle subtended by 30.5 m or 100 ft chord at the center of the curve can be calculated as;

$$2\pi R \rightarrow 360$$

$$\frac{2\pi R}{360} \rightarrow 1° \text{ and} \frac{2\pi R}{360} \rightarrow D°$$

Since arc length of 30.5 m corresponds to D°,

i.e $\frac{2\pi RD}{360}$ = 30.5. $\frac{2\pi RD}{360} \rightarrow$ D° solving the equation we get

$$D = \frac{1750}{R} \quad (3.1)$$

The direction of curve is determined by the change in direction as seen in the direction of movement of trains. Left hand curve is there if the change in direction of the curve is in counterclockwise direction when seen in the direction of travel in multiple lines or in the direction of increasing kilometers in case of single line. Similarly, Right hand curve is there if the change in direction of the curve is in clockwise direction when seen in the direction of travel in multiple lines or in the direction of increasing kilometers in case of single line. For this research clockwise movement of a train is considered throughout the paper.

3.5 Movement of Vehicle on Curved Track

Curved track are either horizontal curved track or canted based on introduction of super elevation on the track to counteract the lateral centrifugal force developed due to curving movement of vehicle by lateral centripetal force. Horizontal curved track is constructed on level horizontal plane without an introduction of super elevation while canted track constructed with introduction of super elevation. This thesis is only focus on canted tracks throughout the paper. The curving motion of the vehicle on horizontal track or canted track is accompanied with wheelset laterally movement. This causes other changes that are going to be discussed in the next subsequent units such as on rolling radii of the wheels, contact angles in between wheel and rail and wheelset roll angle etc.

When a train moves over the curve track, following are to be achieved:

(A). Continuous change in direction: The change in alignment of a vehicle on a curved track is affected by the rails. The rail closer to the center of track is called inner rail and the rail farther away from the center of curve is called outer rail. The vehicle moving over the curve is pushed in proper alignment by the rails which are changing direction continuously.

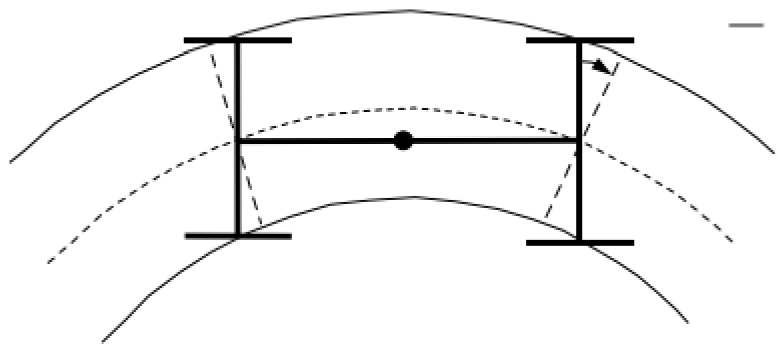


Figure 3.6: Vehicle movement on curve

The leading wheel of a bogie (or trolley) in case of a bogie vehicle and the leading wheel of a vehicle in case of a four wheeler vehicle moves with positive angularity, attacking the outer rail of the curve. The outer rail causes the wheel to change direction as seen in the above figure. Due to this, there are large lateral forces on the track as well as vehicle.

(B). Movement without slip: on a curved track, the outer wheel travels with a large radius as compared to the inner wheel. Therefore, a mechanism is required to ensure that the outer wheel does not slip during movement on a curve. The actual vehicle movement on curve is quite complicated but the vehicle traverses the curved path without appreciable slip due to coning in the wheels, which causes large diameter to travel on the outer rail and smaller diameter on the inner rail by slight shifting of the center of gravity of the vehicle towards the outer rail. This is explained in figure (3.7) and figure (3.8) below.

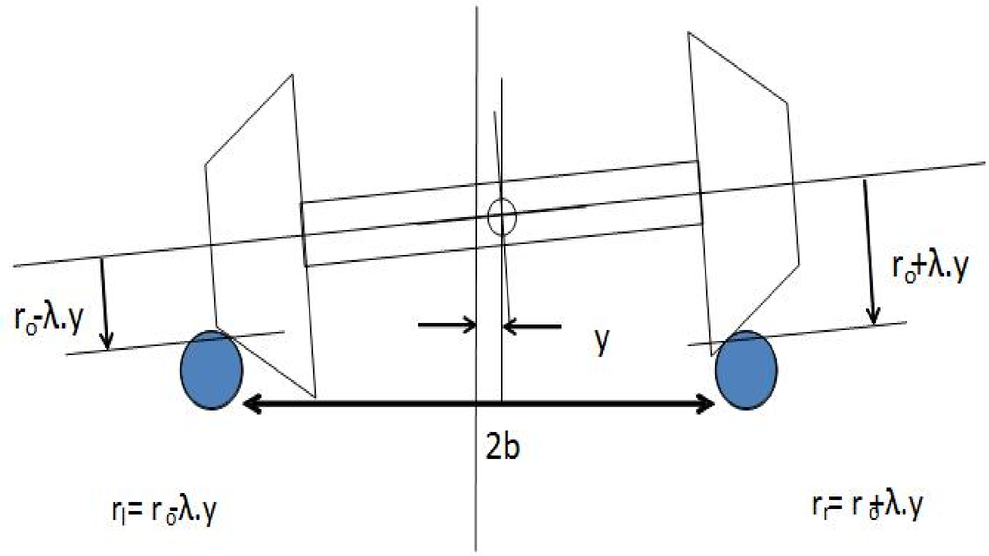


Figure 3.7: Inner and outer rail of track [31].

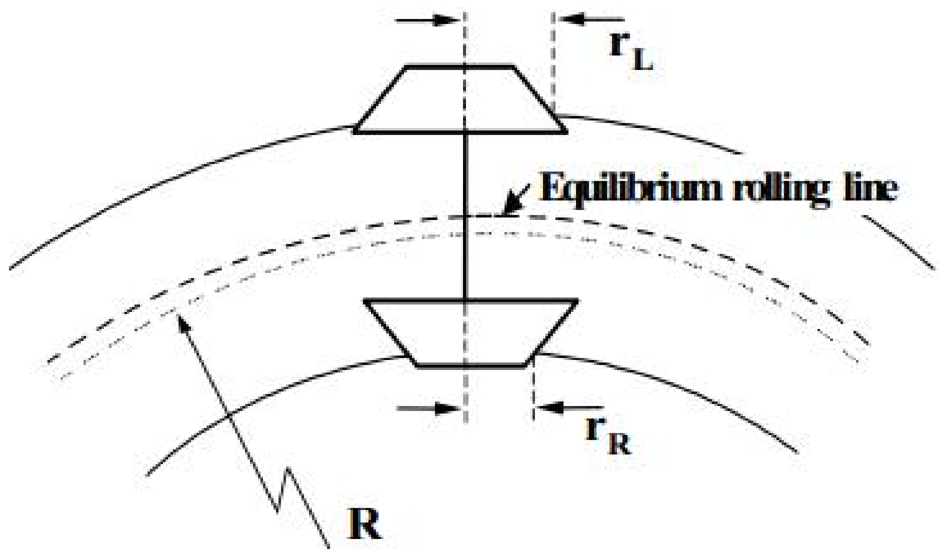


Figure 3.8: Rolling radius difference between inner and outer rail [31]

However, the above is possible only up to a certain radius of the curves and if the radius is sharper, the outer wheel will skid and inner wheel will slip in order to ensure that the axle moves as a unit in the desired direction.

(C). Forces on a vehicle during movement on curve: The vehicle passing over the curve continuously changes its direction over a curve. Due to the inertia, the vehicle tends to continue moving in the straight line but the forced change in direction of the movement gives rise to lateral acceleration acting outwards which is felt by the vehicle and all passenger/ things inside. This acceleration is called centrifugal acceleration and the force due to the same is called centrifugal force.

Therefore, in the railway track, the vehicle experience lateral forces when the vehicle travels in a curved path and if these are not secured, these will tend to move outwards. And, if the curve is not designed or maintained to proper geometry, the unbalanced forces in the lateral direction will go up. These forces will lead to higher maintenance costs and the comfort of the passengers as well as safety of the vehicle on the curve may also be affected. To counteract the effect of centrifugal force, the raising of outer rail with respect to inner rail is done. This raising of outer rail of a curve with respect to inner rail is referred to as a cant or super elevation, due to the vehicle being on slope, a component of weight starts acting towards the lower rail. This component, called centripetal force, acts opposite to the direction of the centrifugal force. The forces on a vehicle on the curve having cant is shown in figure (3.9).

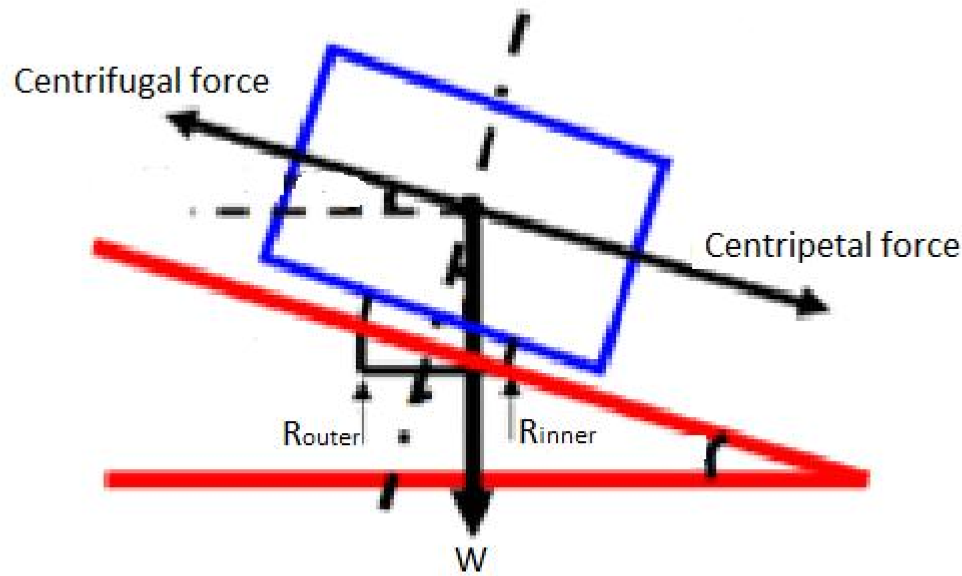


Figure 3.9: Force on vehicle moving on curved track

If, in some situation, outer rail of curve is lower than inner rail, the cant is said to be negative cant. In this case, the centripetal force will act in the same direction as centrifugal force [31].

