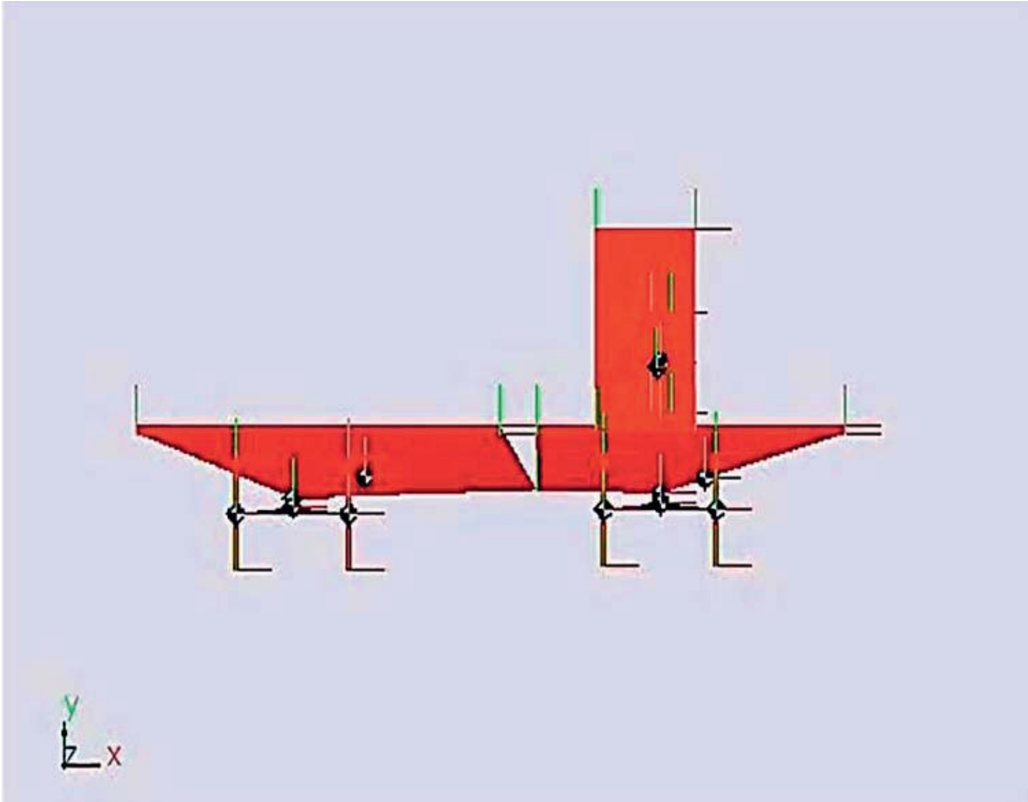
ARBETSRAPPORT

FRÅN SKOGFORSK NR 744 2011



Modellering av åkkomforten i en skotare

Modeling the Ride Comfort a Forwarder

Cheng Cheng

Examensarbete MMK 2011:30 MDA 399
KTH Industriell teknik och management
Maskinkonstruktion
SE-100 44 STOCKHOLM

SKOGFORSK

Bildtext: Visualisering av skotarmodellen i SimMechanics.

Keywords: Dämpning, modellering, skotare, åkkomfort.

SKOGFORSK

– The Forestry Research Institute of Sweden

The mission of Skogforsk is to pursue research and development for a sustainable, profitable and ecologically sound forestry. The Institute is supported by the entire forestry sector in Sweden through member fees and a charge levied on each cubic meter of wood harvested. Financial support is also given by the government according to a special agreement and by scientific funds for certain projects.

RESEARCH

– Forest production

– Wood supply

CONSULTANCY WORK

The Institute undertakes extensive consultancy work in its fields of special competence on behalf of forest enterprises, machine manufacturers and local and central government authorities. Such projects may involve specially commissioned research or the tailoring of new methods and routines to local conditions.

DISSEMINATION OF INFORMATION

The Institute uses a variety of media and channels for disseminating the findings of its research. These include personal contact, Internet, courses and seminares, the trade press, films and its own series of publications.

ISSN 1404-305X

Foreword

This page is to give many thanks to people who give great support and help during my thesis.

Thanks to Professor Jan Wikander for your kind guidance and help. I am very grateful for being able to learn so much from you. I really cherish every Friday having meeting with you.

Special thanks to Program of Mechatronics in KTH and Skogforsk for allowing me to participate in this project. Thanks to Björn Löfgren, Ulf Sellgren, Kjell Andersson for your really kind help and care for me during the entire process of the project and thank you Petrus Jönsson and Bengt O. Eriksson for answering me questions about measurement and SimMechanics with great patience.

Thanks to my three Swedish colleagues in this project, Hannes Åström, Jonas Jonsson and Daniel Grönqvist for your kind and consistent company in the past six months. Also thanks to my friends Xiaowan Su, Zhenrui Wang, Zheng Wang and Xiaolin Pan in Stockholm for their support and help in the project.

Also thanks to Assistant Professor Leping Feng and Professor Hans Boden from Program of Sound and Vibration, KTH.

Thanks very much for the support from different companies such as Komatsu Forest, Gremo, Trelleborg, Rottne, Ponsse and John Deere, I am so grateful.

Finally, I want to give my special thanks to my parents, family, teachers and friends in China, thank you for your support, encourage and patience throughout this thesis.

Dad and Mom, this thesis is a gift from Stockholm.

Cheng Cheng

Stockholm, 2011-03-01

Abstract

This thesis is based on a Forwarder – Valmet 860.4 – from Komatsu Forest, which has high productivity but without any primary suspension system at front and rear axle and without a cabin suspension system to isolate from vibrations caused by uneven ground. A previous project did practical measurement of vertical accelerations of seat, cabin, front wagon and rear wagon, as well as the four bogie angles. In this thesis project a simulation model of the forwarder has been established in SimMechanics based on some assumptions and simplifications, and evaluated against the previous measurement data. The simulation and measurement data is matched both in the time and frequency domains to demonstrate that the simulation model can reflect the main characteristics of the real forwarder.

Based on existing models appearing in some published papers, a seat suspension (with air spring), a cabin hydro pneumatic suspension, three different passive and three different semi-active primary suspensions are modeled in Simulink and inserted to the whole forwarder model in SimMechanics. The effect of each type of suspension is analyzed. The difference between the three passive primary suspensions is the different stiffness and damping coefficients at front and rear axle, which are selected based on previous projects. The semi-active suspensions are controlled by Sky-hook and on-off control principles, and provide variable stiffness and damping coefficients according to changing ground irregularity.

