ADDIS ABABA UNIVERSITY

ADDIS ABABA INSTITUTE OF TECHNOLOGY

SCHOOL OF MECHANICAL AND INDUSTRIAL ENGINEERING

(Railway Engineering Program)

STRUCTURAL ANALYSIS AND DYNAMIC SIMULATION OF HEAVY HAUL TRAIN COUPLER WITH DRAFT GEAR FOR ROTIONAL EFFECT

BY

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Advisor

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A master thesis submitted to the school of Graduate Studies of Addis Ababa University in Partial fulfillment of the degree of Master of Science in Rolling Stock Engineering

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Addis Ababa Institute of Technology [AAIT]

DECLARATION

I hereby declare that the work which is being presented in this thesis entitled **“**STRUCTURAL ANALYSIS AND DYNAMIC SIMULATION OF HEAVY HAUL TRAIN COUPLER WITH DRAFT GEAR FOR ROTIONAL EFFECT” is original work of my own, has not been presented for a degree in any other university; and that all sources of material used for the thesis have been duly acknowledged.

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ABSTRACT

Coupler and draft gear package is the most important parts of heavy haul train .In my thesis basic consideration is the structural analysis and rotational behaviour effects of heavy haul train coupler and draft gear .The structural analysis of the coupler and draft gear was analyzed by ANSYS software and the 3D modeling by CATIA .The important of structural analysis is used to determine the strength of the coupler material, the maximum deformation and stress that acted on the coupler simply determined .Those helps us to determine the critical part needs more maintenance , and also used to determine which part is wear easily. Finally the structural analysis of coupler with draft gear is important in order to increasing the performance of the heavy haul train, decreasing of the train derailment due to coupler effects.

When a locomotive coupler rotates at an angle, the lateral component of the coupler force has an adverse effect on the locomotive's safety, particularly in heavy haul trains. In this paper, a model of four locomotives and three couplers, a 100-ton heavy haul train is developed to analyse the rotation behaviour of the locomotive's coupler system and its effect on the dynamic behaviour of such a train's when operating on straight and curved tracks. The train model includes detailed coupler and draft gear with which to consider the hysteretic characteristics of the rubber draft gear model, the friction characteristics of the coupler knuckles, and the alignment-control characteristics of the coupler shoulder. The results indicate that the coupler's rotation behaviour differs between the lateral force, longitudinal force and vertical forces, significantly affecting the locomotive's running performance under the braking condition. A larger coupler rotation angle generates a larger lateral component, which increases the wheel set's lateral force and the derailment coefficient. Decreasing the maximum coupler free angle can improve the locomotive's operational performance and safety. Based on these results, the recommended maximum coupler free angle is 5°.the dynamic simulation of individual coupler rotational angle due to parameter variation of forces. This parameter variation helps as to understand the coupler maximum rotational angles due to the lateral forces.

Keywords: Structural analysis; coupler Rotational angle; Rotational angle behaviour and effects; lateral force, longitudinal force, vertical forces and parameter variation

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NOMENCLATURE

AAR -Association of America rule

FEM -Finite element methods

DAE- Differential Algebraic Equations

DOF-Degrees of Freedom

FFT- Fast Fourier Transformation

ISO -International Standards Organization

MBS-Multibody System

ODE-Ordinary differential Equations

UIC - Union of International Railway

CHAPTER 1

INTRODUCTION

1.1. Background

[30] The history of rail transport, including systems with man or horse power, and tracks or guides made of stone or wood, dates to as early as Greek times. The first trains were Single wagons pushed or pulled by people or animals, and were used to move goods, such as coal. Steam trains were the first type of trains. They use coal in firebox to boil water until it turns to steam. The steam is forced through powerful pistons to give the engine the power to drive the wheels. At first, steam engines moved mainly goods, but were soon used to carry passengers as well. Steam engines are still in use all over the world, although most have been replaced by diesel or electric trains. Diesel locomotives were first introduced in Australia in the 1930's. They were a powerful addition to the railways. Diesel fuel powers an engine which drives a generator to make electricity. The electricity powers traction motors that turn the wheels. Diesel locomotives are used to transport enormous quantities of materials over huge distances. They are also more efficient and smoother than steam trains, and carry much heavier loads.

[27] Rail way lines construction i n Ethiopia was first started i n October 1897 from Dji bouti in the period of Emperor Menelek II. The first commercial service began in July 1901, from Djibouti to DireDawa. By 1915 the line reached Akaki , 23 kilo meters from the capital , and two y ears later came all the way to Addis s Ababa. The railway links Addis Ababa, the capital of landlocked Ethiopia, to the port of Djibouti in coastal Djibouti. Maintenance shops along the line are located in DireDawa, which grew up as the Imperial Ethiopian Railway depot for nearby Harar. The single track 781 km railways used to a 1,000 mm gauge, most of it on Ethiopian territory, and about 100 km in Djibouti.

Railway vehicles are connected to each other by different kinds of couplers, which have certain elastic and damping characteristics. When a vehicle moves from one place to another place and during the shunting operation in train yard, the motion of the train regards high pull and push force between connected wagons known as the buff and draft force. This force causes damage on the wagon equipment's, or increases the risk of derailment, which is a serious economic and safety problem. To prevent the wagon equipment's from damage it should be mounted on draft gear units which will be designed with the proper parameters. This unit will keep the equipment from damage by absorbing the longitudinal shocks (energy) during train movements and during shunting operation. This is achieved by receiving and reducing the shocks and pulling stresses before transmitting them to the car body and it reduces the amount of force when a decrease the vehicle's speed in impact yard. The purpose of draft gear may include also protection of passengers and freight; they are also used to protect the draw gears and the car structure from excessive loads in heavy mineral haulage applications.

In the railroad history, in 1890 dozens of draft gears designs were already in use to protect the vehicle from damaging by draft forces, it was located within the center sill between the coupler shank and body bolster [26]. Since many cars use this arrangement till now only by changing the friction dampening. Friction type draft gear was introduced in 1900, and it becomes increasingly popular. As the coupler moves the friction draft gear slides against each other on their surfaces, and these surface hold together by the spring.

Coupler and draft gear packages are important components of heavy haul trains because their behavior directly affects the safety of these trains. Due to the high cost of running tests, most researchers have adopted computer simulations and developed mathematical models of coupler and draft gear packages to carry out numerical studies on these systems. The existing draft gear mathematical models can be categorized into three types: linear, piecewise linear and nonlinear hysteretic. The linear model simplifies the draft gear into linear spring and damper system,[22–23] which greatly improves the simulation efficiency. However, because of the conspicuous differences between an actual draft gear and its simplified model when using linear characteristics, this model has rarely been used in research. Some piecewise linear models have been applied to European side buffers.[4,5] To date, however, the most accurate draft gear model is the nonlinear hysteretic model, which was proposed by Duncan [6] in 1989. Duncan's nonlinear hysteretic model included a coupler slack, approximate draft gear impedance characteristics, and rigid impact characteristics applied after the draft gear loses its effectiveness.

1.2 Statement of the problem

When a locomotive coupler rotates at an angle, the lateral, longitudinal and vertical component of the coupler force has an adverse effect on the locomotive's safety. The results indicate that the coupler's rotation behavior differs between the straight and curved tracks, significantly affecting the locomotive's running performance. A larger coupler rotation angle generates a larger lateral component, which increases the lateral force and the derailment coefficient. A large longitudinal compression force makes the coupler yaw unstable when the heavy-haul trains in stopping conditions. The larger yaw angle of coupler will produce a greater lateral force which may lead to the locomotive's derailment. As one of the key equipment items of the railway heavy-haul transport, the yaw stability requirements of the heavy-haul coupler are required to increase as train tonnage and speed increase. The derailment caused by coupler occurs occasionally when the train is in emergency breaking, [1–5]. Several train derailment accidents have occurred in recent years due to the rotation behavior of couplers. [15, 16] As a result, many scholars have built coupler models to analyses coupler rotation behavior and its implementation. The most common method involved calculating the coupler rotation angles and coupler forces individually, the results of which were then applied to a train model. So in order to make solution for this problem I have to consider parameter variation for the force components and easily identified by the dynamic simulation result using SIMPACK software.

1.3. Objective of the research

1.3.1 General objective of the research

The objective of this paper is to set forth methods of structural analysis and the rotational behavior effects of rail freight coupler with draft gear due to the lateral, longitudinal and vertical operational system including the parameter variation.

1.3.2 Specific objectives

- Model of the coupler and draft gear by CATIA.
- Static analysis of the coupler and draft gear by ANSYS software to check the effect of the static load on the coupler and draft gear.

- Model and dynamic simulation of the coupler with draft gear by using SIMPACK.
- The train model includes detailed coupler and draft gear with which to consider the hysteretic characteristics of the rubber draft gear model.

1.4. Significant of the research

The main advantage of this paper is to know the strength of the coupling material in order to preparing for maintenance during damaging of coupling. The coupler system and its attachment to the car body shall respond to normal and overload conditions in a predictable manner. The coupler system shall be capable of absorbing the compression and tension forces encountered in normal vehicle operation in a train, including coupling and uncoupling, without damage. Coupler and draft gear rotation effect on heavy haul trains due to the lateral, longitudinal, vertical operation and parameter variation. To improve the rotational effects of the coupler with draft gear we have to consider basically the railway freight parameterization based on the coupler applied force in the different components in the SIMPACK dynamic simulation software in order to understand the variation of force which acts in the lateral, longitudinal and vertical operational conditions that affect the coupler rotational behavior.

1.5. Scope of the study

As the designer of the coupling system, the scope of work for the Project is mainly as follows:

- Structural Analysis and modeling of rail freight coupler with draft gear.
- Static Analysis of rail coupler with draft gear using ANSYS software.
- Basically we have to consider F-type railway coupling with draft gear standard parameters. And also considering the coupler force parameterization for dynamic behavior of railway freight coupler and draft gear due to lateral force, longitudinal force and vertical forces.

1.6. Limitation of the study

The main limitations of the current study are:

- The rail vehicle model used in the simulation study comprises a rigid car body, i.e. Car body flexibility was not considered in the current study but it does influence rail vehicle dynamics and it is another engineering problem which needs to be solved.
- Track flexibility and mass were not considered in the current study but they do influence rail vehicle dynamics; therefore, the effect of such track parameters on rail vehicle dynamic behaviour is another engineering problem which needs to be studied.
- Effect of wind, traction and braking were not considered in the current study but they do influence rail vehicle dynamics so incorporating traction and braking in rail vehicle dynamics analysis results in better simulation results.
- All wheel profiles were considered identical from left to right on a given axle and from axle to axle and all wheels remains in contact with the rails as well as no wheel flats was considered but in practice all wheel profiles are not identical, there are wheel lifts and wheel flats which influence ride dynamics; hence, consideration of such things will give better result.

CHAPTER 2

LITERATURE REVIEW

2.1. Theoretical review

This part presents review of materials relevant for the study such as:

- ✓ Multi body system dynamics, rail vehicle dynamics modeling and coupler Rotational effect of behaviour of rail vehicles.
- ✓ Structural analysis of coupler with draft gear By ANSYS software.

2.1.1. Multibody system and dynamic analysis

Multibody system (MBS) dynamics studies the motion of mechanical systems composed of several rigid or flexible bodies interconnected by joints and subjected to internal or external forces and torques. Those forces can result from both passive elements such as springs, or active components such as electromechanical actuators.

The calculation of the body motion is one of the principal objectives of the dynamic analysis. This type of analysis also provides a process to estimate external forces which depends on the relative position between system components, such as those generated by springs, dampers and actuators. Also external forces which generated as a consequence of the system interaction with the surrounding environment, such as contact and friction forces, are considered. Another significant result provided by this type of analysis, is the calculation of the internal reaction forces, generated in the kinematic pairs. The first step in many multibody problems consists in describing the system topology in suitable way for mathematical analysis. The main components of a multibody system are:

- ➢ The bodies, characterized by their mass, center of mass location and inertia tensor
- ➢ The joints that determine the relative motion between two bodies or between a body and the inertial frame several formalisms have been developed so as to automatically obtain the equations of motion of any multibody system. The most basic one relies directly on the Newton-Euler equations.

The so called recursive Newton-Euler algorithm allows reducing the computational efforts when relative coordinates are used. Multi body dynamics is based on classical mechanics and has a long and detailed history.

The simplest multibody system is a free particle which can be treated by Newton's equations. D'Alembert considered a system of constrained rigid bodies where he distinguished between applied and reaction forces. A systematic analysis of constrained mechanical systems was established by Lagrange. Modern methods for the dynamic analysis of constrained multibody systems fall into two main categories: differential algebraic equations (DAEs) and ordinary differential equations (ODEs) [15]. DAEs employ a maximal set of variable to describe the motion of the system and use multipliers to model the constraint forces. Pre-multiplying the constraint reaction-induced dynamic equations by orthogonal complement matrix to the constraint Jacobin results in the governing equations as ODEs. A Multibody system is a system which is an assembly of two or more rigid or flexible bodies (also called elements) imperfectly joined together, having the possibility of relative movement between them. This imperfect joining of the two rigid bodies that makes up a multibody system is called a kinematic pair or joint, or simply a joint. A joint permits certain degrees of freedom of relative motion and prevents or restricts others.

2.1.2. Advanced modeling for simulation and computer tools

Various techniques and formulations have led to the implementation of various computer tools vehicles. Some of these have been combined into general purpose packages and some examples of those currently in widespread use are given here [3, 6, and 24]. One of the early complete packages, MEDYNA (Mehrko¨rper-Dynamik) was developed at the German Aerospace Research organization DLR together with MAN and the Technical University of Berlin. MEDYNA was based on a MBS with small rigid body motions relative to a global reference frame which allowed large motions. The linearized kinematic equations of motion for each body are formulated with respect to the global reference frame. SIMPACK was developed later by the same team at DLR and as it was intended for road vehicles and other systems as well as rail vehicles it allowed nonlinear kinematics from the start. The equations of motion are formulated in terms of relative coordinates and can be generated symbolically and numerically in an implicit and explicit form. The kinematics of elastic bodies was developed to allow stress stiffening effects to be taken into account. ADAMS is one of the most popular dynamic simulation codes worldwide and in 1995 entered the railway vehicle simulation market as ADAMS/Rail, initially by including wheel–rail contact methods developed by Ned Train and later by licensing the wheel–rail contact elements from MEDYNA.In the U.S.A the Association of American Railroads (AAR) funded the development of a program to simulate the behaviour of a railway vehicle negotiating a curve. This was developed in to the general purpose simulation package NUCARS (New and Untried Car Analytic Regime Simulation). NUCARS has been used to improve the dynamic behaviour of the three-piece freight bogie.

2.1.3. Rail way engineering systems and coupler Rotation effects behaviour

A rail vehicle often consists of a car body supported by two sets of running gear. In former passenger vehicles and still partly today's freight vehicles, the car body was only supported by single-axle running gear. Modern freight and fright rail vehicles, in particular for fast or high-speed traffic, are mostly designed as bogie vehicles. Note that the positive vertical direction is pointing downwards according to International Union of Railways UIC. The vehicle consists of a car body supported by two bogies through secondary suspension. Each bogie consists of a frame and two wheelsets, connected by the primary suspension. Both the primary and secondary suspensions include spring and damper components that vary depending on the vehicle type. Examples of such suspension elements are air springs, coil springs, rubber springs and hydraulic dampers. The type of component is dependent on the vehicle operational task and the desired suspension characteristics. Furthermore; the suspensions include bump stops, delimiting the suspension motions in the vertical and lateral directions.

2.1.4. Review of researches related to the current study

Sun Xing and Cheng Qing, in 1989[12] .The model was developed primarily to allow use of limited computational capacity. To study longitudinal train dynamics they applied a numerical method. During motion many exciting forces act on the train, these forces are tractive, resistant braking, and air braking force. They simplified these forces to the calculation of train dynamics and simplified the mode to sum up considered that the train longitudinal vibration is composed of the lower frequency expand and contract of train overall and the higher frequency partscaused by shock. In their model, the locomotive and vehicles in the train considered as a rigid join with coupler buffers and train is simplified to an equivalent linear system. They also realized to simulate the train longitudinal dynamics in real time on a common microcomputer.

The longitudinal dynamics of railway vehicles were studied by M. Ansari, D.Younesian and E.Esmailzadeh in 2007[23]. They looked in on longitudinal train dynamic thoroughly in their model. Damping of automatic couplers, train forward speed, coupler clearance, and train acceleration and braking processes) on longitudinal train dynamics were investigated parametrically .Ahmed A. Shabana and Ahmed K. Aboubakr in 2012 [13]developed accurate non-inertial longitudinal train force models applied a linear equation of the draft gear. Aimed nonlinear coupler model allows for arbitrary three dimensional motions of the car bodies and captures kinematic degrees of freedom that are not captured using existing simpler models

Garg and Dukkipati, in 1984[14],presented several examples of longitudinal train dynamics analysis using such wagon connection modeling, draft gear and coupler system was represented as an assemblage of spring and dry friction element. In general, draft gear forces are independent of velocity and travel versus force characteristic curve was adequate for modeling.

Dynamic research in connected trains based on Daqing heavy haul coal line was studied by Sun and Sun in1987 [15]. The longitudinal shock changes along the train length, and depends

on the characters of the locomotive and wagon equipment, handling ways, the longitudinal and lateral profiles of track etc. This longitudinal shock affects the safe operation of the train. They found that; the shock level of two connected trains is higher than the ordinary one, but much lower than a long train with double heads. The clearance between couplers has big influence on the train shock. Sun and Sun demonstrated, it can reduce the shock level by

[32]Freight car coupling arrangements (Wabtec Corporation).This document includes the movement of railway equipment involves heavy pulling and pushing forces to move the weight of the cars as the train moves. This buffing and pulling action between each car and locomotive occurs each time the equipment moves. To ensure the car(s) and locomotive(s) can accept this movement without causing damage to the equipment and/or lading, the equipment must have a system to absorb this punishment. The coupling system is designed like a large shock absorber to

accept the impact of car movement without resulting in equipment or lading damage. With the use of a draft system trains hauling many tons of consumer products and over a mile in length are possible.

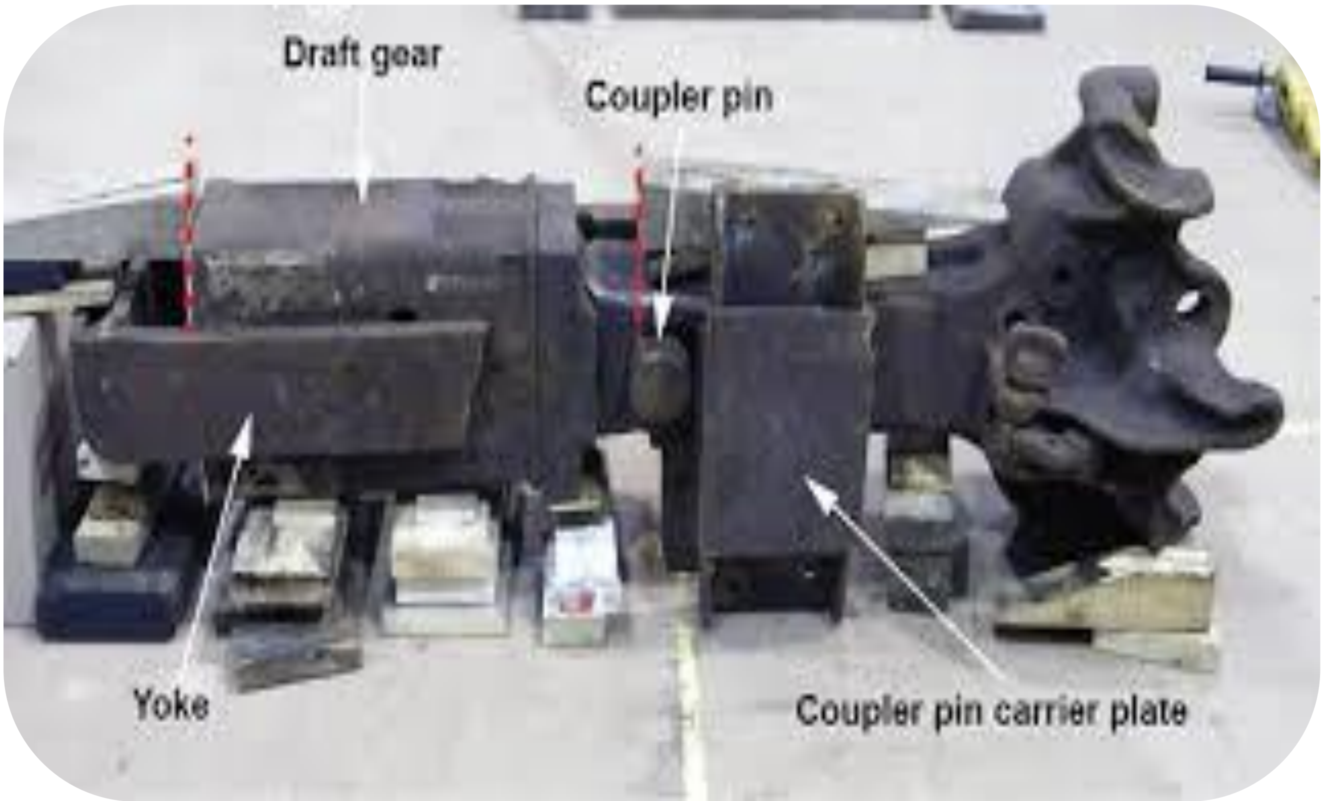


Figure-2.1.railway fright coupler and draft gear position.[32]

Methods of analysis for determine the coupler force and longitudinal motion of along the freight trains in over the road operation by gc.martin and w.w hay. [33].This paper contains a methods of analyzing for determine the longitudinal dynamics of a long train in over the road operation .the methods hinges up on modeling train and its longitudinal cushioning devices as amass spring system. The force acting on the train in its operation are described and applied to the model .A computer simulation is then performed, using a numerical integration scheme ,to find the coupler forces, displacement, velocities and acceleration for the cars in the trains.