3.6 Mathematical Analysis of vehicle on a curve

If a vehicle having mass M (weight W) is moving on a curve of radius R with a speed V, the centrifugal force experienced by the vehicle comes to $\frac{MV^2}{R}$. This force is acting at center of gravity of the vehicle in a perpendicular direction away from the center of the curve. If the curve is having cant, centripetal force will be acting towards the center of curve. When the two forces acting in the lateral direction match with each other, the vehicle is in equilibrium as far as lateral forces are concerned. In this situation, any person sitting inside the vehicle will not be able to differentiate between the motion on a straight or a curve due to the absence of lateral forces. The cant at which the equilibrium is there on a curve is called equilibrium cant. Conversely, the speed corresponding to cant in any curve is called equilibrium speed. By assuming the two forces (centrifugal and centripetal forces) are equal, the relation between equilibrium speed and equilibrium cant can be designed.

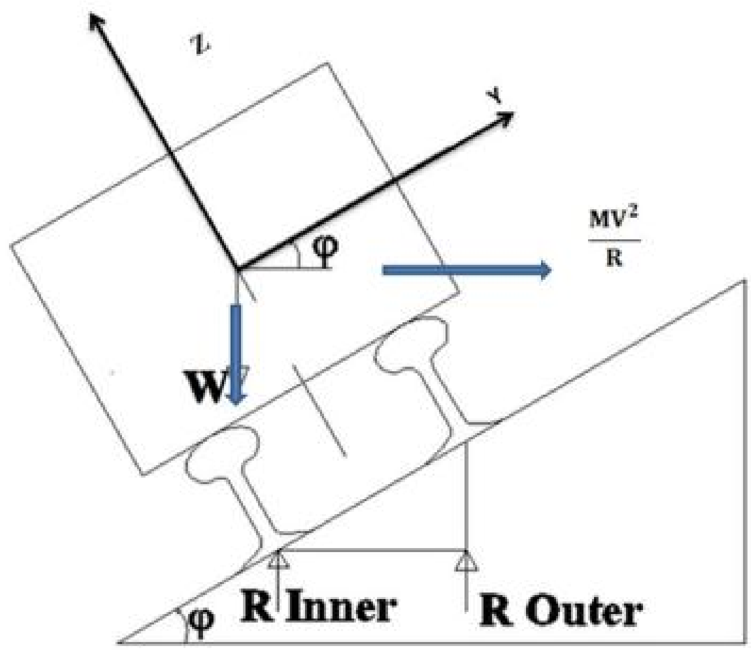


Figure 3.10 Forces on the vehicle on the canted curve

In case of quasi static curving the vehicle is exposed to two accelerations i.e., normal lateral which is vector sum effect of horizontal centrifugal acceleration and vertical gravitational acceleration. The resultant of the acceleration is the vector sum of these two components as shown in fig. above.

$$a_y = \frac{MV^2}{R}\cos(\varphi) - W\sin(\varphi) \quad (3.2)$$

$$a_z = \frac{MV^2}{R} W\sin(\varphi) + W\cos(\varphi) \quad (3.3)$$

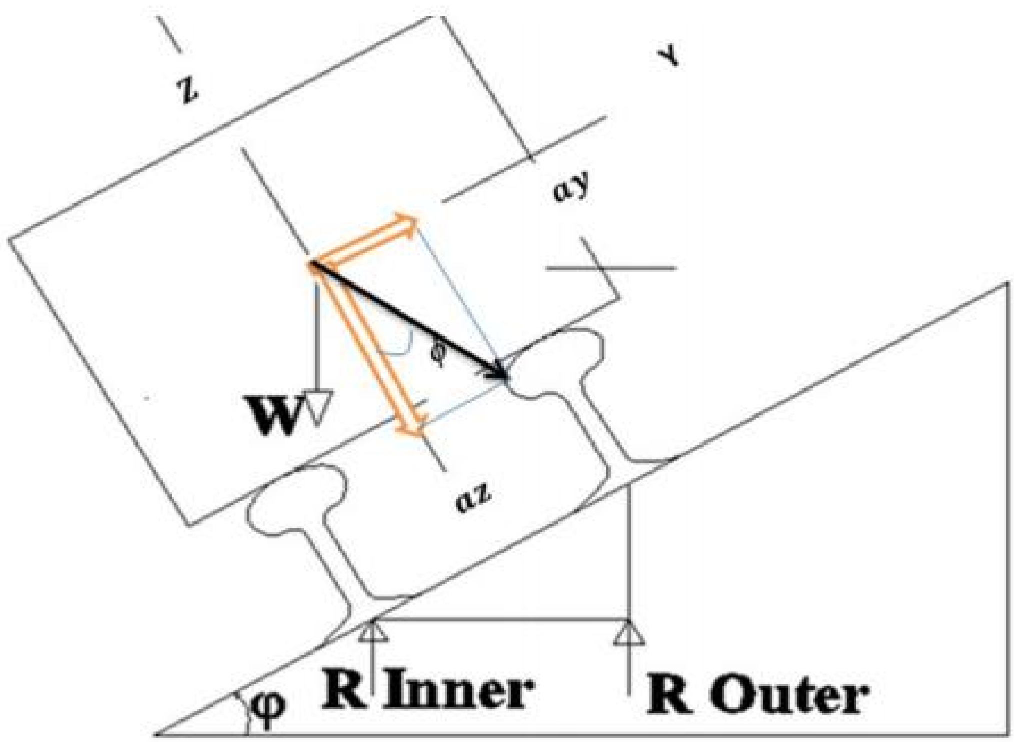


Figure 3.11: Lateral acceleration (a_y) and lateral force angle (ϕ)

The lateral force angle (ϕ) is related to lateral acceleration" a_y" and gravitational acceleration "a_z"

$$\phi = tan^{-1} \frac{a_y}{a_z}$$

The lateral displacement of a wheelset for canted track is calculated as

$$y= \frac{ro \times l}{R \times \lambda} \cos(\varphi) \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (3.4)$$

Where, y= lateral displacement

ro= rolling radius

R=curve radius

$\lambda = wheel\ conicity$

l =half of track gauge length

The lateral shift distance depends on both the cant angle in addition to the curve radius and other requirement that the vehicle should move with equilibrium speed. At low velocity the wheel flanges are in contact with the inner rail, on the other hand at high velocity the wheel flanges are in contact with outer rail so that the wheel travels from one side of conical wheel to the other.

3.6.1 Track cant

The difference between the level of the two rails in a curve is called cant ht (also called superelevation) and is arranged to compensate part of the lateral acceleration, see Figure 3.12. A cant angle arises where a cant is arranged. The angle can be determined by

$$\varphi_t = sin^{-1} \frac{h_t}{2b_o} \qquad (3.5)$$

Where $2b_o$ = 1.500 m on standard track gauge.

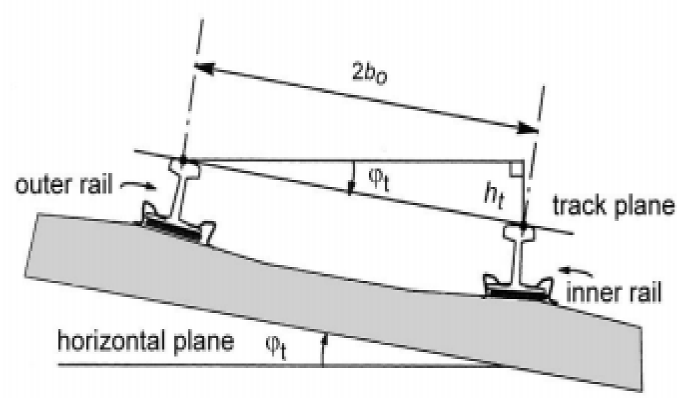


Figure 3.12 Cant h_t and cant angle φ_t

The cant which gives lateral acceleration (a_y) =0, for a given radius and given vehicle speed is called equilibrium cant, h_{eq}. The equilibrium cant is thus

$$h_{eq} = \frac{2b_o}{g} \cdot \frac{V^2}{R} \quad (3.6)$$

Equation (3.8) is based on SI-units. In practice it is useful to express speed V in [km/h] and cant in [mm] shown in Equation (3.9)

$$h_{eq,mm} = \frac{2b_{o,\ mm}}{g} \cdot \frac{V^2}{3.6^2 \cdot R} \quad (3.7)$$

The equation can be simplified further if the values $2b_o$ for standard track gauge and the gravitational acceleration g are used.

$$h_{eq,mm} = \frac{1500}{9.81} \cdot \frac{V^2}{3.6^2 \cdot R} = 11.8 \cdot \frac{V^2}{R} \quad (3.8)$$

The vehicle speed giving a_y= 0 for a give radius and a given cant is called the equilibrium speed or balanced speed, v_{eq} and is defined as

$$v_{eq} = \sqrt{\frac{R \cdot g \cdot h_t}{2b_o}} \quad (3.9)$$

Thus, at equilibrium speed the lateral acceleration in the track plane, a_y is zero. With speed expressed in [km/h] and cant in [mm] this equation transforms to (for standard gauge) [31]:

$$v_{eq} = \sqrt{\frac{R \cdot h_{t,mm}}{11.8}} \quad (3.10)$$

3.6.2 Longitudinal slippage

Longitudinal creepage (ξ): arise inter alia through the difference in effective rolling radii of the wheels, left and right, due to conicity; also through accelerating or braking couples and, very important, through the rotation velocity αv of the yaw angle α, by which the left wheel moves with a different velocity over the rail than the right wheel.

The difference in the circumference of two curved rails (right rail (R1) and left rails (R2)) when assuming is equals

$2\pi(R2-R1) = 4\pi l$

It is assumed to be compensated by the conicity of the wheels up to a certain level of lateral displacement (look fig below) but when maximum possible lateral distance is shifted the wheels are exposed for longitudinal slip. It implies the wheels longitudinal sliding distance are responsible for the two wheels to be placed on the same radial position.