The effects of different suspension systems are compared, and it is concluded that the largest improvement was obtained by on-off controlling of the semi-active suspension at front and rear axle, which gave a reduction of the seat vertical acceleration RMS value by 29.3%. Further, it is possible to reduce the vibration levels of the forwarder by selecting appropriate parameters of the passive suspension system. SimMechanics has proved to be a useful tool to model the vibration of the whole forwarder, but more efforts and research should be devoted to the area.

Contents

Sammanfattning

Denna avhandling avser en Skotare – Valmet 860,4 – från Komatsu Forest, som har hög produktivitet, men saknar såväl primärt fjädringssystem för fram- och bakaxel som hyttfjädringssystem för att isolera mot vibrationer från ojämnt underlag. Tidigare projekt har praktiskt mätt upp accelerationer (vibrationer) i förarstol, hytt, framvagn och bakvagn, liksom fyra boggievinklar. I detta examensarbete har en simuleringsmodell av skotaren utvecklats i SimMechanics baserat på vissa antaganden och förenklingar, och utvärderats gentemot tidigare mätdata. Gjorda simuleringar matchas mot mätdata i både tids- och frekvensdomänen för att demonstrera att simuleringsmodellen reflekterar huvuddragen i den verkliga skotaren.

Baserat på befintliga modeller av olika dämpningsalternativ som förekommer i vissa publicerade artiklar, implementeras dessa modeller i Simulink och integreras med skotarmodellen varefter den vibrationsisolerande effekten analyseras. Dämpningsalternativen är; luftfjädring av förarstolen, hydropneumatisk fjädring av hytten, tre olika passiva respektive semi-aktiva primära dämparsystem. De tre olika passiva primära dämparsystemen har olika styvhets- och dämpningskoefficienter för fram- respektive bakvagn, och har valts på basis av tidigare projekt. De semi-aktiva alternativen är av typen Sky-hook respektive on-off styrning, vilka ger variabla styvhets- och dämpningskoefficienter som funktion av underlagets karaktär.

Effekterna av olika fjädringssystem jämförs, och det konstateras att den största förbättringen erhölls genom semi-aktiv on-off styrning av det primära fjädringssystemet på fram- och bakaxel, som gav en reduktion av den vertikala accelerationens RMS-värdet om 29,3%. Arbetet visar också att det är klart möjligt att minska vibrationsnivåerna i skotaren genom väl anpassade parametrar i ett passivt system. SimMechanics har visat sig vara ett användbart verktyg för att modellera vibrationer i hela skotaren, men mer insatser och forskning bör ägnas åt området.

Nomenclature

Here are the Notations and Abbreviations that are used in this Master thesis.

Notations

Symbol	Description	Units
a	Acceleration	$[m/s^2]$
a_w	Frequency-weighted acceleration	$[m/s^2]$
A	Area	$[m^2]$
c	Damping coefficient	$[N \cdot s/m]$
c_{sky}	Skyhook damping coefficient	$[N \cdot s/m]$
F	Force	[N]
F_s	Sampling frequency	[Hz]
F_{sky}	Skyhook force	[N]
h	Ground vertical displacement	[m]
I	Moment	$[N \cdot m]$
k	Spring stiffness	[N/m]
m	Mass	[kg]
P	Pressure	[Pa]
t	Time	[s]
v	Velocity	[m/s]
z	Displacement	[m]

Abbreviations

A (8)	8-hour energy-equivalent frequency weighted acceleration
ADAMS	Automatic Dynamic Analysis of Mechanical Systems
BS	British Standard
CAD	Computer Aided Design
DOF	Degree-of-freedom
EU	European Union
FFT	Fast Fourier Transform
HAV	Hand-Arm Vibration
ISO	International Standard Organization
PSD	Power Spectral Density
RMS	Root Mean Square
SEAT	Seat Effective Amplitude Transmissibility
VDV	Vibration Dose Value
WBV	Whole-body Vibration
W_d	ISO frequency weighting filter used for lateral and longitudinal vibrations of seated human
W_k	ISO frequency weighting filter used for vertical vibrations of seated human

1 Introduction

This chapter describes the background, the purposes, the limitations and the methods used in the presented project, and introduces the thesis overview.

1.1 BACKGROUND

Ride comfort analysis for modern forestry machines is extremely critical because operators are exposed to high-level and long-time vibration environment, which is strongly dependent upon machine-terrain interactions and dynamics of the machine (P.E Boileau, et al., 1990). In the past, little attention was paid to machine vibrations, especially for off-off machines such as forestry machines and agricultural machines. For previously manufactured forestry machines which did not have a suspension system or soft tires to offer damping, the whole body vibrations became a big problem for operators, who would suffer from shoulder, neck and back pain, or even motion sickness while the vibration frequency range is 0.1 to 0.5 Hz. As a result, vibration has become one of the most significant problems for the forestry industry. For the development of machine design technology and due to the increasing requirement of operating environment, new legislations as ISO 2631(International standard) and EU Directive 2002/44/EC have been introduced to reduce and control the whole-body vibration.

The vibration and shock of forestry machines, primarily originating from engine, drive-train and uneven terrain, are currently more or less directly transmitted from the vibration source to the seat, cab and operator.

The forwarder in this report is Valmet 860.4, a product from Komatsu Forest AB. It does not have any primary suspension between wheels and chassis, and no cabin suspension system, just four bushings under the cabin to take away engine vibrations. Much work has been done to improve its productivity which has contributed to shorten delivery time of harvested logs. However, in another way, increasing productivity i.e. increasing driving speed makes the operator exposed to higher vibration levels.

Generally, three steps of vibration reduction could be applied to machines: tire, seat and suspension systems. Previous researchers have provided different measures to improve the ride comfort, such as suitable tires, primary suspension at front and rear axles, cab suspension and seat suspension (P.E Boileau, et al., 1990).

1.2 GOAL

The purpose was to make a ride comfort analysis for the mentioned forwarder, by modeling the whole forwarder and related suspension systems in Simulink/-SimMechanics. Firstly, evaluate simulation results by comparing with measurement results, and compare vibration reduction effects with respect to different types and levels of suspension. Secondly, find out methods to improve vibration environment for operators, and provide guidance to manufacturers of forestry machines. Thirdly, provide a basic method to model the forwarder in SimMechanics.

