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Prediction of Rolling Noise Due to Wheel/Rail Interaction

Thesis Submitted for the Completion of Masters of Science at Graduate School of Addis Ababa Institute of Technology

In

Mechanical Engineering

(Railway Mechanical)

By

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Advisor

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Declaration

I, Hassen Mohammed, declare that the thesis entitled 'wheel rail interaction and generation of rolling noise' and the work presented in the thesis are my own, and have been generated by me as the result of my own original thesis.

I confirm that this work is done wholly or mainly while in candidature for a master's degree in railway mechanical engineering at Addis Ababa Institute of Technology. No part of this thesis has previously been submitted for a degree or any other qualification at this University.

Hassen Mohammed

Sign -------------------

Date -------------------

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Abstract

Noise, caused by the interaction of steel wheel rolling on the steel rail is always present. The noise radiated by the railway system has significant influences on both passengers inside the trains and the residents along the railway lines. To overcome this problem, first we have to predict the noise level by applying and varying different parameters, like wheel load, materials of the wheel and rail, operating speed and so on. But in this paper i mainly consider only train operating speed to predict the noise generated between the contacting surfaces which is obtained from the contact pressure.

The objective of this research is to predict the noise generated between wheel/rail by applying different operating speed. The maximum contact pressure during the operation of the train is used to predict the noise level. This research is done by developing the 3D model using CATIA V5 R16, simulation using ANSYS v14.5 and analytical calculation.

In this research, the value of average maximum contact pressure obtained by simulation of ANSYS software on the whole area of ellipse (contact patch) is 511.7 MPa. This is due to the operating speed 80km/h and the minimum contact pressure on the whole area of ellipse is 351.68 Mpa when the operating speed is 60km/h, the corresponding sound pressures are ***268.15 dB*** and ***264.9 dB*** respectively. From the results, high speed is the cause of high contact pressure and this high contact pressure is the cause of high noise.

Keywords: wheel/rail interaction, contact pressure, and rolling noise.

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Notations

K_w: Constants that depend on the material properties of wheel

K_r: Constants that depend on the material properties of rail

K_3**:** Geometrical properties of both wheel and rail

a: Minor semi axes of the contact ellipse

b: Major semi axes of the contact ellipse

F: Vertical load

m, n :Hertz coefficients

ν_w: Poisson's ratio wheel material

$\boldsymbol{E_w}$: Young's modulus of the railway wheel material

ν_r: Poisson's ratio of rail material

$\boldsymbol{E_r}$**:** Young's modulus of rail material

K : Stiffness matrix of the system

E_x: Young's modulus in the x direction

Gxy: Shear modulus in the xy plane

Vxy: Major poison's ratio

v_{yx}: Minor poison's

$\boldsymbol{\sigma_x}$**:** Stress in x direction

$\boldsymbol{\sigma_y}$: Stress in y direction

$\boldsymbol{\sigma_z}$: Stress in z direction

Z: Contact point depth

UIC: International union of railways

1. Introduction

1.1Background

Wheel/rail interaction and noise has been studied extensively since the 1970s and models of rolling noise are now well established. A comprehensive reference covering the overall topic of railway noise and vibration, including analysis of rolling noise mechanisms, modelling and control [18].

Now a days in Ethiopia demand for rail transport services is increasing with growing populations and environmental awareness. Populations are becoming increasingly mobile, demanding improved opportunities for travel and tourism, as well as access to increasingly diverse products. Increasing demand for rail services caused by economic and population growth, increasing awareness of the need to reduce carbon emission means that a shift in transport mode towards rail from the more energy intensive modes of road and air is desirable. Rail is an environmentally friendly means of transport when compared with the alternatives of road and air traffic. The average carbon dioxide emission for rail transport are around 18 to 35g/tkm, compared with those for road transport of 62 to 110g/tkm and for air of over 665g/tkm [19].

Noise is not simply an annoyance, it is recognized by the world health organization as being harmful to health, as well as interfering with performance of daily activities [20]. Noise and vibration concerns are often used to oppose new rail developments. In an interview for UIC newsletter, Dr. Matthias Mather (head of Environmental Protection in the Environmental Center Deutsche Bahn AG) states that: “Residents are afraid of that traffic, and hence noise, will increase. So they object to the expansion of rail freight traffic without simultaneous, effective measures for noise abatement. In other words, the residents will only accept the growth in rail freight traffic if the noise pollution falls at the same time” [21]. Reducing rail freight noise is important for citizen’s health as well as ensuring their support for shifting goods transport from road to rail. Higher usage of rail transport for freight contributes to reducing CO_2, air pollutant emissions and fossil fuel dependence.

Noise is generally considered to be unwanted sound. Sound is what we hear when our ears are exposed to small pressure fluctuations in the air. There are many ways in which pressure fluctuations are generated, but typically they are caused by vibrating movement of a solid object. This paper uses the terms ‘noise’ and ‘sound’ interchangeably since there is no physical difference between them. Noise can be described in terms of three variables: amplitude (loud or soft); frequency (pitch); and time pattern (variability).

1.2 Railway Noise Sources

1.2.1 Rolling Noise and Roughness

The dominant source of railway noise is the rolling of the wheels over the rails, except at very high speeds where aerodynamic noise can become the most significant source. This rolling noise has a broad spectrum in the frequency range up to about 5000 Hz, and its level and frequency content increase with train speed. The inherent roughness of the wheels and rails in the contact zone induces a relative motion which results in dynamic interaction forces, vibration of the wheel and track structures, and noise radiation [18]. It is therefore the combined roughness of the wheel and rail that provides the fundamental excitation source for railway rolling noise.

Increasing roughness of the wheel and rail leads to higher noise levels and also to higher contact forces. Ultimately high roughness levels require maintenance attention. Wheels can be removed from service for re-profiling, but grinding the track to control roughness and corrugation is time-consuming and expensive. It is important to understand mechanisms of rail wear and roughness development in order to minimise rail roughness growth rates and hence the costs of noise control and maintenance. The roughness of interest is not limited to corrugation; it includes broadband roughness with wavelengths between 5 and 500 mm, referred to as ‘acoustic roughness.
Noise, caused by the steel wheel rolling on the steel rail is always present. It increases with speed and is dependent on wheel and rail roughness levels. Cast iron tread braked wheels have a higher surface roughness than disc braked wheels or those braked using composite block brakes. Consequently they are noisier.

Rolling noise consists of noise radiated by the track and noise radiated by the wheel. Depending on the design of each and train speed the contribution of each to the total rolling noise level will vary with track noise generally dominating for track designs with soft rail pads and with train speeds typical of freight operation. The wheel contribution increases as rail pad stiffness and train speed increase. Therefore before developing an action plan aimed at reducing rolling noise it will be necessary to know whether the noise comes from wheel or track (or both).

1.2.2 T raction Noise

Power equipment noise comes from a variety of sources including the engine, fans, exhaust outlets and traction motors.This generally has little or no independency on train speed but can be significant in situations where full power is required, especially at low speed when rolling noise will be low. Power equipment noise is potentially a more serious problem for diesel traction compared to electric traction.

Although replacement low noise equipment, such as fans, may be available, it is probable that in the short to medium term, reduction of power equipment noise may only be possible where there is a replacement option with an existing quieter locomotive. For this to be feasible there has to be a quieter locomotive in use on the network and it has to be possible to transfer the duties of that locomotive from one routing to another. That will then mean that the effect of rerouting the noisier locomotive will need to be assessed.

1.2.3 Aerodynamic Noise

As train speed increases noise from the flow of air over the train surface can become significant in the area of pantographs, coach end connections, and bogie areas etc, where changes in cross section affect that air flow. The latest thinking is that this is not the dominant noise source below 300 km/h[2] but as train speeds increase and where it is shown that high speed trains are the major noise source, aerodynamic noise will need to be investigated in relation to other sources.

As with power equipment noise, the mitigation of aerodynamic noise may not be a short to medium term option but again will need to be reviewed as a longer term strategy in action planning at locations where it is shown to be a significant contributor to total noise.
Figure 1 shows the typical importance of each of these sources with speed, although the absolute and relative noise levels are only indicative and will vary with train design. It does show the potential for power equipment noise to be dominant at low train speeds, for rolling noise to be the main source at speeds from 50 km/h to 300 km/h and for aerodynamic noise to become significant at higher speeds.

The latest publications about the contribution of the rolling noise and the aerodynamic noise show that the contribution of the aerodynamic noise is not as high as previously assumed and that the reduction of the global pass-by noise must combine actions on the both sources.

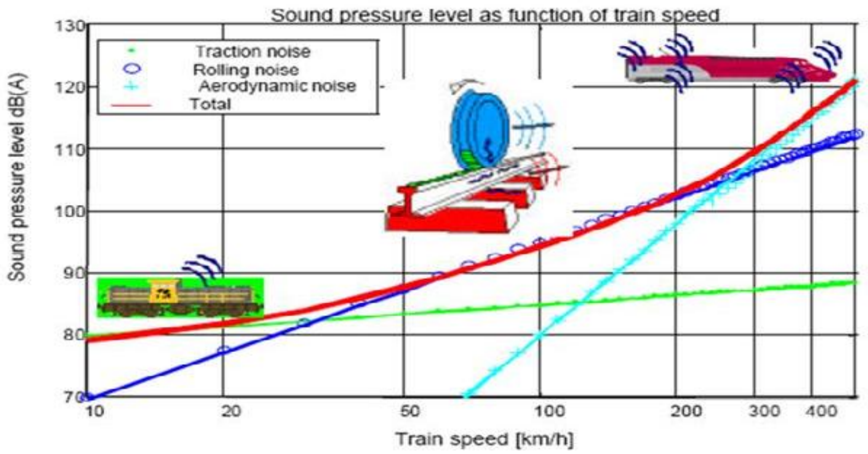


Figure 1.1 types of railway noise [10].

1.3 Statement of Problem

In our country Ethiopia, the railway network is upon construction and in the near future its service will begin. The fundamental understanding of the effects of rolling noise on its effectiveness and side effects on possible optimization is very important for the new railway network beginners like Ethiopia. The wheel-rail contact is a crucial component of the railway system as a whole. Being able to manage the interaction between the wheel and rail is a key to reduce rolling noise in a modern cost effective resilient rail network.

Noisy environment during work and rest time annoys, causes fatigue, by reducing attention, slowing psychical reactions, and wearing nervous system. Noise affects not only hearing, but all human organisms. Noise of trains depends on the category of the train, type of rails, train speed and regime of movement (breaking or non-breaking). The level of noise also depends on spinning of wheels as well as moving throughout connections between rails. The contact rail-wheel has a significant influence for the level of noise. Typical noise of rail-wheel contact is caused of each pair of wheels passing rail connections. Noise level depends on type and condition of rails and wheels, type of breaking system, the way of train is connected and axial loading. Critical impact is done by rail connections, quality and the degree of surface deterioration.

In all instances, noise resulting from traffic may be valued in terms of given average daily and/or nightly figure as a consequence of a 24-hour traffic noise; however, it may also be characterized in more detail. This detailed analysis of noise would then include the features of noise generated by particular trains, vehicles, and the condition of rail tracks. Legislation dealing with noise pollution resulting from railway traffic is in most countries based on average values, calculated for a certain location in accordance with known traffic flows.

1.4 Significance of the Research

This research has a great impact on future analysis and applications of rolling noise reduction in general and in particular in the Ethiopian context. Especially, it will contributes a lot for future analysis of rolling noise generation due to failures like wheel wear, rail roughness, fracture and instability of wheel as well as rail. Showing the position of contact points and the way of load distributions on the contacting interface. Most practical wheel/rail contact installations are seen to be wearied out before the specified design periods. This may be due to improper installations and engagements of wheel/rail contacting surfaces. However, this paper provides the general wheel/rail contacting models and contact pressure to predict the contacting interface noise.

1.5 Objectives

1.5.1 General Objectives

The aim of research is to analyze the wheel/rail interaction and to predict the generation of rolling noise.

1.5.2 Specific Objectives

- Analysis of wheel/rail interaction.
- Analysis of applied load (stress) distribution on the contacting surfaces.
- Determination of contact pressure.
- Prediction of generation of rolling noise.

1.6. General Methodologies

To attain the above objectives, there are a step by step procedures that should be followed:

Analysis of Wheel/Rail Contact Geometry, Applied Load Conditions and contact pressure on the Rail Head Surface and wheel

- Using CATIA V5R16, Modeling of
  - Rail surface geometry
  - Wheel surface geometry

- Simulation of a specific wheel rail contact using ANSYS 14.5

1.7. General Conditions

During the formulations of this paper the general conditions considered are based on the Hertz contact theory. These are:

- The surface of contacting bodies are assumed to be rough
- The contact is assumed to be elastic
- Isotropic and homogeneous material
- Both contacting bodies were considered.