[34]Yuan Yaoa, Xu Liua, considered investigating the stability and mechanical characteristics of a type of heavy haul coupler with restoring bumps top, the geometry and force states of couplers were analyzed at different yaw angles and the longitudinal forces. The structural characteristics of this coupler were summarized. To aid in the investigation, a multi-body dynamics model with four heavy haul locomotives and three detailed couplers was established to simulate the process of emergency braking. In addition, the coupler yaw instability and lateral forces were tested in order to investigate the effect of relevant parameters on the locomotive's wheel set lateral forces. The results show that only when the bumps top force exceeds half of the coupler longitudinal compression force, can the follower be rotated and the yaw angle of the coupler increase. The bumps top preload is the most important stabilizing factor.

[34] The stability mechanism and its application to heavy-haul couplers with arc surface contact. This paper basically considered the stability mechanism of a type of heavy-haul coupler with arc surface contact, the force states of coupler were analyzed at different yaw angles according to the friction circle theory and the structural characteristics of this coupler were summarized. A multi-body dynamics model with four heavy-haul locomotives and three detailed couplers was established to simulate the process of emergency braking. In addition, the coupler yaw instability was tested in order to investigate the effect of relevant parameters on the coupler stability. The results show that this coupler exhibits the self-stabilization and less lateral force at a small yaw angle. The yaw angle of force line is less than the actual coupler yaw angle which reduces the lateral force and the critical instability. An increase in the friction coefficient of the arc contact surfaces can improve the stability of couplers. The friction coefficient needs to be increased with the increase in the maximum coupler longitudinal compressive force. The stability of couplers is significantly enhanced by increasing the secondary suspension stiffness and reducing the clearance of the lateral stopper of the locomotives.

[35] National model railroad association (data a sheet).This documents basically concerning about the different types of coupling with their main purpose and application.

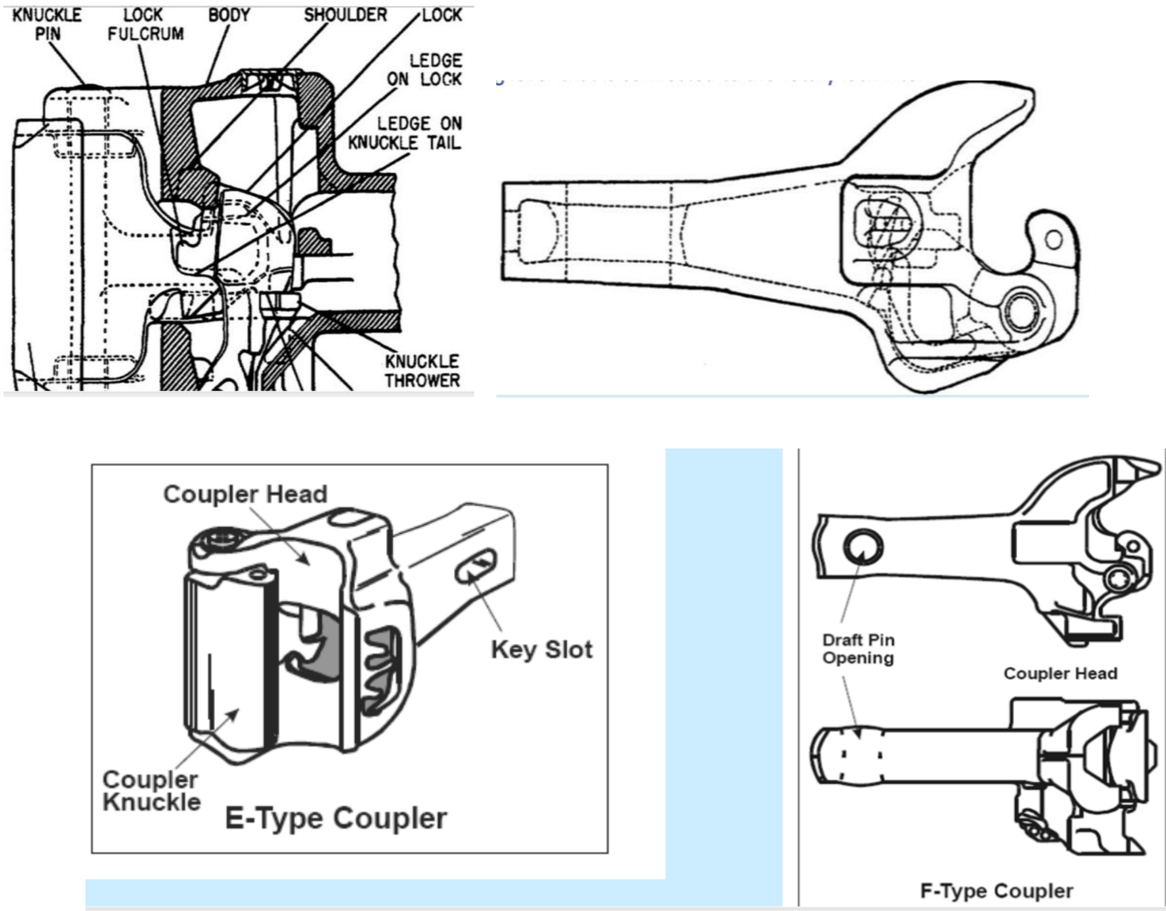


Figure-2.2. different types of rail coupling [35]

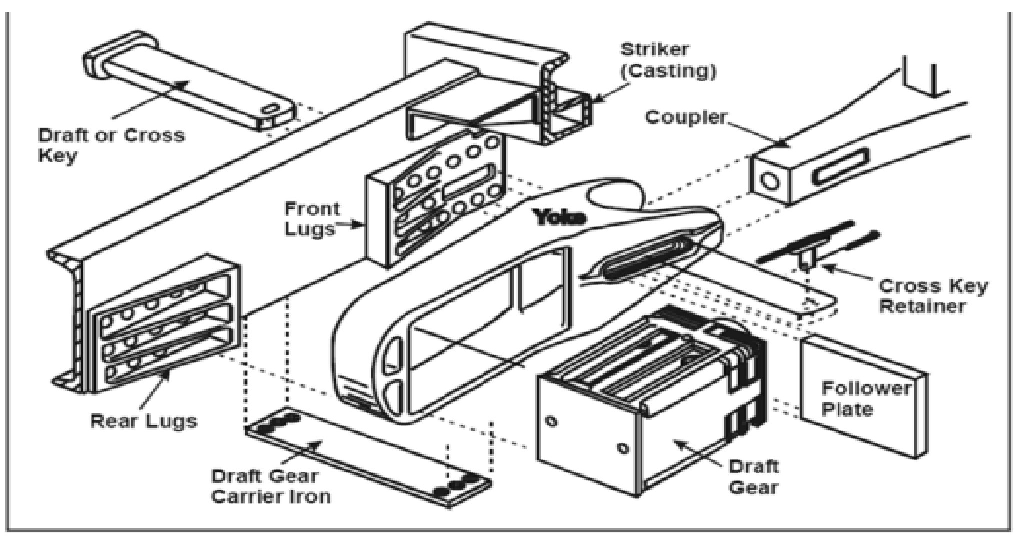


Figure-2.3.Arrangement of draft gear with coupler.[35]

CHAPTER 3

METHODOLOGY

The method to be employed to achieve the objectives of the research includes:

3.1 Data collection

- Visualizing our project by taking a sample from different railway engineering book
- Reading literatures books and browsing from different websites regarding railway engineering.
- Specifying the required specifications from different catalogues.
- This includes reading books, articles, designs, standards and guidelines.

3.2 Data Analysis

- Making the analysis and interpreting the results of the components using Ansys software .And also involves simulating the modeled coupler and draft gear system for different detection schemes using ANSYS software.
- Preparing the modeling using CATIA or solid work Software.
- Analyzing all tracks System modeling: it involves formulating the mathematical relationship governing the related coupling system parameter..

3.3 Data presentation

- Performance comparison, and Analysis and Interpretation of the Results: This involves comparing the performance of the different detection schemes, and analyzing and interpreting the result.
- By Preparing the graph of in different data
- By preparing the chart.
- By preparing the picture of the ANSYS result

3.4. Materials

- ✓ Rail vehicle-considering the freight coupler with draft gear
- ✓ Track- The vehicle was simulated on curvature and straight standard gauge track of length 3000m
- ✓ Simulation tool- The study was based on computer simulations and analysis of rail vehicles using SIMPACK 8.605 . The software is developed at the German aerospace research center (DLR) together with INTEC GmbH, with important practical enhancements added to the software at MAN technology AG, Germany .

CHAPTER 4

STATIC ANALYSIS OF RAIL FREIGHT COUPLER WITH DRAFT GEAR

4.1 Introduction

There is currently worldwide trend towards using continuously welded rails for minimizing the wheel-rail impact forces. Nevertheless, rail joints are still used in some areas. In particular, insulated rail joints are required for track electrical insulation to detect the train location and to isolate sections such as those near road crossings. When a train runs over a joint, large dynamic impact forces are developed which lead to vibrations in the structures and a higher probability of component fatigue and damage. Thus, it is clear that rail joints can affect the maintenance costs, ride comfort and running security a modern railway.

The freight damage incurred during railroad transportation is a serious economic and safety problem. The railroad freight car's dynamic characteristic leads to most of the freight damage. The dynamics of railroad car and freight damage can be divided into two groups:

- ✓ During the marshaling operation in train yard, the car-to-car end impacts from coupling cause high car and lading acceleration;
- ✓ The car-body vibrations come from track irregularities and some extra forces, such as the wind, and so forth. Most of the damage is attributed to car-to-car end impacts in the marshaling yard, so more focus is given on it when working out the load support and load securement method. Railroad freight car impact tests are always carried out for checking if the method can ensure transportation safety and no damage to the ladings.

Draft gear is the most important component of a freight car during impact. Its performance is investigated by mechanics dynamics software .The draft gear's characteristic is analyzed under different impact speeds.

4.2 main parts of draft gear and coupler

Railway cars are made up into trains. They are connected end to end or (coupled). The devices used in the coupling, pulling, pushing, and uncoupling processes are the draft gear and coupler. The draft gear is connected to the car under frame and seated in the center sill pocket. The draft gear receives and reduces the shocks and pulling stresses before transmitting them to the car body. The coupler is attached to the draft gear.

The movement of railway equipment involves heavy pulling and pushing forces to move the weight of the cars as the train moves. This buffing and pulling action between each car and locomotive occurs each time the equipment moves [2]. To ensure the car(s) and locomotive(s) can accept this movement without causing damage to the equipment and/or lading, the equipment must have a system to absorb this punishment. The draft system is designed like a large shock absorber to accept the impact of car movement without resulting in equipment or lading damage [1]. With the use of a draft system trains hauling many tons of consumer products and over a mile in length is possible. This workbook is designed for the draft arrangements associated with freight cars. Locomotive draft systems are similar but are not covered in this booklet.

The main parts of draft gear and coupler

1. Coupler
2. 2. Draft gear casing
3. 3.Front lug
4. Rear lug
5. 5.Spring for draft gear
6. 6. Circular pin
7. Rectangular pin
8. Follower
9. Draft gear support U-channel
10. 10. yoke

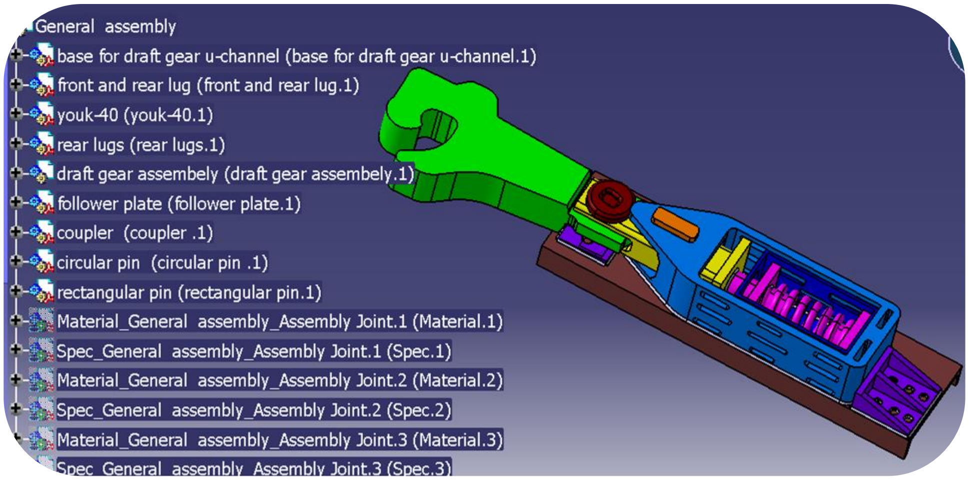


Figure –4.1.draft gears with coupler model

1. Follower Block or Plate

placed inside the coupler yoke at the front of the draft gear. It acts as a stop for the front of the draft gear in the pulling function. It transmits the stress of the draft gear force to the car center sill.

2. Coupler Yoke

It is the metal framework that houses the draft gear and the follower block or plate. It is connected or keyed to the shank of the coupler.

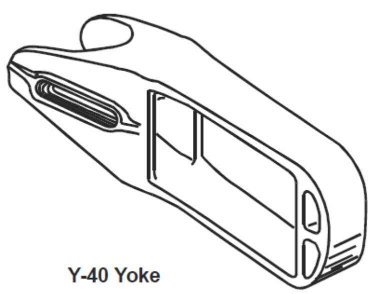


Figure-4.2.Coupler yoke [35]

3. Draft Gear Pocket

The opening in the center sill of the car designed to house the draft assembly. the yoke is inserted into the underside of the center sill at one end of the car. The draft gear and follower plate are allowed a specific size of pocket to allow restricted movement. The size of the draft gear pocket is restricted by the location of the front and rear lugs.

4. Draft Gear Lugs

The rear lugs are secured to the center sill at the rear of the draft gear pocket. In some cases they may be part of the center plate spider casting. The front lugs are generally part of the striker casting.

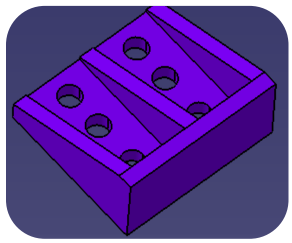


Figure-4.3. front and rear lug

5. Coupler

[35]A coupler does three things: connects one car with another, holds the connection, and then disconnects the two cars. Cars built before 1930s were equipped with what was known as the D coupler. Since then almost all freight cars have had the E coupler; some are being equipped recently with an F coupler and passenger cars have an H coupler. The working principles of the E, F, and H, however, are the same. Link and Pin. The simplest type of coupler Each vehicle has a bar attached to the center of the headstock Each coupler has a bell mouth around the end of the bar to assist in guiding the bar with the hole into place. The loops are lined up and a pin drop ped into them.

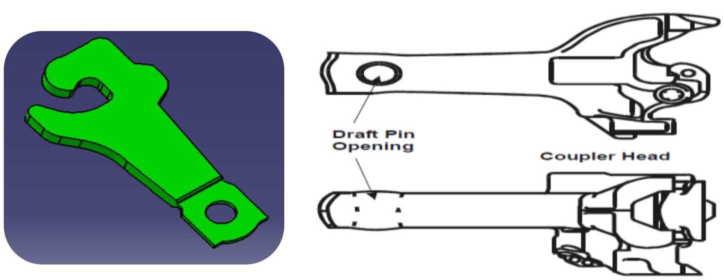


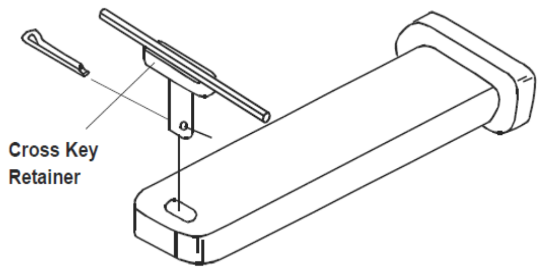
Figure –4.4.coupler [35]

6. Draft Gear Carrier or Carrier Iron

To support the weight of the draft gear and associated components, a metal plate is secured to the Underside of the center sill. It also allows free movement of the yoke as it moves in conjunction to the car movement.

7. Draft Key or Cross Key

The cross key (slang) is a large metal plate that is inserted through the slot in the coupler and yoke to tie the draft system to the coupler. The Draft/Cross Key is used with E-Type couplers and Y-40 yokes. On F-Type couplers and Y-45 Yokes, a large vertical pin is used to connect these parts.



Cross or Draft Key

Figure- 4.5.Draft Key or Cross Key [35]

8. Draft Gear

This is the heart of the draft system. It is a large heavy component that absorbs the energy of the Equipment coupling together and it also provides a rebound force that maintains slack between the cars. In this way the cars are moved without damage to other components or to the product within the cars. There are various types of draft gears designed for specific situations. The most Common heavy capacity draft gear is the Mark 50. This type of draft gear is suited to heavy applications such as for coal cars, open top hopper cars and other bulk commodity cars. In this workbook hydraulic draft gears are covered. These are specifically designed for cars where the lading requires extra protection against impact.

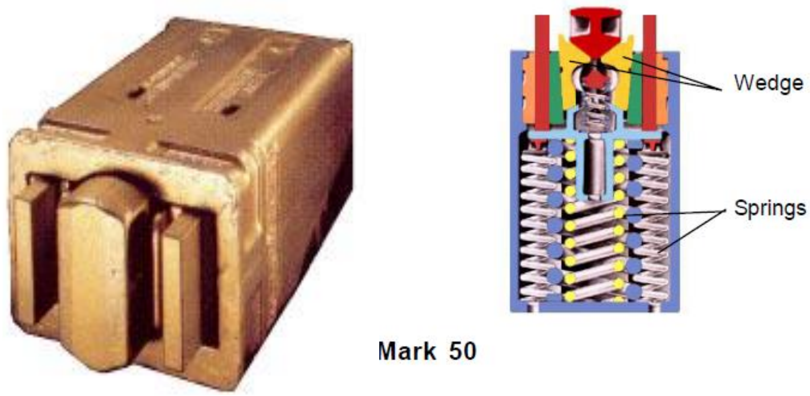


Figure-4.6. Cardwell Westinghouse Mark 50 Draft Gear [35]

9. Draft Key (Cross Key) Retainer

The draft/cross key must be secured to prevent it coming out unintentionally. The securing mechanism used must have a positive locking mechanism. In this diagram it is a pin inserted in the hole of the draft key with a cotter key and a bar across the top of the pin. The bar is heated and bent to wrap around the draft/cross key.

10. Striker Casting

The striker casting is a large metal casting secured to the inside end of the center sill. The casting has three main functions. First, the front lugs restrict the movement of the draft gear. It also acts as a support for the coupler shank. Third, it is the main component to absorb heavy impacts to the center sill. To do this, the striker casting is a heavy metal block that lines up with the horn of the coupler. When the car receives excessive pushing force exerted against it, the coupler will compress the draft gear. This allows the horn of the coupler to contact the face of the striker. If the gear becomes severely worn or receivers too much impact it may become

4.3 The mass of draft gear and coupler parts

These masses are not identical to all types of coupler and draft gear but they are average masses of the available coupler and draft gear.

The data below required for the Multi body model of the coupler and draft gear concerns the mass, inertia, initial positions and initial orientations of all bodies in the system. The type and location of all kinematic joints that connect the different bodies of the system must also model carefully. The force element characteristics and type of motions are identified for the correct modeling and simulation of the coupler and draft gear. The right springs and dampers must also be specified and located in the right place and their direction of force also identified. Because of the main concern here is the spring system all the forces and another effects on the spring must be identified and modeled correctly.

Table-4.1. mass of draft gear and coupler.

	Name	Mass(Kg)	I_{XX}	I_{YY}	I_{ZZ}
1	coupler	185.214	1.662kgxm2	7.909kgxm2	8.924kgxm2
2	base for draft gear	95.738	1.457kgxm2	13.534kgxm2	14.889kgxm2
3	circular pin	9.267	0.015kgxm2	0.034kgxm2	0.034kgxm2
4	follower plate	53.412	0.205kgxm2	1.304kgxm2	1.306kgxm2
5	spring with support	34.232	0.111kgxm2	0.712kgxm2	0.714kgxm2
6	front and rear lug	11.609	0.047kgxm2	0.06kgxm2	0.097kgxm2
7	youk-40	95.946	1.434kgxm2	7.58kgxm2	8.229kgxm2
8	Draft gear	23.373	0.239kgxm2	0.528kgxm2	0.576kgxm2
9	rectangular pin	8.538	0.015kgxm2	0.031kgxm2	0.044kgxm2

4.4 Overview of the Draft Gear Connection and Components

The movement of railway equipment involves heavy pulling and pushing forces to move the weight of the cars as the train moves. This buffing and pulling action between each car and locomotive occurs each time the equipment moves. To ensure the car(s) and locomotive(s) can accept this movement without causing damage to the equipment and/or lading, the equipment must have a system to absorb this punishment. The draft system is designed like a large shock absorber to accept the impact of car movement without resulting in equipment or lading damage. With the use of a draft system trains hauling many tons of consumer products and over a mile in length is possible.

4.4.1. Pushing Position

In the draft or pushing position, the car or locomotive is moved by pushing on the coupler. Having slack between the cars and in the draft system allows for movement of the equipment without having to move the train as a solid block. In the diagram below you will see the pushing force on the front of the coupler will cause the end of the coupler shank to make contact with the follower plate. The follower plate makes contact with the front of the draft gear. At the opposite end of the draft gear are draft stops or lugs attached to the center sill. They form the rear end of the draft pocket. With the follower plate forcing the front of the draft gear back, the rear of the draft gear cannot move any farther back because of the lugs. The result is the internal components of the draft gear are compressed to act as a shock absorber to the pushing movement. This is a typical E-Type Coupler arrangement. For other types of coupler and draft systems, the basic function remains the same. The only difference is the type and capacity of the draft gear.