1.3 DELIMITATIONS

In this report, the following main requirements and delimitations have been defined:

(1) Simulation results should be reasonably consistent with measurement results, e.g. the vertical acceleration of front wagon, rear wagon, cabin and seat both in time domain and frequency domain.

(2) Simulation should be performed using a ground surface similar to the real situation, in this case a test track with three different types of bumps is given.

(3) The simulation model should be based on the Forwarder Valmet 860.4.

(4) Only vertical direction of vibration is focused in this project.

1.4 METHOD

With the dimensions and mass parameters of Valmet 860.4, and measured data (vertical accelerations of front wagon, rear wagon, cabin and seat, and four bogie angles) from previous projects, a model of the whole forwarder was built in Simulink/SimMechanics based on some assumptions and simplifications, and the model has mass, moment of inertia and coordinate system defined in relation to the real machine. Meanwhile, the test track from Skogforsk has been modeled in terms of vertical displacement of ground with respect to time in Matlab, and imported into the whole model by the block "from file" in Simulink. According to the measurement results, the stiffness and damping coefficients of seat suspension, cabin bushing and tire have been tuned. Then, a hydro pneumatic suspension, a seat suspension (with air spring), 3 different primary passive suspensions and 3 different primary semi-active suspensions have been added to the whole forwarder model separately. Finally, comparisons of the effects of different types of suspension systems, and effects of different levels of suspension systems have been evaluated. Moreover, a combined condition, which is the initial forwarder with hydro pneumatic cabin suspension, seat suspension (with air spring) and primary semi-active suspension (condition 3) was simulated. Because of the limited time, those new suspension systems are designed on the basis of some suspension systems which have appeared in related published papers (listed in reference). All the data are processed in Matlab, e.g. low-pass filtering and Fast Fourier Transform (FFT).

1.5 THESIS OVERVIEW

The background and purposes were presented in this chapter. Chapter 2 focuses on literature review, regarding topics such as legislation, whole body vibration, suspension systems and simulation software.

Chapter 3 focuses on the simulation model and deals with the modeling of the whole forwarder, the ground irregularity, tire and bogie.

Chapter 4 gives the validation of the model, by first describing about data processing in Matlab, and then showing the validation results of the simulation model with measurement data. The chapter also discusses the advantages and disadvantages of using SimMechanics.

Chapter 5 discusses new suspension systems for this forwarder. A hydro pneumatic cabin suspension, a semi-active seat suspension, 3 different primary passive suspensions and 3 different semi-active suspensions have been modeled in Simulink and added to the whole forwarder model, separately.

Chapter 6 includes the results with different levels and different types of suspension system, and presents results of combined suspension configuration.

Chapter 7 presents conclusions and recommendations for future work.

2 Frames of reference

2.1 VALMET 860.4

The forwarder Valmet 860.4 manufactured by Komatsu Forest AB, is the research subject of this report, which is A forwarder is a kind of forestry machine that carries felled logs from the harvesting area to the road side of the landing area. The forwarder Valmet 860.4 has a load capacity of 14 metric tons with market-leading maneuverability on different terrains due to a new refined bogie design (Komatsu Forest, 2010).

The general dimensions of the Valmet 860.4 are shown in figure 2.1 (Komatsu Forest, 2010). The forwarder's width *A*, varies between 2 760 and 2 990 mm. In the simulation model, the width *A* was set to the average of this range, 2 875mm.

The total mass of the forwarder is 16 060 kg and the load capacity is 14 000 kg, with the design of 8 wheels. Each pair of wheels is connected to a bogie attached to the front and rear frame on bogie axles. The propulsion of the wheels is mechanically applied on all eight wheels from a diesel engine with 145kW through a hydraulic system (Komatsu Forest, 2010).

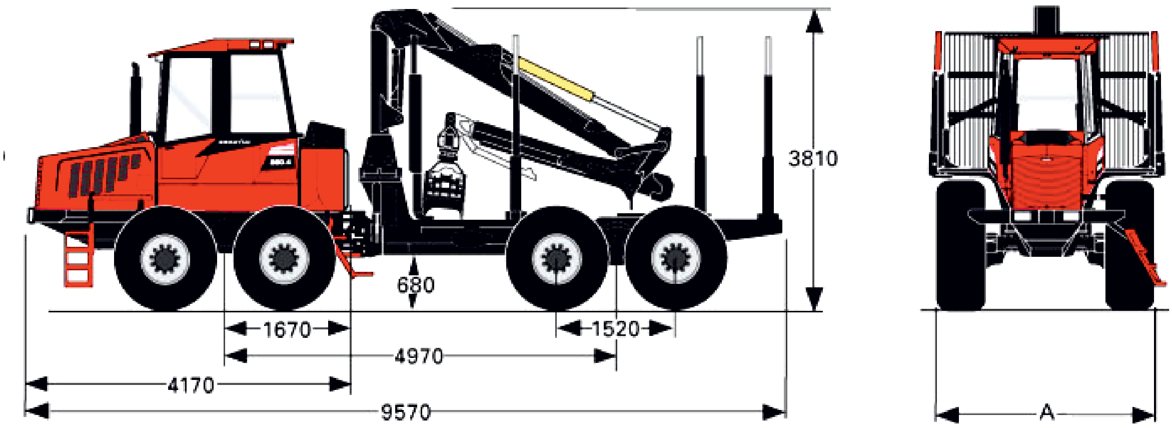


Figure 2.1.
Dimensions of Valmet 860.4 (Komatsu Forest, 2010).

The forwarder is a type of forestry machines which are exposed to high levels of vibrations. Unfortunately, Forwarder Valmet 860.4 does not have a primary suspension system at front and rear bogie or a cab suspension system that isolates the operator from the inputs from vibrations induced by drive-train and uneven terrain, but there is a seat suspension system. This is not enough to satisfy the requirement of ride comfort and vibration environment.

2.2 LEGISLATION

International and national legislations have been applied to provide guidance to whole-body vibration evaluation, such as ISO (International Standard Organization) 2631-1: 1997, BS (British Standards) 2841-1987, the European Directive 2002/44/EC and Swedish ARBETSMILJÖVERKET AFS vibrations 2005:15, etc.

ISO 2631-1: 1997, Mechanical vibration and shock-Evaluation of human exposure to whole-body vibration, developed from ISO 2631-1985, is the general requirement to evaluate whole-body vibration, and is intended to define methods of quantifying whole-body vibration in relation to human health and comfort, probability of vibration perception and the incidence of motion sickness. In ISO 2631-1, two methods have been given for vibration evaluation, a basic evaluation method using weighted root-mean-square acceleration and additional methods with running RMS (root mean square) and fourth power VDV (frequency weighted vibration dose value). The frequency range considered in ISO 2631 is 0.5-80 Hz for health, comfort and perception, and 0.1-0.5 Hz for motion sickness (ISO2631-1, 1997).