1.8. General Parameters

During the analysis of wheel/rail interaction there are different types of external and internal parameters affecting the interactions. These parameters can be broadly classified as:

- Vehicle parameters: vehicle type, operating speed, axle load, rail &wheel profile.
- Track parameters: Track geometry (gauge, cant, alignment and curve), Rail head profile and shape etc.
- Environmental parameters: Wheel /rail coefficient of friction conditions (dry, wet, contaminant etc.), wind force, soil property, forest around the railway track etc.

In this paper all the above parameters are not taken into consideration due to its broad applications. Therefore most of the listed parameters are assumed to be constant except the rail head profile and wheel profile.

Analysis of Wheel/Rail Contact Geometry, Applied Load Conditions and contact pressure on the Rail Head Surface profile, and axle load since the main aim of the research is to find and understand the effects of axle load and train speed on the wheel rail contacting surfaces. Therefore in this paper common assumptions are made, that is:

- The wheel/rail interaction is assumed to be elastic contact that is there is no plastic deformation and time dependent variation of stress and strain during contact
- The material of rail and wheel is assumed to be identical (steel)
- No friction between contacts (Hertz contact theory)
- The analysis of the rail wheel contact at the macro level (no surface irregularities)
- No derailment occurred (nonlinearities due to rail-flange contact is assumed to be zero)

By considering such assumptions it is possible to avoid some complexity which is difficult to conduct without the use of some powerful computers and software packages. In this paper most of the analyses are based on analytical operations and simple ANSYS software simulations.

1.9. Organization of the Paper

The body of this study is divided into six main chapters. The first chapter discusses background, objectives and methodology of the study. In addition, the details of rolling noise. The second chapter covers the review of some of the journal articles, conference papers and publications which are related to wheel/rail interaction and rolling noise prediction. Also, in relation and comparison with previous works, what is done in this study will be stated. Detail discussion of noise generation and prediction is in the third chapter. Model and analysis of wheel/rail interaction is discussed in the fourth chapter. Modeling contact between wheel and rail, stress model using hertzian theory [16], noise generation and prediction and wheel rail simulation presented. In addition, it covers discretization of the solid model of wheel/rail used for the analysis and load condition. The results obtained from the static analysis and analytical calculation of noise prediction and discussions based on these results are included in the fifth chapter. Finally, the sisth chapter cover conclusions drawn based on the results of the analysis, recommendations and future work.

1.10 Scope and Limitation of Research

Numerous types of simulations can be carried out regarding wheel rail interaction. In this research, as it is mentioned earlier, transient analysis and analytical calculation of the contacting bodies are carried out to predict rolling noise.

Since the railway in our country is under construction, the limitations of the researches are lack of data, lack of high performance computer and lack of enough reference books. And also the study of rolling noise generation and prediction experimentally is difficult and because of luck of noise prediction or noise reduction software like TWINS (track wheel interaction noise software).

Chapter 2

2. Literature Review

2.1 Introduction

There are many journals, conference papers, proceedings, design works and books related to the railway engineering, railway vehicle dynamics and particularly wheel rail dynamics, contacts, interactions etc. but to save time and to manage the paper work the review of literatures mainly considers more related works to the paper. This intensifies the deep analysis of the previous related works and selection of appropriate conditions, approaches and methodologies for the successful accomplishment of the paper.

2.2. Wheel-Rail Contact

The history of wheel/rail contact mechanics is an integrated part of contact mechanics which goes back to the middle of the 19th century. Problems of wheel/rail contact (damage phenomena and influence of contact mechanics on vehicle dynamics, especially on vehicle stability) have been investigated since the middle of the 19th Century. Knothe [9] studied that in 1855, Redtenbacher was the first to consider head checking. However, the scientific foundations of our present investigations are based on the work of Heinrich Hertz, Frederick William Carter and Hans Fromm. Basically the work of this paper is based on the Heinrich Hertz contact theory assumptions.

The Hertz contact theory leads to an elliptical contact area and a semi-ellipsoid contact pressure distribution in the contact region. Due to its efficiency and simplicity, this theory has been extensively applied since its publication. However, there are two limiting conditions for the applications of the Hertz contact theory [16]:

a) The contact between elastic bodies should be frictionless,
b) The significant dimensions of the contact area should be much smaller than the dimensions and the radii of curvature of the bodies in contact.

Contact is necessary in any engineering application to transfer force and power and hence it is an indispensable field of study. The major characteristics of contact mechanics are the localized deformation and the variation in the contact area with the contact force [16**]**. Being part of contact mechanics, the wheel/rail interactions encounters both localized deformation and contact area variation corresponding to the variation of the size of applied load. Escalona [1] studied that the calculation of wheel-rail contact forces in the dynamic simulation of railroad vehicles involves the following steps:

- Location of the position of the contact points on the surfaces of the wheel and rail.
- Calculation of the normal contact forces.
- Calculation of the tangential (creep) forces and moments

From the above three steps the first two steps are greatly essential for this research. Because in this paper the first priority is given to the analysis of wheel rail interaction which is critical for the accuracy of the final output of the wheel/rail contact numerical results. In addition to that this paper gives high priority to the analysis and modeling of wheel/rail geometry which is basic to indicate the position of the contact point. Escalona [1] studied that the search of the contact points requires the surfaces of the wheel and rail to be mathematically parameterized.

Different researchers use different approaches to analyze the wheel/rail contact surface geometry and to indicate the position of contact points. Matsumura [14] use table-update and table-interpolation algorithms on the accurate analysis of vehicle/turnout interactions. Based on this method for modeling change in the rail cross-section, multiple look-up contact tables can be used. The numerical procedure of using multiple look-up contact tables can be classified into two different approaches. In the first approach, changes in the rail cross-section around the neighborhood of the contact point are assumed to be small, and the contact table used to determine the location of contact points is discontinuously changed as a function of the distance the wheel set traveled. Since only small modifications are required, the simulation of vehicle/turnout interactions can be performed in a straightforward manner.
However, since discontinuations change in contact points are inevitable due to the discontinuous table updating, a large number of look-up contact tables need to be prepared in advance for reliable simulations.

In the second approach, on the other hand, the location of contact point is determined by interpolating nearest two look-up contact tables. That is, the location of contact point that should be obtained using interpolated rail profiles can be determined using interpolation of two contact tables. Karwacki [7] studied that in the elastic approach, no kinematic contact constraints are

imposed; the wheel has six degrees of freedom with respect to the rail; and small penetrations at the contact points are allowed. In the elastic contact formulations, a compliant force element that consists of stiffness and damping forces is used to determine the normal contact force.
In this method, the location of the contact points is determined by solving a set of algebraic equations. For each contact, four algebraic equations are solved to determine the four parameters that describe the geometry of the wheel and the rail surfaces.

2.3. Rail Roughness

Rail dampers are an example of a change in the track parameters that could affect rail roughness development. Rail damping devices have been shown to be highly effective for noise control, both in theoretical simulations and in installations on operating lines [15]. Adding rail damping components to the track to control noise will change the track dynamics, and a variation in the track dynamics may result in a change in the propensity of the system to develop roughness or corrugation. Therefore the long term effectiveness of rail dampers for noise control may be affected by the propensity of the upgraded system to develop roughness or corrugation over time.

2.4. Wheel/Rail Contact Stress

The stress field created by the contact stresses was first introduced by Henrick Hertz in 1881 [16]. Assessment of contact stresses at the wheel–rail interface is one of the most important aspects of railway research, considering the many phenomena involved (wear, adhesion, surface fatigue damage, etc.). For this reason, many scientists have approached the problem mainly by means of theoretical or numerical solutions based on the Hertz’s theory, which can be considered the basic starting point for all subsequent research. Pau [13] uses the ultrasonic method to study the wheel rail contact parameters.

The principle of the method is to send high-frequency ultrasonic waves (usually in the range 1–20MHz depending on the thickness to test) over the contact interface, evaluating the amount of energy it reflects by calculating the reflection coefficient $R = H_i/H_0$. Where H_i is the amplitude of the ultrasonic wave reflected from the interface subjected to an external load, and H_0 is the amplitude measured in the absence of contact. Since engineering surfaces are rough, a contact interface can be figured as a series of parts in contact and voids that act as reflector for the sound energy, due to the fact that only a very minor part of the ultrasounds can be transmitted in gases.

Thus, an increase of pressure will result in a reduction of the number and size of these voids and, at the same time, a decrease in the energy reflected from the interface.
The process extremes occur:

- When the reflection coefficient *R* is 1: in this case no energy is transmitted through the interface (i.e. no contact exists);
- When the value of *R* is 0: all the energy is transmitted through the interface, implying that each point of the two surfaces is in (perfect) contact.

In practice the normal load carried on the contact causes both wheel and rail to deform locally to the contact and the load is spread over a contact area or contact patch. For normal railway loads and wheel diameters each contact patches has roughly the size of a finger nail. The wheel cross sectional profile is either coned or a circular arc and the rail head profile is also a circular arc then the load distribution can be calculated analytically and is of the form given by Hertz [16].
If at any time there is a radius in contact with another radius or flat, contact stresses will occur. In the case of two spheres, contacting each other, the entire force will be imparted into a theoretical point. Due to elastic properties of the materials this point will deform to a contact area. The deformation that occurs will produce high tensile and compressive stresses in the materials. Even if a singular loading does not produce a failure, it can lead to future fatigue or surface damage. The maximum pressure within the contact area occurs as a compression in the center. Knowing the maximum pressure then allows you to calculate out the principal stresses along the vertical axis. With the principal stresses and the shear stresses known, evaluations can be made. The first evaluation should be to compare the maximum stresses to the yield or shear strength of the material. The total stresses developed in the rail are the sum of the stresses at the wheel/rail contact (named as Hertz stresses), stresses resulted from rail bending about the ballast, stresses resulted from rail head bending about the web, stresses resulted from thermal effects, and plastic stresses remaining in the rail after the removal of the external load. With the exception of the last two categories all other stresses can be calculated on the assumption of an elastic behavior.

2.5. Models for the Prediction of Railway Rolling Noise

Rolling noise models can be used to compare different wheel and track designs and to optimize the components selected for the best acoustic performance [23]. Developed one of the first analytical models of the interaction between wheel and rail in order to predict the radiated noise [24]. In this model the combined roughness of wheel and rail acts as a vertical excitation to the system. The vibration response of the wheel and rail to this excitation is calculated and used to predict the resulting noise. Extended Remington's theory and implemented the model as the computer program TWINS (Track-Wheel Interaction Noise Software) [22].

The TWINS software has been validated by a series of comparisons with measurements, both for conventional wheel/track systems [10] and for various noise-reducing track and wheel designs [15]. These experiments have found the TWINS model to be capable of predicting noise levels from wheel and track components to within about 2 dB. A number of intermediate stages of the calculation have also been examined, including the track response to excitation by the combined wheel-rail roughness. Thus the wheel-rail dynamic interaction model has also been validated, and the linear relationship between roughness, wheel and rail vibration and the resulting noise radiation has been clearly established.

2.6. Rolling Noise Sources and Reduction Techniques

The sources of rolling noise are well understood, thanks largely to the development and validation of TWINS. For typical ballasted track, the wheels and the rails are both significant contributors to the overall rolling noise level. In this case the rail is the major contributor to the overall noise in the frequency range from 500 to 2000 Hz. At lower frequencies, noise from the sleepers dominates, and at higher frequencies wheel noise takes over from the rail. The sleeper noise is increasingly significant in cases with stiffer rail pads [15].

Rolling noise reduction techniques have been summarized [2]. At the source, a noise reduction may be achieved by reducing the roughness of the wheels and rails. The next option is to deal with the vibration response to the roughness excitation, for example by using wheel or rail dampers. Finally, noise barriers may be used to limit propagation.

To reduce the roughness of the rails, the track maybe ground. Grinding is usually applied as part of track maintenance rather than purely to control noise, although in Germany a programme of acoustic rail grinding is in place [25]. For the wheels, using disc-braked vehicles or composite brake blocks rather than cast-iron blocks in tread-braking systems gives a significant reduction in wheel roughness [26]. The railways are phasing out the use of cast-iron brake blocks mainly due to legislation in the form of the TSIs; however, it is expected to take many years before they are eliminated.

The rail dampers proposed by Thompson [15] are tuned, damped mass-spring absorber systems, with either a single mass or two masses enclosed in an elastomeric material. In their current form these dampers are attached to the rail in pairs in the middle of each sleeper bay. The dampers reduce the effective radiating length of the rail by increasing the track decay rate. To control the response of the wheel, wheel dampers can be used. Wheel dampers are most useful in controlling curve squeal, but they also have some effect on rolling noise [23]. Wheel shape optimization can also give some benefit.

Noise barriers restricting propagation should be seen as a last resort for noise control, erected at specific problem sites. They are expensive, are not effective at all locations, and are often visually intrusive.