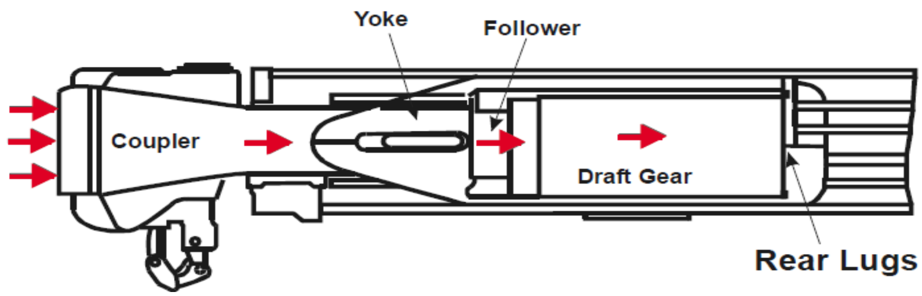


Figure-4.7. Draft Gear in Pushing Position [36]

4.4.2. Pulling Position

When moving a group of cars or a train each car is pulled by the coupler and the internal operation of the draft arrangement must work to allow the draft gear to work like a shock absorbers. The coupler is connected to the yoke by a cross or draft key. When the coupler is pulled forward, the draft key pulls the yoke forward. Inside the yoke is the draft gear and follower plate. With the yoke pulling the draft gear forward from the backend, the draft gear movement is restricted by contact the follower plate. The follower plate move forward slightly but is now restricted by the front draft lugs. The result is the draft gear now gets compressed again. Thus the draft gear is compressed in both the pushing and pulling action of the coupler

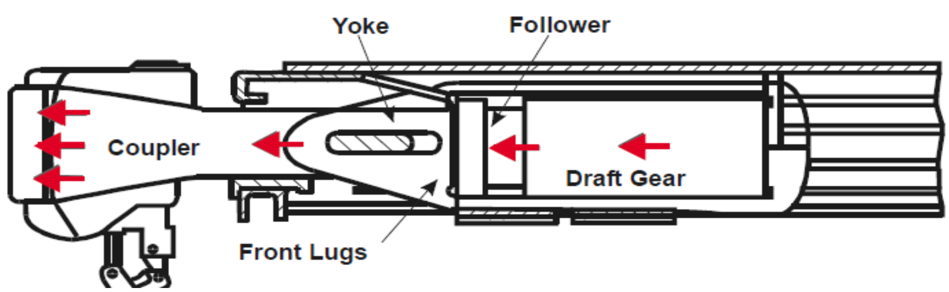


Figure-4.8. Draft Gear in Pulling Position [36]

4.5. Technical data

Overall coupler length is determined by measuring the distance from the center line of the yoke pin to the pulling face of the knuckle. Coupler shank length is determined by measuring the distance from the center line of the yoke pin to the back of the coupler head. The shank length is the important functional dimension to consider when choosing the correct coupler size to suit a particular wagon. The coupler butt and draw gear front follower must also match.

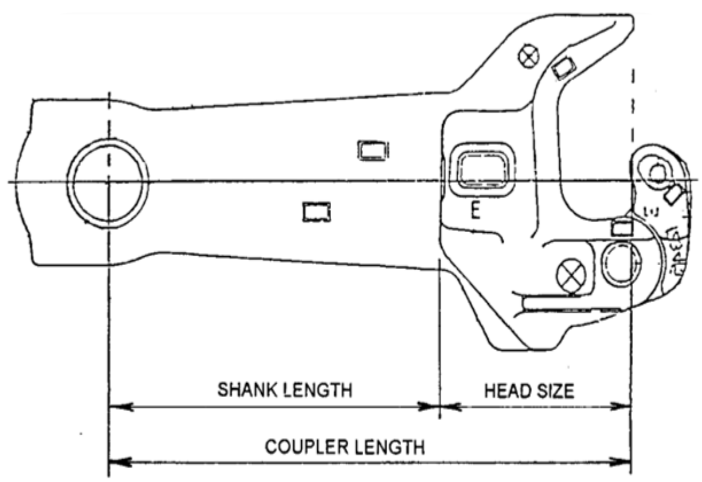


Figure-4.9. coupler standard dimension [12]

The technical data of the coupler and draft gear used for modeling and simulating are somehow different from one coupler and draft gear to another one due to the speed and condition criteria's of a user. For example; for high speed trains, low speed trains and the weather condition differences. The following data are used in this modeling and simulating conditions.

4.6. Definition of the design mass

- ✓ Design of the vehicle body in the working order, M1
- ✓ Design of one bogie or running gear, M2
- ✓ Normal designs pay load or carrying weight (cargo), M3

4.7. Superposition of static load case for the vehicle body

In order to demonstrate a satisfactory static strength as a minimum superposition.

- ✓ **Superposition case**

1. Compressive force and vertical load

$= g * (M1 + M3)$……………………………………………………………….5.1

2. Compressive force and minimum vertical load

$= g * (M1)$……………………………………………………………………..5.2

3. Tensile force and vertical load.

$= g * (M1 + M3)$……………………………………………………………..5.3

4. Tensile force and minimum vertical load.

$= g * (M1)$`……………………………………………………………..……5.4

4.8. Modeling of coupler with draft gear by CATIA

The coupler is a large metal casting used for the connection of cars and locomotives. There are many couplers identified in the AAR Rules, however, they are generally considered as Type E, F or E/F in design. The type of coupler required depends upon the type of car and its use and the draft gear is the heart of the draft system. It is a large heavy component that absorbs the energy of the equipment coupling together and it also provides a rebound force that maintains slack between the cars. In this way the cars are moved without damage to other components or to the product within the cars. There are various types of draft gears designed for specific situations.

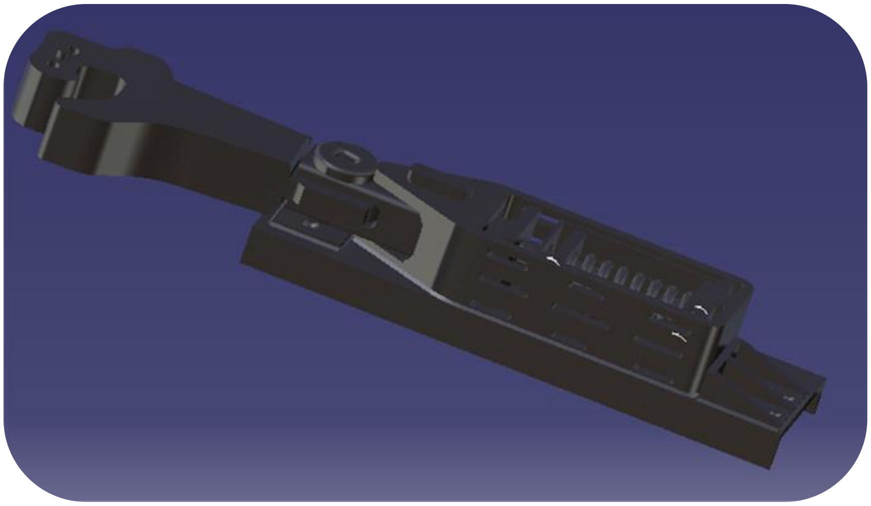


Figure-4.10.modeling of coupler with draft gear

4.9. Static analysis of coupler (by Finite element)

The finite elements method (FEM) is a solution based on an integral of a function, as shown in Spyrakos (1994).Basically, its solution is an approach made for polynomials defined in the discretization of the system; Therefore, the FEM is not an accurate method, but brings solutions of high precision when used correctly. The procedure of the FEM consists basically on structuring the system in a matricidal form, on applying boundary conditions and then solving the system. The matricidal system to be solved is:

$[K] * [U] = \{F\}$, Where……………………………………………………………………5.5

K-Rigid matrix

U-Displacement vector

F- The force vector

For the coupler analysis we need to know the forces and moments involved on the longitudinal direction. When an external force is applied into any direction, the internal forces will balance the external forces. These internal forces can present different magnitudes and directions, but they can be decomposed in parallels and perpendiculars components. As our purpose is to measure exclusively the longitudinal forces in the coupler, we have to minimize the moments 'effects over the instrumentation area. Making the free body diagram we can check the efforts and the reactions involved. If we have a perfect coupling, the force will be on the neutral line of the coupler, and a force in the restrain will balance the external force, which is the force of the other coupler. Therefore, in a real coupling we have two moments that can happen. One in the direction z, if the centerlines of the wagons are not aligned, and another in the direction y, due to the difference of height between the wagons. In Figure below), one can see the perfect coupling and a coupling with a moment in direction z. This two moments influence the distribution of the stress on the coupler, therefore the interest of the analysis of this case in the present study

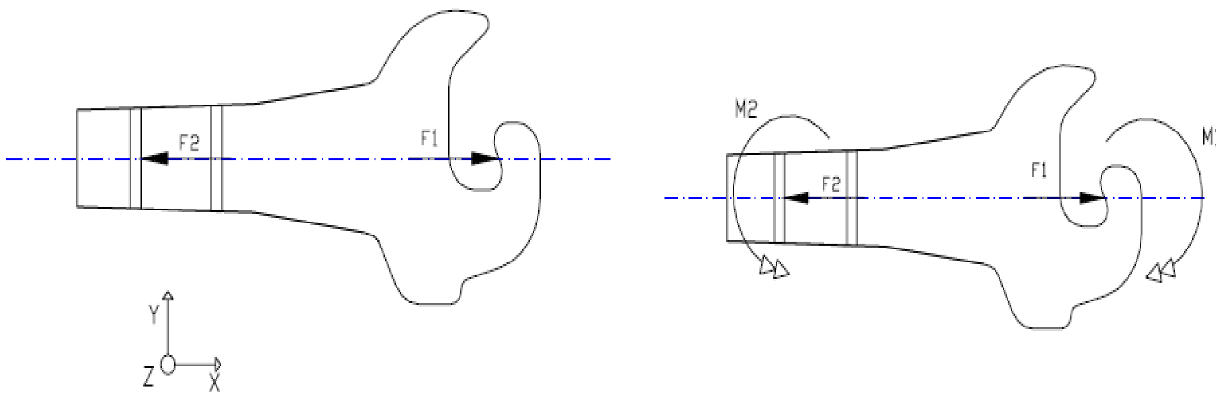


Figure-4.11. Forces involved on the coupling.[37]

The static balance would be

$\sum F = 0$..5.6

$F1 = F2$

$\sum M = 0$..5.7

$M1 = M2$

$M1 = F1 * d$..5.8

$\sigma_x = \frac{F}{A} + \frac{M*Y}{I}$ 5.9

Where

- ✓ d is the distance between the point of the force application and the neutral line;
- ✓ F1,F2,F are forces;
- ✓ M,M1,M2 are Moments; A is the transversal area; y is the distance between the point where we calculate the stress and the neutral line of the area;
- ✓ I- is the inertia of the transversal area; σx is the stress in x direction. The analysis is analogous for the moment in y direction.

4.9.1 Geometrical Model

In order to facilitate the finite elements analysis the real model of the coupler can be simplify. The coupler type F was used on this study because is one of the most used in railroads in world. Its basic dimension was taken of the American railway standard – AAR (1978).The most important details are those who are not symmetrical according to the neutral line, the others can be omitted. As the study is qualitative, the main importance is to maintenance the proportion between the dimensions, not the dimension itself. To make the FEM analysis it is necessary to make the geometrical model. This can be done using drawing software that is compatible with the FEM software. In this case, CATIA was used. The geometrical model it is shown in Figure below.

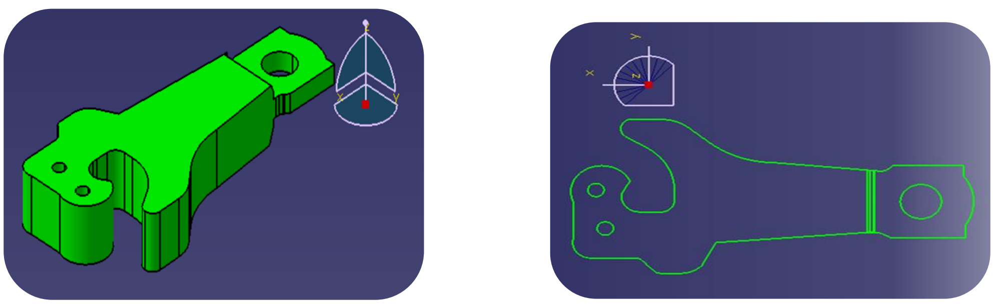


Figure-4.12. Geometrical model of the coupler

4.9.2 Effects and boundary conditions

For the traction load it was applied 1500KN normal to the face were the other coupler would couple as it can be seen in Figure below). The boundary condition is fixed, similar to the force that the pin would apply against the hole. Due to the fact that the hole is not a perfect cylinder we have a small area of contact. For the compression load it was applied -1500KN normal to the face of the contact with the other coupler, Figure below).The boundary condition is the same of the traction but in the opposite side.

NOTE-the numerical value of tensile force and compressive force is given from European standard - **EN 12663-2**

4.9.3. Input data for finite element Analysis

Young's Modulus	2.e+011 Pa	Thermal Expansion	1.2e-005 1/°C
Poisson's Ratio	0.3	Tensile Yield Strength	2.5e+008 Pa
Density	7870. kg/m³	Compressive Yield Strength	2.5e+008 Pa
Young's Modulus	2.e+011 Pa	Tensile Ultimate Strength	4.6e+008 Pa
Thermal Conductivity	60.5 W/m·°C	Relative Permeability	10000
Specific Heat	434. J/kg·°C	Resistivity	1.7e-007 Ohm

Table-4.2.Input data for finite element Analysis

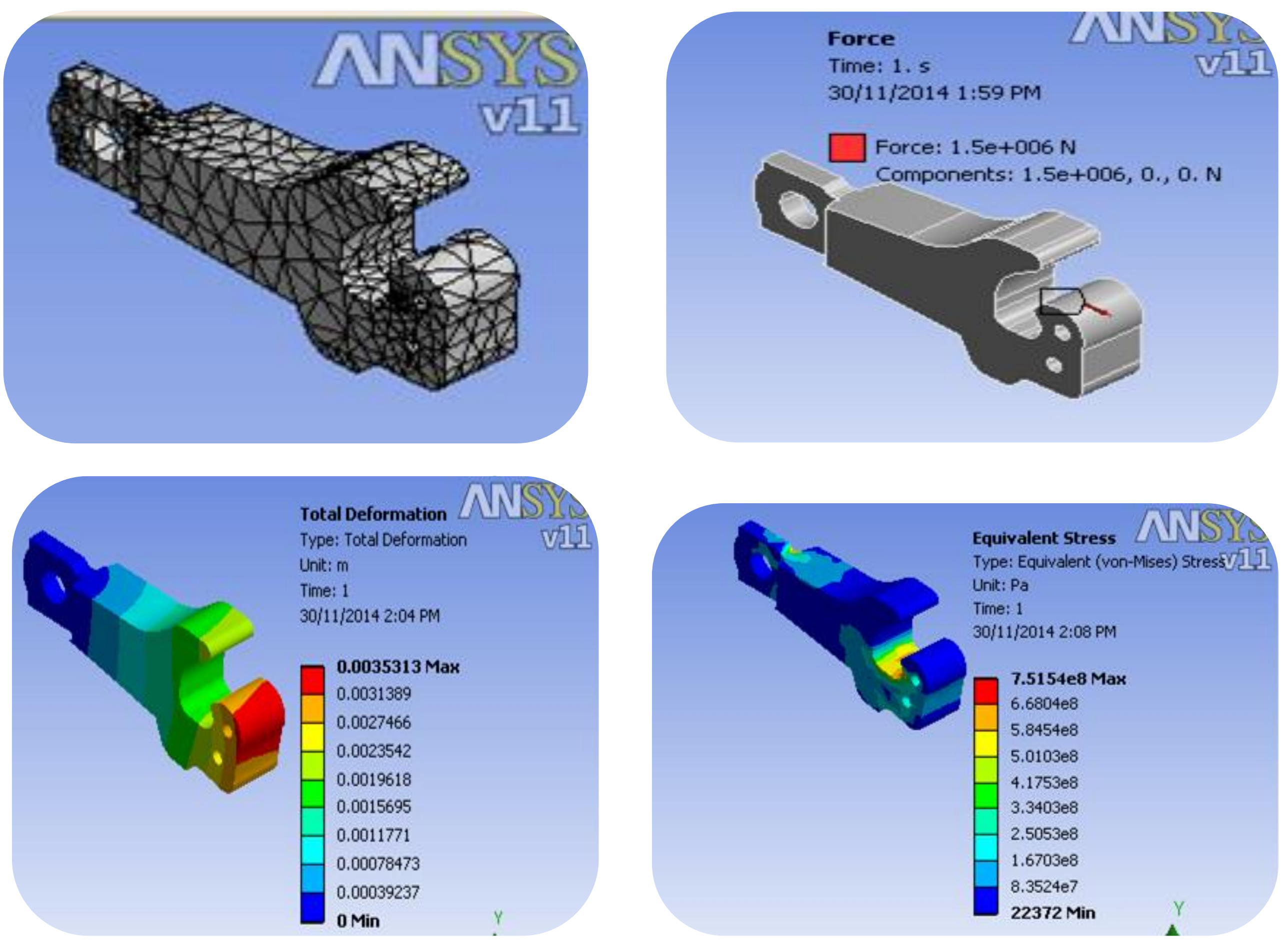


Figure -4.13. Mesh and boundary conditions for traction load (tensile load) for coupler

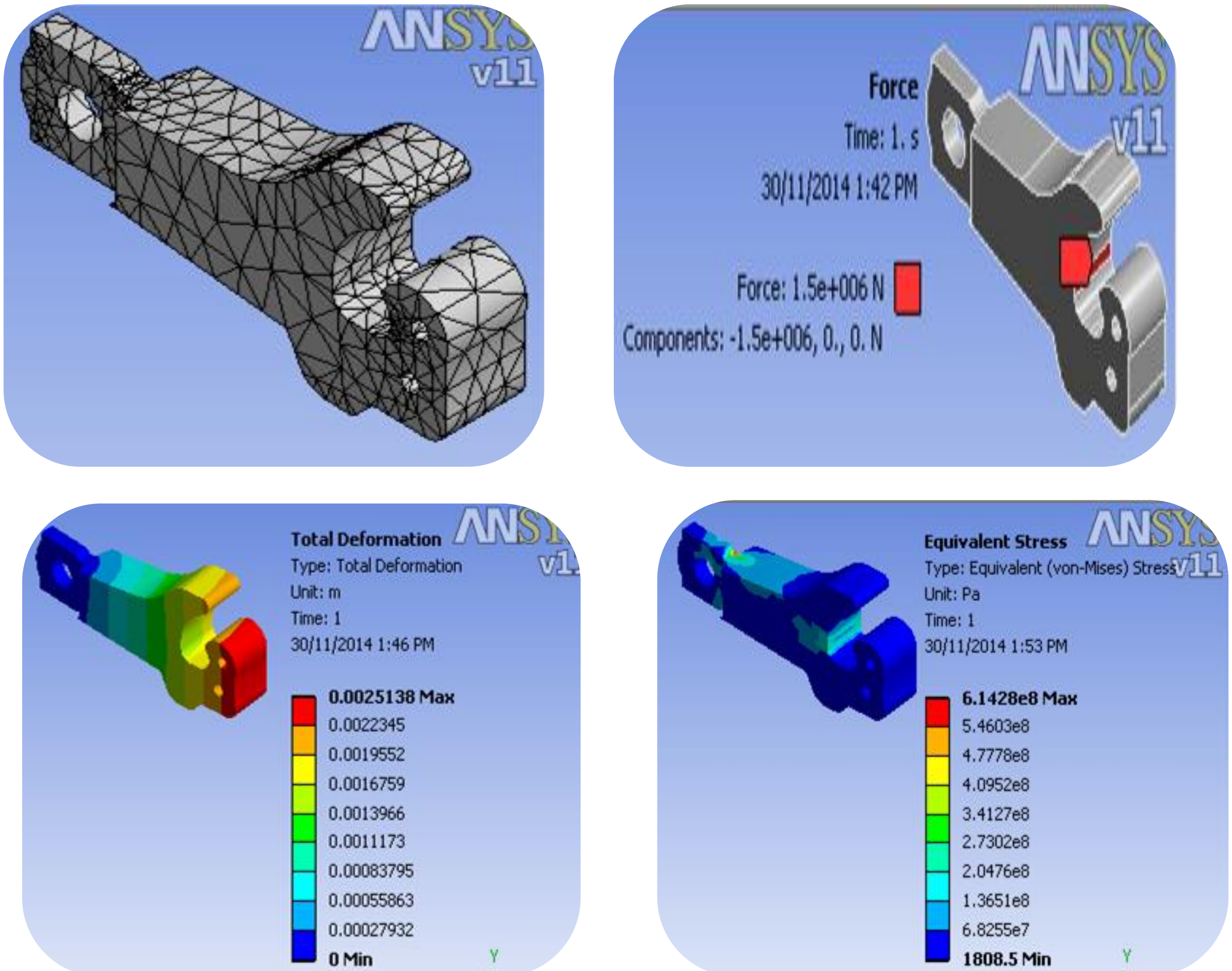
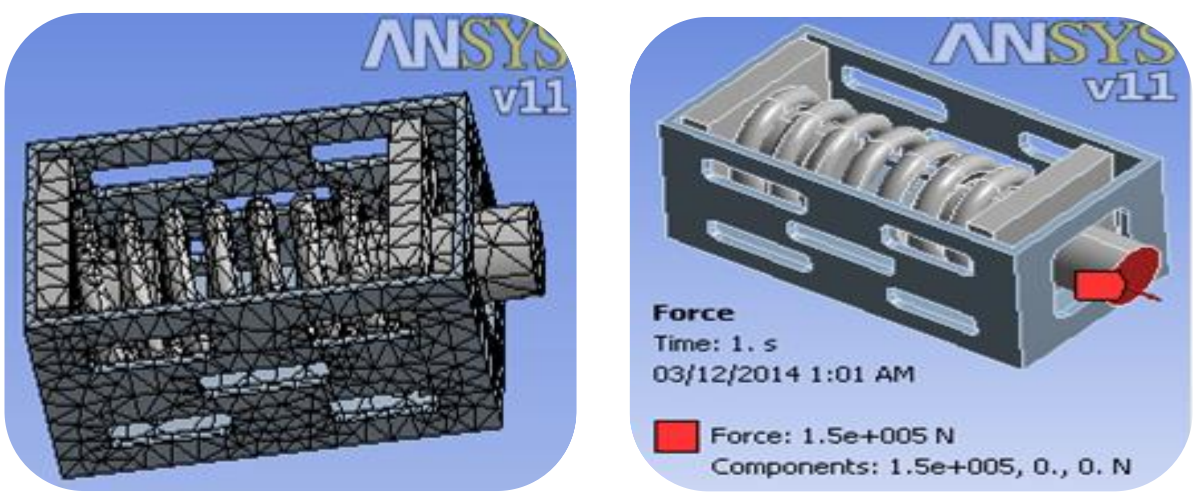