BS 6841-1987, similar to ISO 2631:1997, also gives methods for quantifying vibration and repeated shock in relation to human health, interference with activities, discomfort, the probability of vibration perception and the incidence of motion sickness. BS 6841 provides greater guidance on vibration effects without defining vibration limits, but gives the methods for vibration measurements and evaluation methods.

The European Union Directive 2002/44/EC, different from ISO 2631-1 and BS 6841, define the exposure minimum limit values and action values for both HAV (hand-arm vibration) and WBV (whole-body vibration). For HAV, the daily exposure limit value and action value standardized to an eight hour reference period are $5m/s^2$, 2.5 m/s^2, respectively. For WBV, the daily exposure limit value and action value standardized to an eight hour reference period are 1.15$\mathrm{m/s^2}$, 0.5$\mathrm{m/s^2}$, respectively. The daily exposure action and limit values in the directive are all specified as an 8-hour energy-equivalent frequency-weighted acceleration (known as A(8) value), although vibration dose value (VDV) alternatives are given for whole-body vibration (WBV) (European Union Directive 2002/44/EC, 2002).

Following the European Union Directive, Swedish ARBETSMILJÖVERKET (Work Environment Authority) introduced AFS vibrations 2005:15 to regulate the amount of vibration that an operator may be exposed to during 8 hours in a working day. Similar to the EU Directive 2002/44/EC, if the vibration exceeds 1.1 $\mathrm{m/s^2}$, the operator should stop working immediately to avoid the whole-body vibration. And especially in Sweden, the limit lower than 1.1 $\mathrm{m/s^2}$ has been required.

2.3 WHOLE BODY VIBRATION OF FORESTRY MACHINES

Whole-body vibration (WBV) refers to mechanical energy oscillations which are transferred to the body as a whole (in contrast to specific body regions), usually through a supporting system such as a seat or platform. Machines (air, land and water), machinery (e.g. those used in industry and agriculture) and industrial activities (such as piling) expose people to periodic, random and transient mechanical vibration which can interfere with comfort, activities and health.

Regular exposure to whole-body vibration over many months or years can lead to damage and back pain. The longer you are exposed and the higher the level of whole-body vibration, the greater the risks of suffering a back injury. Once you begin to suffer back pain, continued exposure to vibration is likely to make the pain worse. Prompt action to protect workers from vibration should stop the damage from getting worse. Especially for the operators of forestry machines, who have to be exposed to the vibration for a total 8 hours per day, so whole-body vibration is a serious problem, which can cause long-time pain in shoulder, neck and back. Whole-body vibration usually contains many frequencies, occurs in several directions and varies over time. According to ISO 2631-1: 1997, vibration between the range 0.1-0.5 Hz increases the risk of motion sickness.

Whole-body vibration is the reflection of ride comfort limitations of the machine, and is evaluated by the subjective feeling of the drivers or passengers sitting in the machines. The effects of mechanical vibration are determined by vibration frequency, magnitude, directions and duration time. Different people have various sensitivities to vibrations and different parts of human body resonate at different frequencies. Generally, a seated human has a horizontal vibration natural frequency at about 1~2 Hz while the vertical vibration natural frequency is 4-8 Hz. In these frequency ranges, the human body is most easily feeling the vibration, and the vibration is most easily transmitted to the body through (from the uneven terrain via tire, bogie, chassis, cabin and seat).

In the listed literature, two basic methods are mentioned for vibration evaluation, weighted RMS acceleration method and fourth power VDV method. According to ISO 2631, for vibration with crest factors below or equal to 9, the basic weighted RMS acceleration method is sufficient; while if the crest factor is larger than 9, it is better to use fourth power VDV method, which is more sensitive to peaks than the RMS acceleration method.

The Seat Effective Amplitude Transmissibility (SEAT) value is used to evaluate the seat isolation efficiency. SEAT is the ratio of the vibration experienced on top of the seat and the vibration that one would be exposed to when sitting directly on the vibrating floor (G.S. Paddan et al., 2002). SEAT value below 1 means that there is a ride comfort improvement through the seat, because the vibration on the seat is lower than that on the floor. SEAT % equal to 1 means there is no such effect at all.

Solutions to reduce the whole-body vibration for operators in the machines can be achieved in the following three ways: 1) Isolation of the vibration sources by ground improvement, careful selection of machines or machinery, correct loading, proper maintenance; 2) Improving transmission of vibration, e.g. improving suspension systems between the operator and the source of vibration; 3) Improving structures of the cab and seat or optimizing posture according to ergonomics (P.Donati, 2002). In the past few years, much research has been studied to reduce vibrations from uneven ground and improve ride comfort.

2.4 SUSPENSION SYSTEM

The suspension system is one of the most important systems in modern machines since it has great effects on vibration reduction and comfort. The suspension system is the system of springs, shock absorbers and linkage that connect a machine to its wheels (or one mechanical body to another). The suspension system not only contributes to good handling and braking performance, but also isolates the machine from ground irregularity, noise and vibration, and keep drivers and passengers feel comfortable.

Usually, a suspension system can be categorized into passive, semi-active and active suspension system types; it can also be divided into different levels such as primary suspension system (front and rear axle suspension system) and secondary suspension system (cab suspension and seat suspension).

However, according to different purposes of the machine and the number of human in machines, most forestry machines don't have suspension system due to cost reasons. With the development of modern technology and focus on ride comfort, an increasing number of manufacturers have kept an eye on improvement of ride comfort of off-road machines due to new legislations. JCB Fast-Tract has developed a full primary suspension for agricultural tractors (P. Donati, 2002), which has four link front suspensions with coil spring and damper and self-leveling for rear suspension, and anti-roll bars controlling stiffness. In this report, we focus not only on primary suspension system at front and rear axle, but also seat and cabin suspension.

2.4.1 Primary suspension system

Primary suspension refers to the suspension at front and rear axle, which serves to support the whole machine body, connects wheels with chassis, and controls vibrations from terrain irregularities to the body. Secondary suspensions include seat suspension and cab suspension, which connect other major components, such as engine, cabin and seat to the machine body (Mehdi Ahmadian, et al., 2002).