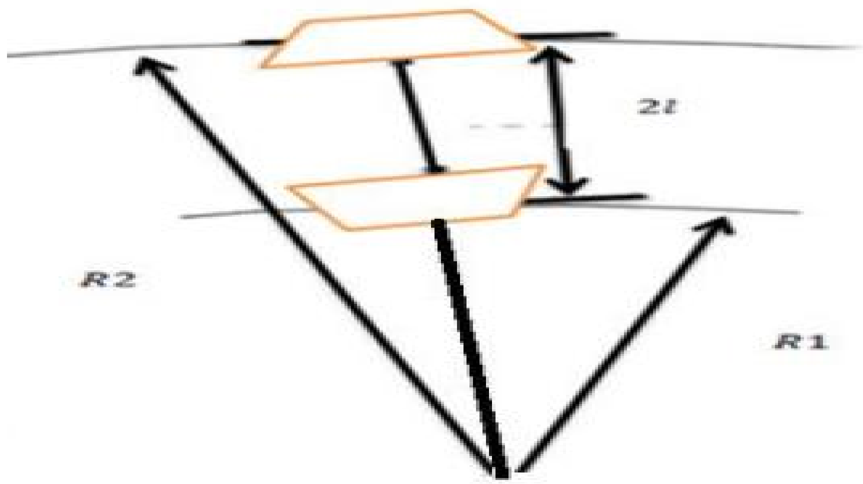


Figure3.13: The two Rail Radius's with Gauge Width 2l.

Then the longitudinal slippage can be calculated from rolling radius differences and the two curved rails as follows:

$r_r = r_o + y\lambda$ and $r_l = r_o - y\lambda$

their difference becomes , $\Delta r = 2y\lambda$

From the two wheels the distance travel difference for same time and same angular velocity of the wheelset, the rolling radius difference creates this amount linear displacement

$$\text{Linear displacement} = \omega t \times 2\lambda y$$

Let the wheel makes displacement equals the circumference of curved track with the same angular velocity angular displacement becomes

$$\boldsymbol{\theta} = \omega t = \frac{s}{ro}$$

But this also related with the radius of curved road.

$$S = 2\pi R.$$

Then rolling radius difference becomes

$$\boldsymbol{\Delta} r = 2y\boldsymbol{\lambda} \times (2\pi R/ro)$$

And it is the amount of reduction not to slide on the rail by the help of conicity from full curved motion. So,

Sliding for one full revolution= $4\pi\ (1- \boldsymbol{\lambda} y\frac{R}{ro})$

Dividing both side for the full circumference of the track and equally shared for both rails since there are two rails. Therefore longitudinal slip at right rail and left rail respectively as follows

$$\xi xr = (-\frac{1}{R} + \frac{\lambda y}{ro}) \quad (3.11)$$

$$\xi xl = (\frac{1}{R} - \frac{\lambda y}{ro}) \quad (3.12)$$

Therefore the vehicle wheels are slipping the inner rail forward and the outer rail is slip backward. But when the vehicle wheels are in flange contact to the inner side of the rail the advantage gain by conicity affects the system negatively since this creates with larger rolling diameter at the inner rail. So slip for left flanging becomes

$$\xi xr = (-\frac{1}{R} - \frac{\lambda y}{ro}) \quad (3.13)$$

$$\xi xr = (\frac{l}{R} + \frac{\lambda y}{ro}) \quad (3.14)$$

The sliding distance on two point contact is by wheel flange is determines by assuming the rail side contact is completely sliding. So the longitudinal creepage at side point contact becomes

$$\xi xs = 1 \quad (3.15)$$

3.6.3 Lateral Slippage

The velocities $v2'$ and $v2$ makes an angle α one to another. Hence the difference velocity $v2' - v2$ has a component in the y direction; velocity Vy give rise to lateral creepage.

$$\xi y = \frac{v2' - v2}{v} \quad \text{...(3.16)}$$

3.6.4 Spin Creepage

It is the difference in angular velocity of wheel within contact patch. The angle of contact is high on the flange to give lateral curving forces that guide the wheelset around the curve of the outer rail. At this point the region of contact is inclined and it produces the significant difference in rotational velocity within the area of contact. That means the innermost and the outer most parts of the region spent an equal time in contact with the rail.

$$\xi sp = \frac{\omega 2 - \omega 1}{v} \quad \text{...(3.17)}$$

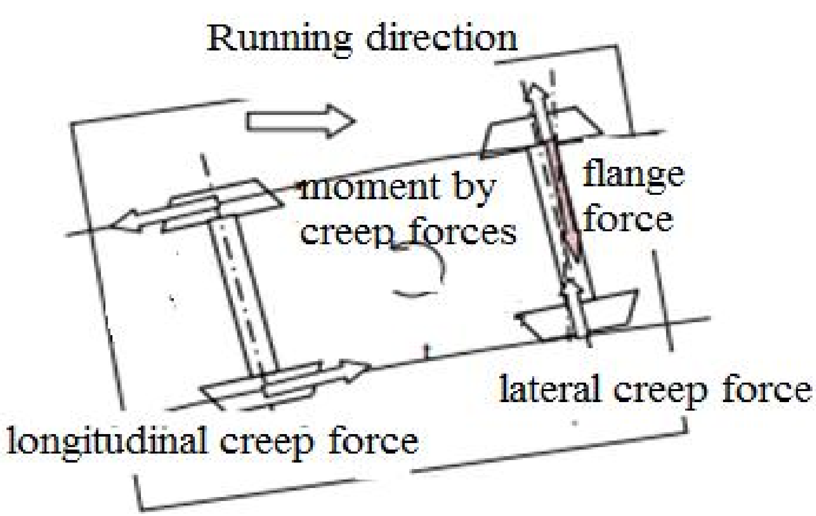


Figure 3.14: Bogie attitude and contact force in curving [33].

3.7 Dynamic Vehicle load analysis

3.7.1 Vehicle loads

When the vehicle moving on the curved track, the lateral centrifugal force are added on the system and it makes a change in rolling radius difference, angle of contact, and lateral force angle. For the case ϕ= 0, it represents horizontal curved track, at same time lateral force becomes $\frac{mv^2}{R}$, normal force Fz =mg, and the motion would be either of one point contact or two point contact depending on lateral loading and friction conditions. Whereas if ϕ> 0 a track is called canted track. Similar to the above the motion would be either of one point contact or two point contacts depending on vehicle speed. The difference with the former case is the flange contact may happen on both sides of rails (at inner rail when the vehicle speed is lower than equilibrium speed and outer rail at speed is greater than the equilibrium). The general case for the vehicle moving on the curved track is shown as in fig. below. Center of mass a vehicle considered from A located (750j + 1653k) and from B located at (-0.75j + 1.653k).

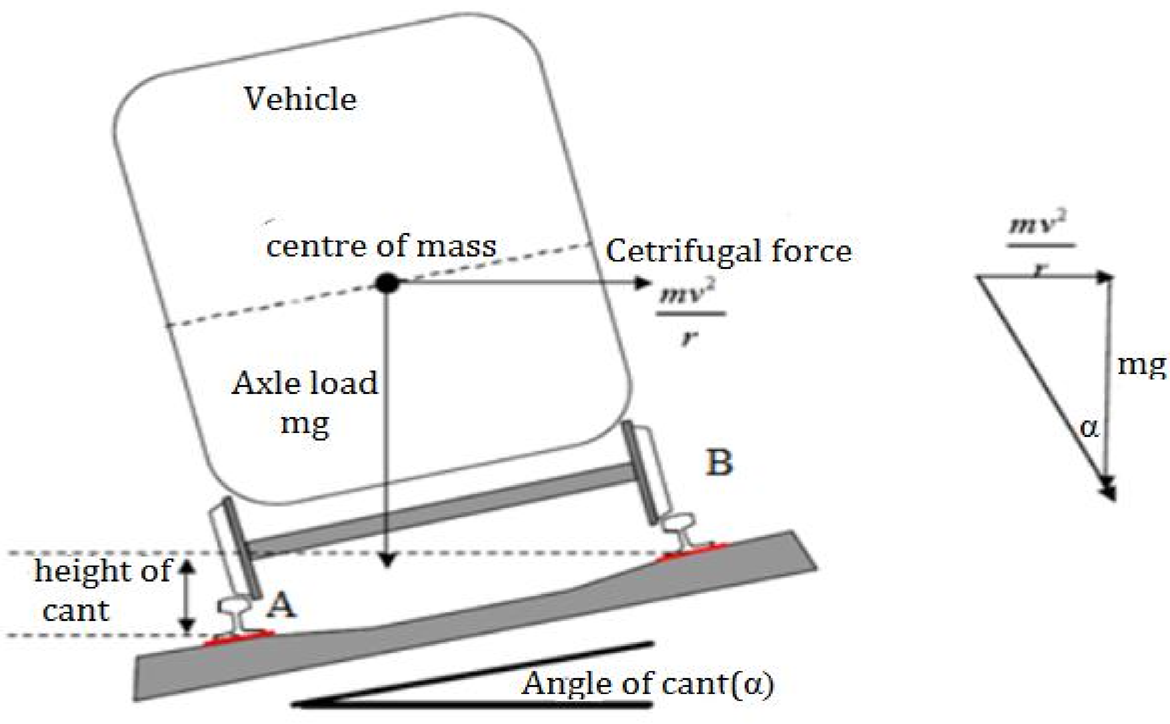


Figure 3.15: Vehicle Loads

3.7.2 Contact force at left and right rails

A) Normal load: As can be seen from figure 3.15, represented as FA and FB

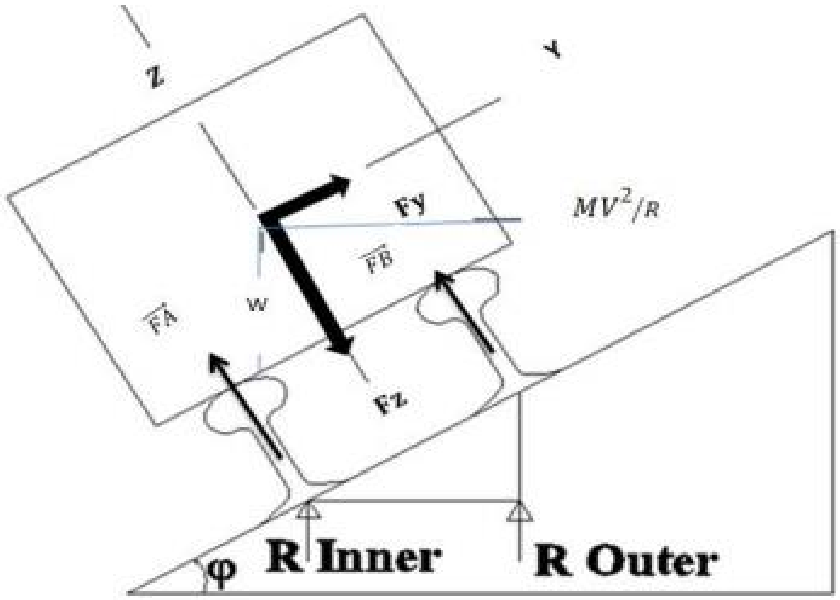


Figure 3.16: Loadings in the inner and outer rails

Ø= tan^{-1}(Fy/Fz)……………………………………………………………..………. (3.18)