For a given roughness level, the noise from the system can be reduced by increasing the damping of the track, by acoustic optimization of wheels and track components or by barriers. In the long term, however, the noise level is dependent both on the noise produced for a given roughness and on the rate of roughness development.

The roughness response for rolling noise has wave length between 5 and 500mm [Thomson, 2009] [18] and amplitude from less than 1µm up to about 50 µm. For a train speed of 100 km/h, this wavelength range corresponds to a frequency range of around 50 to 5000 Hz. Roughness is usually expressed as decibel levels with a reference value of 1µm root mean square (rms) roughness amplitude is equivalent to 0 dB.

Linear noise prediction models are based on the assumption that the roughness from the wheel and the rail can be combined by simple incoherent addition. This was found to be valid [10] for typical roughness levels. However if either the wheel or the rail is significantly rougher than the other component, the resulting noise spectrum will be dominated by the rougher surface. For example, the type of braking system employed by the rolling stock has a significant effect on the roughness of the wheels. If cast-iron block brakes are used then the wheel roughness will tend to dominate unless corrugation is present on the rail [28].

Chapter 3

3. Generation of Wheel/Rail Rolling Noise

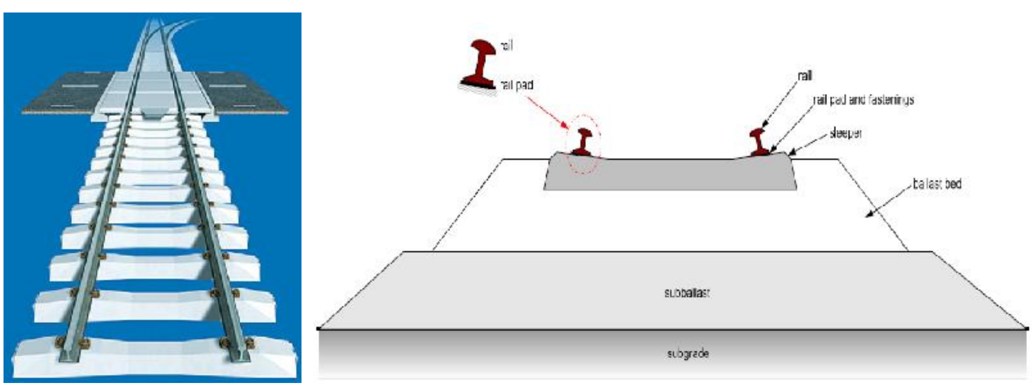
3.1. Introduction

There have been substantial advances in the theories concerning wheel/rail noise generation during the 1970s and 1980s, and research in the area is continuing, especially with respect to high speed rail. The theory of wheel/rail noise generation, though highly developed, is incomplete. The effect of wheel and rail profiles on wayside noise from tangent track is being studied, but contradictions exist. Wheel squeal theory is also subject to some inconsistency with respect to field experience.

3.2. Railway Track

It is laid on the appropriate ground level and geographical sites based on the appropriate design specification. It has different components interrelated with meaningful arrangements for the general railway dynamic operations. Rail track is a fundamental part of railway infrastructure and its components can be classified into two main categories: superstructure and substructure. The most obvious parts of the track as the rails, rail pads, sleepers, and fastening systems are referred to as the superstructure while the substructure is associated with a geotechnical system consisting of ballast, sub-ballast and sub grade (formation). Both superstructure and substructure are mutually important in ensuring the safety and comfort of passengers and quality of the ride [27]. The typical shape and construction profiles of a ballasted track are illustrated in figure 3.1.

The rail track sleeper is used to transmit the wheel load to the ballast medium. In addition, it has functions such as maintaining track alignment and gauge, restraining longitudinal and lateral rail movements, and providing strength and stability to track structure [29].



a b

Figure 3.1: Rail track components and their arrangements [6]

During the interaction between the wheel set and the rail track there are different conditions created on the contacting surface. These dynamic contact behaviors depend partly on the track geometry. The most common track geometries having great impact on the railway vehicle dynamic behavior are:

- Track gauge: the distance between the right and left rail inner gauge corners
- Track cant (supper elevation): the difference between the level of the two rail on a curve
- Track curvature: it is the inverse of the radius of the curved track
- Rail head: it is the surface of the rail having a direct contact to the railway vehicle wheel set.
- Wheel and rail tread profile

By managing those geometrical parameters of the track it is possible to control the dynamic behaviors of the wheel-rail dynamic contact, especially on the curved track.

3.3. Wheel Set

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. 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

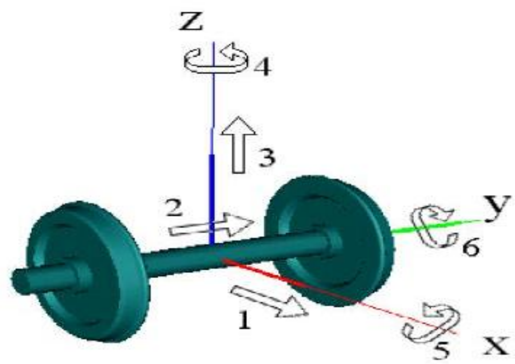


Figure 3.2: Wheel set degrees of freedom [15]

Similarly the rotational degrees of freedom consist of three components that is rotation about: X-axis, Y-axis, and Z-axis

In conventional railway vehicles the wheels, assembled in a wheel set, 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 [4]. Performance, safety and maintenance cost strongly depend on the wheel set dynamics and particularly on how good is design of wheel and rail profiles. Using geometrical characteristics of a contact between wheel and rail it is possible to judge about dynamic parameters of wheel set and ultimately parameters of vehicle since a wheel set represents a source of disturbances from track to vehicle. The wheel-rail geometry plays a dominant role in vehicle lateral dynamics. The rolling radii, contact angles and the wheel set roll angle vary as the wheel set moves laterally relative to the rails. The nature of the functional dependency between these geometrically constrained variables and the wheel set lateral position depends on the wheel and rail cross-sectional shape [3]. The formulation of the wheel-rail contact problem is a complex task since it requires the study of the contact geometry, which is the problem of determining the location of the contact point on the profiled surfaces of the bodies, of the contact kinematics, which involves the calculation of normalized relative velocities at the point of contact, and of the contact mechanics, which is the problem of determining the contact forces [6]. The spatial distribution of contact forces caused by the form of tread surfaces of a wheel and a rail is the reason of a differential creep in contacts and occurrence of moving resistance forces. Schematically, the point-to-point contact is a statically indefinable system with a parameter of non-definability, equal to a unit. Therefore, in mathematical modeling of distribution of loadings in contacts it is necessary to take deformations of contact areas into account. The geometrical parameters of the point-to-point contact in the numerical integration of the equations of wheel set movement are defined as a result of coordinate analysis of contact points of wheels with rails. The radiuses of tread contact surfaces and profiles grade in points of contacts are related to such parameters [28].

3.4. Rail

Railway lines are made of straight sections and curves. Train driving on the curves essentially differs from that one on straight sections. On the curves railway gauge is widened (when curve radius is less than 350 m), and cant are mounted [30]. Rails are longitudinal steel members that are placed on spaced sleepers to guide the rolling stock [5]. 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 [11]. 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:

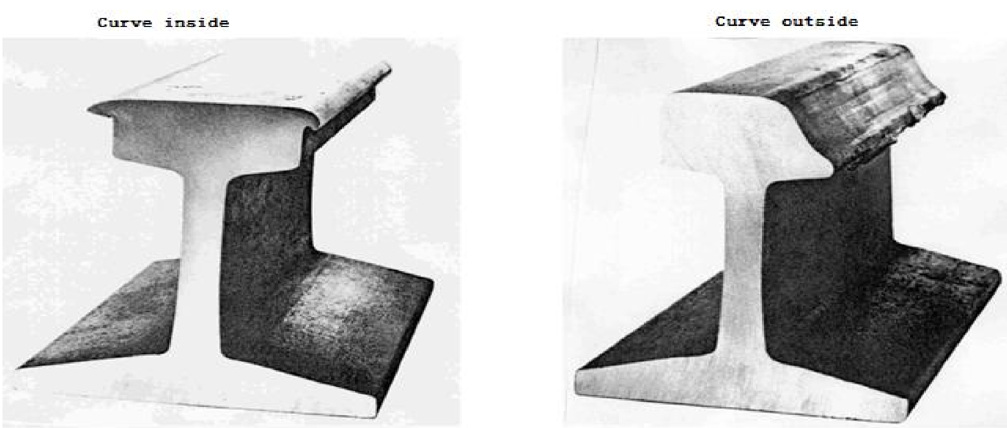


Figure 3.3: Flat bottom rail parts [17]

- Rail head: the top surface that contacts with the wheel
- 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.

3.5. 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, 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 [16]. 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 contact profiles 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.

In railway transport system, the difficult problem frequently occurred is in the interaction between wheel and rail. In particular, the complexity of the interface geometry and the general operating geometric parameters (wheel lateral movement, cant angle, wheel conicity, and wheel rail profile variation and irregularities) makes this interaction very complex. There are different approaches to analyze this complex interaction. Some researchers focus on the dynamic behavior and dynamic response of each component during complex operational interaction. However, finding the locations of point of contacts, and analyzing the manner of interaction are the very priority.

Therefore the contact parameters (wheel/rail profile, rail inclinations, wheel conicity, wheel lateral movement and track gauge) are the main inputs for the analysis of interaction.

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 contact profile and applied load condition on the rail head surface. The figure below shows the general wheel rail interactions from the front and side view respectively.

Figure 3.4: wheel/rail contact interface (17)

Generally the main objective of this research is to find optimum solution to the wheel rail interaction which increases the safety and durability of these components. It is by indicating and showing the point of contact, contact patches and load distribution which is guidance for the future analysis of the system.

The wheel rail interaction is mainly occurred between the wheel and the rail head surfaces. Both surfaces have a complicated geometry and micro structural complicated profiles. In addition to this the horizontal alignments of the track also affects the rail wheel dynamic interactions. These and other factors have increased the significance of this research.

3.6. Generation of Rolling Noise

Tangent track noise is due to wheel and rail roughness, which may include rail corrugation and random surface defects. Tangent track noise is subdivided into

1. Normal rolling noise ;

2. Rolling noise due to excessively roughened rails and wheels;
3. Impact noise due to pits and spalls of the rail and wheel running surfaces, wheel flats, and rail joints; and
4. Rail corrugation noise.

In this paper normal rolling noise will be discussed. These categories form a basis for discussion of tangent track noise and selection of noise control procedures. The term wheel/rail roar is often used to refer to rolling noise of all types, while roaring rail has been used to refer to corrugated rail noise.

Tangent track normal and excessive rolling noise are discussed within the context of a linear theory of wheel/rail interaction, relying on rail and wheel roughness as the primary cause. Another cause of wheel/rail noise may be heterogeneity of the rail steel moduli, for which the linear theory presented here may not be entirely adequate. Further, with sufficiently large amplitude roughness, as with severely corrugated rail, or with severely flatted wheels, wheel/rail contact separation may occur, producing impact noise.

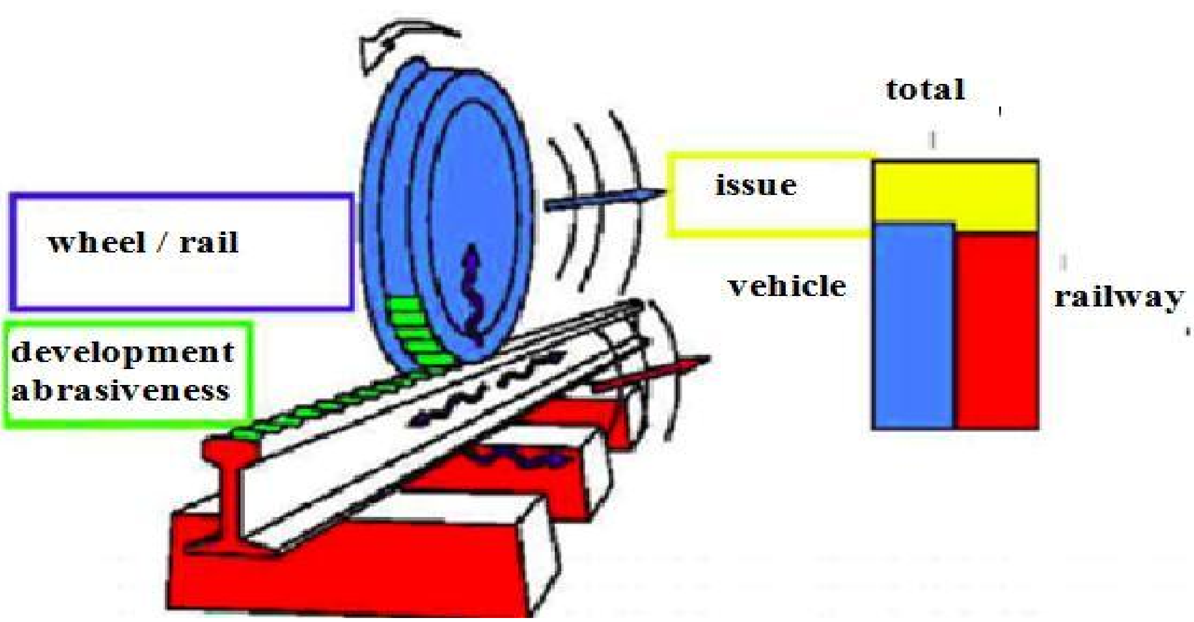


Figure 3.5: rolling noise, its development, transmission and emission [20]

3.6.1. Wheel/rail Rolling Noise

Noise in the absence of wheel and rail imperfections or discontinuities, such as wheel flats, spalls, rail corrugation, joints, and so on is termed normal rolling noise. Normal rolling noise is the baseline condition for which most new rail transit systems are designed, and departures therefrom constitute a degradation of condition and performance with respect to noise control. This baseline condition is the state that operating rail transit systems should be striving for or preserving. There is also an important side benefit associated with a quiet system: smooth rolling surfaces produce less vibration of various track work and truck components, thus extending life and, possibly, reducing operational costs.