Primary suspension systems at the axles can significantly improve the ride by increasing the ratio of sprung mass to unsprung mass of the machine (P.E Boileau, et al., 1990). Different types of suspensions have been applied to modern machines, such as passive, semi-active and active suspension system. The conventional suspension is a passive suspension system with passive springs to absorb impacts and dampers to control spring motions. Semi-active suspension has air springs and switchable shock absorbers, and can only change the viscous damping coefficient of the shock absorber, and does not add energy to the suspension system. An active suspension system, of course, is the most advanced suspension system, and it can use separate actuators which can exert an independent force on the suspension to improve the riding characteristics.

Unlike cars and lorries, most all-terrain machines such as forwarders, skidders and harvesters do not have any suspension system between wheel-axles and chassis, to reduce the effects of ground roughness. Specific details about the primary suspension are discussed in Chapter 5.3.

Due to the inherent limitations of primary suspension systems, a number of efforts have been devoted toward the development of effective secondary (cab and seat) suspension systems for off-highway machines (P.E Boileau, et al., 1990), which may have higher efficiency but with a low cost.

2.4.2 Cabin suspension

Cabins on forwarders are designed to create better working environment for operators, so the vibration of cabins should be minimized to improve the ride comfort of drivers. A cab suspension system locates between the cab and the chassis, can provide operators with a stable floor and isolate vibrations from noise, chassis vibration and ground irregularity. Different from passenger cars and commercial machines, the cabs in the forestry machines have no suspension system and this means transmission of the vibration from the ground goes directly to the operator.

On Valmet 860.4, only four bushings are placed between the chassis and cabin. The bushings can provide a little effort to reduce high frequency vibration from chassis to the cabin. However, these four bushings cannot help a lot in terrain related to vibration problem, and should be modified into a more advanced suspension. Specific details about the cab suspension are discussed in Chapter 5.1.

2.4.3 Seat suspension

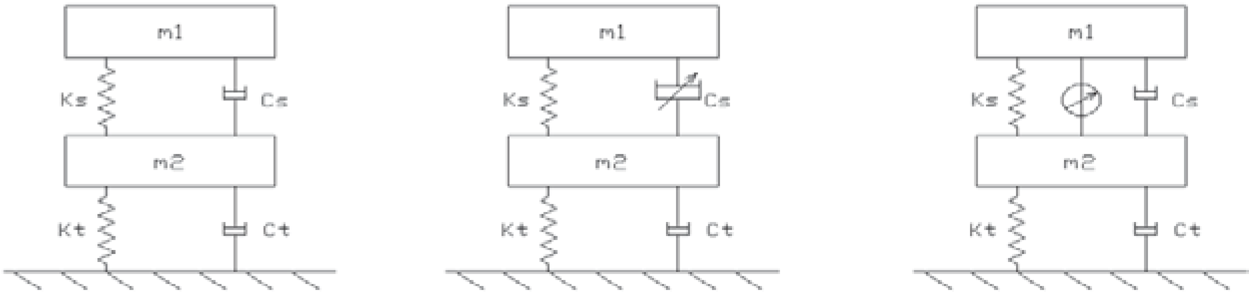
Forest operators have to sit in the seat during work for almost 8 hours per day. Vibrations from the cab floor are transmitted through seat suspension system and seat cushion, and finally arrive to the operators. Thus, suspension of the seat itself constitutes the final stage of suspension before the operator (P. Donati, 2002). In previous research studies, a lot of measurements have been taken on a wide range of seats and some measures have been taken out to minimize seat transmissibility to improve comfort.

Suitable seats have a close relationship with the structure of suspension, type of suspension, layout of the seat according to different sizes and postures of drivers. In this report, only structure and type of seat suspension system have been considered.

For the forwarder Valmet 860.4, seat suspension is the only used suspension system. Thus, necessity of a seat suspension system is obvious and research on the transmissibility of vibration input and output is motivated in order to reduce the vibration amount transmitted to the human body and significantly improve ride comfort. Specific details about the seat suspension are discussed in Chapter 5.2.

2.5 TYPES OF SUSPENSION SYSTEM

As mentioned in Chapter 2.4, three types of suspension system have been applied in the automotive industry, active suspension, semi-active suspension and passive suspension. Figure 2.2 shows different structures for suspension systems.



a) Passive suspension b) Semi-active suspension c) Active suspension

Figure 2.2.
Different structures for suspension system.

2.5.1 Active suspension system

Active suspension uses separate actuators which can exert an independent force by the suspension and have an optimal feedback control according to input and output, and enables the suspension with best damping characteristics to improve ride comfort and handling performance.

The biggest drawback of active suspension in modern industry is high cost, frequent maintenance and complicated structure. In forestry machines, it is rare to apply active suspension because of those problems. Manufacturers would like to use cheaper suspension to replace active suspension although it has a rather good performance.

2.5.2 Semi-active suspension system

The concept of semi-active suspension was proposed by D.A. Crosby and D.C. Karnopp in 1973, and Karnopp also proposed a sky-hook damping control model and its implementation. Until early 1970s, products of active suspension came into the world. However, semi-active suspension has developed much faster than active suspension in application, and more manufacturers prefer to choose semi-active suspension due to its high performance and relatively cheap price.

Semi-active suspension can be regarded as a suspension system consisting of spring and damper with variable characteristics. Although it does not have optimal control and adjustment, it can adjust damping coefficient according to optimal parameters of springs and dampers. Different from active suspension, semi-active suspension does no add energy to the suspension system, and it can only change the viscous damping coefficient of the shock absorber. With less expensive cost and less energy-consumption, research on semi-active suspension has been continued regardless of its limitations.