R= -Rcos Ø$\vec{k}$ + RsinØ$\vec{j}$ ………………………………………………..………… (3.19)

From first law of equilibrium ∑Fz = 0, ⟶ FA$\vec{k}$ +FB$\vec{k}$ = Fz

From second low of equilibrium

∑MA = 0, ⟶ (0.750$\vec{j}$ + 1.653$\vec{k}$) × (-RcosØ$\vec{k}$ +RsinØ$\vec{j}$) + 1.5$\vec{j}$ × FB$\vec{k}$ (0.750$\vec{j}$ + 1.653$\vec{k}$) ×

(-FZ$\vec{k}$ + FY$\vec{j}$) +1.5$\vec{j}$ ×FB$\vec{k}$ (-0.750FZ$\vec{\imath}$ – 1.653FY$\vec{\imath}$) +1.5FB$\vec{\imath}$

FB$\vec{k}$ = 0.5FZ + 1.1Fy………………………………………………………………..….. (3.20)

∑MB =0, ⟶ (-0.750$\vec{j}$ + 1.653$\vec{k}$)×(-RcosØ$\vec{k}$ +RsinØ$\vec{j}$)+ (-1.5$\vec{j}$×FA$\vec{k}$) (-0.750$\vec{j}$ + 1.653$\vec{k}$) ×

(-FZ$\vec{k}$ + FY$\vec{j}$) + (-1.5$\vec{j}$ ×FA$\vec{k}$)(0.750FZ$\vec{\imath}$ – 1.653FY$\vec{\imath}$) – 1.5FA$\vec{\imath}$

FA$\vec{k}$ = 0.5Fz – 1.1 Fy ……………………………………………………………….…. (3.21)

3.7.3 Modeling of Wear rate

The wear that inevitably occurs between the wheel and the rail while the vehicle is moving depends on a large number of factors, among which sliding phenomena inside the contact patch, the normal force transmitted, the friction coefficient (lubrication conditions), size and shape of the contact patch etc. Wear on wheels and rails makes it necessary for equipment to be replaced when the upper safety limits have been breached and, as a general rule, the vehicle also sustains losses in terms of dynamic performance. Worn profiles tend to be less stable and show lower performance levels when negotiating curved tracks, and this makes reducing the wear index a major factor in the design of railway vehicles. At present there are a number of models attempting to quantify the wear index that include some of the abovementioned parameters. In any case, first it is essential to solve the contact problem. This leads to correct location of all contact patches and provides knowledge as to the size and shape of each patch, data which are essential for the application of any wear model.

Most used theories in railway dynamics assume that wear is proportional to the energy dissipated within the contact patch, calculated as the scalar product of tangential force on the contact and the value of creepage. Models which assume the material loss is proportional to the frictional energy dissipated in the contact patch ($T\gamma$). $T\gamma$ is expressed as the sum of the products of the creepage and creep force for the lateral, longitudinal and spin components, as illustrated below:

$$T\gamma = [\, T_y\gamma_y] + [\, T_x\gamma_x \,] + [\, M_z\omega_z \,] \quad \ldots\ldots (3.22)$$

Where T_y, T_x are lateral and longitudinal creep force respectively, and M_z is spin creep moment. γ_y, γ_x are lateral and longitudinal creepages, and ω_z is spin creepage, the spin creepage contribution is dismissed; the result is called the ***wear number*** [23].

3.8 SIMPACK Model

In order to perform the computer simulations, a suitable model have to be created first. This could be done using different Multi Body Simulation software. In this project, one of the most well-known software accepted by industrial and academic communities called Multi Body Simulation (MBS) software SIMPACK is used. It should be noted that SIMPACK 9.6 version is used for both modules, Pre and Post-Processor.

3.8.1Wheelset model

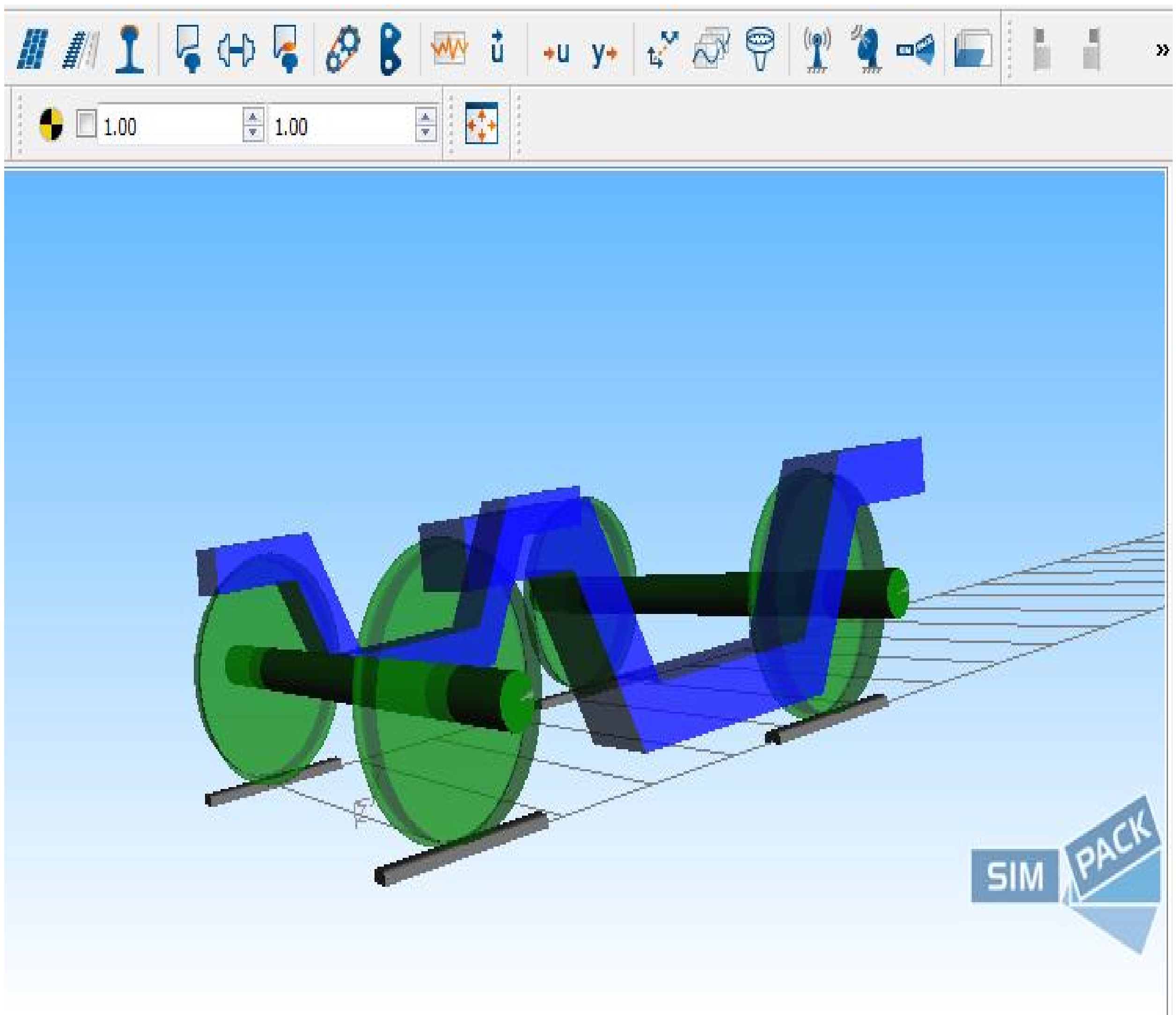


Figure3.17: A Wheelset with Bogie Frame Model.