Figure -4.14. Mesh and boundary conditions for compressive load for coupler



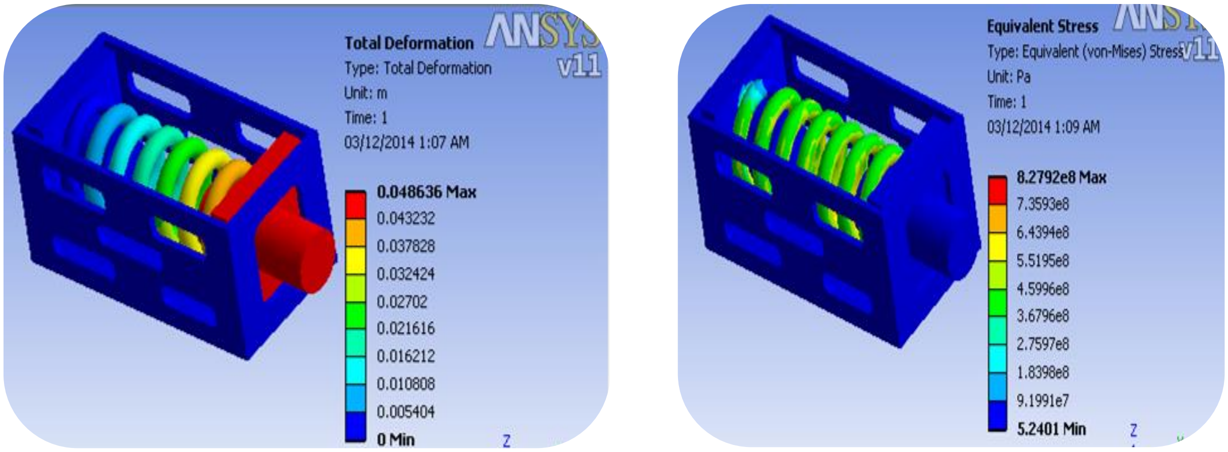


Figure -4.15.Mesh and boundary conditions for tensile load for draft gear

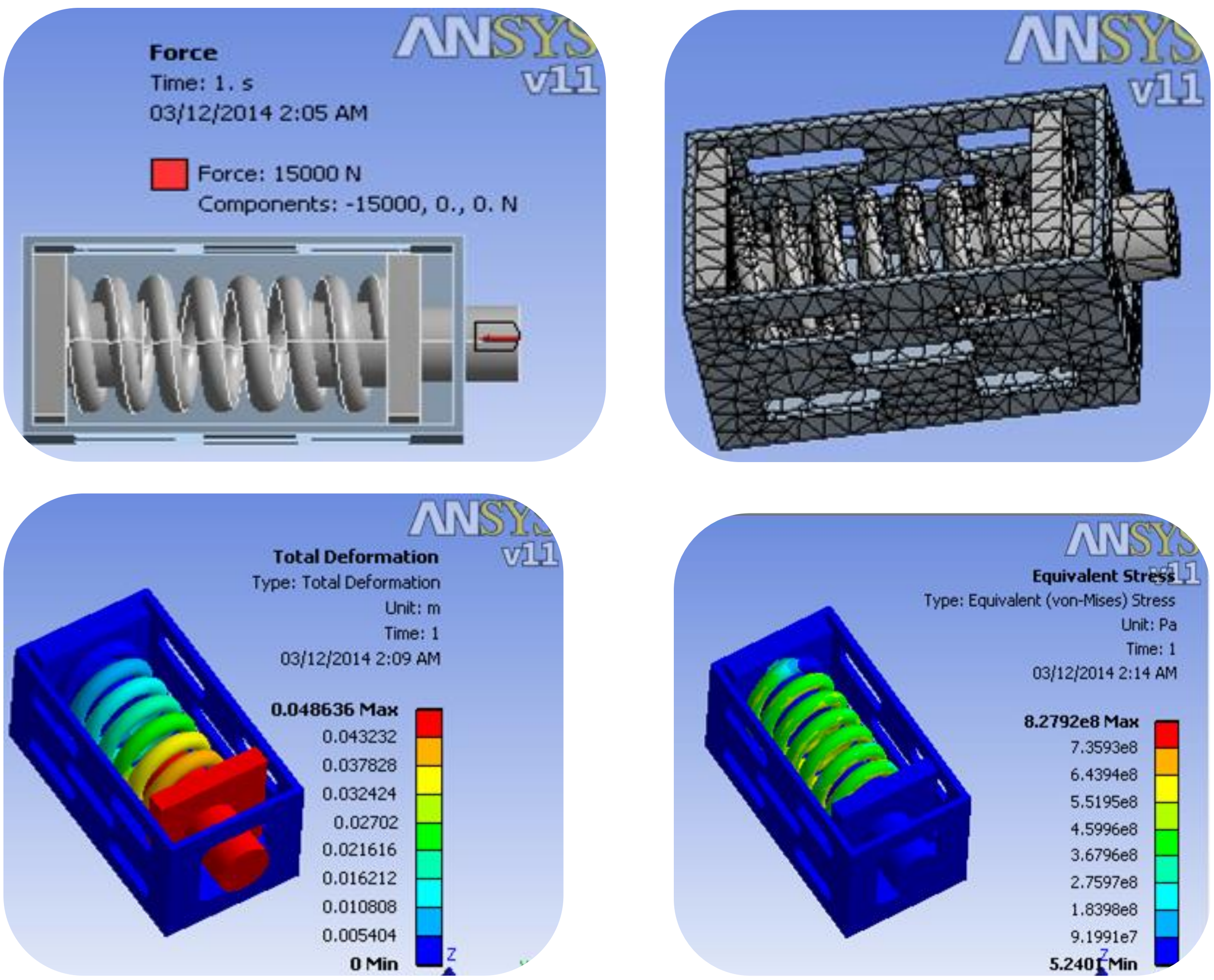
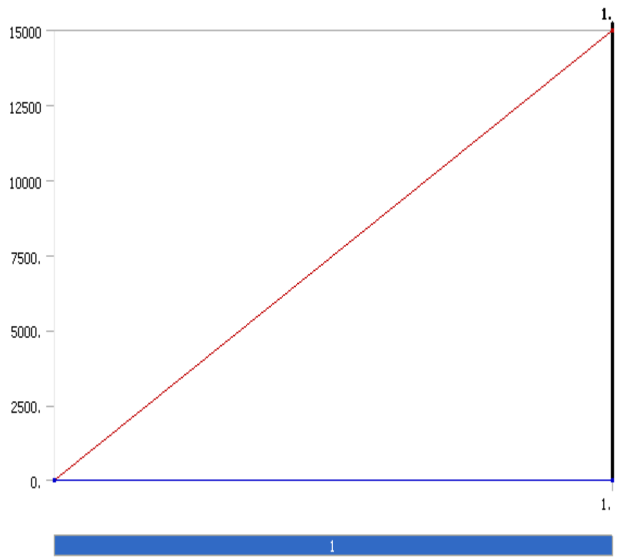


Figure -4.16.Mesh and boundary conditions for compressive load for draft gear

Compressive Force



Tensile Force

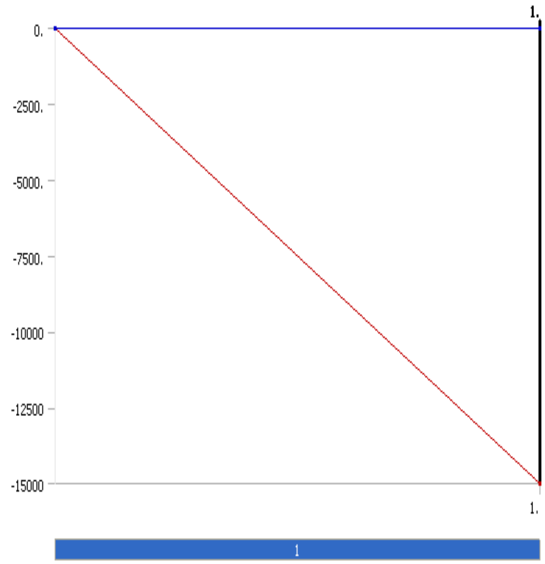


Figure -4.17. Model -Static Structural

Structural Steel > Alternating Stress

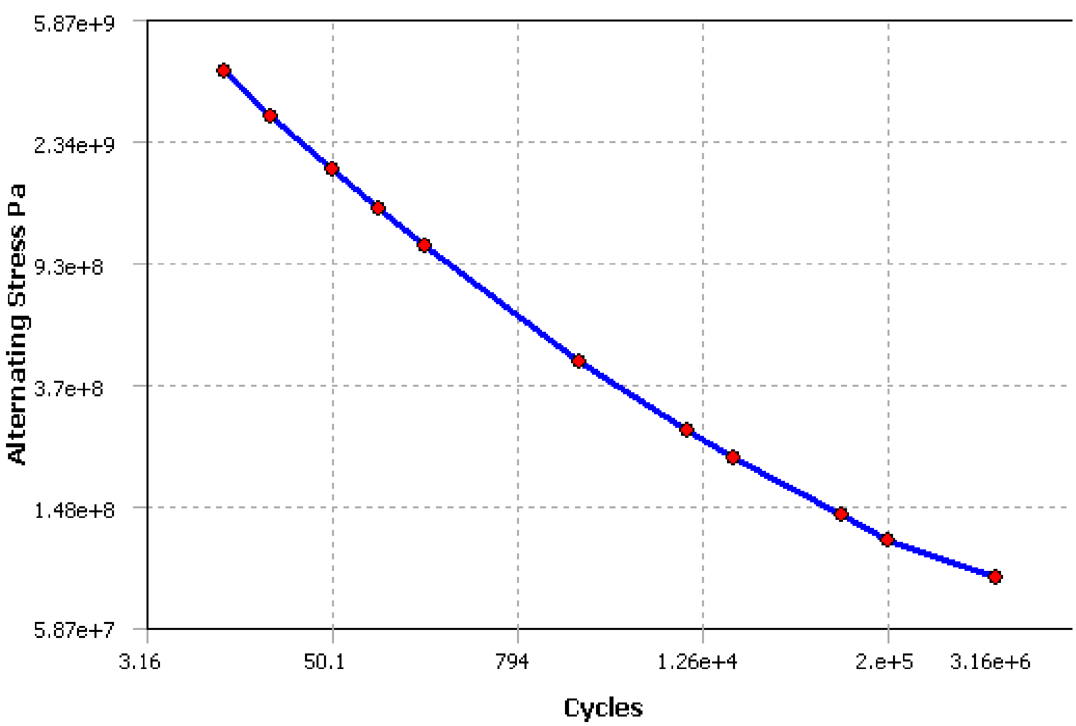


Figure-4.18. Structural cast iron > Alternating Stress > Alternating Stress vs. Cycles

Table -4.3, Alternating Stress vs. Cycles

Cycles	Alternating Stress Pa	Cycles	Alternating Stress Pa
10.	3.999e+009	2000.	4.41e+008
20.	2.827e+009	10000	2.62e+008
50.	1.896e+009	20000	2.14e+008
100.	1.413e+009	1.e+005	1.38e+008
200.	1.069e+009	2.e+005	1.14e+008
		1.e+006	8.62e+007

Structural Steel > Strain-Life Parameters

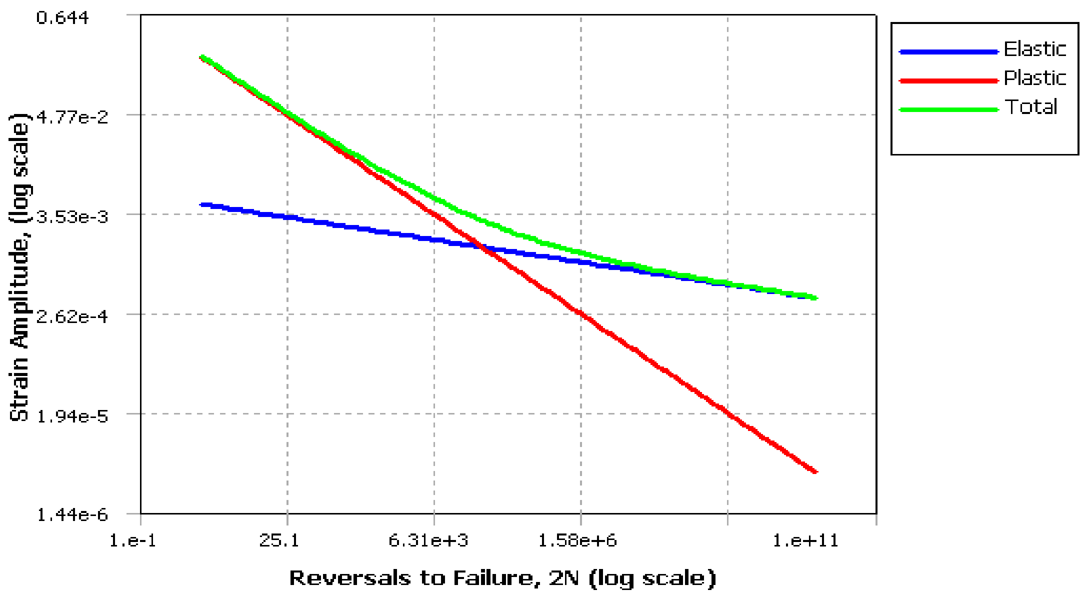


Figure-4.19. Structural Steel > Strain-Life Parameters

4.10. ANSYS result

The main objective of this structural Analysis at static conditions of coupler and draft gear is basically to determine the part which is easily affected during the impact load that created highly the compressive and tensile force due to these un necessary forces most part of the coupler and draft gear is damaged so according to the FEM (finite element method) or the ANSYS result in the coupler head highly deformation and maximum stress is happened and also in the draft gear the maximum stress and deformation is happened at the front part of the spring and the part which is connected the draft gear with coupler is known as **follower** also highly damaged.

Finally according to the above result it is important to

- Understand the material which is critically needs maintenance.
- To reduce the maintenance and material cost.
- To minimize rail derailment during operations.

CHAPTER 5

MULTI-BODY MODEL OF THE RAIL COUPLER WITH DRAFT GEAR SIMULATION USING SIM-PACK

5.1 Introduction

Multi-body simulation, known also as multi-body system simulation, is used to predict and optimize the behavior of any type of multi-body system by solving the equations of motion. The name multi-body can refer to a wide range of systems including machinery, vehicles, robotics, etc. Usually the bodies of a multi-body system are linked by means of joints which allow certain relative motions and restrict others. The bodies themselves can be rigid or flexible. The movements within these degrees of freedom are defined by the force and torque elements using an arbitrary force or torque law. The system is completed by excitation elements and sensors for measuring the desired outputs. In describing the kinematic behavior, the motion or the position of the multi-body system is studied with respect to the kinematic joints. Dynamic problems describe the motion of the system due to the applied forces and the inertia characteristics of the bodies, i.e. their mass, moments of inertia and position of the center of gravity.

5.2. Train models

The accurate simulation for modeling of draft gear with coupler is necessary trains, especially for heavy haul train. The longitudinal behavior of trains is a function of train control inputs from the locomotive, train brake inputs, track topography, truck curvature, rolling-stock, bogie characteristics and wagon connection characteristics. This thesis will focus on the wagon connection characteristics which affect the longitudinal behavior of the train. A wagon connection characteristic is one of the causes which influence the longitudinal dynamic behavior of a train. This behavior can be described by a system of differential equations. The differential equations can be formulated by considering the three mass trains as showed in Figure below .It will be found that the in-train vehicle, whether locomotive or wagon, can be classified as one of the only three connection configurations, leading, in-train, and tail. For the purpose of modeling, the wagons assumed having no lateral or vertical movement. For all vehicles have retardation and grade forces, for powered vehicles traction and dynamic brake forces are added

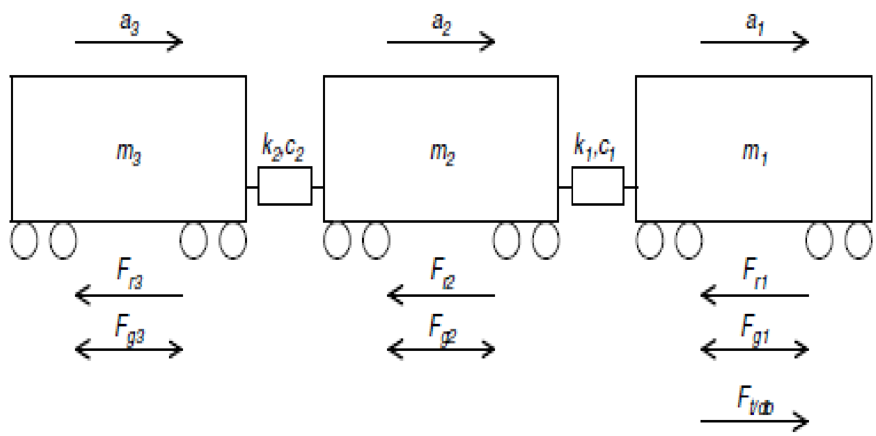


Figure-5.1. Three mass train model [31]

5.3. Locomotive connection models

The most common wagon and locomotive connection component is the coupler with friction type draft gears. This connection component is important for any longitudinal train dynamics modeling and simulation [9, 11, and 12]. The most challenges for modeling and simulation is due to the nonlinearities of air gap (or coupler slack), draft gear spring characteristic, (polymer or steel), and stick – slip friction provided by a wedge system. A schematic auto coupler and draft gear unit are illustrated in Figure bellow.

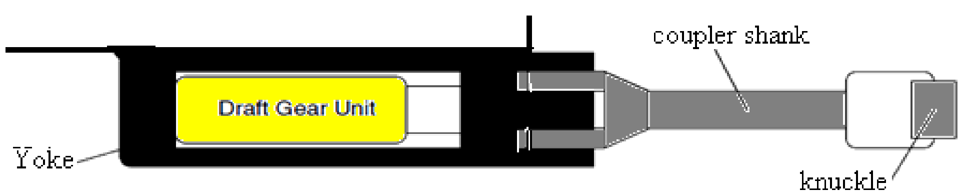


Figure – 5.2.Draft gear spring characteristic [9]

When considering a wagon connection, the two coupler assemblies must be considered along with gap elements, and also stiffness elements describing bend in the wagon body. A wagon connection model will therefore seem as something similar to the schematic in Figure above Modeling the coupler slack is straightforward, a simple dead zone. Modeling of the steel components including wagon body stiffness can be provided by a single linear stiffness.

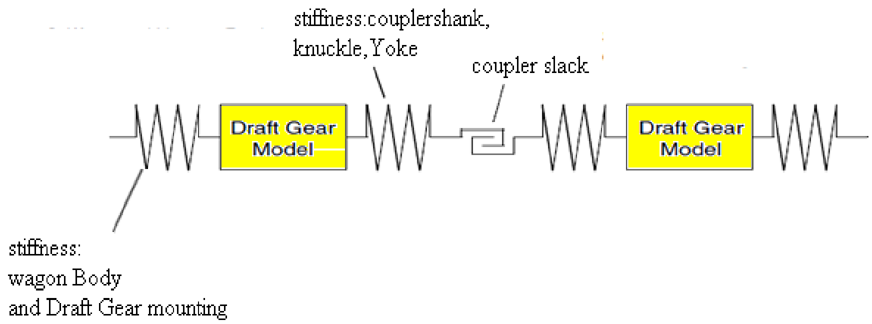


Figure – 5.3.Components in a locomotive connection model.[9]

For the purpose of modeling, the locomotive connection can be considered as three elements. These are a dead zone for coupler slack, a linear spring modeling the stiffness of all steel and/or aluminum components in the connection and a model equivalent to the two draft gears, as showed in Figure below

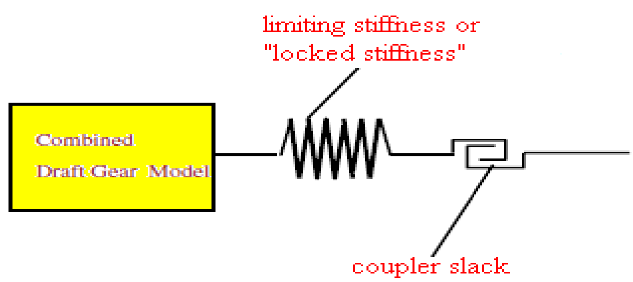


Figure-5.4. draft gear model [9]

5.4 Modeling of Friction Type Draft Gear

Wagons are linked by a coupler-yoke-draft gear mechanism that serves to relieve a part of the load found in train action and switchyard impacts. A good understanding of draft gears is necessary to the study of longitudinal train dynamics and impact mechanism in train dynamics of draft gears. Therefore acceptable numerical model of draft-gear force-deflection characteristics is essential to provide a basis for design, manufacturer and evaluation of draft

Gears, before engineers can adequately predict the coupler loads in impact situations. In modeling of the draft gear, the coupler is considered to be a part of the system. [9] A conventional wedge arrangement of the draft gear unit is shown in Figure

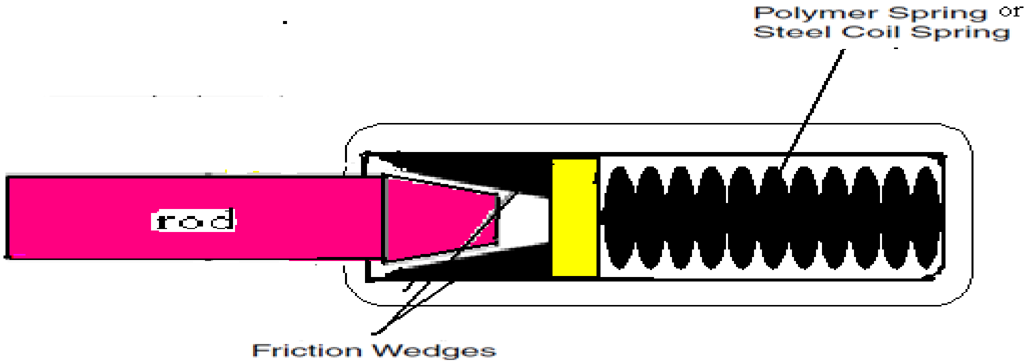


Figure – 5.5.Friction type draft gear Unit.[9]

5.4.1 Mathematical model of draft gear

The draft gear package can be considered as a single wedge spring system as shown in fig below. The rollers provided on one side of the compression rod can be justified in that the multiple wedges are arranged symmetrically around the outside of the rod in the actual unit. It will be realised that different equilibrium states are possible depending the direction of motion, wedge angles, and surface conditions. The free body diagram for increasing load (i.e., compressing) as shown in Figure below. The state of the friction m1N1 on the sloping surface can be any value between ^m_1N_1: The fully saturated cases of m_1N_1 are drawn on the diagram. If there is sliding action in the direction for compression, then only the Case $_1$ friction component applies. Case $_2$ applies if a prejammed state exists. In this case, the rod is held in by the jamming action of the wedge. If the equations are examined, it can be seen that for certain wedge angles and coefficients of friction, wedges are self-locking. Examining the rod:

Case 1 : $F_c = N_1(\sin\phi + \mu_1 \cos|\phi)$

Case 2 : $F_c = N_1(\sin\phi - \mu_1 \cos\phi)$

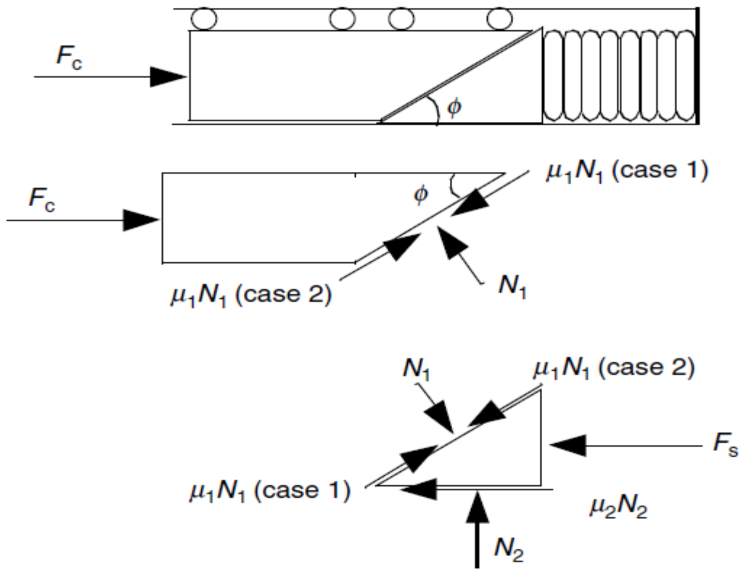


Figure – 5.6.free body diagram of a simplified draft gear rod – wedge – spring system.