2.5.3 Passive suspension system

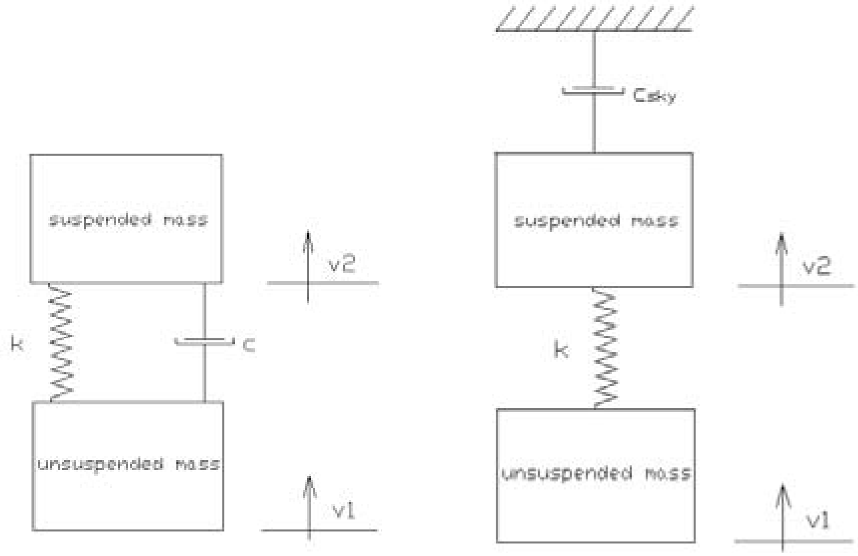
Passive suspension includes traditional springs and dampers with spring stiffness and damping coefficients which are not possible to vary according to the ground irregularity. In another way, a passive suspension system cannot be controlled externally, and does not have to add energy to the suspension system. Right now, most machines are suspended by passive suspensions system. The biggest advantage of passive suspension is that it is cheap, reliable and simple. The problem with passive suspension is the compromise between handling and ride comfort. A soft spring results in small vibration but poor handling, while a tougher spring provides higher acceleration but better handling. With the development of automotive industry, manufacturers give more emphasis on the performance improvement of passive suspension system, so semi-active and active suspension come into practice.

With the help of control system, various semi-active/active suspensions realize an improved design compromise among different vibrations modes of the machine, namely roll, pitch and yaw modes. However, the applications of these advanced suspensions are constrained by the cost, packaging, weight, reliability, and other challenges.

2.6 SKYHOOK CONTROL AND ON-OFF CONTROL

Until now, various controlling principles have been applied to suspension control system, such as skyhook control, optimal control, predictive control, fuzzy logic control, self-adaptive control, neural network control and compound control.

The skyhook damping control principle was proposed by D.C.Karnopp in 1973. In order to eliminate the resonance between the suspended mass and suspension system, the damper between the suspended mass and the base in figure 2.3.a is now connected to a reference in the sky that remains fixed in the vertical direction, shown in figure 2.3.b.



(a) Passive suspension. (b) Ideal skyhook suspension.

Figure 2.3.
Skyhook control.

It is assumed that the suspended mass is moving upwards with a positive velocity v_2. If we consider the force that is applied by the skyhook damper to the suspended mass, the damping force can be regarded as,

$$F_{sky} = kC_{sky}v_2 \tag{2.1}$$

Where, F_{sky} is the skyhook force, C_{sky} is the damping coefficient, v_2 is the velocity of suspended mass.

In the ideal skyhook control principle, the control force is determined by the feedback of suspended mass's absolute velocity, and it has a high reliability with less sensors and simpler model.

However, the skyhook control is just an ideal control principle in theory. As a result, semi-active on-off control was put forward based on skyhook control, and it adjusts the damping coefficient of damper according to a control signal, so as to adjust the magnitude of damping force. The damping force in semi-active on-off control policy can be summarized as,

$$\begin{cases} F_d = c(v_2 - v_1) & v_2(v_2 - v_1) > 0 \\ F_d = 0 & v_2(v_2 - v_1) < 0 \end{cases} \tag{2.2}$$

Where, F_d is damping force, v_1 is the vertical velocity of unsuspended mass, v_2 is the vertical velocity of suspended mass.

In this report, the typical skyhook control and semi-active on-off control principle have been applied to seat suspension and primary suspension system, providing a good result to reduce the vibration from the forwarder. Especially for primary suspension system, the damping coefficient of one damper is adjusted to control the total equivalent stiffness and damping to achieve the realization of semi-active suspension adjustment.

2.7 SOFTWARE FOR SIMULATION

SimMechanics is a toolbox of Simulink, which extends Simscape with tools for modeling three-dimensional mechanical systems with the Simulink environment. However, different from Simulink, SimMechanics does not require deriving equations prior to modeling, but builds a model composed of bodies, joints, constraints and force elements that reflect the structure of the system (MathWorks, 2010). Meanwhile, it provides visualization of the system dynamics with a 3-D animation. Models in SimMechanics are identified with mass, moments of inertia, coordinates, constraints and geometries. CAD (Computer Aided Design) models can be imported from some other CAD platforms such as SolidWorks and Pro/Engineer into SimMechanics.

SimMechanics is widely used in different fields such developing active suspensions, robotics, surgical devices, landing gear and a variety of other systems. With Simulink environment, SimMechanics can be combined with other MathWorks physical modeling products to model complex interactions in multi-domain physical systems (MathWorks, 2010). And the data produced in SimMechanics can be exported directly to WorkSpace in Matlab to process, which is very efficient. Moreover, it has the ability to convert models to C code with Real-Time Workshop.

In the previous project, some related work has been done in ADAMS to deal with primary suspension system and its dynamics, because Komatsu provided CAD models and ADAMS can take advantage of CAD software, such that parts are modeled in CAD and then implemented in ADAMS. In this project, using SimMechanics is kind of innovative way to try, and it has a good integration with Matlab and Simulink since they share the same environment, so SimMechancis becomes the main modeling tool.

3 Simulation model descriptions

3.1 SIMPLIFIED MACHINE MODEL

In the simulation, it is impossible to model every component of the real machine, so as to reduce the complexity of the model the following assumptions were made to construct the whole model:

- All structural components are considered to be rigid;
- Steering is not included;
- The operator and the seat is modeled as a simple 1-DOF lumped parameter model, whose shape is similar to cuboid;
- Tires are modeled as simple spring damper systems;
- Engine, hydraulic tank, drivetrain are simply included into front wagon, meanwhile bunks, grind and crane are simply included into rear wagon;
- Only parts or assemblies with a mass of 200kg or more are considered (except for the operator and seat).

To some extent, the above assumptions will contribute to some deflections of the simulation result off the real situation. In real situation, tire is not a simple spring-damper system, rear wagon should have separated parts such as bunks, grind and crane, whose movement will also affect the change of vibration, and some small components such as exhaust, intake installation, and pivot should contribute to the total mass of the forwarder. Considering these simplifications, the simulation model will differ from the real machine but it is anticipated that the model will capture the main behavior of the real machine.

3.1.1 General description

The whole machine model was established in Simulink/SimMechanics. In the measurement, the forwarder speed is varying along the bumped ground surface, and the average speed is 0.3 m/s during 90 seconds. In the simulation, it was impossible to control the forwarder speed changes as that in the real situation because no velocity data from tests are available, so a constant speed 0.3m/s has been assumed.