3.8.2 Bogie model

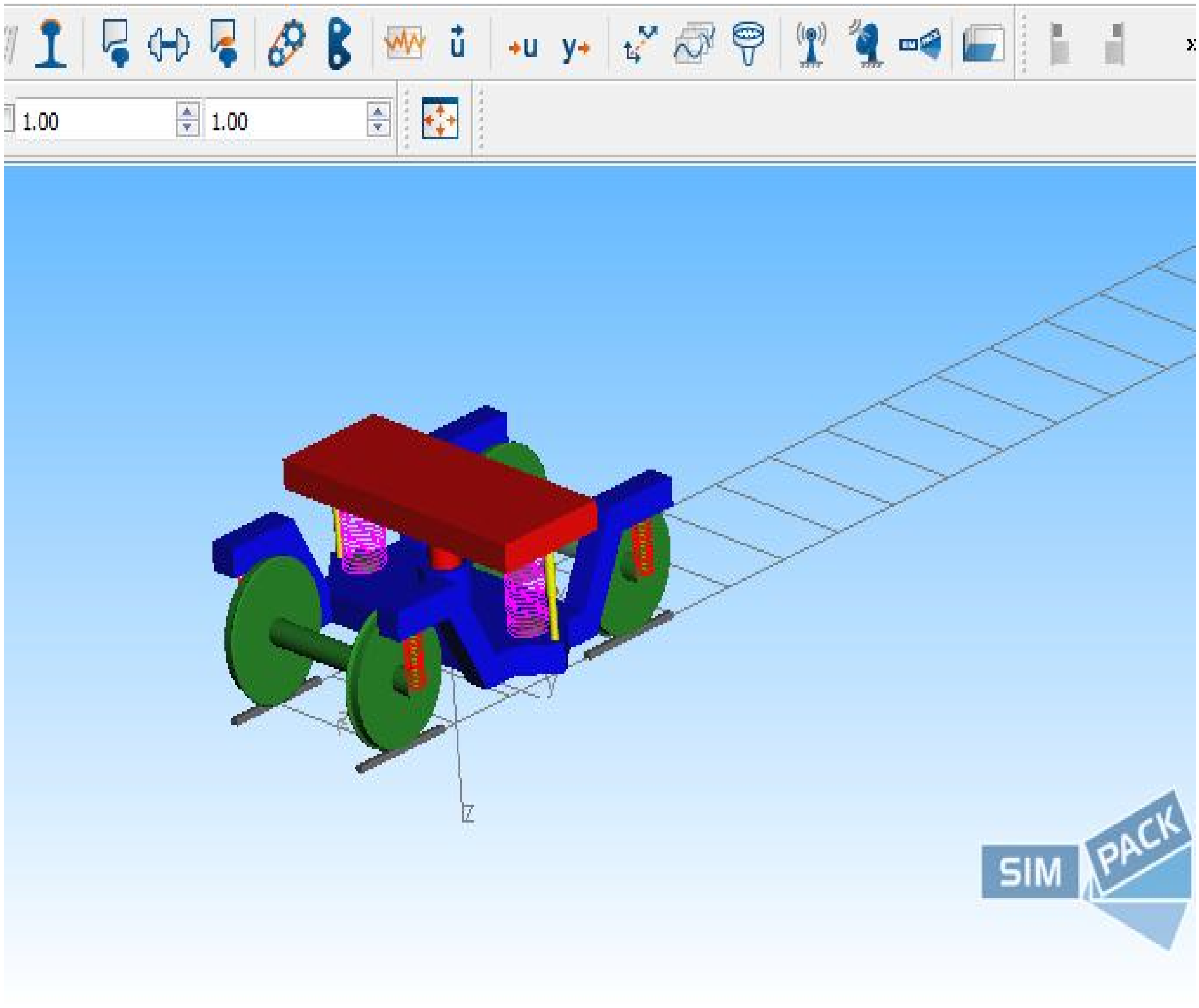


Figure3.18: Bogie Model

3.8.3 Vehicle model

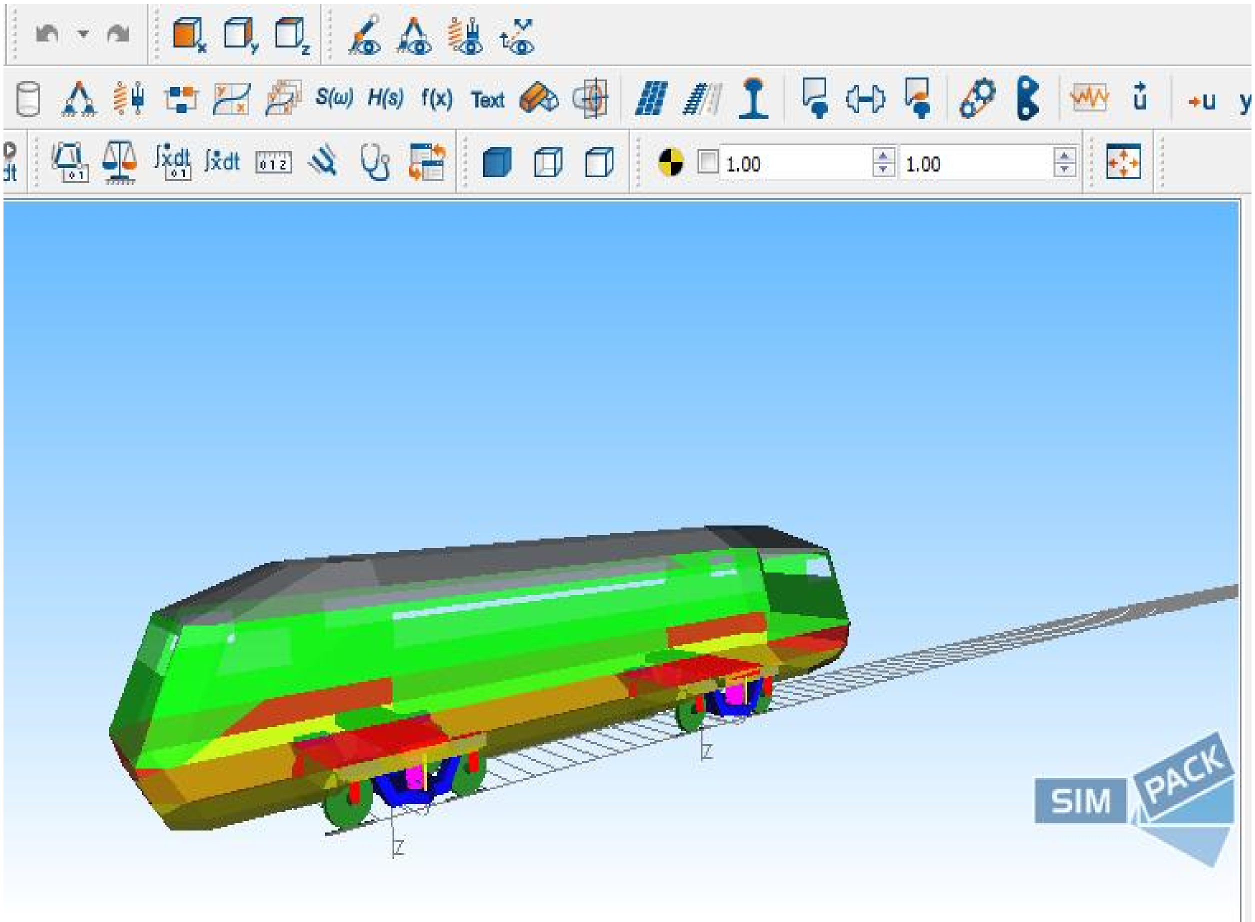


Figure 3.19: Vehicle model used during simulations.

CHAPTER 4

4. MATERIAL AND CONDITIONS

4.1 MATERIAL

In this paper the material property is taken from standard rails. UIC 60 Rails that is made from high carbon steel. Since the rails have to withstand the impact load, friction and stress; they should have sufficient strength, hardness, toughness and good welding performance.

Table 4.1: Rail material selection

Serie no.	Parameters name	Values
1	Rail strength	900-1200 MPA
2	Poison ratio	0.3
3	Younge's modulus	207 GPA

4.2 CONDITIONS

The general conditions considered during the wheel rail contact simulation are listed below.

- Throughout the paper a vehicle represents a vehicle which has two bogies one front the other behind. Each bogie consists of two wheelsets or four wheeler of expected 10 tons service. The mass is assumed equally shared for each axle of the bogie.
- The vehicle on the track assumed right hand travel and the wheel set on the left is as outer rail and right is the inner rail.
- The other important assumption is curving motion with maximum curving potential of the vehicle and the perfect driver otherwise their effect significant than that the curved nature of the track.
- The tribology of curved rail determinate from the leading wheelset. Since the trailing wheelset is assumed always move to the center of rail axis. The leading wheelset only affect the behavior of curved track. The calculation is based on centrifugal loading only neglecting other forces such as wind. So all calculation from the point of axles loads and forces generated due curving motion of the vehicles.
- Wear occurs on both wheel and rail as a result of the relative velocity difference of the two contacting bodies in the contact zone, where part of the contact is in adhesion and the rest is sliding.
- Coefficient of friction under dry condition is assumed 0.4 throughout simulation of vehicle model.

CHAPTER 5

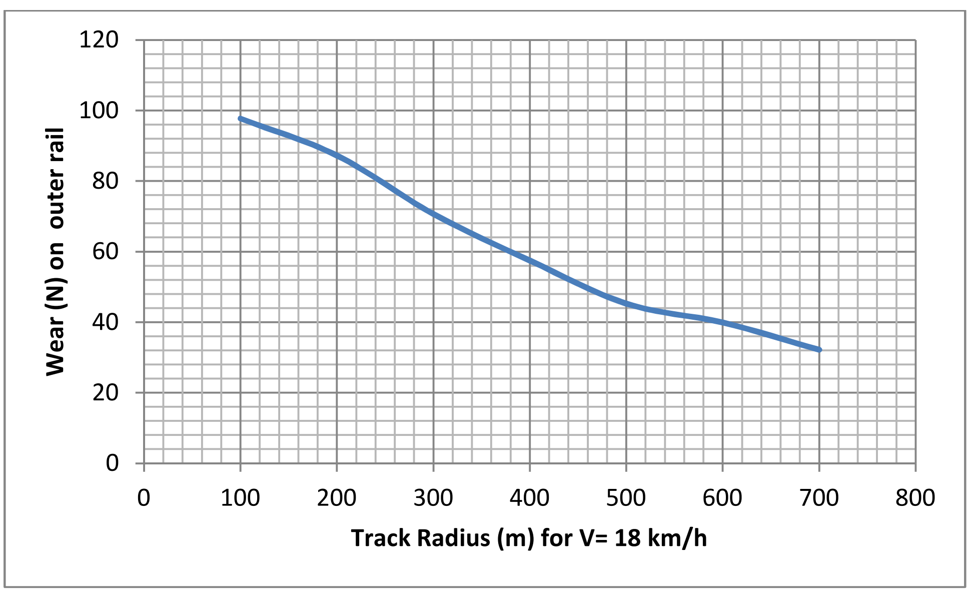
5. RESULT AND DISCUSSION

5.1 Introduction

This paper analyses the mechanism of curving motion, forces generated and their influence on rail wear condition. Here in this section it discussed the influences of the curve radius of track using mathematical equations and Simpack vehicle model. Simulation of vehicle model is conducted using Simpack software. Relations between different parametric results from simulation are analysed using graphs and most results have positive relation with theoretical background of wheel rail dynamic interaction.