5.5. Coupler

Is a component that connects two railcars together Coupler designs are important part in the wheel rail industry. The coupling couplers have clearances in the coupler knuckles, and the summations of these clearances are called coupler slack. The coupler slack is important for coupling couplers and coupler/train relative motion in the train negotiation, but it develops impact forces during train operations [32]. If the braking is applied excessively, coupler can and led a train derailment, [31]. When the railcars are subjected to compressive forces, they are in the buff load condition. In contrast, when they are subjected to tensile forces, they are in the draft load condition [28]. Longitudinal train dynamics are important in investigating ride comfort, rolling stock design, coupler design, and braking control design [31].second model, the shortcoming is that couplers' effects are the input characteristics which cannot reflect the actual behaviors of couplers during the train simulations. The most detailed coupler system models consider coupler as a rigid body which has the rotation DOF (Degree of Freedom), these kinds of models are used to research for real time behaviors of coupler systems and their effects. Luo Shihui and Ma Weihua [37] constructed detailed coupler models in which the coupler structure and rotation limit were considered. Based on this model, the coupler rotational behaviors and locomotive safety problems on the braking condition have been studied. Xu Ziqiang and Wu Qing [38] improved the Luo's model which contained the hysteretic draft gear model and friction-pair coupler, they analyzed the running behaviors of the couplers on different conditions. This chapter shows simulation results of the train and typical coupler 13A/QKX-100 models together. The dynamic models of train and type 13A/QKX-100 coupler systems has been developed and applied to the train model. The analyses include coupler angle and force on the curved and tangent track.

5.5.1. Coupler rational behaviour and its effects on heavy haul train

Coupler and draft gear packages are important components of heavy haul trains because their be heavier directly affects the safety of these trains. Due to the high cost of running tests, most researchers have adopted computer simulations and developed mathematical models of coupler and draft gear packages to carry out numerical studies on these systems. The existing draft gear mathematical models can be categorized into three types: linear, piecewise linear and nonlinear hysteretic. The linear model simplifies the draft gear into linear spring and damper system,[1–3] which greatly improves the simulation efficiency. However, because of the conspicuous differences between an actual draft gear and its simplified model when using linear characteristics, this model has rarely been used in research. Some piecewise linear models have been applied to European side buffers.[4,5] To date, however, the most accurate draft gear model is the nonlinear hysteretic model, which was proposed by Duncan[6] in 1989. Duncan's nonlinear hysteretic model included a coupler slack, approximated Draft gear impedance characteristics, and rigid impact characteristics applied after the draft gear loses its effectiveness.

5.5.2. Coupler model

5.5.2.1 Coupling surface characteristics

Coupler knuckles were not considered in previous coupler studies. Scholars generally simplified the coupler as a straight bar, and the relative motions of the two couplers were neglected. In this study, the coupling surface between the two coupler knuckles is built using the Coulomb friction model. The coupling surface has rotation and translation motions in the direction of the *z*-axis. The coupler body can rotate around the *y*- and *z*-axes and translate in the *x*-direction. The followers have longitudinal translation freedom. Figure 9 presents the structure of the coupler and draft gear package model. The angle between the car body and coupler's centerline is called the coupler rotation angle. When the coupler shoulder contacts the plunger casting, the angle between the two is called the maximum Coupler free angle.

5.5.2.2. Coupler shoulder alignment control characteristic

A Type 102 coupler, commonly used in Chinese heavy haul locomotives, has a shoulder alignment control function (shown in Figure 9(c)-A). When the coupler rotation angle reaches the maximum coupler free angle, the plunger casting is in contact with the coupler shoulder. The counterforce of the draft gear provides the aligning moment and constrains further rotation of the coupler by balancing the overturning moment, which is generated from the longitudinal

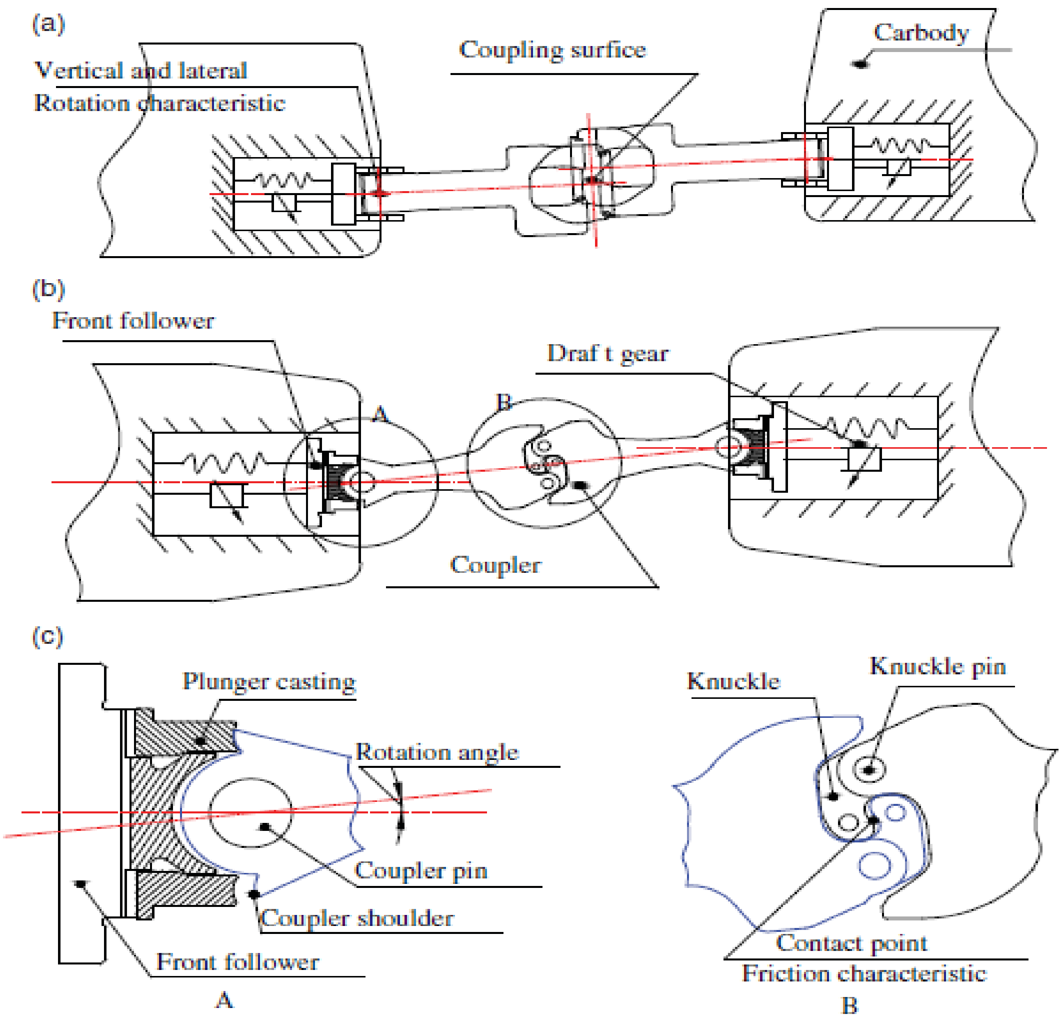


Figure -5.7.Structure of the coupler and draft gear models. (a) Side view, (b) vertical view and (c) detailed view. Compressive force. [8]

5.5.2.3. Coupling surface friction model

Before the coupler reaches the profile limit, the coupling surface generates a friction force and constrains the relative motion between the two coupler knuckles. The friction forces between the couplers consist of the rotation friction force on the horizontal plane and the transition friction force on the vertical plane.

5.6. Profile limit of the coupling surface

The coupler head profile can constrain the relative motion between the two coupler knuckles. Therefore, nonlinear stop force elements must be constructed. According to the structure of the coupler profile, the maximum horizontal angle is 10.7◦ and the maximum vertical angle is3.7◦.

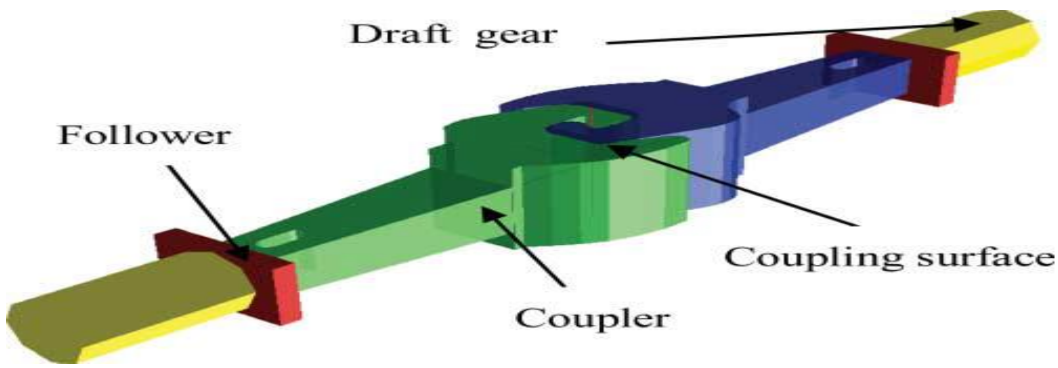


Figure-5.8.dynamic model of coupler

5.7. Multi locomotive dynamic simulation models.

In order to verify the theoretical analysis and accurately find the relationship between a coupler's bearing capacity and a locomotive's secondary suspension parameters, the multi-body dynamics software, SIMPACK, was used to build the locomotive and the detailed coupler simulate the processes of emergency braking and lateral coupler over turning forces with appropriate boundary conditions. The main impact factors of the coupler's bearing capacity and the effect of the suspension parameters of the locomotive on the wheel set lateral forces had been analysed.

5.8. The locomotive dynamics model

This locomotive model was established according to an actual heavy haul locomotive in China, as shown in Table below. The model consists of four locomotives and three couplers as shown in Figure bellow. The longitudinal forces are applied to both ends of the locomotives separately and the braking torques is applied to each wheel set to simulate the longitudinal impulse caused by emergency braking. These studies focus on the middle two locomotives.

Table-5.1. The main parameters of locomotive.

S/N	Name of body	Mass(kg)	Ixx(Ns/m)	Iyy(Ns/m)	Izz(Ns/m)
1	Mass - car body	62970	275721	14337364	1221057
2	Mass-wheel set	5396	2483	1081	2955
3	Mass - Bogie	7724	5390	13110	16778
	Primary spring Stiffness	c-x 164000000000	c-y 595000000	c-z 156900000	
4	Secondary spring Stiffness	c-x 130500	c-y 130500	c-z 535000	
5	Coupler length	1.588m			
6	Length of straight track	300m			
7	Length of transition track	120m			
8	Radius's of curvature	800m			
9	Total track length	3000m			

5.9. Multi-body Model Simulation using SIM-PACK

Multi-body simulation, known also as multi-body system simulation, is used to predict and optimize the behavior of any type of multi-body system by solving the equations of motion. The name multi-body can refer to a wide range of systems including machinery, vehicles, robotics, etc. Usually the bodies of a multi-body system are linked by means of joints which allow certain relative motions and restrict others. The bodies themselves can be rigid or flexible. The movements within these degrees of freedom are defined by the force and torque elements using an arbitrary force or torque law. The system is completed by excitation elements and sensors for measuring the desired outputs. In describing the kinematic behavior, the motion or the position of the multi-body system is studied with respect to the kinematic joints. Dynamic problems describe the motion of the system due to the applied forces and the inertia characteristics of the bodies, i.e. their mass, moments of inertia and position of the center of gravity.

The procedure of the simulation analysis study in the SIMPACK software is shown in Figure -below.

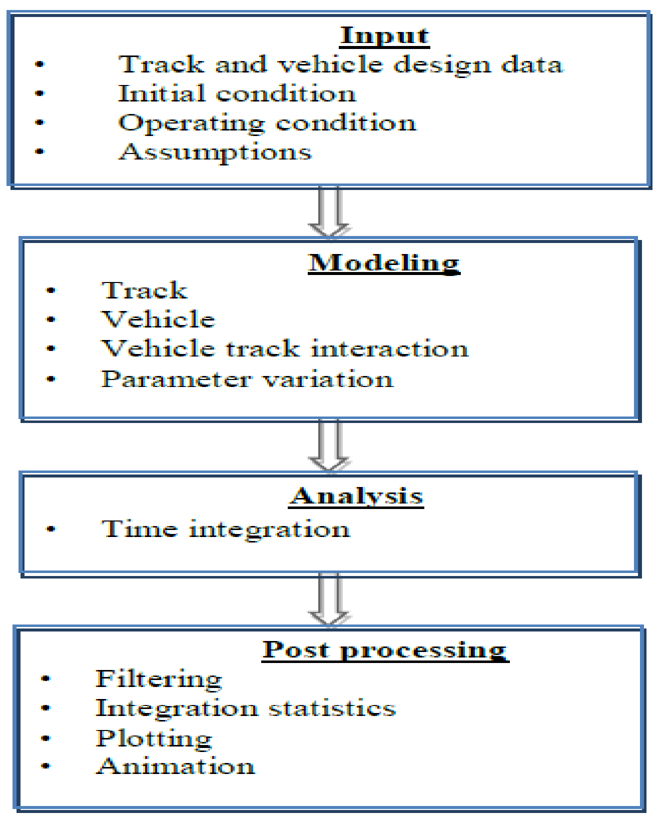


Figure-5.9.Flow chart of the simulation procedure

Modeling coupler with draft gear by SIMPACK

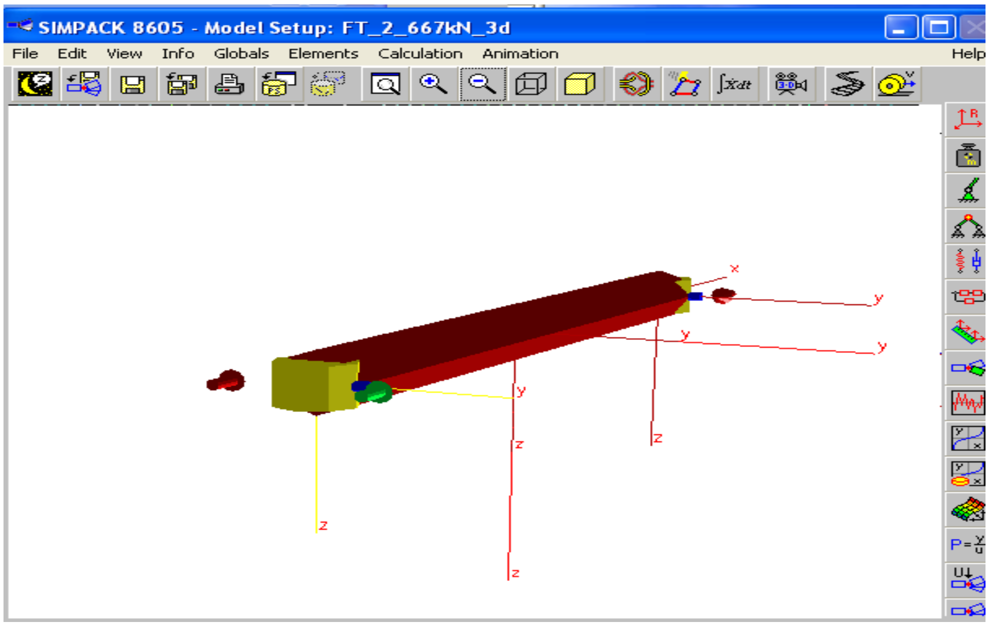


Figure-5.10. Modeling coupler with draft gear by SIMPACK

As shown in the above coupler with draft gear modelling we have to consider basically two trains contains one coupler, two draft gear.so according to the model Structural Steel > Strain-Life Parameters.The red color part body indicates the coupler and the yellow color part body indicates the draft gear.