According to the test ground surface, bump shape and distances between bumps, vertical displacements of the ground surface with respect to time can be obtained, and used as the input to the whole simulation system.

3.1.1.1 Coordinate system and the origin of the forwarder

The construction of the whole forwarder in Simulink/SimMechanics is based on a global coordinate system, which is shown in figure 3.1 (Olof Karlsson, et al., 2010). Three translational axes X (red), Y (green) and Z (blue), and three rotational axes roll (yellow), pitch (purple) and yaw (white) are used, to represent the global (reference) coordinate system. The origin of the global coordinate system lies at the junction between the front wagon and rear wagon, where roll axis and rotation point between front wagon and rear wagon intersect.

Figure 3.1.
Global coordinate system of the forwarder (Olof Karlsson, et al., 2010).

3.1.1.2 Constraints of the forwarder

The initial simulation model without primary suspension system and cabin suspension system has some structure constraints, which are summarized as:

- Translational movements of eight wheels along Y-axis;
- Translational movements of four bogies along Y-axis;
- Rotational movements of four bogies around X-axis and Z-axis;
- Translational movements of two bogie axles along Y-axis;
- Translational movements of front wagon and rear wagon along Y-axis;
- Translational movement of cabin along Y-axis;
- Translational movement of seat and driver along Y-axis.

In this project, vertical displacements of seat, cabin, front wagon, rear wagon and angles of four bogies are variables mainly taken into the consideration, and hence used as the evaluation parameters in Chapter 4.

3.1.1.3 Construction of the whole forwarder

The whole forwarder has multiple subsystems such as cabin, seat, front frame, rear frame, wheels, bogies, bogie axles, hydraulic tank, engine, crane, grind, bunks, and etc. Big components such as front frame, rear frame, cabin, seat, bogies are focused in this report. Front frame and rear frame support the whole machine and connect unsprung mass and sprung mass.

The cabin is attached to the front frame with four anchor points, which consist of rubber bushings with spring and damper characteristics along three translational axes. In the cabin, the seat bottom is attached to the cabin floor with a simple seat suspension system (spring and damper system).

Four bogies are connected with eight wheels at front and rear, and two wheels comprise one pair which is installed at both sides of one bogie. The bogie can freely rotate around Y-axis and is attached to the bogie axle.

Front wagon and rear wagon are connected to each other by an articulated joint, which allows rotation about Z-axis and rotation about X-axis relative to each other. This articulated joint separates front wagon and rear wagon, which means the rotations of the front and rear wagon of the forwarder are independent.

3.1.1.4 Whole forwarder in Simulink/SimMechanics

Parameters for each subsystem such as mass, moment of inertia and coordinate system are shown in Appendix B, and have been applied to the model in Simulink/SimMechanics. Figure 3.2 shows the model of the whole forwarder in Simulink/SimMechanics, red blocks represent components with mass and moment of inertia, yellow blocks represent 4 pair of wheels, which will be explained in Chapter 3.2.3, green blocks represent spring and damper systems for the components with spring stiffness and damping coefficient, which are shown in Appendix C, and blue blocks represent connecting degrees of freedoms between two subsystems. All the data for analysis such as vertical displacements of front wagon, rear wagon, cabin and seat, and bogie angles of 4 bogies are exported to workspace in Matlab, and will be processed as explained in Chapter 4.1.

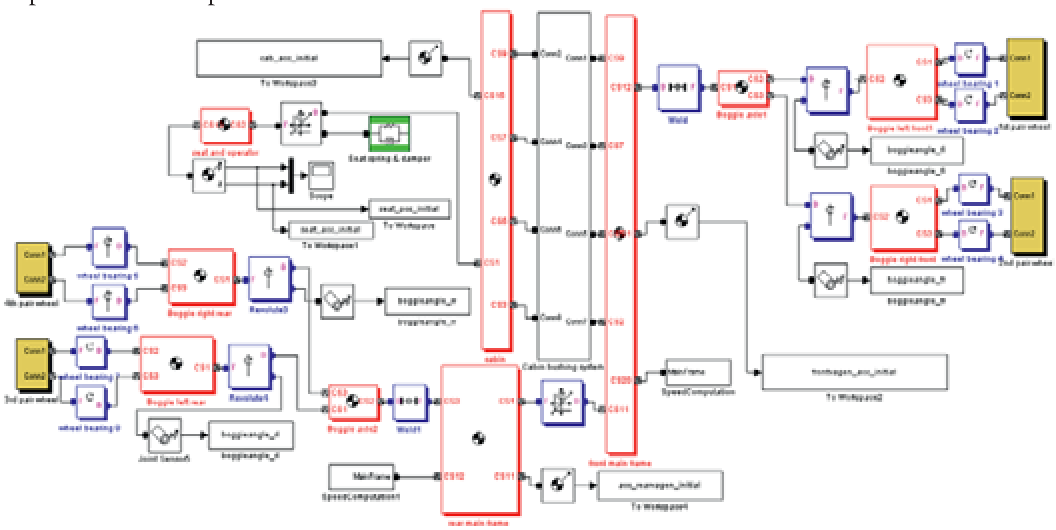


Figure 3.2.
Whole forwarder in Simulink/SimMechanics.