5.2 Result Analysis

5.2.1 Wear property Versus Curved Track Radius



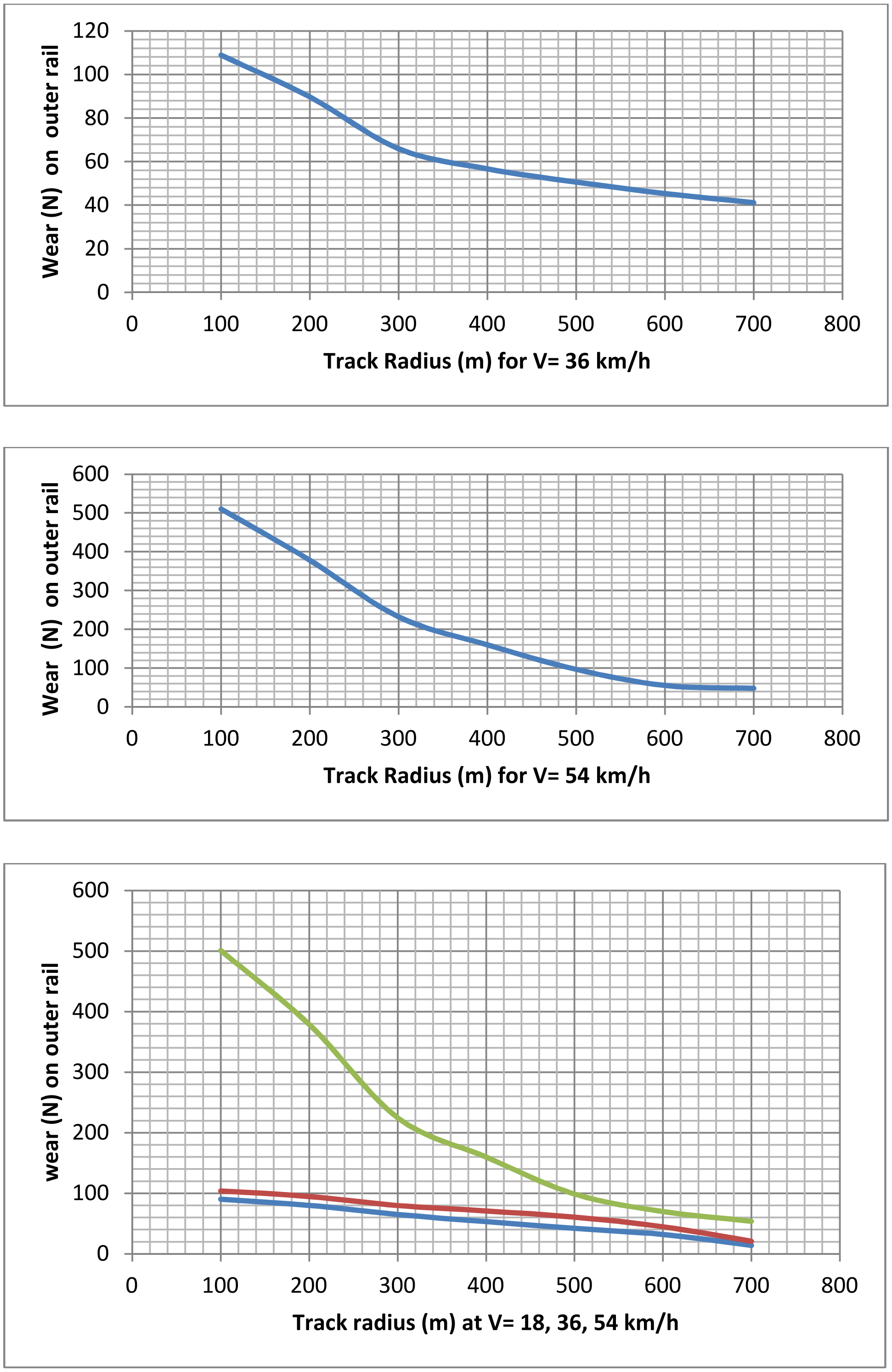
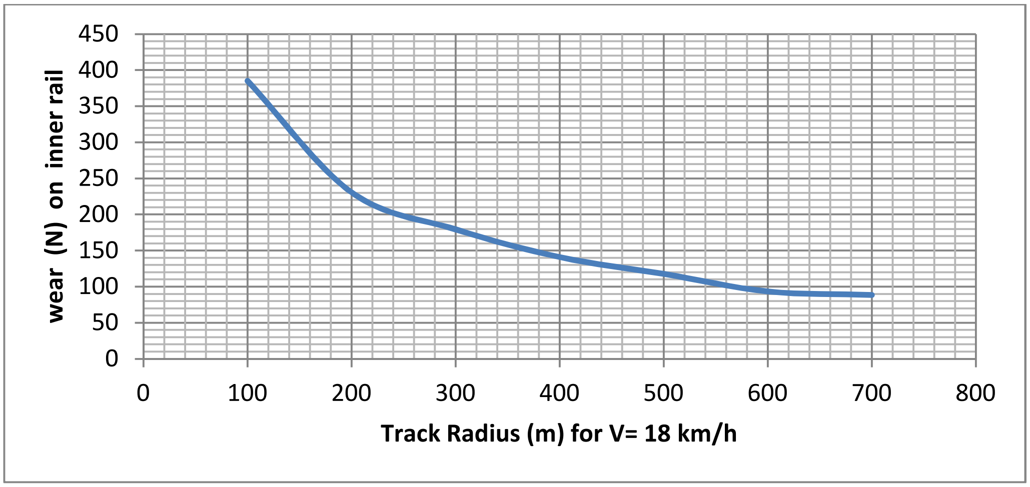
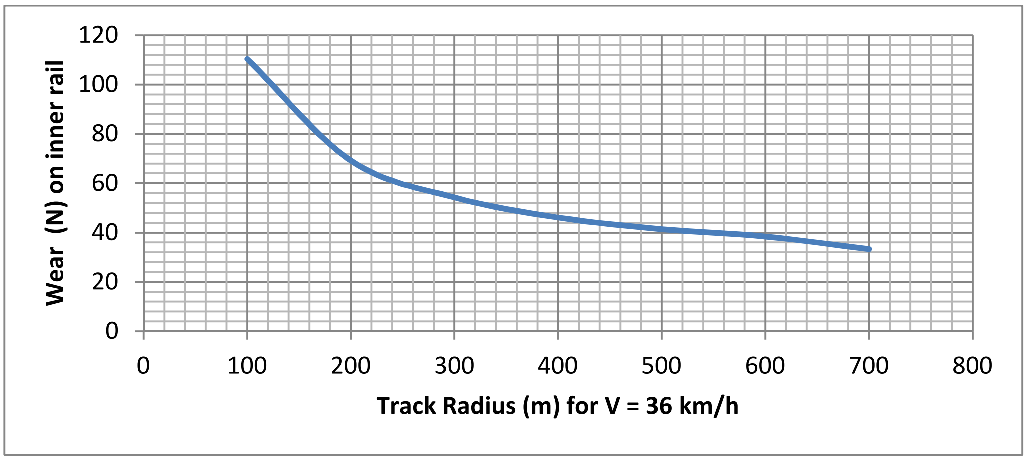
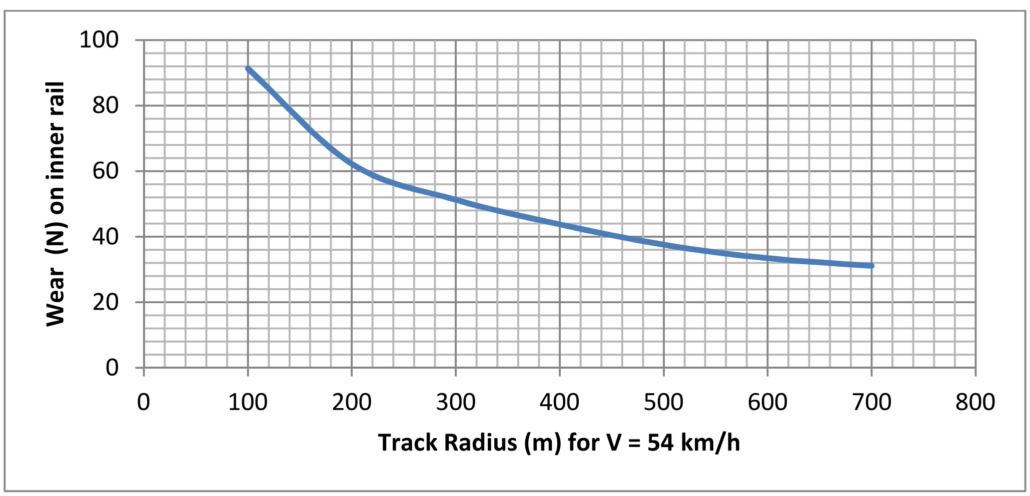


Figure 5.1: The Graph shows Left (Outer) Rail wear as a function of Radius of curved track in different forward velocity.


450
400
350
300
250
200
150
100
50
0
wear (N) on inner rail
0
100
200
300
400
500
600
700
800
Track Radius (m) for V= 18 km/h


120
100
80
60
40
20
0
Wear (N) on inner rail
0
100
200
300
400
500
600
700
800
Track Radius (m) for V = 36 km/h


100
80
60
40
20
0
Wear (N) on inner rail
0
100
200
300
400
500
600
700
800
Track Radius (m) for V = 54 km/h

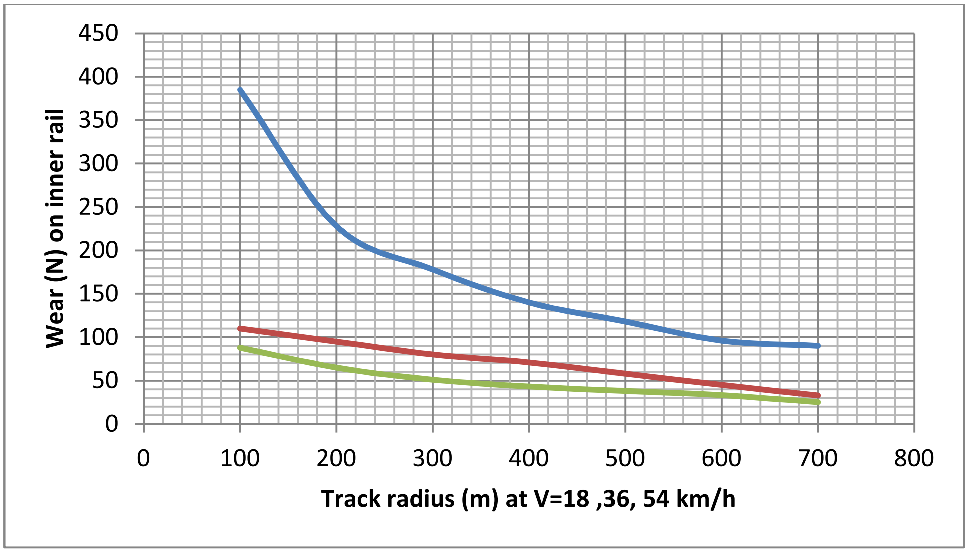


Figure 5.2: The Graph shows Right (Inner) Rail Head wear as a function of Radius of curved track in different forward velocity under dry condition.

The wear property of rail at left (outer) side of a canted curved track as shown at **fig. 5.1** and can be summarized as the left rail wear have different property with the vehicle speed (less than and greater than equilibrium speed). When the vehicle speed is above equilibrium speed the outer rail is highly damaged as the radius becomes sharper but for the vehicle speed lower than equilibrium speed the wear has decreased with decreased of radius. The wear property of rail at right (inner) side of a canted curved track as shown from the graph above **fig. 5.2** can be summarized as the right (inner) rail wear have a property of decreasing with increase of curve radius. The higher the vehicle speed the less head wear at right rail head.

5.2.2 Pressure on contact patch versus Curved Track Radius

From **fig. 5.3** the slope of the graph increases with the increase of radius of the track. This indicates that the pressure on the contact patch increases with increase of radius. As the radius increases the contact area on the contact patch decreases. When a vehicle moves on sharp curves two point contact will occur, these are flange and tread contact. Due to this the contact area increases and pressure on contact patch decreases. As the radius of curve increases the flange contact decreases and tread contact exist, as a result the contact area of contact patch will decrease and the pressure on the contact patch increases.