Coupler with draft gear Part modeling

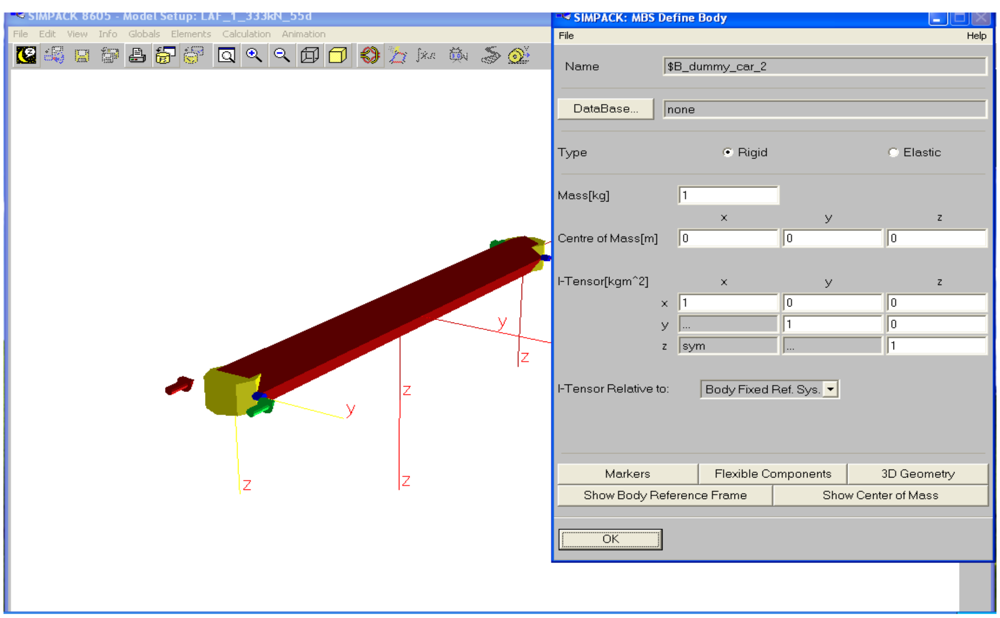


Figure-5.11. Coupler body model by SIMPACK

Coupler with draft joint elements

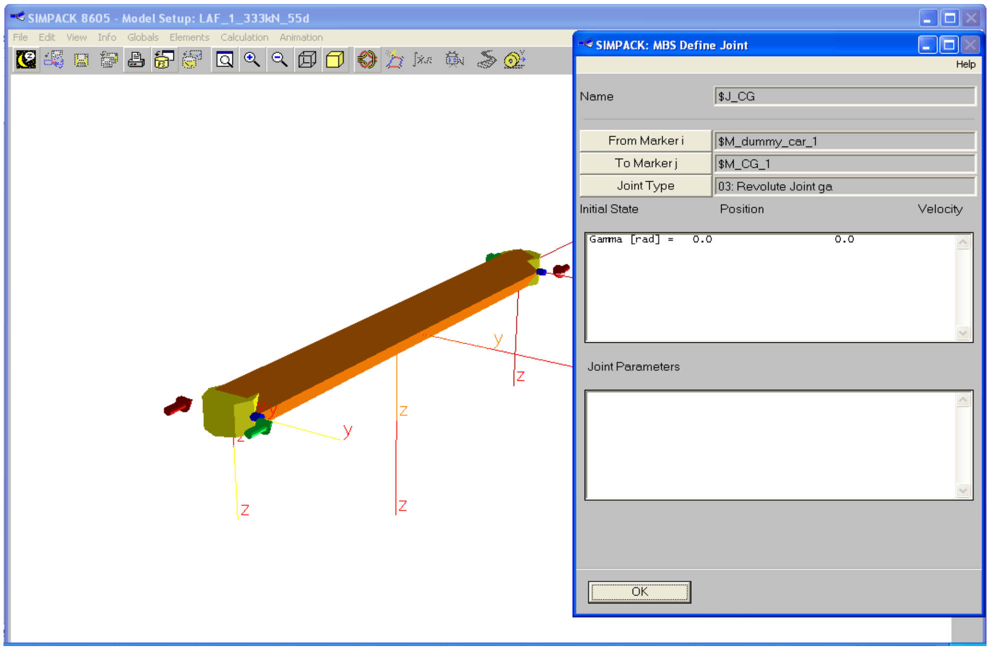


Figure -5.12. Coupler with draft joint elements

Coupler with draft joint elements

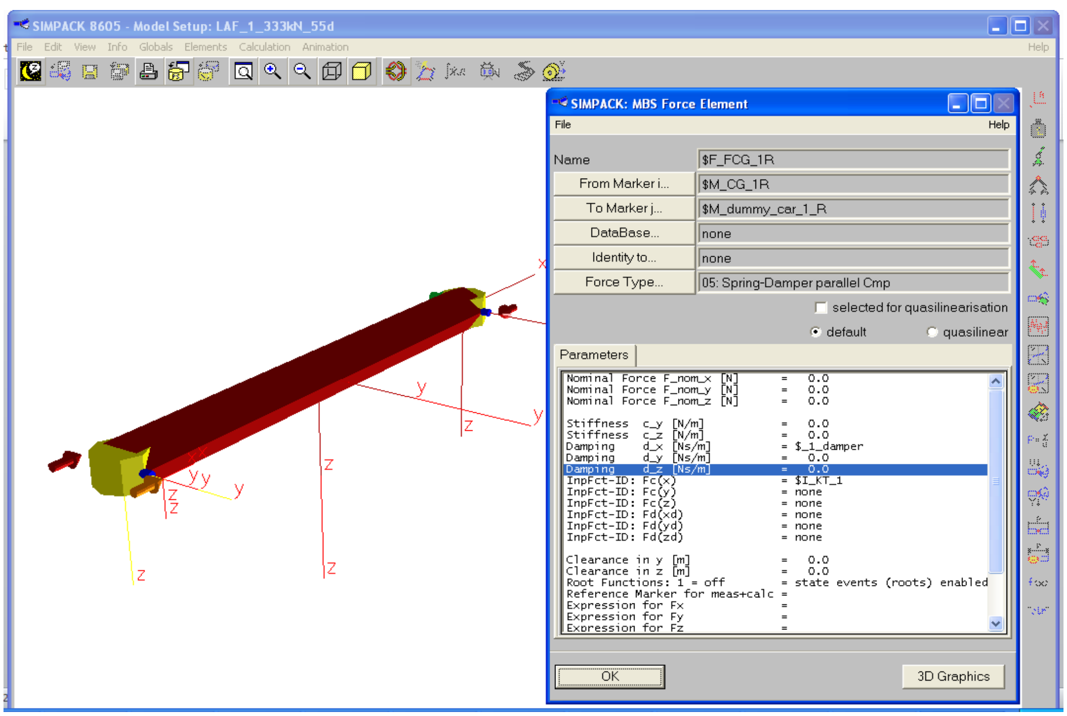


Figure -5.13. Coupler with draft force elements

Modeling Train by SIMPACK

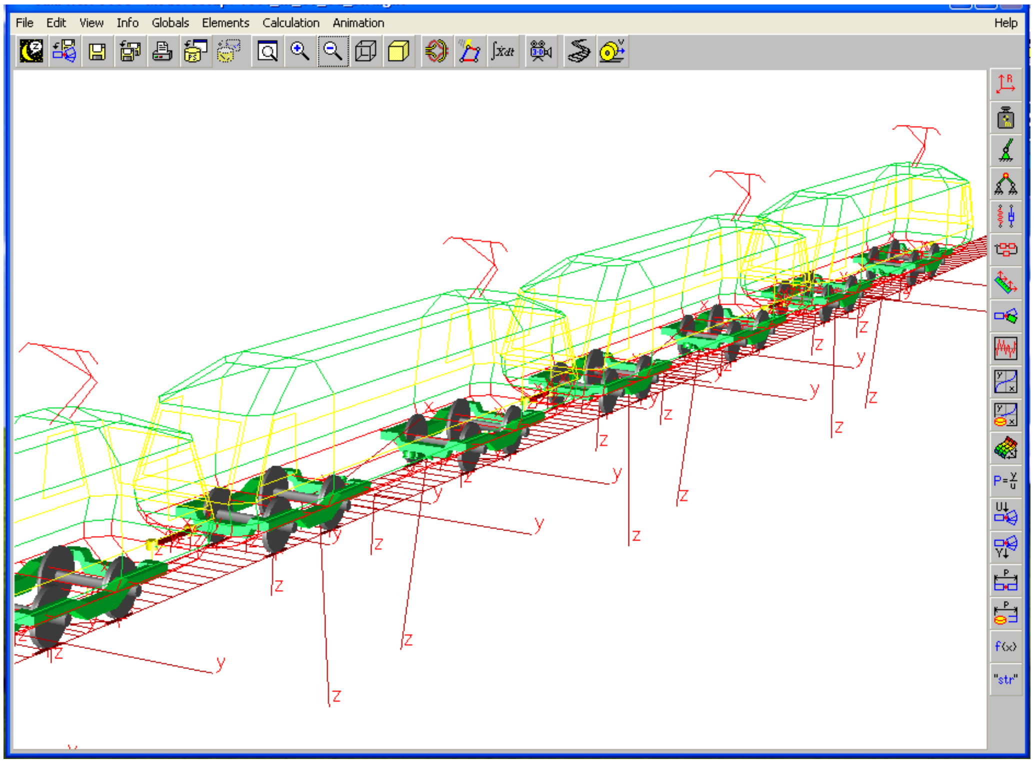


Figure-5.14. Modeling train by SIMPACK

Train body elements

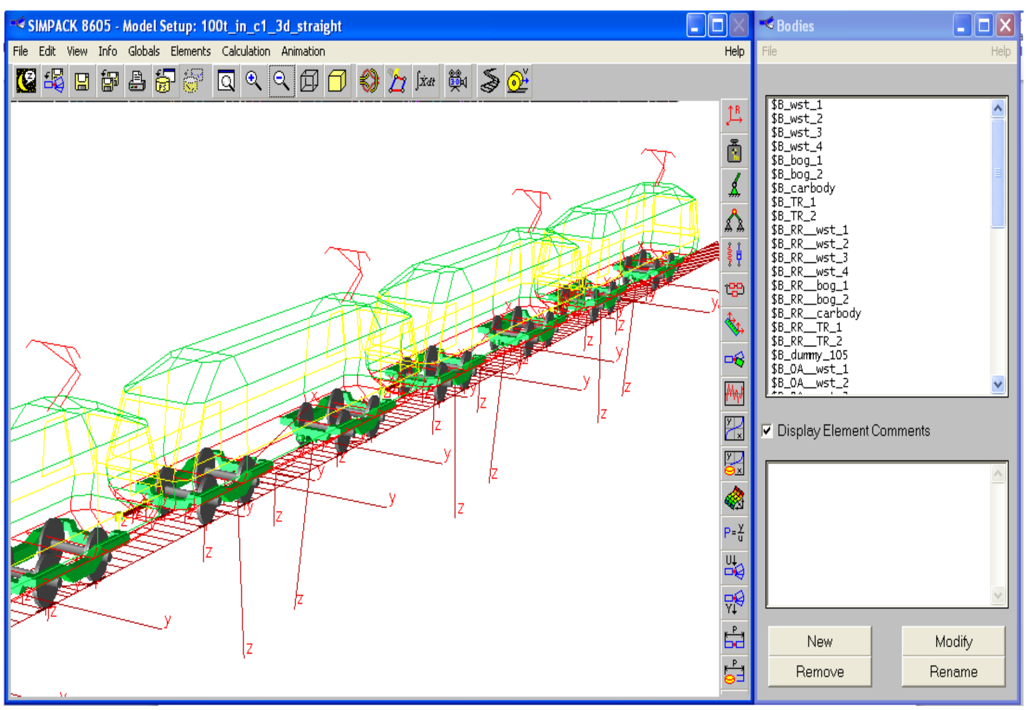


Figure-5.15. Train body elements

Train joint elements

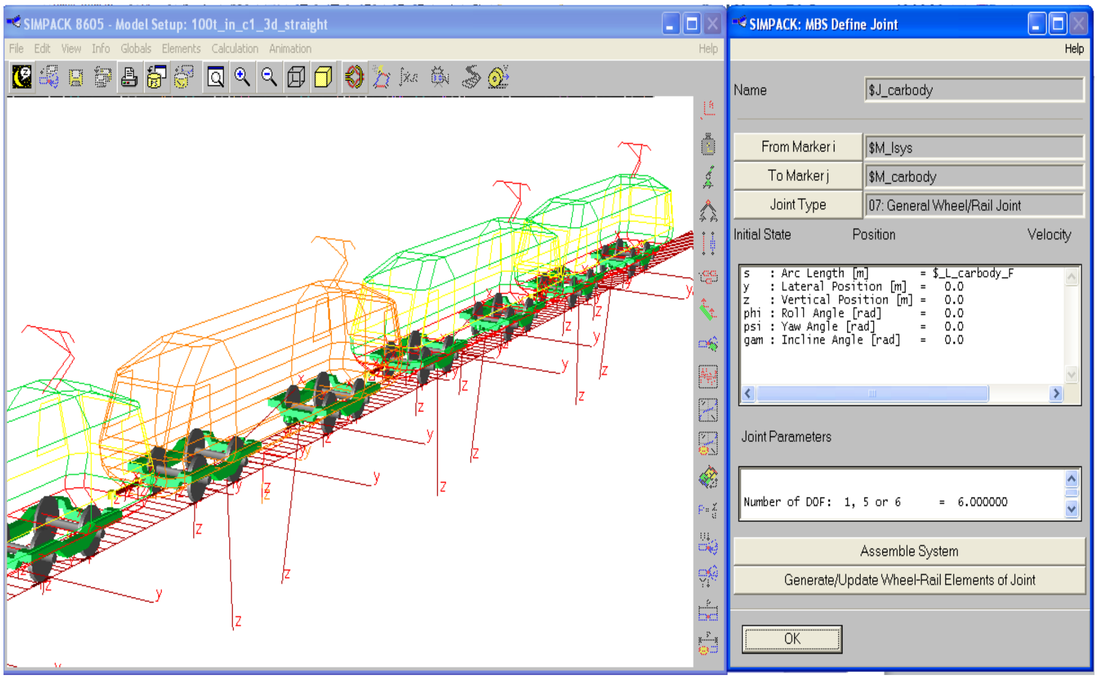


Figure- 5.16.Train joint elements

Train force elements

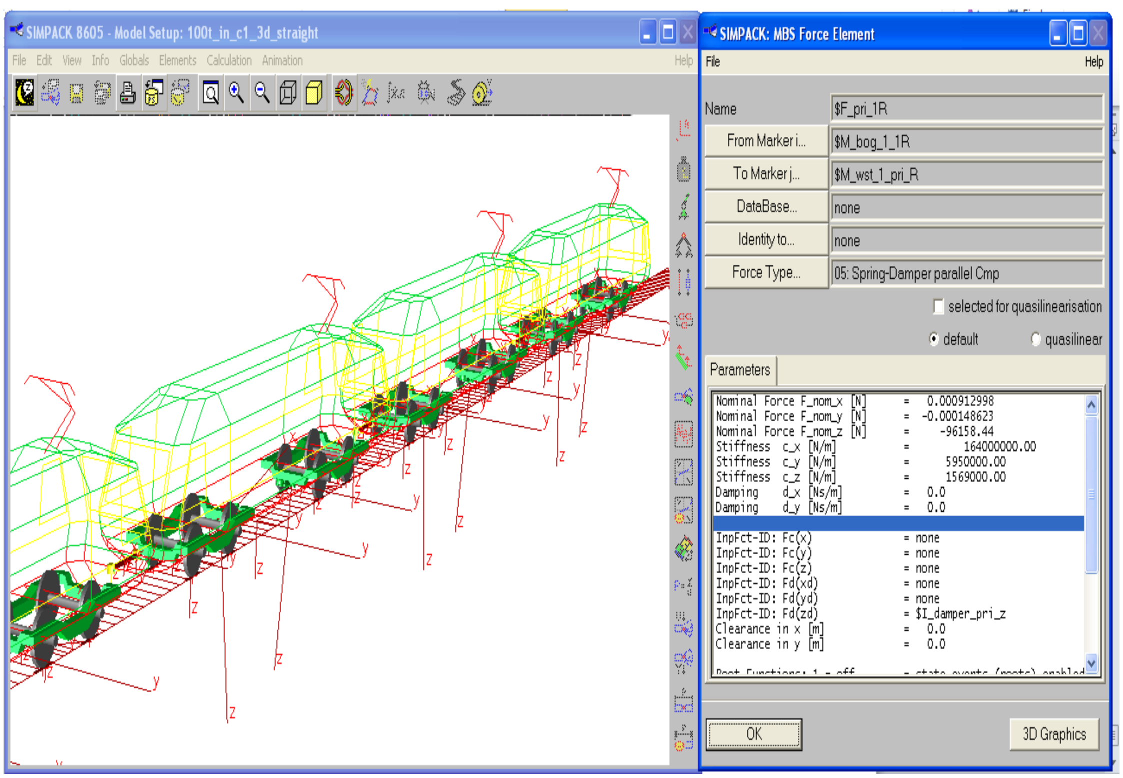


Figure-5.17.Train force elements

5.10 SIM-PACK simulation results.

As shown from the above modeling locomotive and simulation results in according to the locomotive input parameters, in the same track irregularities condition and at speed of 80km/h, R=800m curve radius . The coupler systems and train models already modeled, it included draft gear, followers and fright locomotive coupler mechanisms are presented in this chapter. The coupler system simulate together with train models in the chapter five in the simulation of the train, and show the coupler rotational angle, coupler compressive force, dynamic behaviour and performances of fright locomotives coupler systems. 13A/QKX-100 type is most typical in china and it has been shown good longitudinal dynamics behaviour in the fright locomotives. This coupler system serves both in curved and tangent tracks in good conditions.

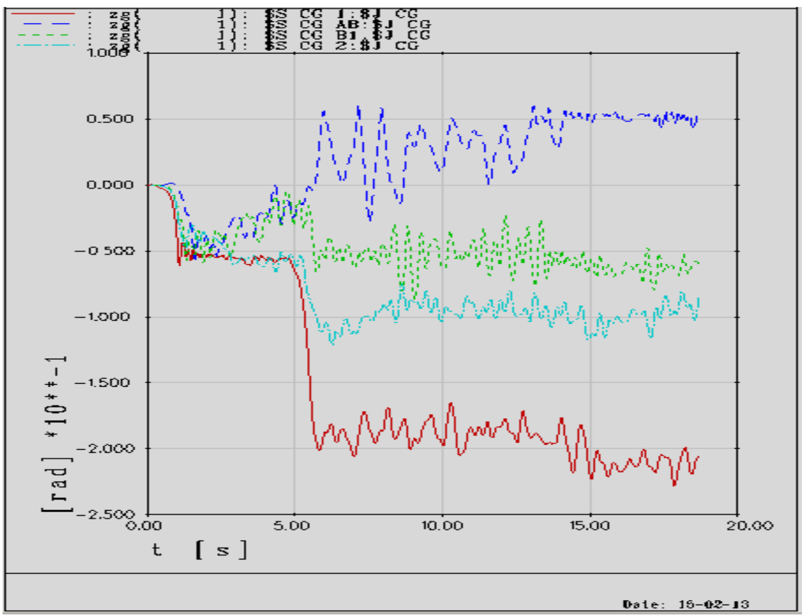


Figure-5.18.Coupler with draft gear joint position

As shown in the figure-5.18 the joint position of the four coupler at the initial condition the couplers starts to increasing joint position until it reaches -0.05rad after it reaches these value the four coupler makes a constant joint position until it reaches the time in 5sec .According to joint position with respect to time coupler -1 have high degree of freedom and coupler -2 also have high degree of freedom relative to coupler-1 the remaining coupler-3 and coupler-4 have a small degree of freedom. There for from the graph indicates that the maximum degree of freedom causes to increasing the rotational angles .as the rotational angle excess beyond its limit the wheel set moves out of line then it causes the deraignment

5.10.1. Coupler rotational effects due to longitudinal force

Figure 5.19 presents the compressive forces on the four locomotive couplers. When the locomotives begin dynamic braking, the braking wave of the first locomotive is delivered to the back end. Longitudinal in-train impact forces have little effect on the first coupler, the force of which increases from 0 to 500 kN and then remains almost stable at 500 kN, which is almost equal to the braking force. The next two couplers connected to locomotive 2, which is located in the middle of the train, suffer larger in-train impact forces. The coupler forces present periodic sinusoidal forms. The maximum forces of the second, third and the fourth couplers are 700, 1200 and 2000kN, respectively. Therefore, longitudinal in-train forces increase the coupler forces. At the same time, a distinct disturbance can be observed when the coupler forces just overcome there-load force of the draft gear.

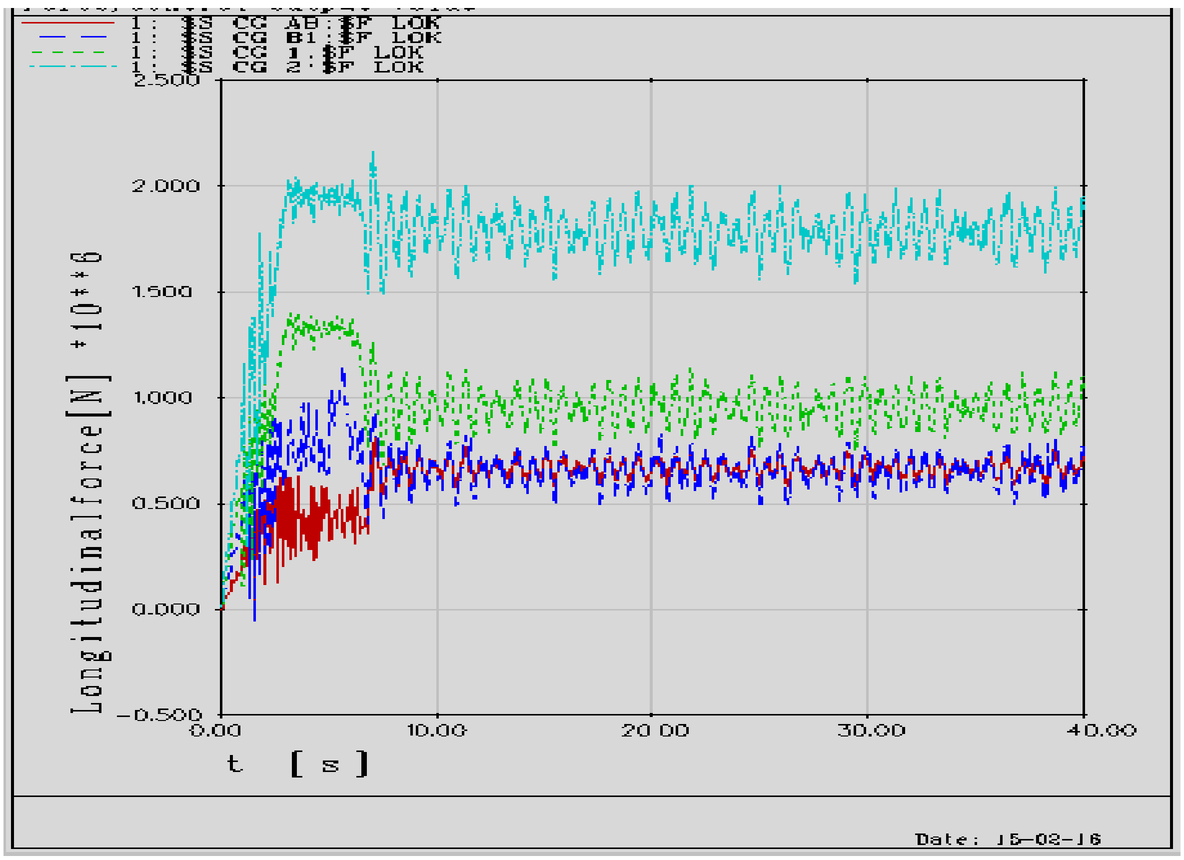


Figure-5.19.longitudinal force

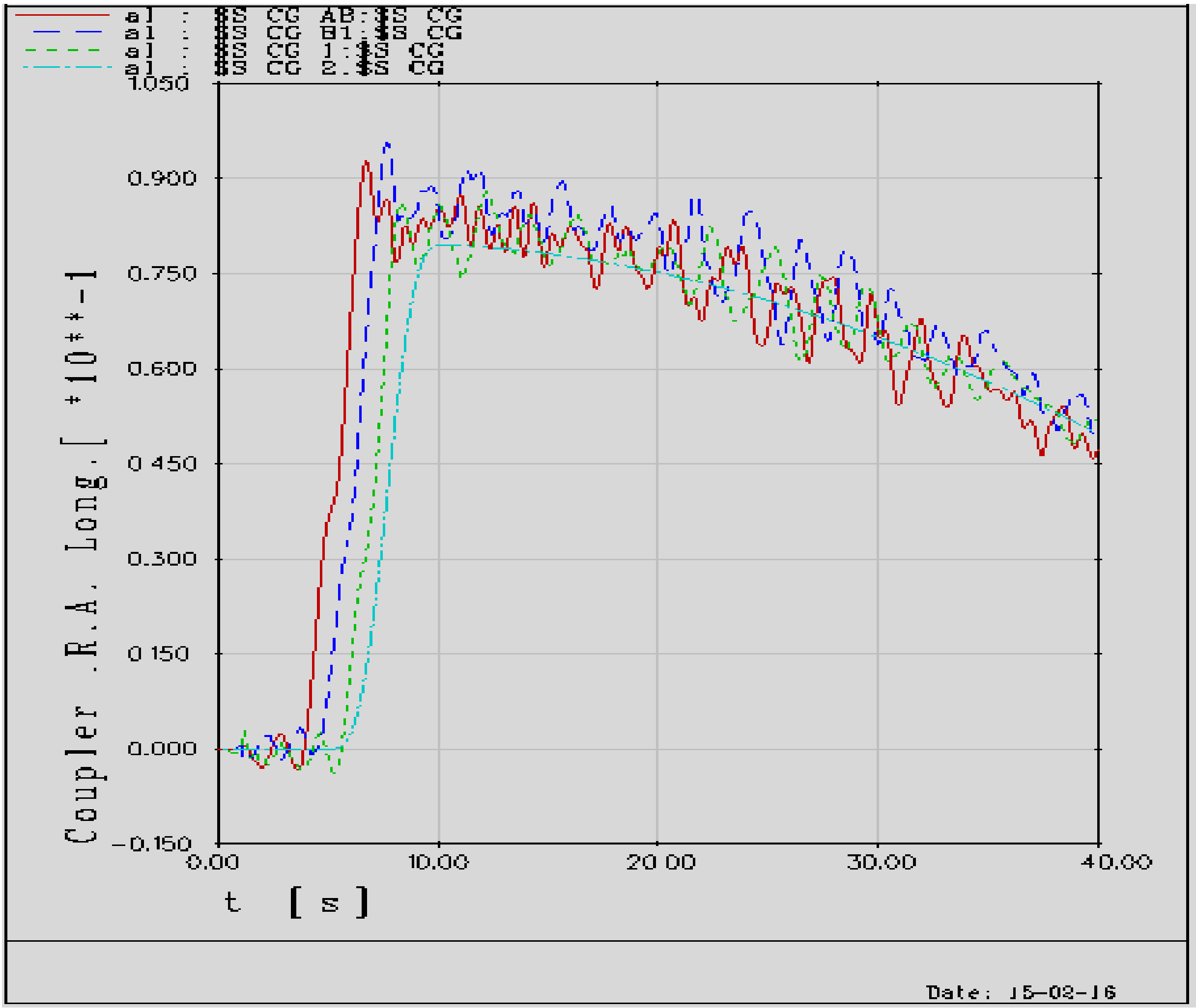


Figure-5.20.Coupler Rotational angle due to longitudinal force

Figure 5.20 Illustrates that the rotation angle of the first coupler is between ±2.8◦, which does not reach the maximum coupler free angle. The third coupler bears a maximum compressive force, and its rotation angle reaches the maximum coupler free angle of 6◦. The rotation behaviour of the third coupler drives the locomotive and causes a small rotation of the second coupler. The rotations of the second and third couplers change periodically with the coupler compressive force increase.