3.6.2 Characterization of Wheel/Rail Rolling Noise

Track rolling noise has a broadband frequency spectrum, often with a broad peak between 250 and 2,000 Hz. At systems with adequate ground rail and trued wheels, traction motor cooling fan and gearbox noise and undercar aerodynamic noise may contribute significantly to operational noise, and some care must be exercised in identifying the wheel/rail noise component. Representative samples of wayside noise produced by a Bay Area Rapid Transit (BART) vehicle at various speeds are presented in Figure 3.6. In this example, traction power equipment produces the peak in the spectrum at 250 Hz. At higher frequencies, wheel/rail noise dominates the spectrum, peaking at about 1,600 Hz. With trued wheels and smooth ground rail on ballast and ties, BART is one of the quietest vehicles in operation at U.S. transit systems. A whine may also occur because of a periodic grinding pattern in the rail running surface, as evidenced by the peak at about 1,600 Hz in the 130km/h data shown in Figure 3.6 below.

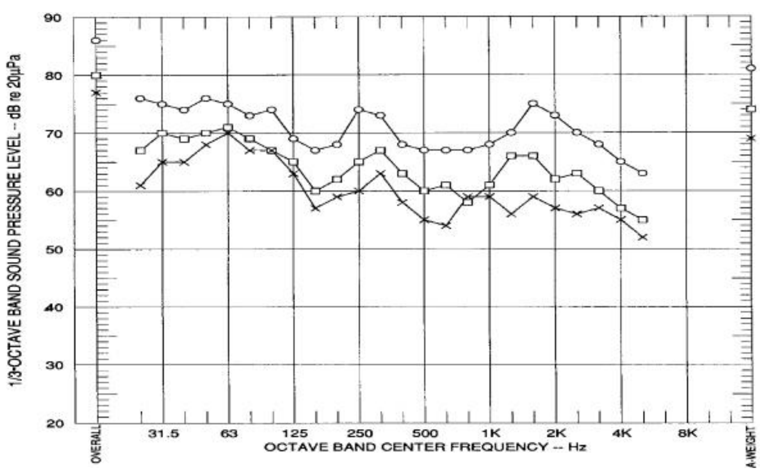


Figure 3.6 vehicle with aluminum centered wheel on ballast and tie truck [23]

At lower speeds, this peak is reduced in frequency, consistent with a wavelength in the rail of about 20 mm long. Without this component, the pass by noise would have been less than measured by a few decibels. The high levels of low-frequency noise below 125 Hz are likely due to aerodynamic sources in view of the relatively large difference between the 60 and 80 mph data in this frequency range. If there is difficulty in distinguishing the wheel/rail noise component from traction power equipment noise and other under car equipment noise, then wheel/rail noise is probably not excessive, and efforts at noise control might be directed toward traction power equipment [1] With normal rolling noise, the rail running surface will be free of spalls, checks, pitting, burns, corrugation, or other surface defects, which may not be entirely visible. The wheel and rail provide running surfaces which, under ideal conditions, should have similar characteristics for smoothness and low noise as any anti-friction bearing. From a practical perspective, ideal bearing surfaces are difficult to realize and maintain in track due to lack of lubricant, corrosion and contamination, and dynamic wheel/rail interaction forces.

3.6.3 Rolling Noise Generating Mechanisms

Four generating mechanisms have been suggested in the literature as sources of normal rolling noise. These include

• Rail and wheel roughness,

• Parameter variation, or moduli heterogeneity,

• Creep, and

• Aerodynamic noise

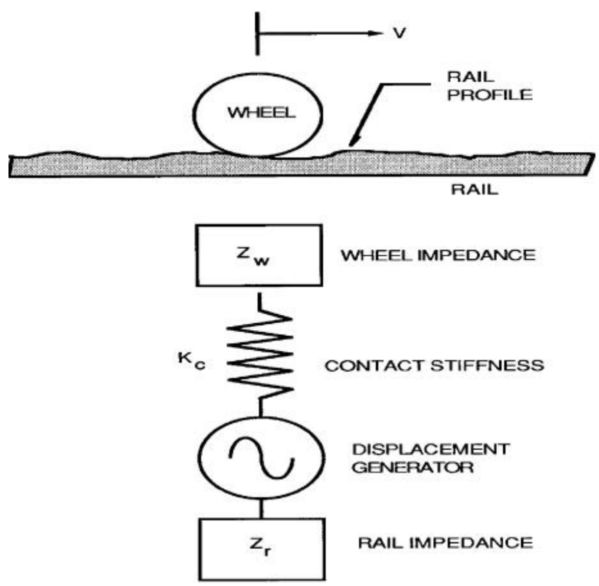


Figure 3.7: model of wheel/rail noise generation [4]

3.6.4 Wheel/Rail Roughness.

Wheel and rail surface roughness is believed to be the most significant cause of wheel/rail noise. The surface roughness profile may be decomposed into a continuous spectrum of wavelengths. At wavelengths short relative to the contact patch dimension, the surface roughness is attenuated by averaging of the roughness across the contact patch, an effect which is described as contact patch filtering. Thus, fine regular grinding marks of dimensions less than, perhaps, 1/ 16 in., should not produce significant noise compared to lower frequency components.

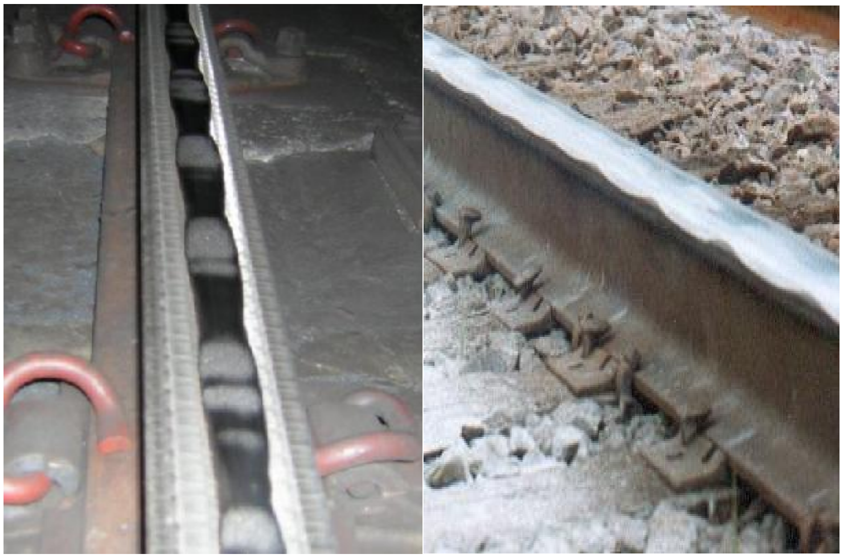


Figure 3.8: Rail roughness [9]

3.6.4.1 Parameter Variation.

Parameter variation refers to the variation of rail and wheel steel moduli, rail support stiffness, and contact stiffness due to variation in rail head transverse radius-of-curvature. The influence of fractional changes in elastic moduli and of radius-of-curvature of the rail head as a function of wavelength necessary to generate wheel/rail noise equivalent to that generated by surface roughness. The wavelength of greatest interest is 25.4mm to 50.8mm. Corresponding to a frequency of about 500 to 1,000 Hz for a vehicle speed of about 60 mph. Over this range, a variation in modulus of 3% to 10% is required to produce the same noise due to rail roughness.

Experimental data for the effect of modulus variation at this frequency have not yet been found. Rail head ball radius heterogeneity also induces a dynamic response in the wheel and rail. The variation of rail head curvature would have to be on the order of 10% to 50% to produce a noise level similar to that produced by rail roughness alone. Data on rail head radii of curvature as a function of wavelength have not been obtained nor correlated with wayside noise. Also, rail head ball radius variation will normally accompany surface roughness, so that distinguishing between ball radius variation and roughness may be difficult in practice.

3.6.4.2 Dynamic Creep.

Dynamic creep may include both longitudinal and lateral dynamic creep, roll-slip in a direction parallel with the rail, and spin-creep of the wheel about a vertical axis normal to the wheel/rail contact area.

3.6.4.3 Longitudinal Creep

It is not considered significant by some researchers, as rolling noise levels are claimed to not increase significantly during braking or acceleration on smooth ground rail. However, qualitative changes of the sound of wheel/rail noise on newly ground rail with a grinding pattern in the rail running surface is observable to the ear as a train accelerates or decelerates, in contradiction to the notion that longitudinal creep is of no significance.

3.6.4.4 Lateral Creep

Occurs during curve negotiation, and is responsible for the well-known wheel squeal phenomena resulting from stick-slip. Lateral creep may not be significant at tangent track, but lateral dynamic creep may occur during unloading cycles at high frequencies on abnormally rough or corrugated rail. Lateral dynamic creep is postulated by some to be responsible for short-pitch corrugation at tangent track. Therefore, lateral creep, at least in the broad sense, may be a significant source of noise.

3.6.4.5 Spin Creep

It is caused by wheel taper which produces a rolling radius differential between the field and gauge sides of the contact patch. Spin-creep has been suggested as a source of wayside noise at the Vancouver, though, no data supporting a strong dependence of noise level on spin-creep have been obtained.

3.6.5 Prediction of Rolling Noise Using Analytical Calculation

Figure below shows a block diagram on prediction of rolling noise. The input condition is the roughness of wheel/rail tread. The wayside noise is predicted by calculating the noise radiated by using the contact pressure from the wheel and rail interaction.

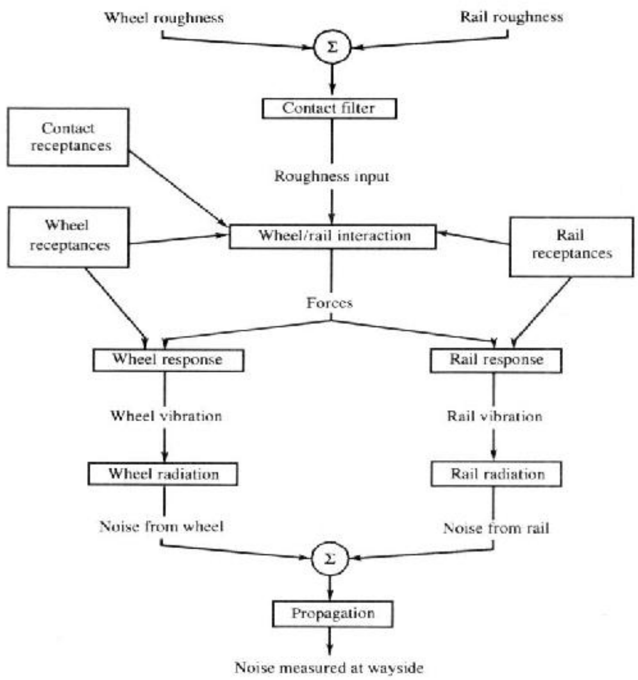


Figure 3.9 prediction of rolling noise [26]

When the wheel and rail contact each other, the local deformation occurs in the contact area by the wheel/rail interaction.

Noise is a fluctuation of air pressure. The number of times the fluctuation occurs in one second is termed as its frequency. Frequency is quantified in cycle per second or (Herz).

$$\boldsymbol{f = \lambda c} \quad \text{--}3.1$$

where: f is frequency (Hz)

λ is wave length (m)

c is speed of sound (m/s)

and $$\boldsymbol{\omega = 2\pi f} \quad \text{--}3.2$$

The effective magnitude of sound pressure in a sound wave can be expressed by the average of the oscillating pressure measured in Pascal.