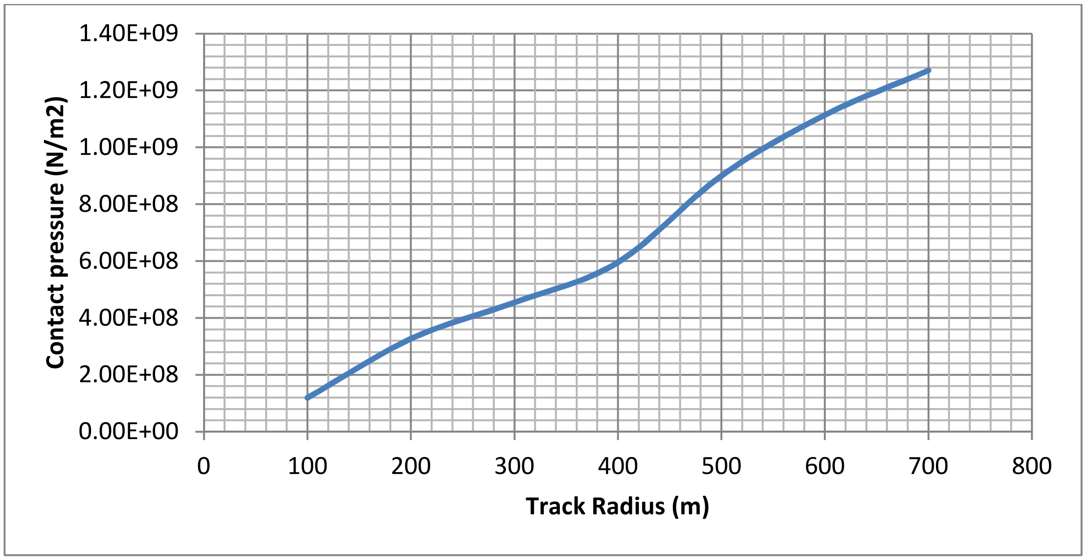


Figure 5.3: The graph shows the effect of curved rail track on pressure on the contact patch

5.2.3 Contact Patch area as a function of Curved Track Radius

From **fig. 5.4** it can be concluded that area of contact patch and radius of curve have inverse relation. As the radius of the curve increases the area of contact patch will decrease. When a vehicle negotiates a sharp curve the wheelset tends to move laterally due to the conicity of wheel and lateral force. As the curve becomes sharper flange will contact with the rail side. This phenomenon increase the area of contact but when the radius of curve increases flange contact will decrease.

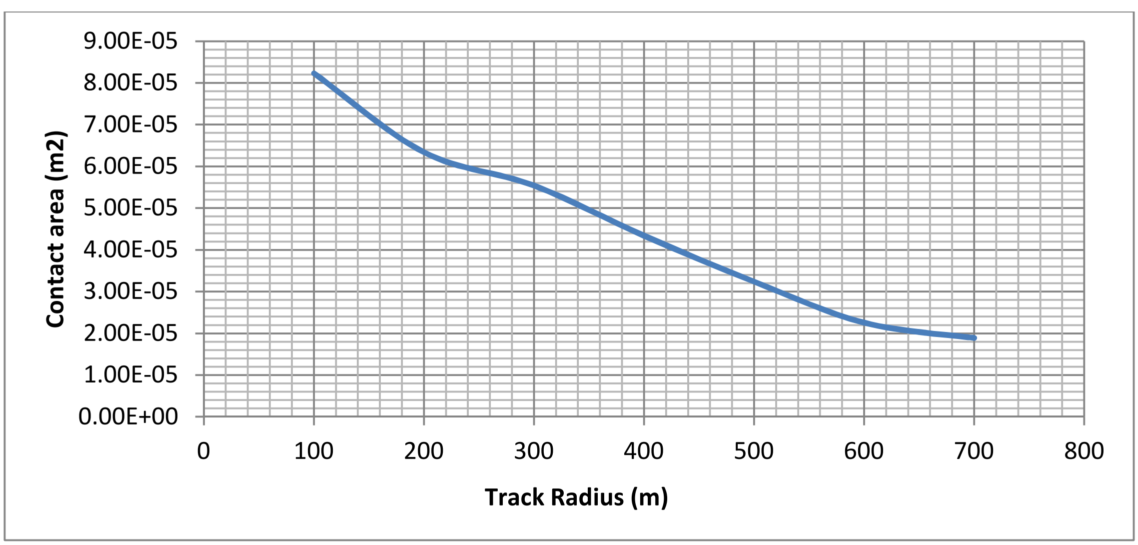


Figure 5.4: The graph shows the effect of radius of curved rail track on area of contact patch.

5.2.4 Wear property versus Contact patch area

Below **fig. 5.5** shows the interaction between wear property of rail and contact patch area , so it can be concluded that wear property and contact patch have a direct relation i.e as the area on the wheel rail contact patch increases the material removal rate also increases.

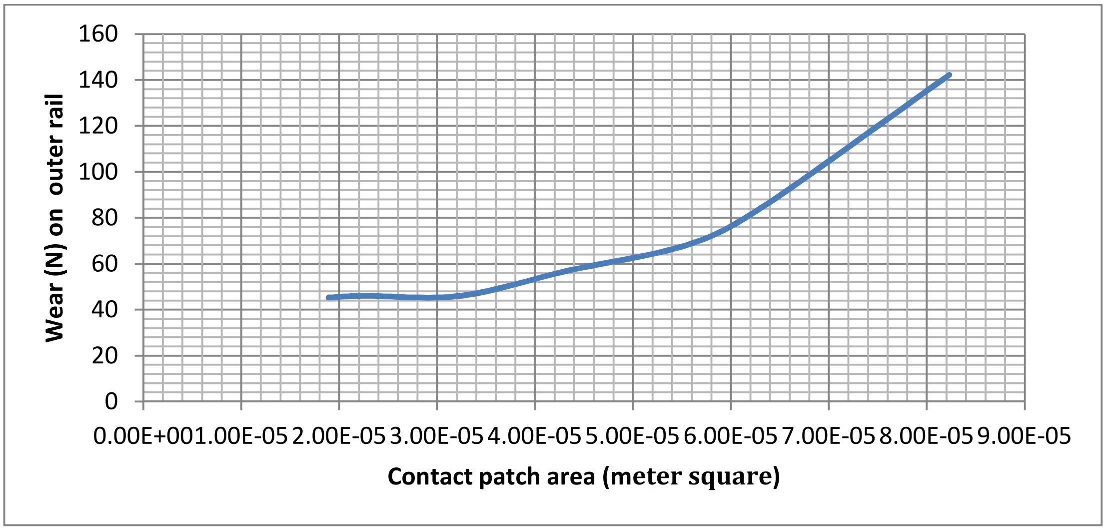


Figure 5.5: The graph shows the effect of contact patch on rail wear.

5.2.5 Wheelset yaw angle versus Track Radius

Fig.5.6 shows the interaction between yaw angle and radius of curved rail. From the graph it can be concluded that yaw angle and radius have an inverse relationship. As the radius of curved rail increases wheelset yaw angle decreases. On sharp curves a wheelset makes large yaw angle to traverse a curve but as the radius increases yaw angle decreases.

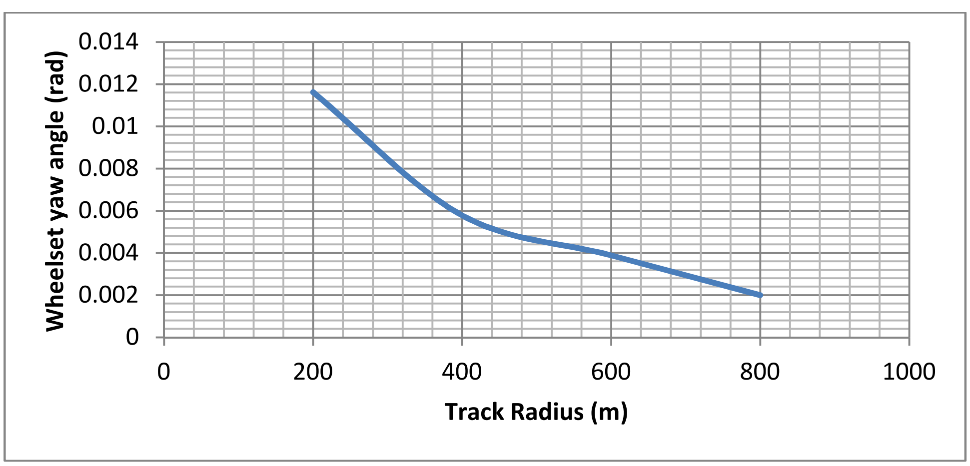


Figure 5.6: The graph shows the effect of Track Radius on yaw angle.

5.2.6 Wheelset yaw angle as a function of contact area

From fig. 5.7 it can be concluded that contact patch area and yaw angle have direct relationship. As the yaw angle increases area of contact patch also increases. When a vehicle is moving on sharp curves a wheelset rotates to some amount of yaw angle to traverse. The amount of yaw angle depends on track radius. On narrow or sharp curved track the wheelset makes large yaw angle. At large yaw angle two point contact occurs, wheel flange-rail gauge and wheel tread-top of rail, so area of contact patch is more but as the radius increases flange contact decreases and when radius is going larger tread contact exists, that means area of contact patch getting reduced.

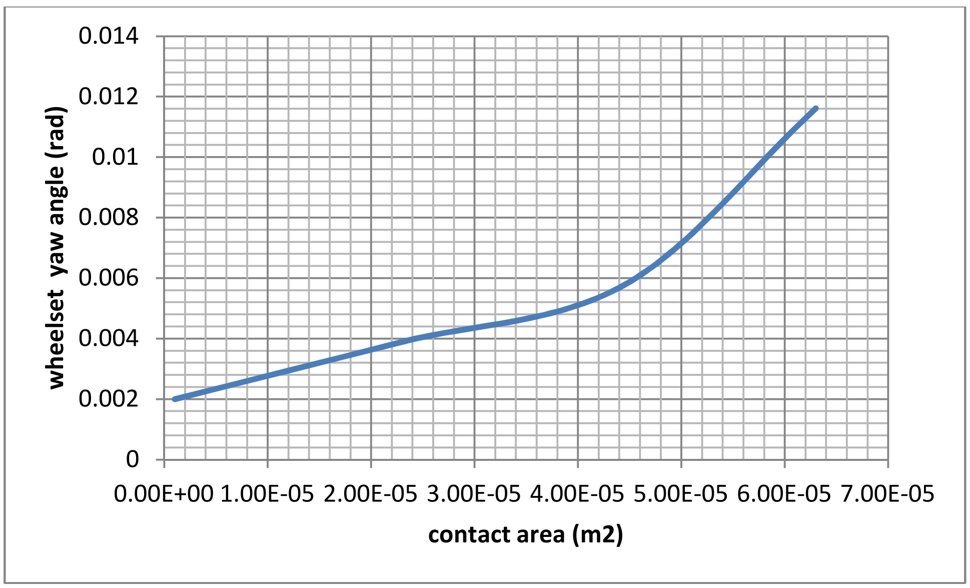


Figure 5.7: The graph shows the effect of yaw angle on contact patch area.