5.10.2. Coupler rotational effects due to lateral force

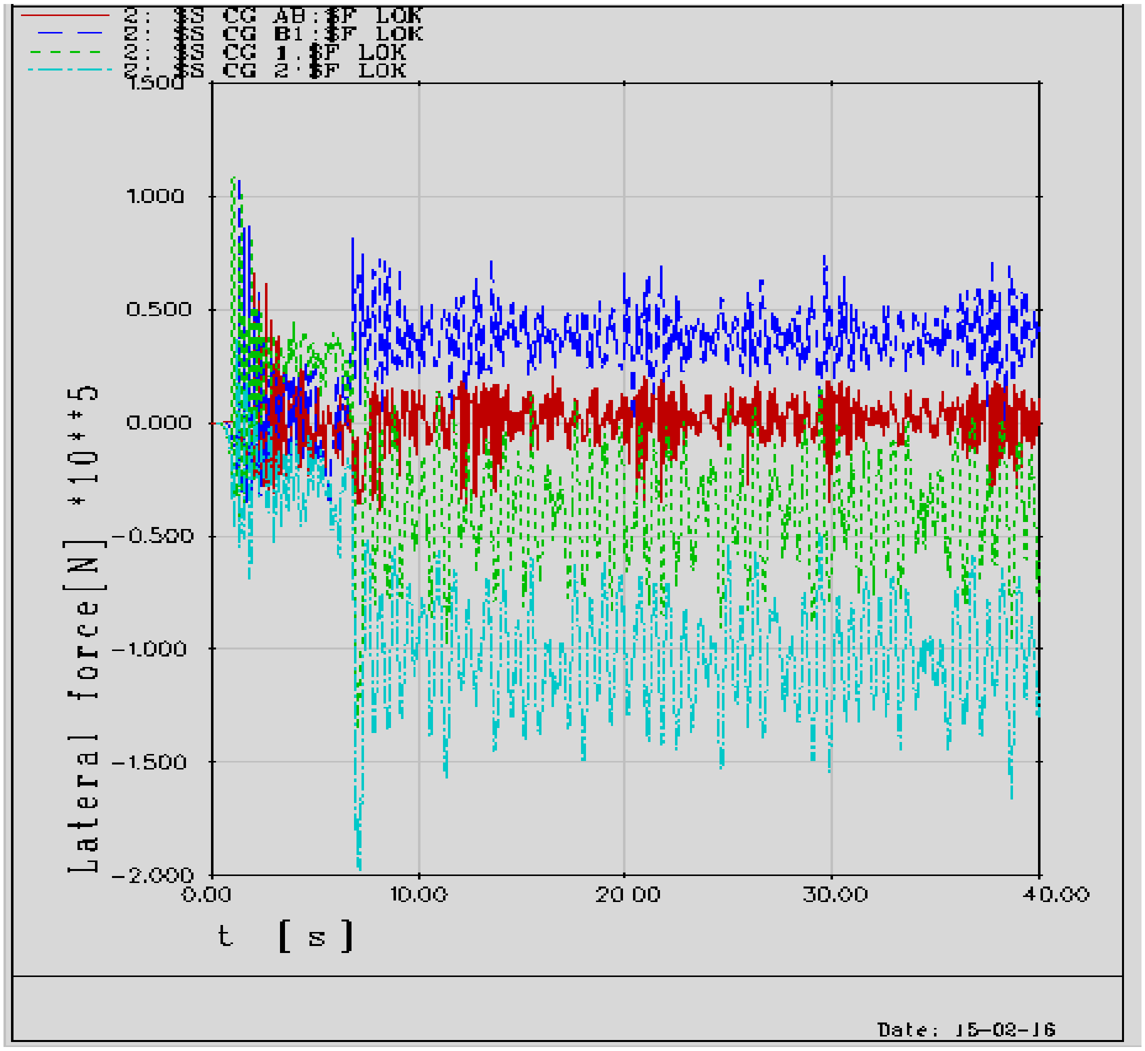


Figure-5.21.Lateral force

Figure 5.21presented in the simulation, the locomotive's initial speed was 80 km/h and. The maximum forces of the middle and rear couplers are 75kNand 200kN for the middle double locomotives according to the simulation results. The simulation coupler's longitudinal forces are shown in Figure 5.19, and the coupler's lateral forces at different track irregularities are shown in Figure 5.21.Under these calculation conditions, the coupler was stable. When the AAR4 class irregularity was used, the yaw angles of the coupler and the force line are larger because of a bigger lateral relative displacement of the car bodies, which results in a larger coupler lateral force. Due to this large lateral force the centrifugal force is maximum and the train could appear the deraignment.

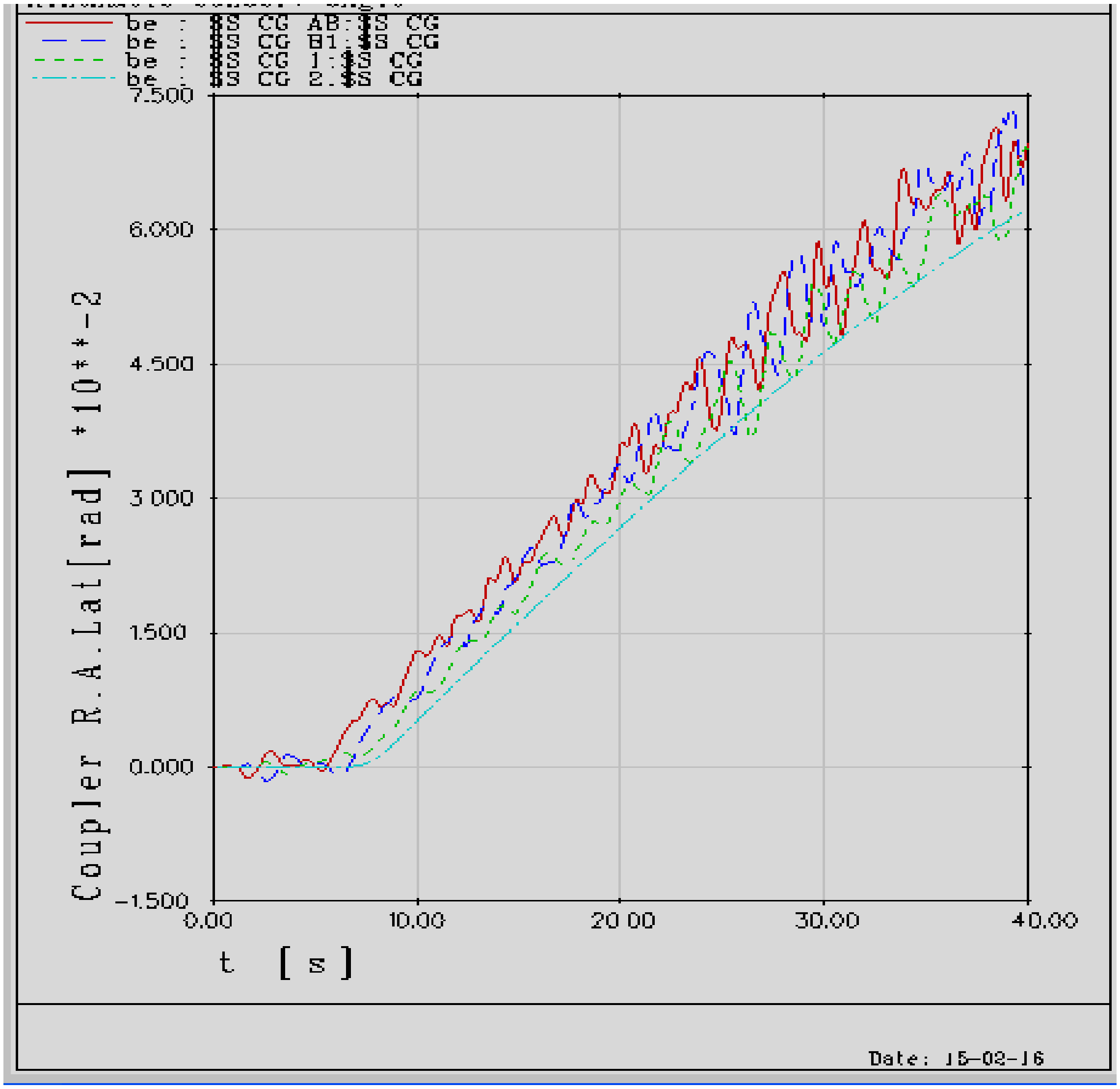


Figure-5.22.Coupler rotational angle due to lateral force

The simulation result shown in the Fig.5.22.Coupler rotational angle due to lateral force. Simulation result showed maximum rotational angle at speed of 80km/h on R800m curve track is 4◦ (0.069rad). Because of AAR6) track irregularities the angle of rotation is safe But in the higher lateral forces during this curve negotiation this leads to generate more rational angle. Due to this reason excessive the wheel set out of line and the coupler also twisting it leads to train dralignement.

5.10.3. Coupler rotational effects due to vertical force

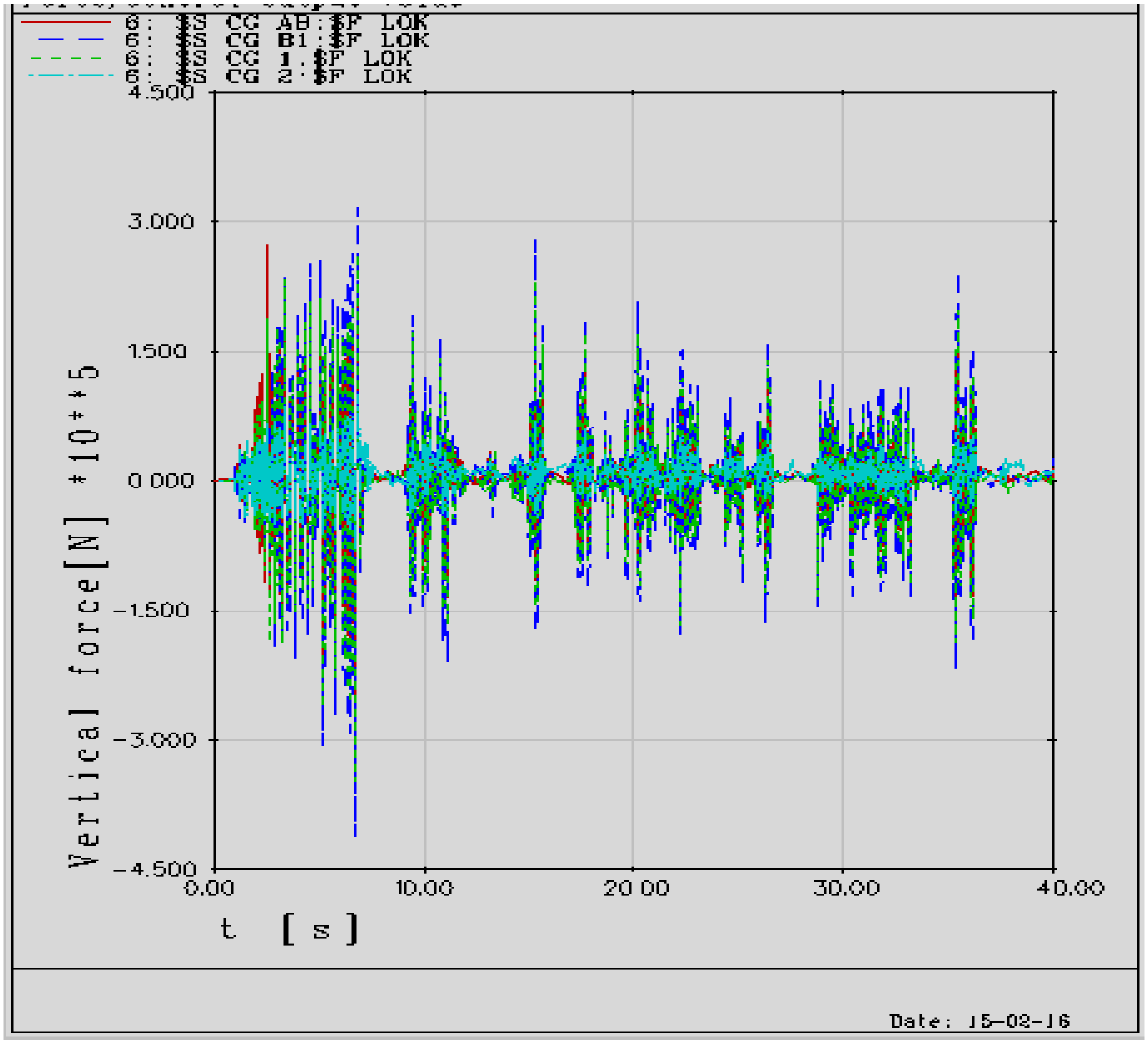


Figure-5.23.vertical force (yaw)

Vertical force of the coupler simulation result is shown Fig.5.24 at speed of 80km/h on curved track with R=800m (AAR6) track irregularities the peak values are less than the maximum specification limit value.in this case the maximum vertical force is 430kN ,410kN for 2nd and 3rd coupler respectively.

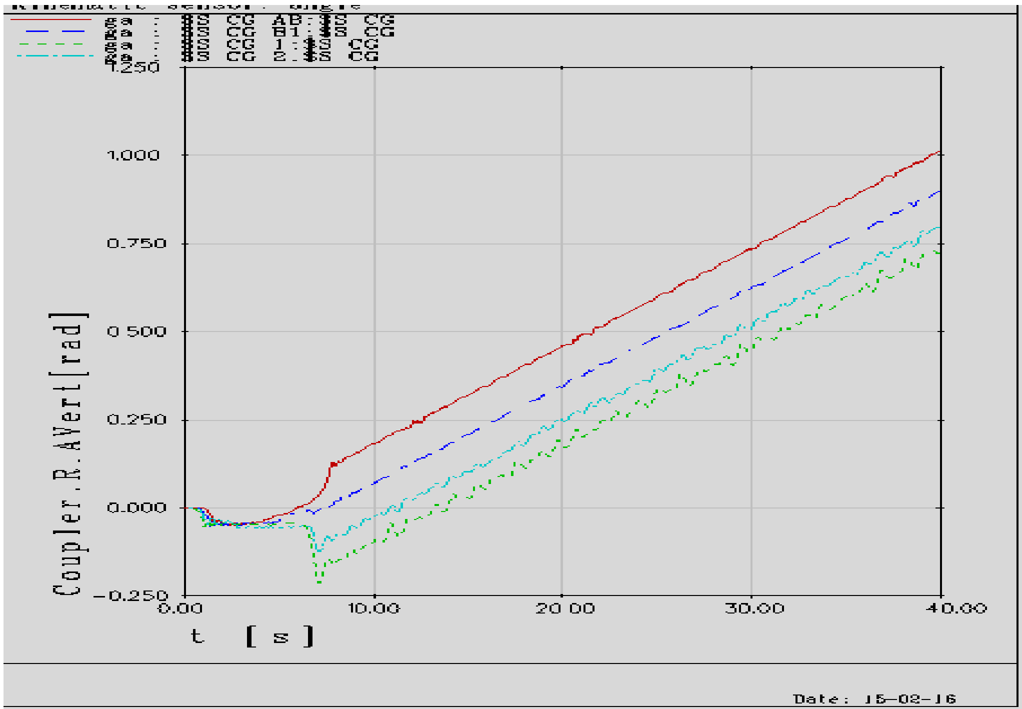


Figure-5.24.coupler rotational angle due to Vertical force. (Yaw angle)

As shown in the Fig.5.24 coupler rotational angles of coupler 1 and 2 simulation results are shown, the applied dynamic braking caused coupler force is 430kN on R800m curved track without track irregularities at speed of 80km/h. the coupler angle of 1is greater to the other couplers. When the coupler suffers from a huge longitudinal compressive force, the dynamic amplitude of the yaw angle of the coupler is small. At the AAR4 class track, the maximum yaw angle of the coupler reaches 3.4 and the maximum lateral relative displacement of the car bodies connected by a coupler is 84.45 mm, which is less than the car body critical lateral displacement of 86.20 mm. That is, the yaw angle of the force line is ideally 0, but the force line deflects the ideal position which then produces a lateral force due to the inertia of the coupler. There is a certain phase difference between the angles of force line .The larger the yaw angular velocity of the coupler, the more obvious deflection angles of the force line. This is inconsistent with the premise of the theoretical analysis because of the sliding between the arc contact surfaces. The force line yaw angle increases with increase in coupler yaw angle. After the brake releasing, the coupler is pulled instantaneously which can lead to larger fluctuations of the force line yaw angle because the contact points of arc surfaces slip due to the decrease in the contact preload

5.10.4 Wheel set and Derailment Coefficient

5.10.4.1 Wheel set lateral force

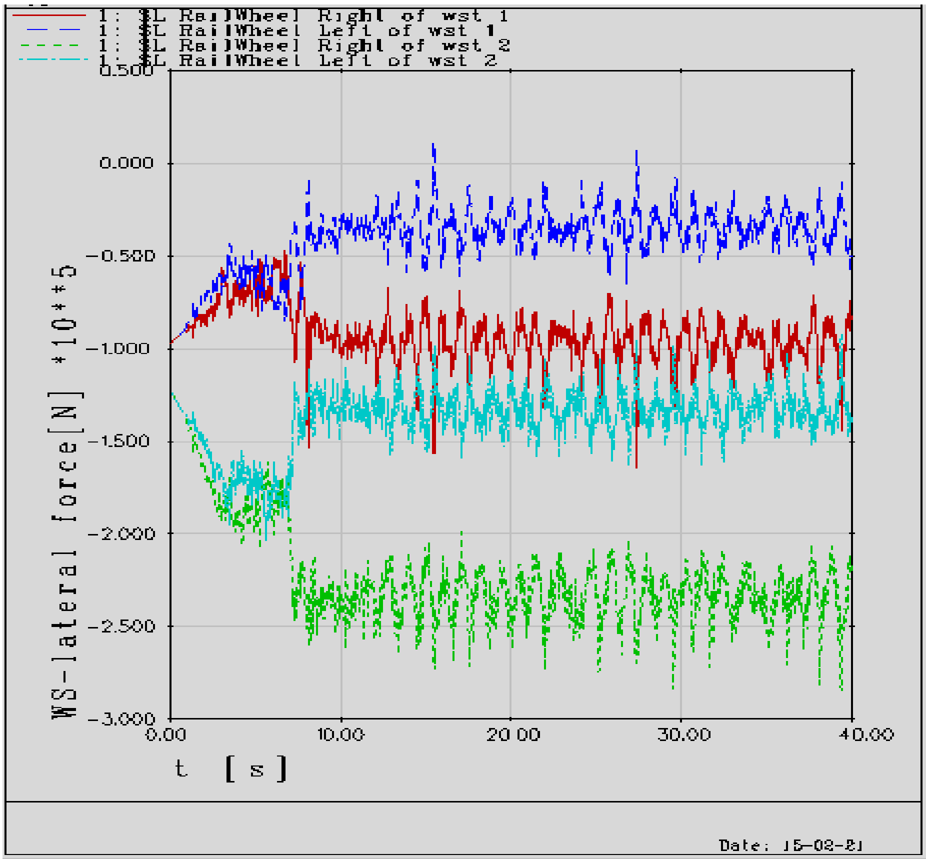


Figure-5.25.wheel set lateral force.

Wheel set lateral force of the leading wheels are shown in the Fig.5.25. Low rail side wheel unloading of leading wheel set maximum peak values of the wheel set force 300kN and 200kN higher values, when the locomotives running R800m curve track with irregularity (AAR6) at speed of 80km/h and applied dynamic braking force caused coupler 200kN.Figures 5.25 illustrates that the coupler rotation behaviour significantly affects the locomotive dynamic performance on the tangent track. Individually, the wheel set lateral force increases significantly with the increase in the third coupler's rotation angle, particularly on the rear bogie's wheel sets. The derailment coefficient exhibits the same trend as the wheel set lateral force increases.