The level of noise (dB) is given by:

$$\boldsymbol{Lp = 20\log 10\left(\frac{p}{po}\right)} \ \textbf{(dB)}\text{--}3.3$$

Where: L_P level of sound pressure (dB)

p is contact pressure (pa)

Po is reference sound pressure (pa)

<u>Note</u>: the value of reference pressure is $2x10^{-5}$pa.

Chapter 4

4. Modelling and Simulation

4.1 Introduction

Different theories, analytical, numerical formula and finite element method developed in different time to solve the problem related to contact component and to increase the performance of the contact area. Although, railway systems are a transportation system still now it has a large number of unsolved problems that related to contact surfaces. As we have discussed in previous chapters, the hertz theory has been developing since 1881[16]. Hertzian contact theory, explain a relationship for determining the contact pressure distribution and contact area of a solid bodies while in contact with an elastic sphere or cylinder under an applied load. This theory has its own assumption to solve the contact area and stress distribution of two solid bodies.

The following are the hertz theory assumption:-

- The contact between elastic bodies should be frictionless.
- The significant dimensions of the contact area should be much smaller than the dimensions and the radii of curvature of the bodies in contact.
- The contact between elastic bodies should be described by second-order polynomials.

The hertz theory has different type of contact for different shape of the solid body contact. The contact between the wheel/rail is elliptical contact. For wheel/rail contact hertz theory is much preferable than others theory.

The analysis of the wheel/rail contact is covered by using the elliptical contact shape to solve the stress, strain, deformation and fatigue wear caused by the vertical load.

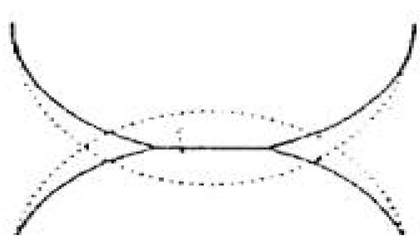


Figure 4.1 Full elastic contact mechanics model of hertz [30]

The wheel profile consists of a flange to guide the trains along the rails and a conical tread that contacts rail head, and rail has many curvatures to guide wheel properly. The contact positions of the wheel / rail are different in the different situation. However, this paper uses the contact between the wheels tread and rail head.

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

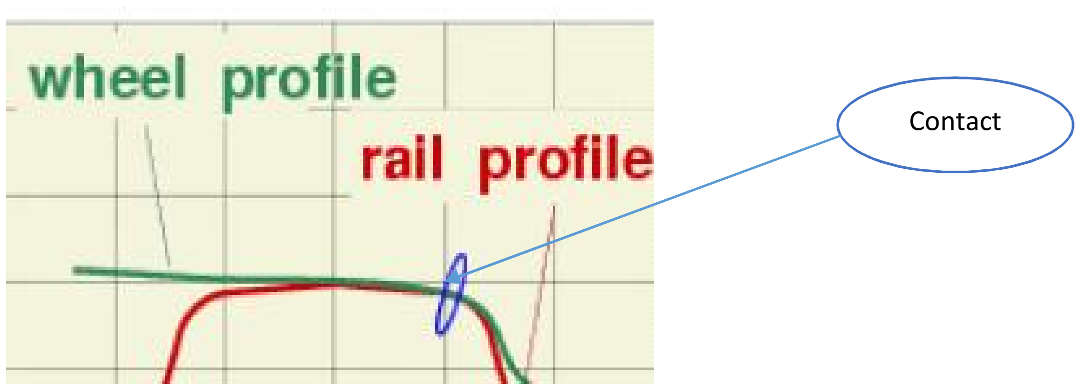


Figure 4.2 Contact area of wheel/rail [25]

High stresses are inducing when a vertical load applied. This can cause serious problem on wheel/rail contact area. Hertz developed a theory to calculate the contact area and stress between the two contact surfaces. When two elastic non-conforming bodies get together, according to the Hertz contact theory, the contact area is elliptical in shape with a major semi-axis “a” and a minor semi-axis “b” [16].

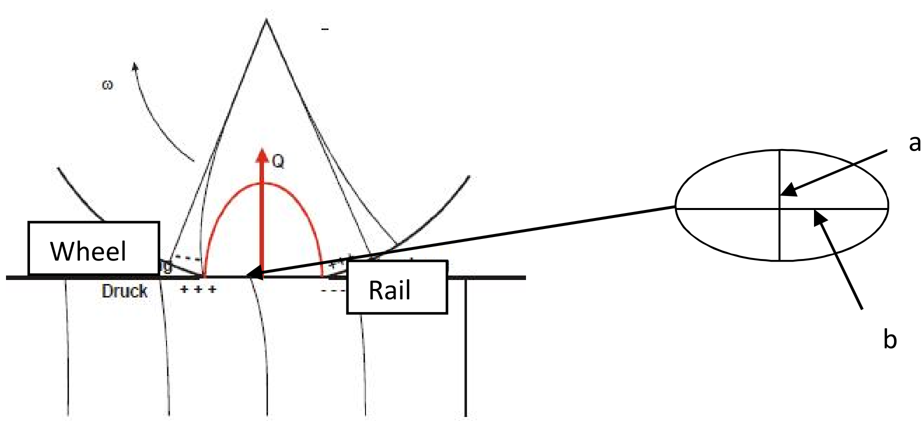


Figure 4.3 wheel/rail contact Shape [18]

The contact pressure distribution in this area represents as semi-ellipsoid, which can be expressed as:

$$P(r) = \boldsymbol{p_o}\sqrt{\mathbf{1} - \frac{x^2}{a^2} - \frac{y^2}{b^2}} \qquad 5.1$$

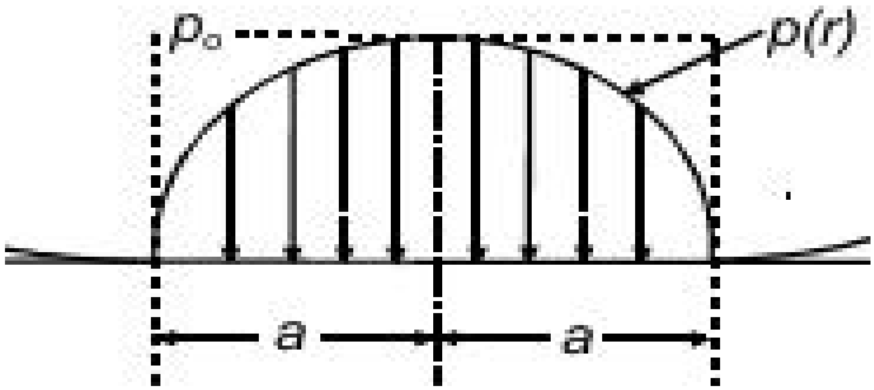


Figure 4.4 Pressure distribution at contact area [11]

Based on the Hertz contact theory, the contact point is very small relative to the overall dimension of wheel and rail surfaces. This very small contact point has elliptical shape.

From the above formula a and b are semi axes of the contact ellipse whereas x and y are the required coordinates to specify the point of contacts on the rail head based on the lateral rail surface parameter.

4.2 Finite Element Modeling

Finite element method is used to analyze the response of wheel/rail interaction the static and dynamic load. The FE analysis is performed using structural analysis of the ANSYS R14.5 work bench software after imported 3D assemblies form CATIA software.

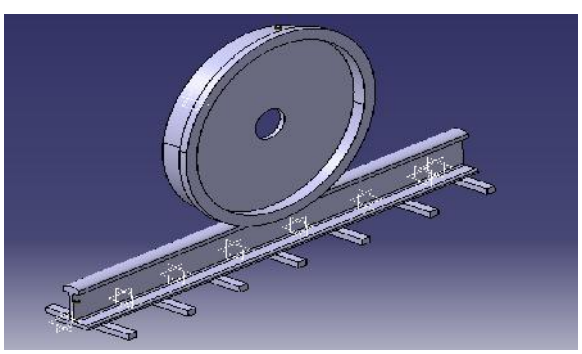


Figure 4.5 Catia 3D model of wheel/rail interaction

ANSYS is a general purpose finite element of modeling package for numerical solved problems. Like any finite element software, ANSYS solves governing differential equations by breaking the problem into small elements. When the wheel is contact element and the rail is target element, the wheel and rail will be different.

4.3 Wheel/Rail Contact Simulation

4.3.1 General Assumption

As much as possible, the general assumption during simulation is to run out of time and simplify the model without any affection for the result.

The following assumption is made analyzing in $\text{ANSYS}_{R14.5}$ software.

- The hertz theory consider only at contact between wheel and rail.
- The analyses of wheel/rail interaction is linear and flexible.
- The place where the sleeper contact rail is considered as fixed support.
- Standard gravitational acceleration is used approximately to $9.8 m/s^2$.
- Analysis is performed by considering the small deformation.

The assembly is done on the modeling package of CATIAV5R16. CATIAV5R16 is 3D mechanical design software for creating 3D prototypes, used to design, visualize and simulate of the analysis.

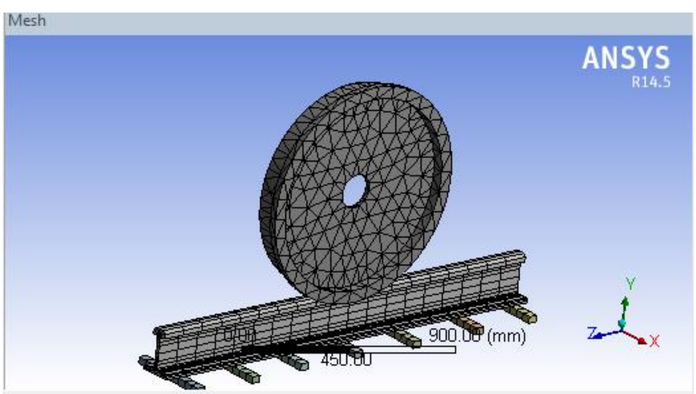


Figure 4.6 Mesh of assembled part on ANSYS workbench.

Assembly model created in assemble workbench of the CATIA after each component created. The assembly model consists; rails, sleeper and wheel. The dimensions and specifications of each part are, based on the relevant standard.

4.4. Material Selection

4.4.1. Rail Material Selection

The railway tracks are mostly steel material. Steel is the most common and widely used metallic material in modern society. It can be produced with tensile strengths exceeding 5 GPa. Steel contains 50% iron and one or more alloying element. These elements generally include carbon, manganese, silicon, chromium, phosphorus, sulpher etc. Each chemical element has a specific role in the steel making process or in achieving particular properties or characteristics. E.g. Strength, hardness and quality. Rails are grouped according to their standards, strength, grade, quality and length.

The rail steel qualities can be distinguished in to two categories.

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

Concerning their chemical compositions rails have great varieties of carbon, manganese, chromium and silicon contents depending on their requirements. Since the rails have to withstand the impact load, friction and stress of freights, they should have sufficient strength, hardness, toughness and good welding performance. However a large increase in rail mechanical strength may result brittle failure and as a result a further increase is not desirable. Similarly, the same material property is selected for wheel materials.

In this study the wheel has approximately similar material property to rail. In this paper GB50 is standard

Table 1: Rail Parameters

Rail (UIC 60)		**Value**
1	Vertical bending stiffness (N/m)	6.42 × 10
2	Vertical Timoshenko shear coefficient	0.4
3	Vertical loss factor	0.02
4	Lateral bending stiffness (N/m)	1.07 × 10
5	Lateral Timoshenko shear coefficient	0.4
6	Lateral loss factor	0.02
7	Mass per unit length (kg)	60
8	Cross receptance factor (<0 dB)	-7
9	Sign cross spectrum	-1
10	Length of track (integration length) (m)	20

Table 2: Wheel Parameters

Wheel (UIC 920)		**Value**
1	Wheel radius (m)	0.46
2	Wheel transverse radius (m)	0.0
3	Inner radius of tyre (m)	0.41
4	Radius of hub (m)	0.15
5	Width of tyre (m)	0.135
6	Width of web (m)	0.025
7	Lateral offset of contact point	0.03

Table 3 Material properties of Wheel/ Rail

1	Poison’s Ratio	0.3
2	Young’s Modulus (Gpa)	207
3	Ultimate tensile strength (MPa)	780
4	Yield strength (Mpa)	640
5	Density (kg/m^3)	7850
6	Elongation (m)	12

4.5 Rail Support

There are different types of load vertical, longitudinal and lateral. Each of these loads produces stresses on the rail and wheel, which change a wheel and rail profile. There is support mechanism to reduce a load. The support by itself has an effect to maximize and minimize the stress the material that has been used.

This paper is formulated by using the strength of material, thought, how positions of rail support to contribute how to maximize and minimize the stress.

Railway system uses sleeper to provide a support and transmit the vertical, lateral and longitudinal load from the rail down to ballast bed.