5.2.7 Wear property versus yaw angle

Below fig.5.8 shows the interaction between yaw angle and wear property of rail. It can be concluded that yaw angle and wear property have direct relationship. As the yaw angle increases wear of rail also increases. When the yaw angle increases contact patch area increases as discussed in fig.5.7 above that is due to wheel rail contact condition. As area of wheel rail contact patch is large wear on rail is also high. So it is generalized as yaw angle is larger wear on rail also high.

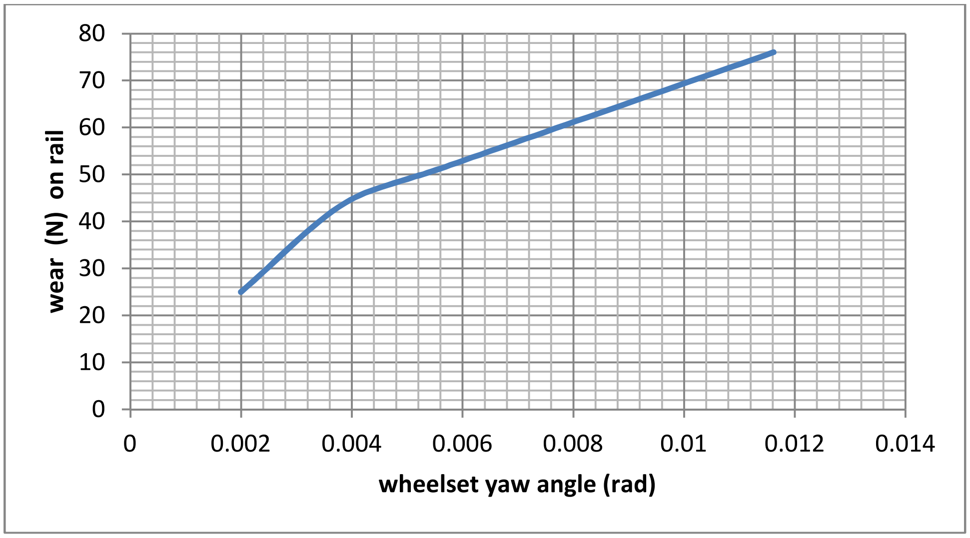


Figure 5.8: The graph shows the effect of yaw angle on wear of rail.

5.3 Validation of simulation results

To validate the software simulation results, the simulated results were compared with results obtained from other related research materials. Results are obtained from MIT international journal of mechanical engineering [34]. Finite element model of rail and wheel was created in 'ABAQUS' and simulated by considering yaw angle, camber angle and surface roughness. After simulation effect of wheel yaw motion (negotiating the curve) i.e yaw angle on contact area is discussed. From the analysis yaw angle increases contact area also increases.

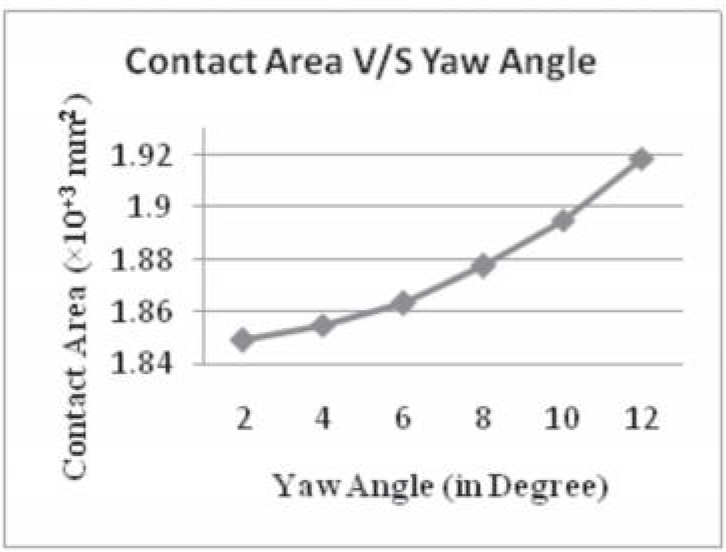


Figure 5.9: Graph between contact area and yaw angle [34].

Variation of pressure with yaw angle also analyzed and results from the figure 5.10 reveals that as the yaw angle increases, contact area increases and contact pressure decreases.

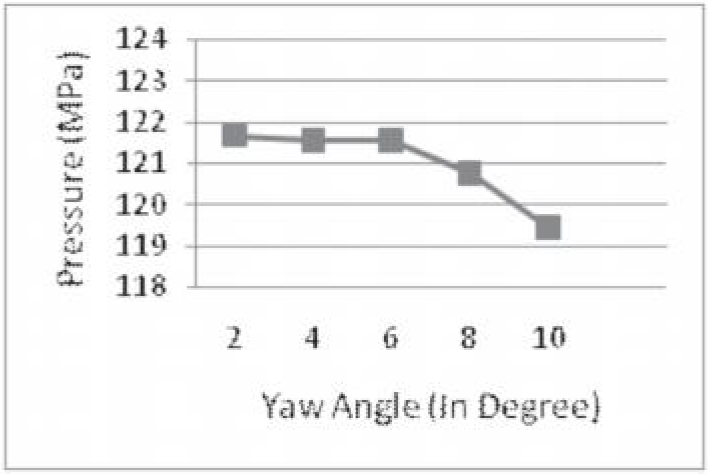


Figure 5.10: Graph shows contact pressure versus yaw angle [34].

CHAPTER 6

6. CONCLUSION AND RECOMMENDATION

6.1 Conclusion

The paper focused mostly on how radius of curved rail affects wear property of rail under dry condition during curving motion of a vehicle. These require understanding of mechanisms of kinematic and dynamic motion a vehicle on curved track and generation of contact forces in between the wheel and the rail.

The dynamics in the vehicle to the rail on canted curve was analyzed using a quasi-static approach which at instant time the vehicle is loaded vertically by its weight and laterally by centrifugal force. This lateral force is balanced by using centripetal force and its direction may be inwards and outwards depending on velocity and weight of the vehicle. It is applied on outer rails while the vehicle is moving above equilibrium speed and it is applied at the inner rail while the vehicle is moving below equilibrium speed.

The simulation is done using SIMPACK software which is more applicable in wheel rail dynamic interaction and results from each simulation is analyzed using Microsoft office excel. Simulation is done at different forward speed of a vehicle, wheel rail contact patch area, pressure on contact patch, yaw angle for different track radius under dry condition of friction coefficient 0.4. The result shows a very good agreement since wear of a rail associates with different factors.

Under dry condition the increase in radius of the rail has exponentially decrease the wear conditions at constant velocity on curved track at both rail head, and the outer rail side is damaged severely as the forward velocity increases. The inner and outer rail wear interchangeably increases as the vehicle speed is out of equilibrium velocity. The outer rail damages more than the inner rail when the vehicle is moving above equilibrium speed and the reverse happen when the vehicle moves below equilibrium speed. Wheel rail contact patch area increases with decreasing of radius of rail at the same time the wear volume also increases. Contact patch area and wear rate have direct relationship. Contact pressure and track radius have direct relation i.e as track radius increases contact pressure decreases due to decrease of contact patch area. Wheelset yaw angle has inverse relation with radius and direct relation with that of contact patch area and wear of rail.

Generally knowing the effects of this parameter on overall system helps for understanding the mechanism and can be added for practical rail way curved track design and speed restriction on curves.

6.2 Recommendations

Railway operating companies, rail track design engineers, rolling stock producing companies, rolling stock operators and any group related to this business area are recommended as follows:

- Curved railway tracks requires more attention than that of straight track due to the introduction of centrifugal force which makes the vehicle tends to move laterally during curving motion.
- Severity of wear can be reduced using equilibrium speed of the vehicle.
- Vehicle drivers must follow a vehicle speed restriction on curves if otherwise that leads to excessive wear on the contacting surfaces.
- Interchanging of inner rail and outer rail can be considered as a means to lengthen the service life of the rail.

6.3 Future Work

However, there are some good achievements were obtained, there are other works remaining which are equally influence wear property of rails like curve radius. To obtain a complete generalization to the point of rail wear, the following works are remaining

- Analysis of aerodynamic effect on vehicle moving in curved track.
- Influence of braking and acceleration on rail wear.
- Prediction of rail life under dry and lubricated condition.

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APPENDIX

Appendix: Definitions

Wear: damage to one or both surfaces, involving loss or displacement of material from a contacting surface.

Degree of the curve (D): the number of the angles subtended by 30.5 m chord of a curve.

Angle of attack: -angle between the wheel and the rail made by wheel to shift direction of movement.

Hardness of materials: - defined as the resistance to penetration of a material by an indenter.

Fatigue: - Most material will fracture when a small load is applied repeatedly.

Inertia force: a tendency of an object to continue its condition. If there changing direction the object tends to continue changing direction.

Equilibrium super elevation (cant): - The track super elevation (cant) needed to neutralize the horizontal acceleration due to curving.

Horizontal plane: - Plane of earth horizon.

Tractive Effort: the force applied to the rail by the wheel of the train to cause movement

Derailment: - overturning of the vehicle

Quasi-static condition: Condition which is static under a certain period, here typically in a circular curve. The equations are formulated based on the assumptions of constant radius at constant velocity of the vehicle.

Centripetal force: a component of weight acting towards the lower rail when the vehicle moving on canted track.

Balancing speed; where the cant is exactly balances the lateral acceleration.

Adhesion: the grip or force of attachment, produced by friction between the wheels and rails. It is required to keep the wheels from slipping.

Inner rail: The rail closer to the center of track.

Outer rail: rail farther away from the center of curve.

Creep: Creep forces are generated at the wheel/rail contact patch, by the much localized action of the wheel rolling on the rail.

Two-point contact; the situation (include contact between the wheel flange and rail gauge face in addition to the rolling contact at the rail head.

Sharp Curved track: the length of curve radius below 300m.

MGT: the amount of traffic in terms of million gross tone mass flow over the rail annually