5.10.4.2 Derailment coefficient.

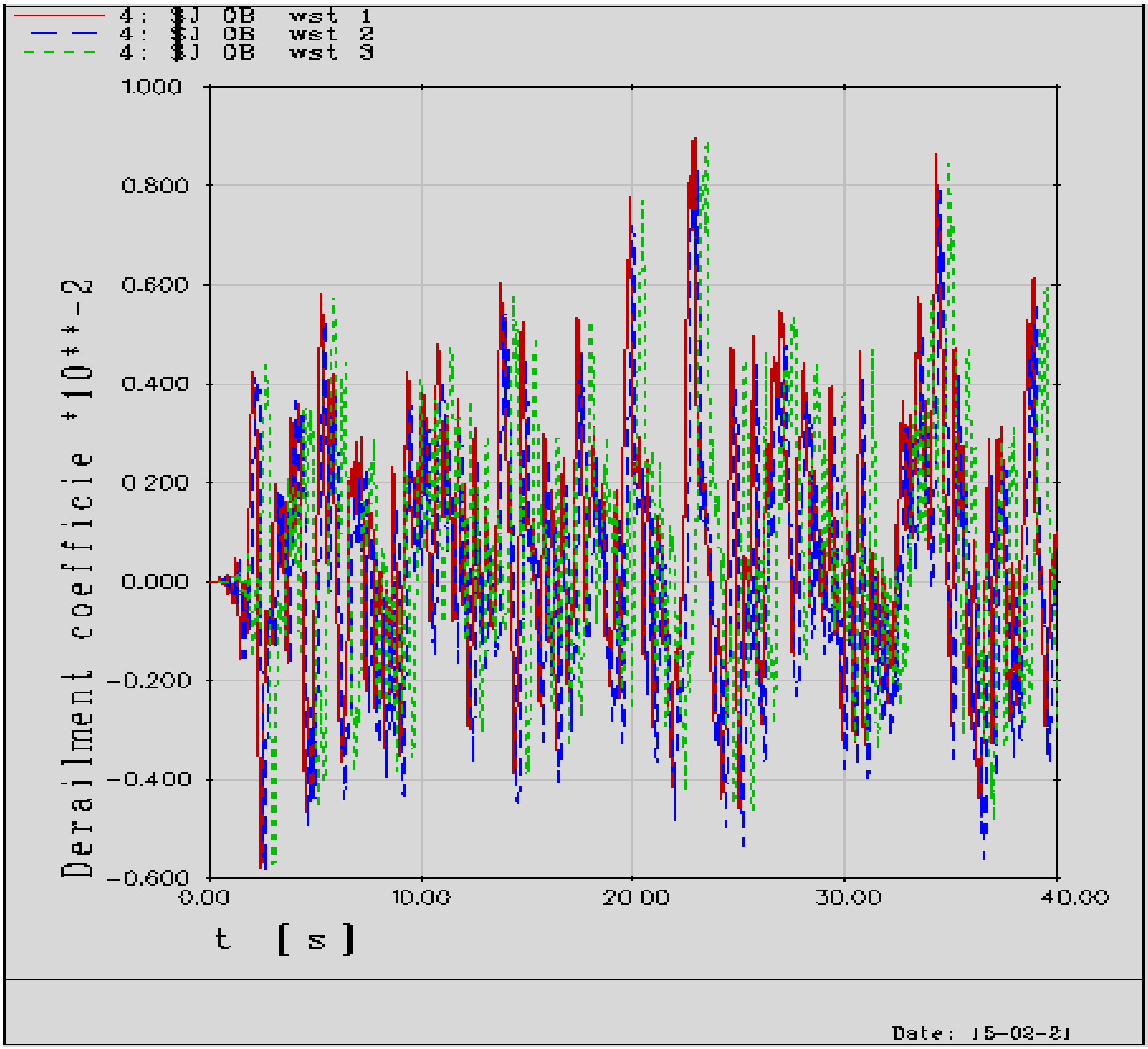


Figure-5.26. Derailment coefficient

As shown in the Fig.5.26 the high rail and low rail side of wheels derailment coefficient simulation results are less far from its maximum limitation values, and the running performance of the train is in safe condition due to this simulation results at speed of 80km/h on straight track with AAR6 irregularities and applied dynamic braking caused Coupler force 200kN.The derailment coefficient exhibits the same trend as the wheel set lateral force increases. The maximum values of the derailment coefficient are 0.5 this occurs during the maximum compressive force.

5.11. Parameter variations of individual couplers

Parameter variation due to lateral force-coupler_1

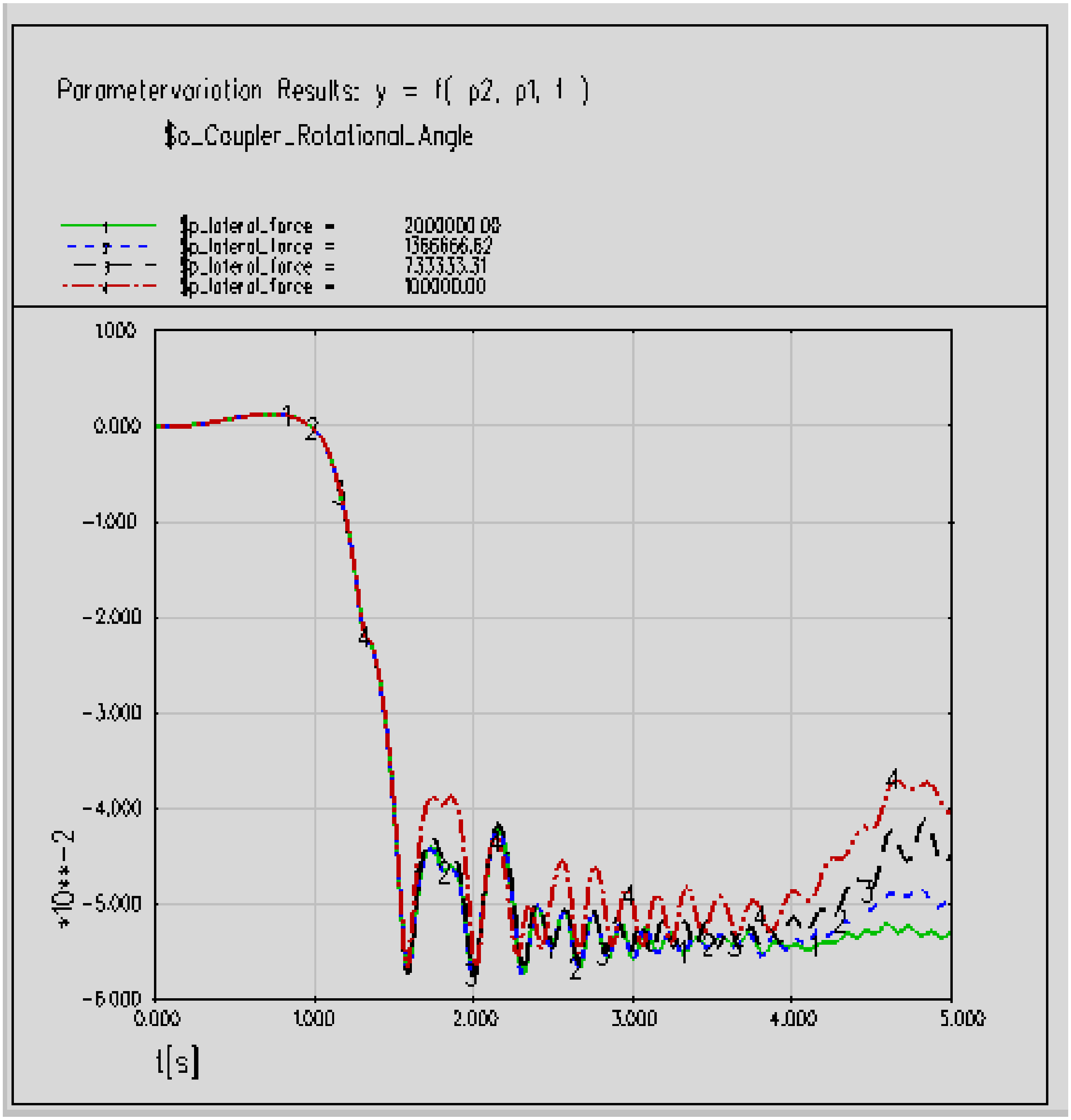


Figure-5.27.coupler_1_rotational angle due to parameter variation of lateral forces

Parameter variation due to lateral force-coupler_2

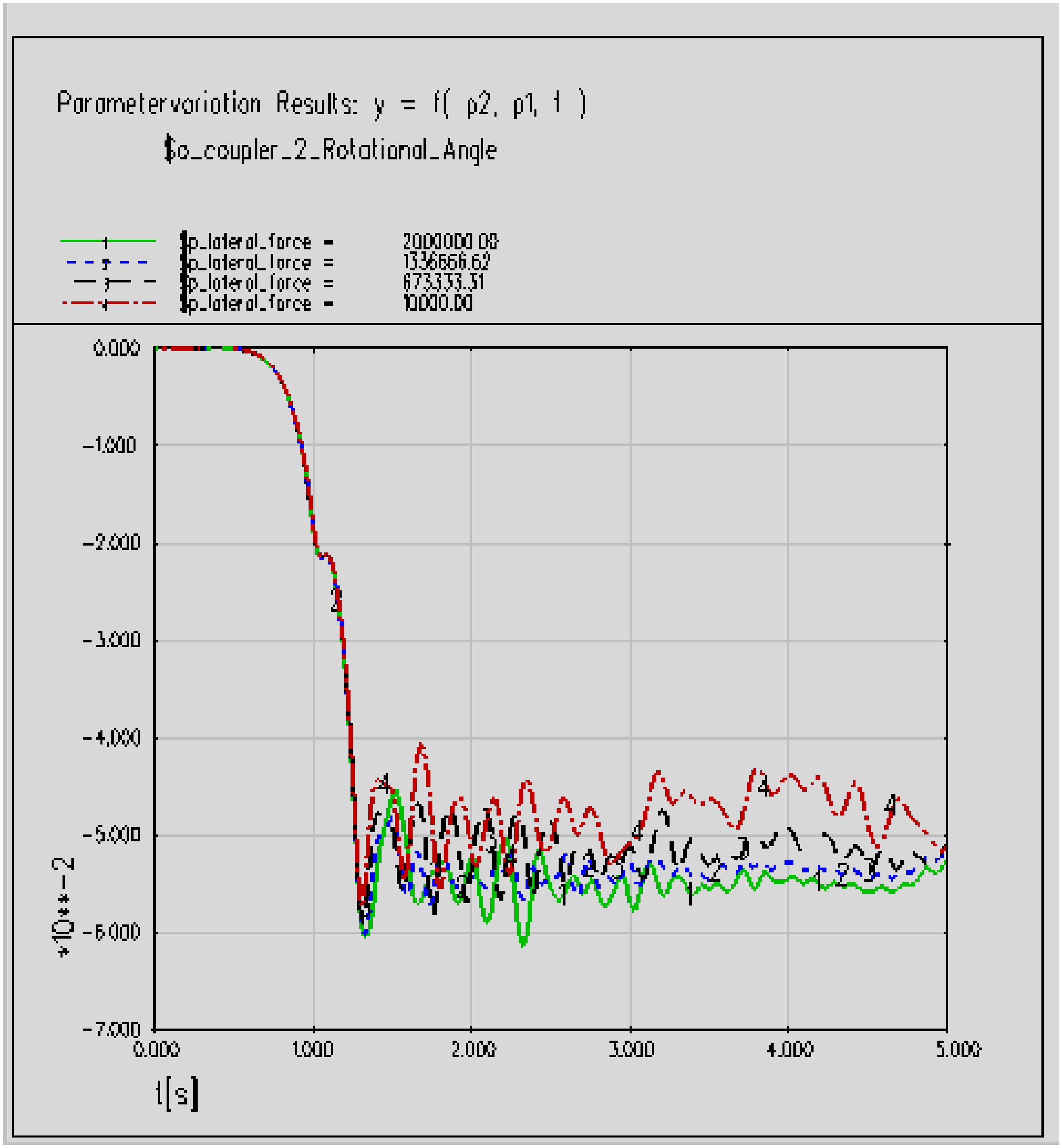


Figure-5.28.coupler_2_rotational angle due to parameter variation of lateral forces

Parameter variation due to lateral force-coupler_3

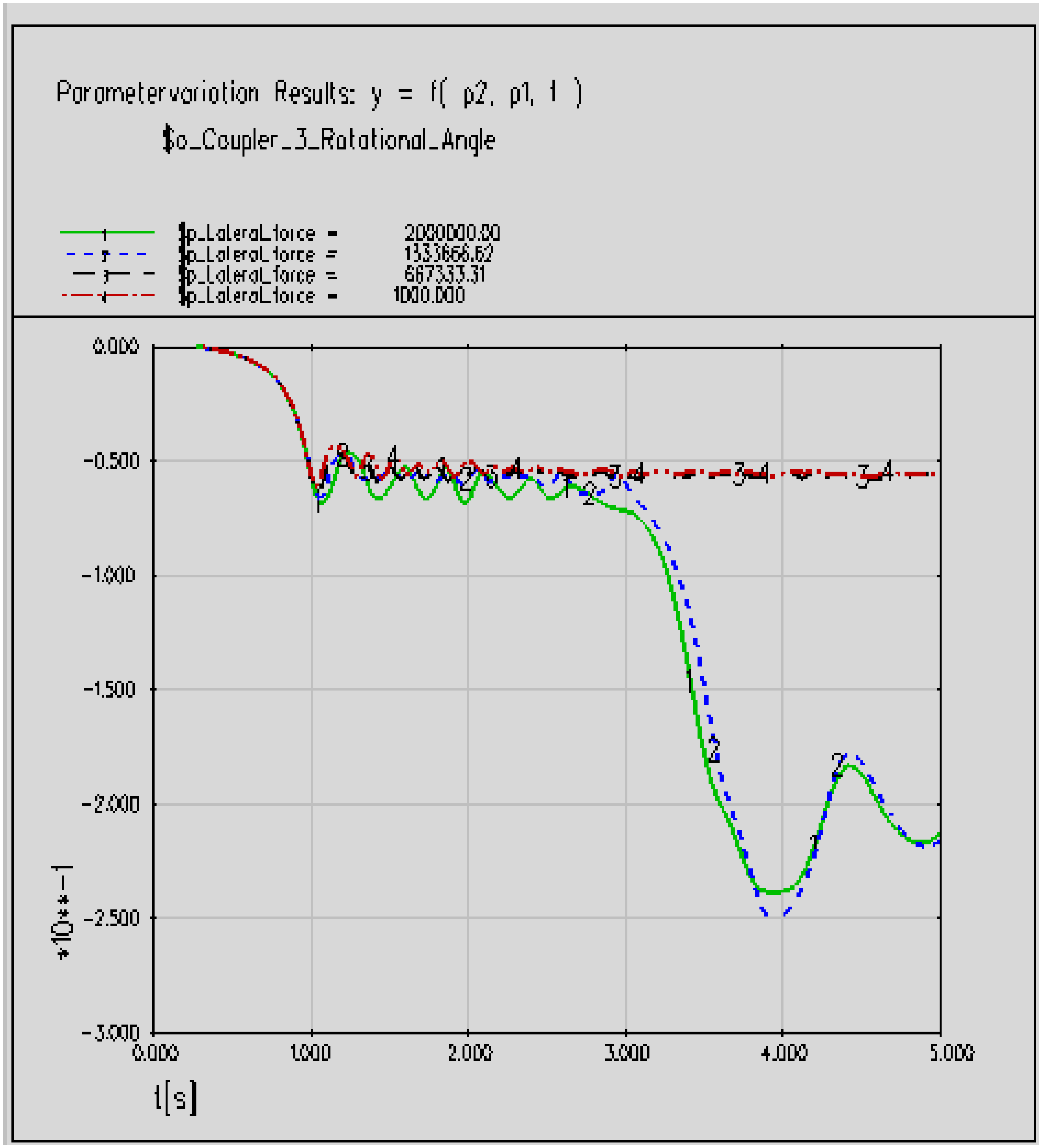


Figure-5.29.coupler_3_rotational angle due to parameter variation of lateral forces

Parameter variation due to lateral force-coupler_4

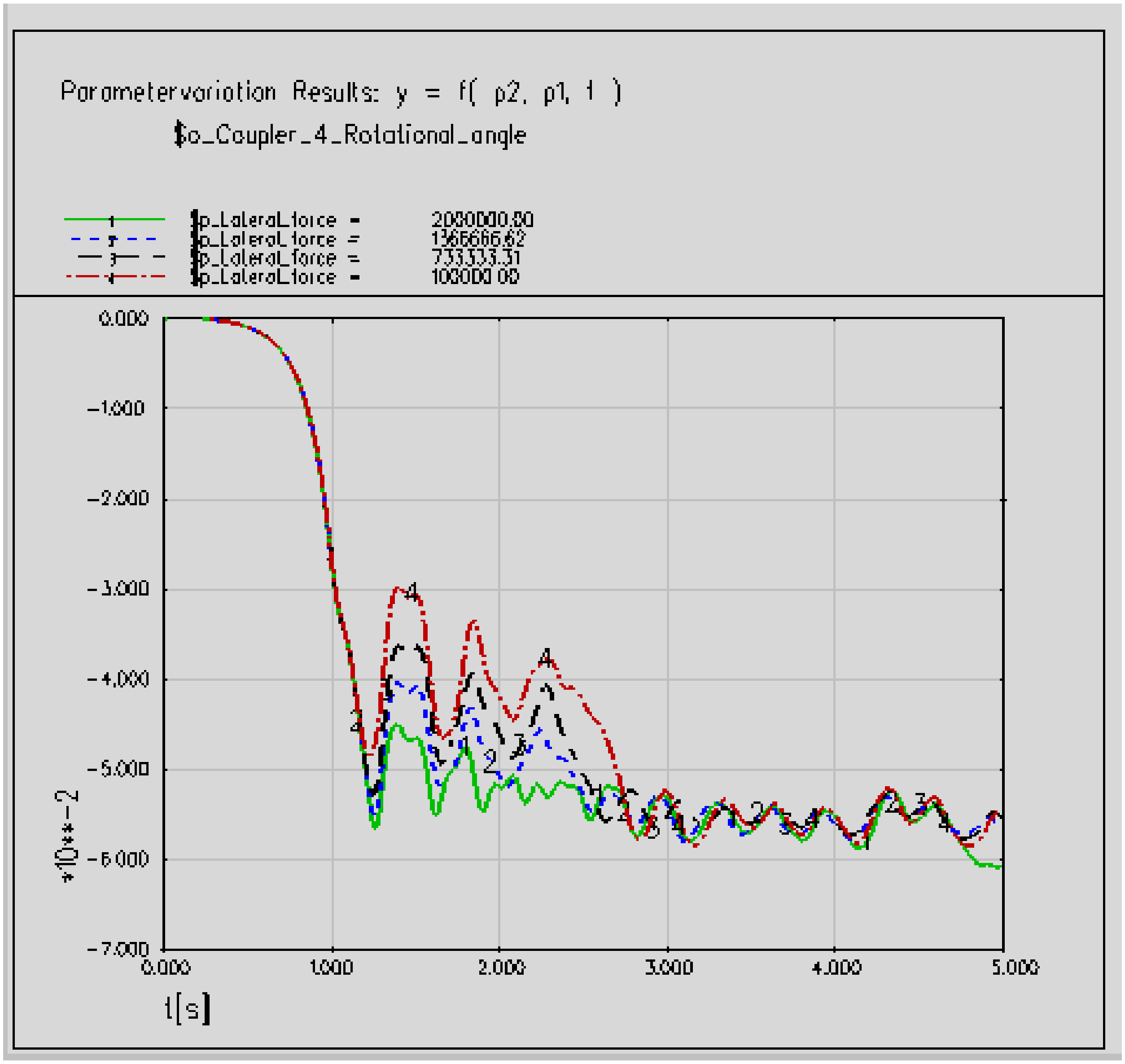


Figure-5.30.coupler_4_rotational angle due to parameter variation of lateral forces

From the above four figures represents the dynamic simulation of individual coupler rotational angle due to parameter variation of forces. This parameter variation helps as to understand the coupler maximum rotational angles due to the lateral forces. There for according to the above four figures the maximum coupler rotation angle is happened due to the maximum lateral forces. And also the deraignment coefficients is increasing that leads the wheel set out of alignment the accident will be happened.

CHAPTER 6

CONCLUSION AND RECOMMENDATION

6.1 CONCLUSION

The modeling, simulation and analyses results showed relevant knowledge's to understand easily about hauling Bo-Bo axle locomotives. The coupler systems and train models have been modeled in order to consider the hauling operation of locomotive; it included draft gear, follower plates and freight locomotive coupler mechanisms. The coupler systems simulated together with modeling of locomotives in heavy haul train service in chapter four and also chapter five, simulated result showed the coupler angle, coupler compressive force, dynamic behaviour and performances of fright locomotives and coupler systems. as shown from the above modeling locomotive and simulation results in according to the locomotive input parameters of table 5.1, in the same track irregularities condition and at speed of 80km/h, R=800m curve radius .13A/QKX-100 type of coupler system is most typical in china and it has been shown good longitudinal ,lateral and vertical dynamic simulation behaviour in the fright locomotives

This paper focuses on rotation behaviour and its effect on locomotive couplers in heavy haul trains. The performed simulations indicate that in-train impact forces can increase the coupler force of middle locomotive couplers, which leads to varying performances among middle and head couplers. As a result, the dynamic performance of a middle locomotive is worse than that of a head or front locomotive. Compared with those on longitudinal, lateral and vertical forces, coupler angles are much larger on a curved track due to maximum lateral force, but the performance of a locomotive remains the same. Decreasing a coupler's maximum free angle can reduce the lateral component of the coupler force, and the coupler's shoulder aligning force can also offset some of the coupler force. The performance of a locomotive and the running safety of a train can be improved significantly by decreasing in the coupler's maximum free angle. Based on calculations of coupler natural angles on a curved track and the maximum free angle simulation results obtained in this study.the dynamic simulation of individual coupler rotational angle due to parameter variation of forces we can easily understand the maximum and minimum coupler rotational angle

6.2. RECOMMENDATION

When a locomotive coupler rotates at an angle, the lateral component of the coupler force has an adverse effect on the locomotive's safety, particularly in heavy haul trains. In this paper, a model, of a 100-t heavy haul train is developed to analyses the rotation behavior of the locomotive's coupler system and its effect on the dynamic behavior of such a train's middle locomotive when operating on straight and curved tracks. The train model includes detailed coupler and draft gear with which to consider the hysteretic characteristics of the rubber draft gear model, the friction characteristics of the coupler knuckles, and the alignment-control characteristics of the coupler shoulder. The results indicate that the coupler's rotation behavior differs between the straight and curved tracks, significantly affecting the locomotive's running performance under the braking condition. A larger coupler rotation angle generates a larger lateral component, which increases the wheelset's lateral force and the derailment coefficient increasing. Decreasing the maximum coupler free angle can improve the locomotive's operational performance and safety. Based on these results, the recommended maximum coupler free angle is 4◦-5◦.

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