4.6 Load

All material properties and data are used based on Ethiopia Railway Corporation. Vertically downward load is sum of 3% allowance, maximum axle load. The overall load is the sum of the total tram weight and carrying capacity of the vehicle. The overall load is classified to each axle of the vehicle then the axle load is classified to each wheel vehicle. Carrying capacity of vehicle has calculated by take average of 60kg/person and the total rate of passenger inside of the tramcar is 317.

Table 4: Seating capacity of vehicle

Item No	Number of passengers (persons)	Seated	Standing	Total
1	Seats (AW_1)	65	0	65
2	Seating capacity (AW_2) (standing: 6 persons/m^2)	65	189	254
3	Overload capacity (AW_3)(standing: 8 persons/m^2)	65	252	317

Table 5: Vehicle weight in tone

Item No	Loads	Carboy weight	Passenger weight	Total weight
1	Empty vehicle (t)	44	0	44
2	Seating capacity (t)	44	15.24	59.24
3	Overload capacity (t)	44	19.02	63.02
4	Axle load	≤(11+3%maximum axle load)t		
5	Axle number	6		

Note: Take 60kg as average weight of each passenger

Table 6: Operating speed of train

Item No	Parameter	Speed
1	Maximum operation speed	80 km/h
2	Average travelling speed	70 km/h
3	Minimum operating speed	≤60 km/h

The total vertical load is calculated as follows:

a. Tram car weight = 44 ton
   - The load apply on each axle = 7.333 ton = 73333kg
   - The load apply on each wheel = 3.667 ton = 36667kg

b. Carrying Capacity = 60kg/person *317 person = 19020kg

c. Over all capacity = Tram car weight on each wheel + Carrying Capacity

d. Maximum Axle load = 11,000 kg × 9.81m/s

   =107,710N

e. The total vertical load = maximum Axle load +3% maximum Axle load

   = 110947.3N

f. The load on each wheel = 55573.63N

Note: The wheel load is taken to perform the analysis

The maximum parameter is used to perform the analysis. Material properties of the rail, wheel and plate are assumed the same for the simplicity of the problem.

In order to attain the accurate stress analysis for any application, the support and the load must be defined. This will ensure that the solution incorporates to support of the domain. The place where the sleepers contact the rail considered as fixed support.

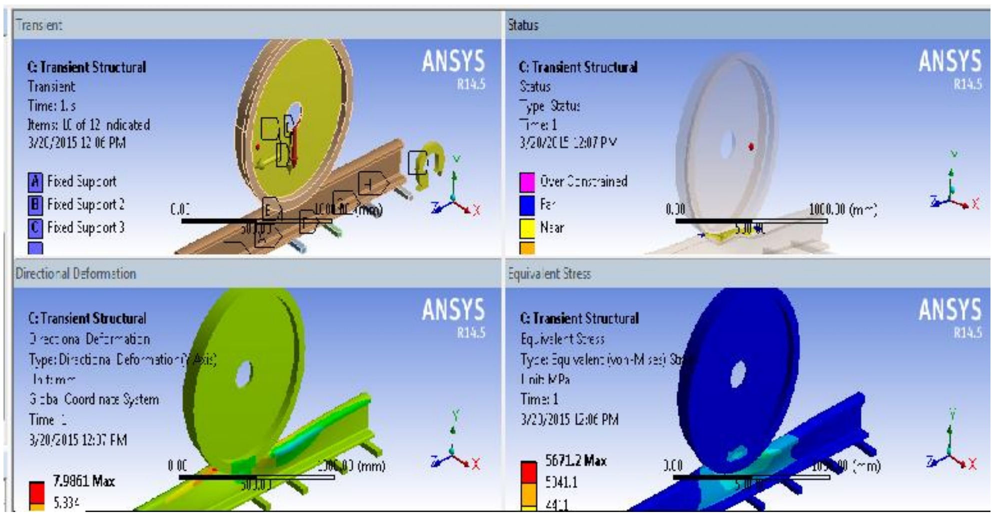


Figure 4.7 Load applied

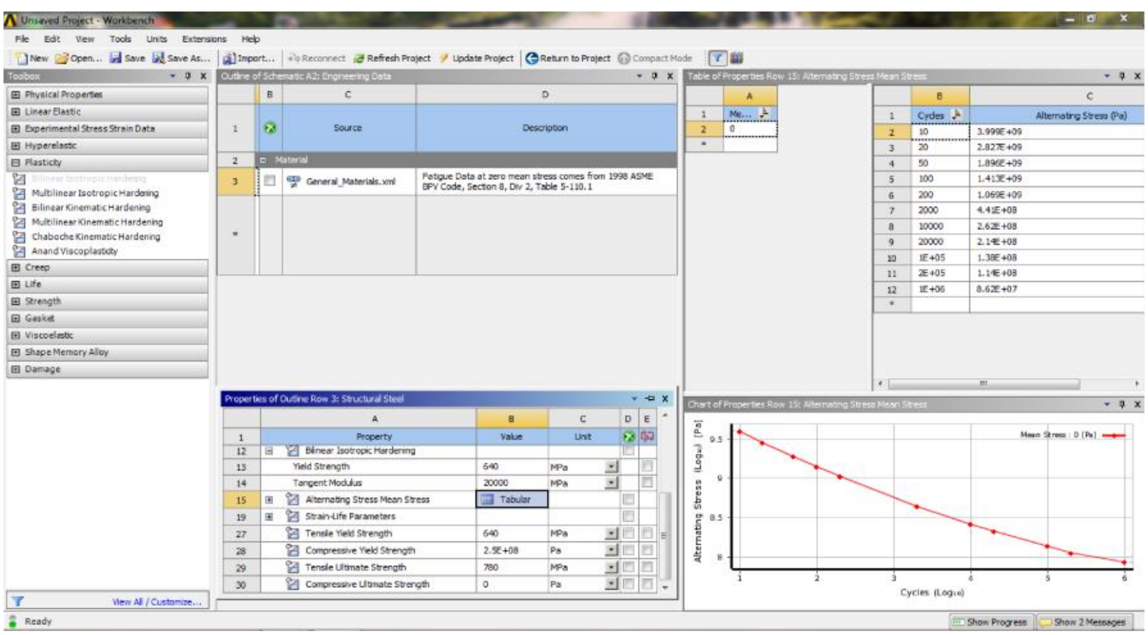


Figure: 4.8 Material property

Chapter 5

5. Result and Discussion

5.1 Result

The analysis is performed by using finite element model consist of analytical calculation, static analysis, and dynamic analysis to determine the impact of the wheel load. Different vehicle speeds are used to perform static and dynamic analysis, although the geometry and load application are the same for all speeds.

5.2 Transient Structural Analysis

Transient structural analysis is determine the deformation, stresses, and contact pressure caused by loads that do not induces significant inertia and damping effects. The load and the structure responses are assumed to vary slowly with respect to time that means steady loading and response condition are assumed.

The types of loading that can be applied in a static analysis include:

- Vertical wheel load (force).
- Standard earth gravity
- Rotational velocity
- Displacement

Case 1. When the Operating Speed is 80km/h

A. Equivalent Stress

As we can see from the figure below the maximum stress is 493.57Mpa

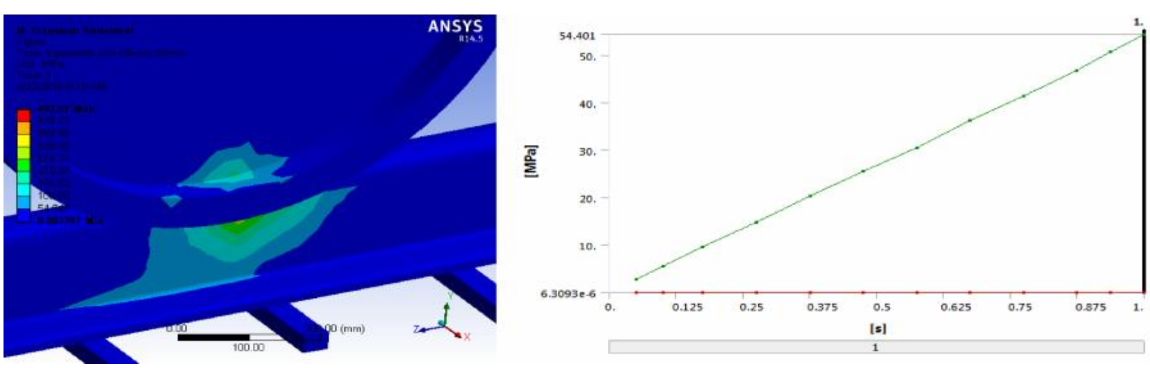


Figure 5.1 equivalent stress

B. Principal Stress

The maximum principal stress is 113.29Mpa

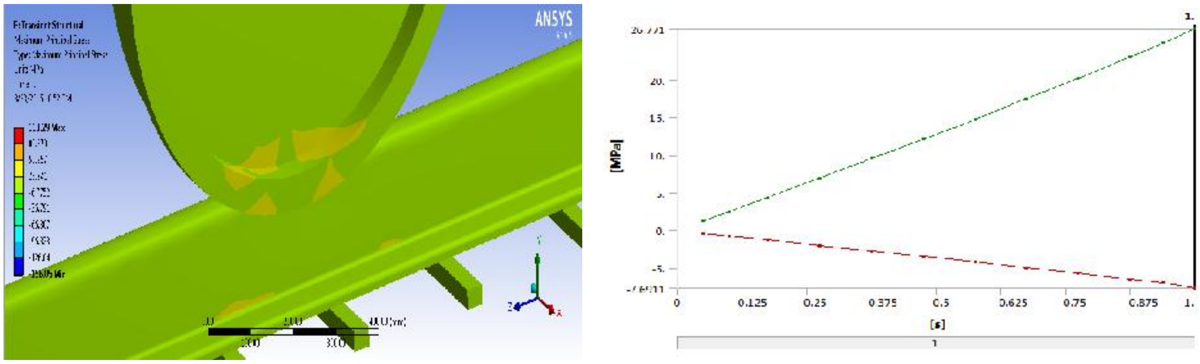


Figure 5.2 principal stress

C. Contact Pressure

As shown in the figure below the maximum contact pressure is 511.17Mpa

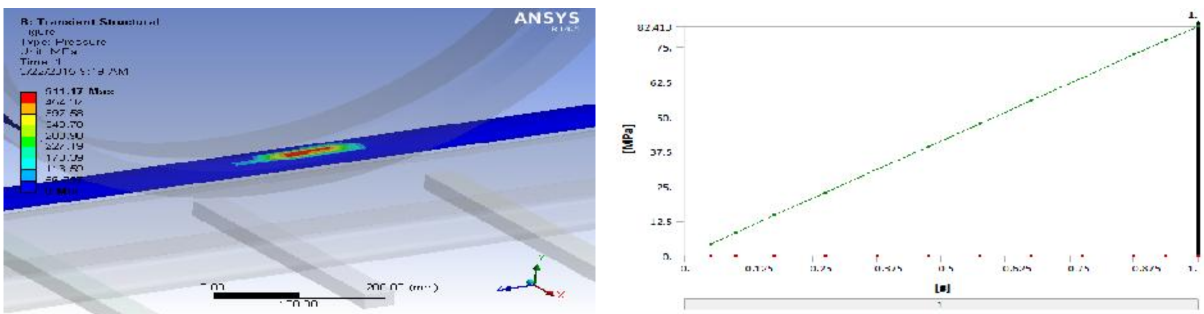


Figure 5.3 contact pressure

D. Deformation

The maximum deformation is 0.0087473mm

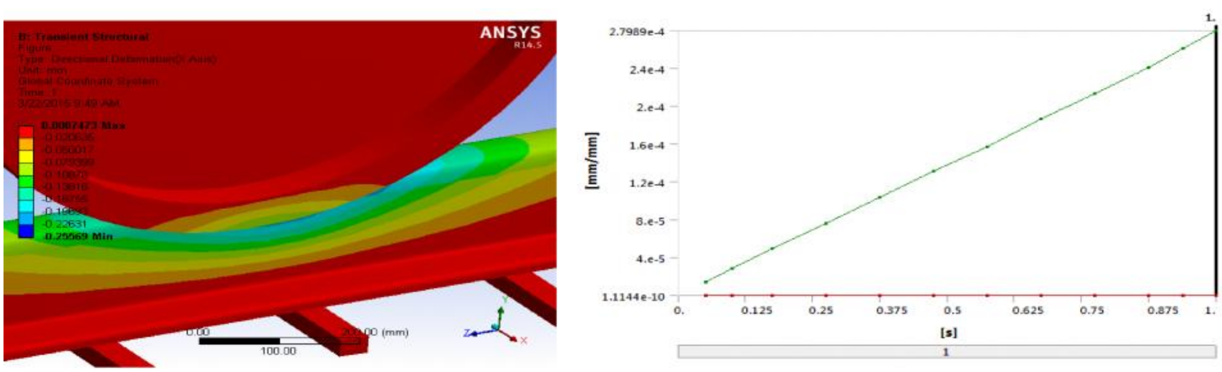


Figure 5.4 deformation

Case 2. When the Operating Speed is 70km/h

A. Equivalent Stress

As we can observe from the figure below the maximum stress is 486.39Mpa

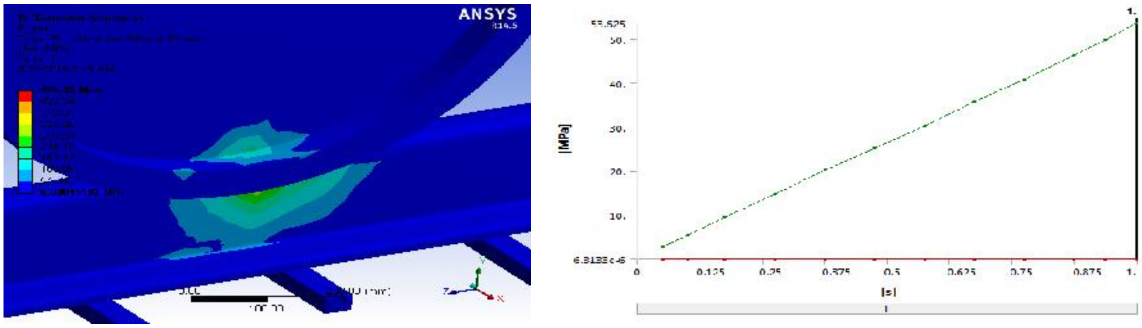


Figure 5.5 equivalent stress

B. Principal Stress

Maximum principal stress is 96.56Mpa

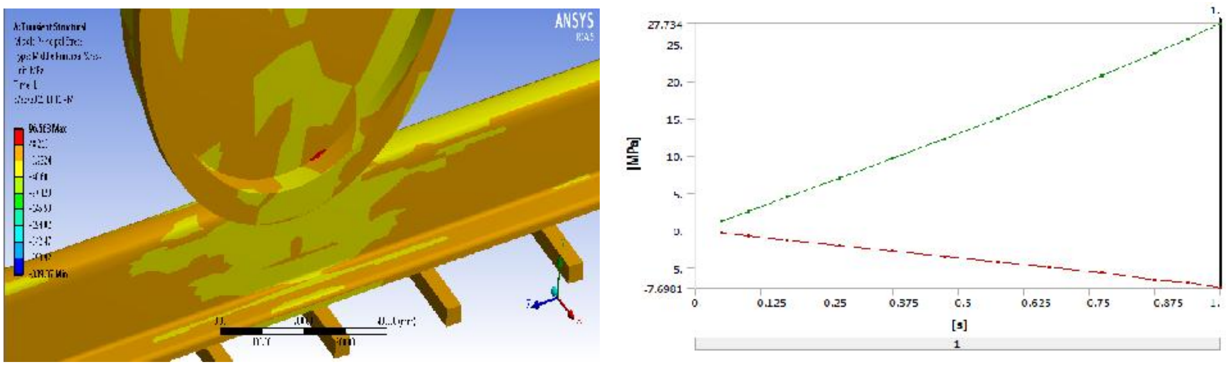


Figure 5.6 principal stress

C. Contact Pressure

As shown in the figure below the maximum contact pressure is 508.04Mpa.

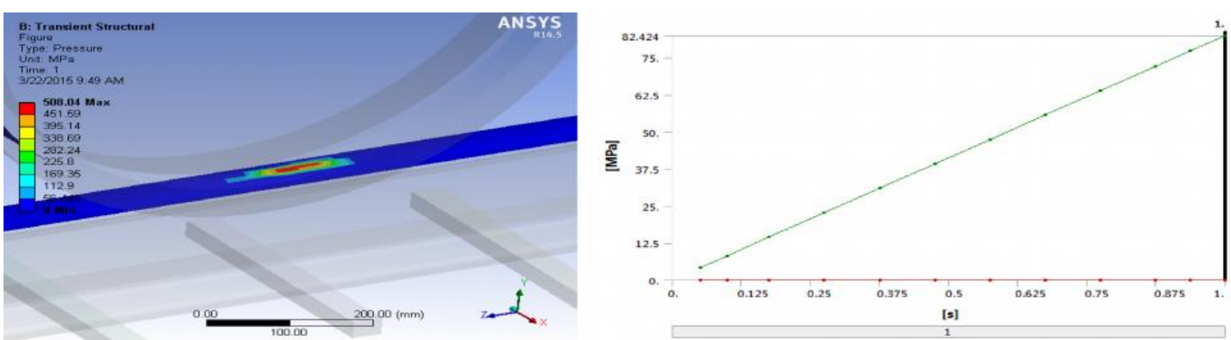


Figure 5.7 contact pressure

D. Deformation

The maximum deformation is 0.34865mm

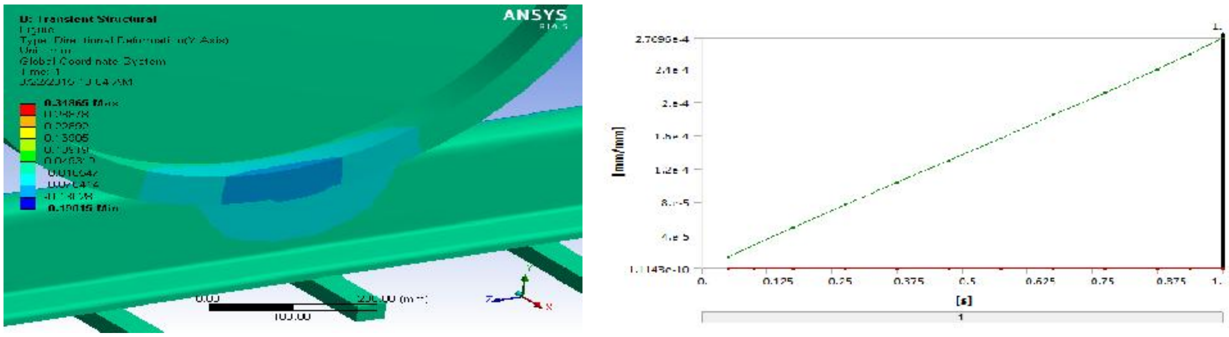


Figure 5.8 deformation

Case 3. When the Operating Speed is 60km/h

A. Equivalent Stress

As we can observe from the above figure the maximum stress is 486.53Mpa

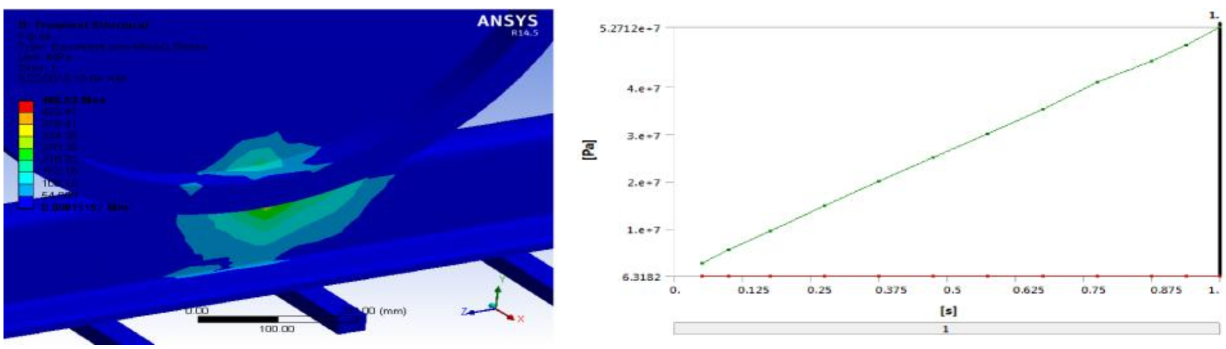


Figure 5.9: Von-mises stress

B. Principal Stress

Maximum principal stress is 227.25Mpa

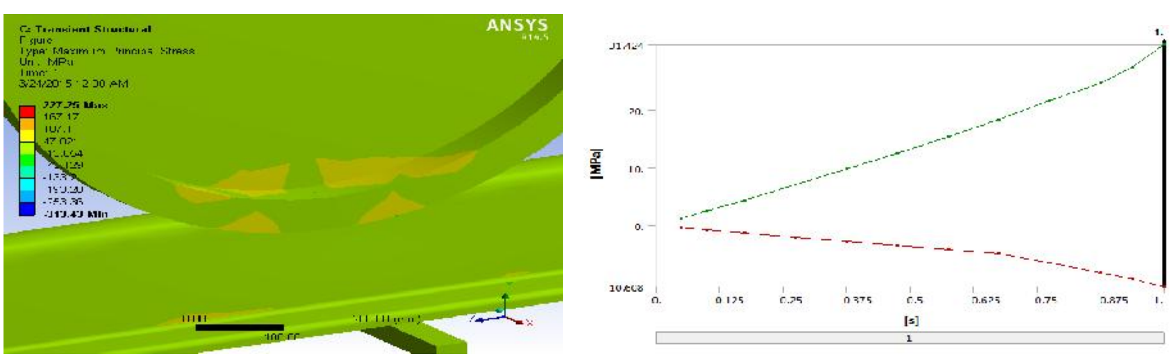


Figure 5.10 principal stress

C. Contact Pressure

As shown in the below figure the maximum contact pressure is 351.68Mpa.

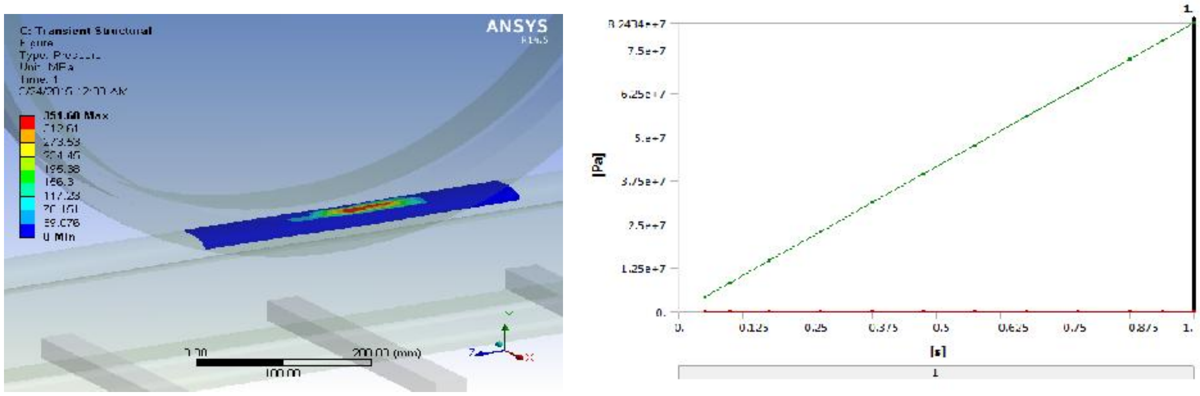


Figure 5.11 contact pressure

D. Deformation

The maximum deformation is 0.007245 mm

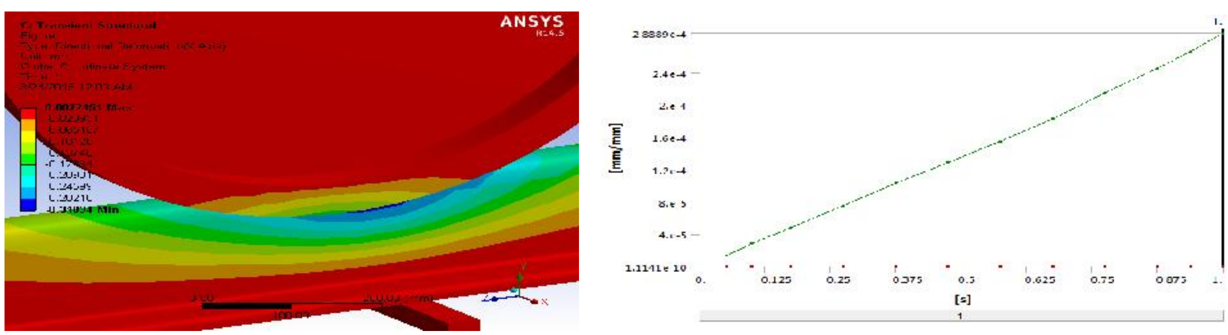


Figure 5.12 deformation

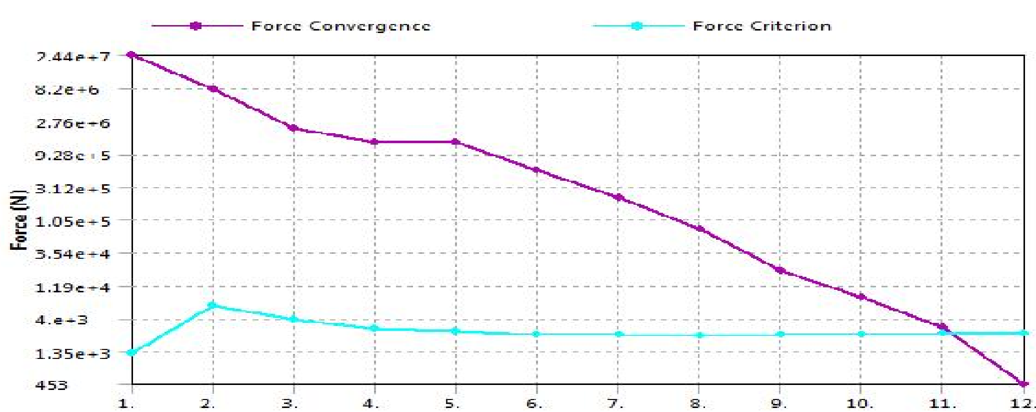


Figure 5.13 force convergence

5.3 Analytic Result

From equation 3.3 the level of sound pressure is given by:

$$\mathrm{Lp} = 20\log 10(\frac{p}{po})$$

The maximum contact pressure occur during wheel rail interaction is 511.7 Mpa when the operating speed is 80km/h. From this value of contact pressure, the level of sound pressure is calculated. By putting this value in the above equation:

$$\mathrm{Lp} = 20\log 10(\frac{p}{po})$$

$$\mathrm{Lp}= 20\log10(511.17\mathrm{Mpa}/p_o)\ (\mathrm{dB})$$

But the value of p_o (reference pressure) is $2x10^{-5}$pa. then it becomes:

$$\mathrm{Lp} = 20\log10(5111.7x10^{5}\mathrm{pa}/2x10^{-5}\mathrm{pa})\ (\mathrm{dB})$$

$$= \underline{268.15}\ \mathrm{dB}$$

And, the minimum contact pressure between the contacting surfaces when the operating speed 60km/h is 351.68Mpa. so

$$\mathrm{Lp} = 20\log10(3516.8x10^{5}\mathrm{pa}/2x10^{-5}\mathrm{pa})\ (\mathrm{dB})$$

$$= \underline{264.9}\ \mathrm{dB}$$

5.4 Discussion

This section of the paper specifies the result obtained from the ANSYS software based on hertz contact theory. The above result shows different stress types and failure criteria due to vertical applied load and different speeds between the interactions.

The above result include both rail and wheel for the reason of the analysis performed using the contact mechanism. As shown from ANSYS result the value of contact pressure is vary for one case to other case. In case1 as shown in the figure 5.3, the contact pressure is 511.7 MPa. In the case 2 as shown in the figures 5.7, the contact pressure is 508.04MPa. In the case3 as shown in the figure, 5.11 the contact pressure is 351.68 MPa.

Form the analytic result the maximum sound pressure is 268.15dB and the minimum is 264.9dB. As the speed varies the contact pressure also varied.

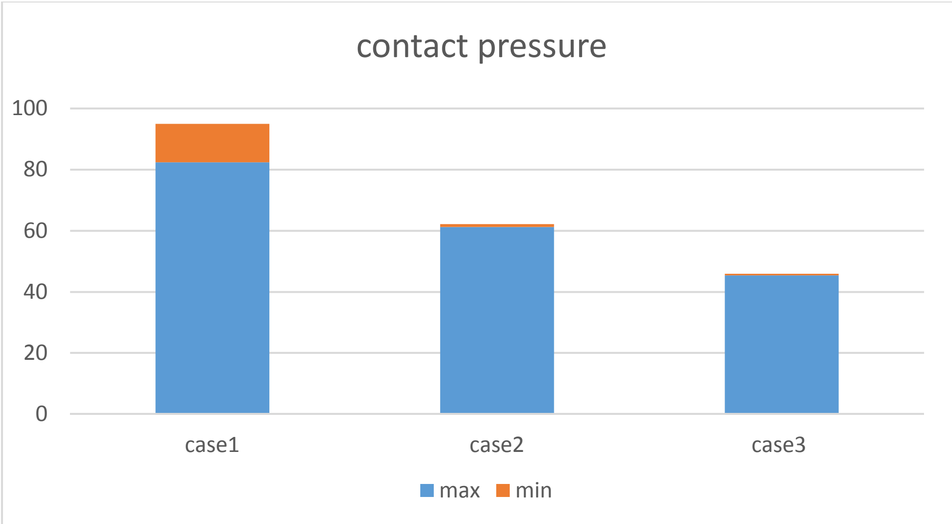


Figure 5.14 contact pressure

As we can see from figure, the maximum and minimum contact pressure distribution on rail head is given for three different operation speed. Case1 has greater contact pressure than the remaining cases.Case3 has the smallest contact pressure distribution than the rest of case.

Chapter 6

6. Conclusion, Recommendation and Future Works

6.1 Conclusion

The responses of a wheel rail contact are determined under transient structural analysis and analytical calculation results, the results are assumed to be significant. The analysis can include stress, strain and contact pressure responses of the interaction caused by vertical wheel load. The analysis is taking into account by using various operational speed.

The results obtained in the transient structural analysis and analytical calculation, the three cases have different stress distribution and different contact pressure on rail head and the wheel. The contact pressure when the operational speed 80km/m is high in comparison to 70km/h and 60km/h. The contact pressure is small when the operational speed is 80km/h in comparison to the remaining.

The maximum contact pressure on the elliptical contact patch is assumed to be uniform pointing its extreme value on the center of the contact patch where the radii of the contact patch approaches zero and finite on the circumference of the ellipse. The value of average maximum contact pressure on the whole area of ellipse (contact patch) is 511.7 MPa. This is due to the operating speed 80km/h and the minimum contact pressure on the whole area of ellipse is 351.68Mpa when the operating speed is 60km/h. the corresponding sound pressures are ***268.15 dB*** at 80km/h and ***264.9 dB*** at 60km/h. From this by reducing the operating speed of the train, we can reduce the noise generated between the contacting surfaces.

6.2 Recommendation

Due to high speed, rolling noise becomes serious case for passengers and near the railway line. In my opinion, to reduce this noise increasing wheel rail material and reducing operating speed of the train is recommended.

6.3 Future Works

During the analysis of wheel/rail contact, there are a lot of problems to be solved. However, due to the broad applications of wheel/rail contact mechanics, in this paper the analysis is limited to the geometry of wheel/rail contact just to predict the generation of noise during contact and to show the effect of different operating speed.

During the work of this paper and review of literatures there are a lot of related problems expected to be solved parallel or after the accomplishment of this paper. However, expecting that, identifying the place of failure and failure causes play a great role on the future works of minimizing wheel/rail contact problems, this paper gives priority for the analysis of wheel/rail contact and generation of noise.

Future railway demands and expected shortage of raw materials in the world and in our country Ethiopia, the broad study of contact mechanics invites the future works of researchers, especially on wheel/rail interaction mechanics. There are so many problems on the wheel/rail interaction mechanics:

- Analysis of Wheel and rail Surface irregularities
- Wheel/rail shape optimization for a specific application
- Design of wheel derailment preventive mechanism etc.
- Analysis of rolling noise reduction
- Identifying the type of stress induced on the wheel/rail contact interface
- Analysis of rolling noise reduction etc.

References

1. Escalona, J. and R. Chamorro (2008). Efficient On-Line Calculation of the Wheel-Rail Contact Forces in Multibody Dynamics. Symposium of Advances in Contact Mechanics. Delft, The Netherlands.
2. Thompson, D. J. and P. E. Gautier (2006). Review of research into wheel/rail rolling noise reduction. Proceedings of the Institution of Mechanical Engineers, Part F: Journal of Rail and Rapid Transit 220: 385-408.
3. Shevtsov, I.Y., V.L. Markine and C. Esveld (2003). Optimal Design of Wheel Profile for Railway Vehicle. Conference on Contact Mechanics and Wear of Rail/Wheel Systems. Gothenburg, Sweden June 10–13, 2003.
4. Evans, J. R., T.K.Y. Lee, and C.C. Hon (2008). Optimising the Wheel/Rail Interface on a Modern Urban Rail System.Vehicle System Dynamics. 46:51, 119-127.
5. Kaewunruen, S. and A.M. Remennikov (2008). Dynamic Properties of Railway Track and its Components : A State-of-The-Art review. Faculity of Engineering Papers. University of Wollongong.
6. Pombo, J., J. Ambrósio and M. Silva (2007). A new Wheel–Rail Contact Model for Ralway Dynamics. International Journal of Vehicle Mechanics and Mobility. 45:2, 165-189.
7. Karwacki, J. and S. Duda (2009). Methodology for the Dynamic Analysis of the Single Wheelset Rrunning on Turnout. Modeling and Optimization of Physical Systems.
8. Kataoka, H. (2008). Design Concept and Material Demand Characteristics of Rail. Welding International . 22:6, 411-415
9. Knothe, K. (2008). History of Wheel/Rail Contact Mechanics: From Redtenbacher to Kalker, Vehicle System Dynamics. International Journal of Vehicle Mechanics and Mobility . 46:1-2, 9-26.
10. Thompson, D. J., B. Hemsworth and N. Vincent (1996). Experimental validation of the TWINS prediction program for rolling noise, Part 1: description of the model and method. Journal of Sound and Vibration.
11. Frich, A. and Dr. Wolfgang, S. (2008). Target Profiles for Grinding A Never Ending Story.

12. ARM Conference. Swedish Rail Administration. Jones, C. J. C. and D. J. Thompson (2003). Extended validation of a theoretical model for railway rolling noise using novel wheel and track designs. Journal of Sound and Vibration.
13. Pau, M., F. Aymerich and F. Ginesu (2002). Distribution of Contact Pressure in Wheel–Rail Contact Area. Wear. 253 : 265-274.
14. Matsumura, R., H. Sugiyama and Y. Suda (2011). Analysis of Vehicle/Turnout Interactions of Railroad Vehicles Using Multiple Contact Tables. Journal of System Dsign and Daynamics.5:3.
15. Thompson, D. J., C. J. C. Jones, T. P. Waters and D. Farrington (2007). A tuned damping device for reducing noise from railway track.
16. Yan, W. and D. Fischer (2000). Applicability of the Hertz contact theory to rail-wheel Contact problems. Archive of Applied Mechanics. 70: 255-268.
17. Addisu Negash, October, 2012, Analysis of Wheel/Rail Contact Geometry and Applied Load Conditions on the Rail Head Surface, Addis Ababa institute of technology, Addis Ababa Ethiopian
18. Thompson, D. J. (2008). Railway noise and vibration: mechanisms, modelling and means of control. Oxford, Elsevier.
19. European Environment Agency (2009). Transport at a crossroads TERM 2008: indicators tracking transport and environment in the European Union. EEA Report No 3/2009.
20. Berglund, B., T. Lindvall and D. H. Schwela (Eds.) (1999). Guidelines for community noise. World Health Organisation, Geneva.
21. UIC Freight Noise Focus (March 2008) No.2:.
22. Thompson, D. J. (1993). Wheel-rail noise generation, Journal of Sound and Vibration
23. Vincent, N., P. Bouvet, D. J. Thompson and P. E. Gautier (1996). Theoretical optimization of track components to reduce rolling noise. Journal of Sound and Vibration 193(1): 161-171.
24. Remington, P. J. (1976). Wheel/rail noise part IV: rolling noise. Journal of Sound and Vibration.
25. Asmussen, B., H. Onnich, R. Strube, L. M. Greven, S. Schroder, K. Jager and K. G. Degen (2006). Status and perspectives of the "Specially Monitored Track". Journal of Sound and Vibration 293(3-5): 1070-1077.

26. Oertli, J. (2008). Railway noise abatement: the case for retrofitting freight vehicles with composite brake blocks. In Noise and Vibration Mitigation for Rail Transportation Systems Proceedings of the 9th International Workshop on Railway Noise, Munich, Germany, 4 - 8 September 2007:236-242, Springer: Berlin.
27. Golubenko, A., S. Sapronova, and V.Tkachenko (2007). Kinematics of Point-to-Point Contact of Wheels With a Rails. Transport Problems. East Ukrainian Volodymyr Dal National University.
28. Dings, P. C. and M. G. Dittrich (1996). Roughness on Dutch railway wheels and rails. Journal of Sound and Vibration 193(1): 103-112.
29. Kumaran, G., D. Menon and K.K. Nair (2003). Dynamic Studies of Rail Track Sleepers in a Track Structure System. Journal of Sound and Vibration. 268: 485-501.
30. Povilaitiene, I. and A. Laurinavicius (2004). Reduction of external rail wearing on road curves. Journal of Civil Engineering and Management. 10:2, 123-130.