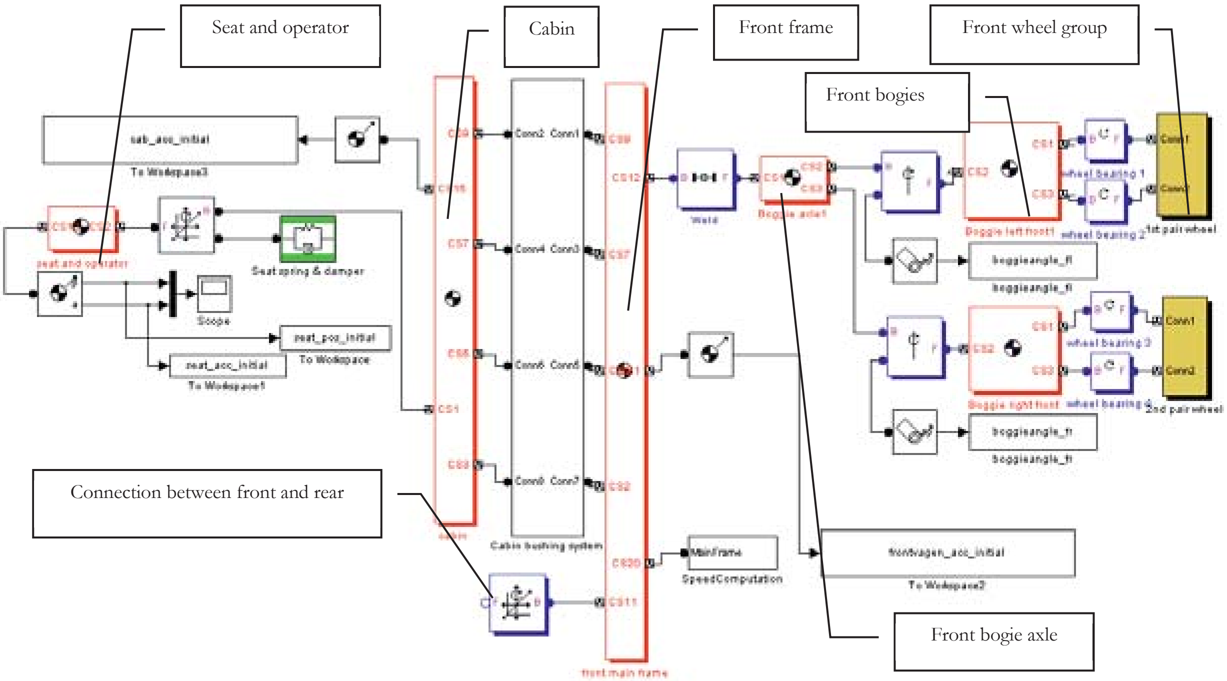


Figure 3.3.
Front wagon in SimMechanics.

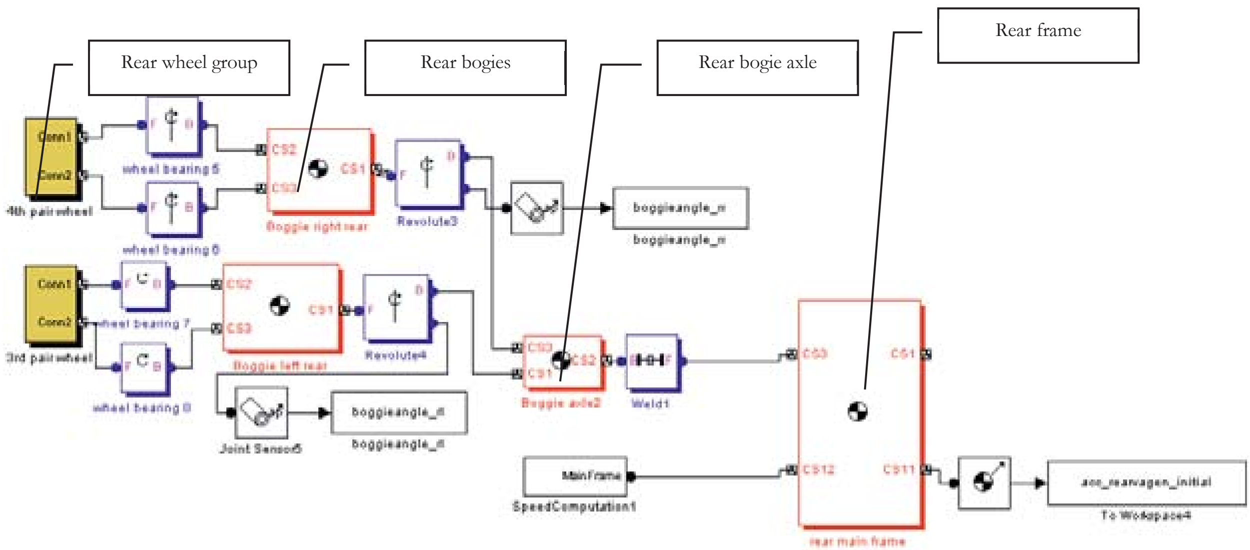


Figure 3.4.
Rear wagon in SimMechanics.

The simulation model in SimMechanics is based on the sketch of the whole forwarder, shown in figure 3.5, and the coordinate systems for each component are shown in Appendix B.

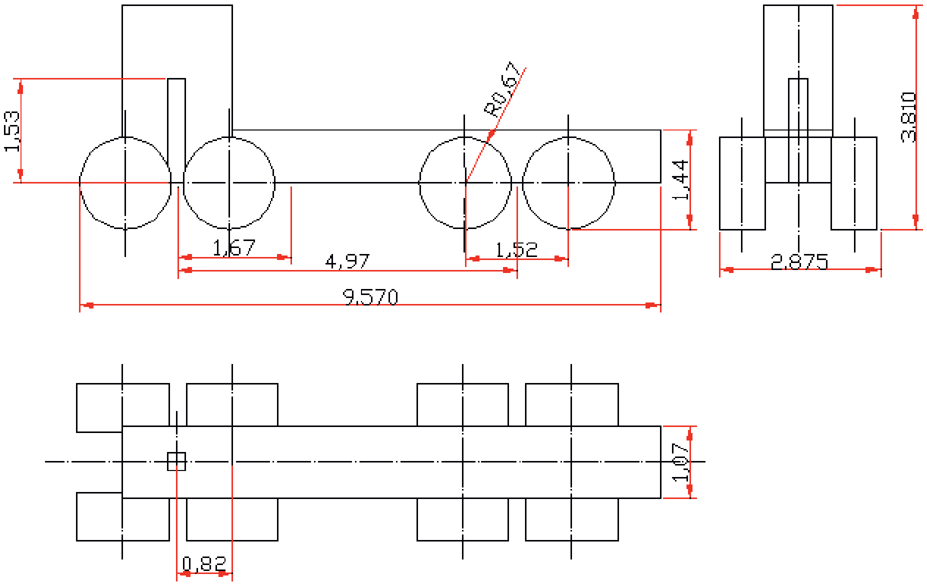


Figure 3.5.
The sketch of the whole forwarder in SimMechancis (unit: m).

A typical SimMechanics simulation runs for 90 seconds, uses Solver ode23 (stiff/Mod. Rosenbrock) with variable integration step, and provides a visualization of the whole forwarder as shown in figure 3.6. Due to that the simulation model has a lot of spring and damper systems, the Solver ode 23 is more suitable than Solver ode45 (Dormand-Prince). The problem with SimMechanics is that it usually provides variable steps instead of fixed step, which will cause some problems of sampling for data processing.

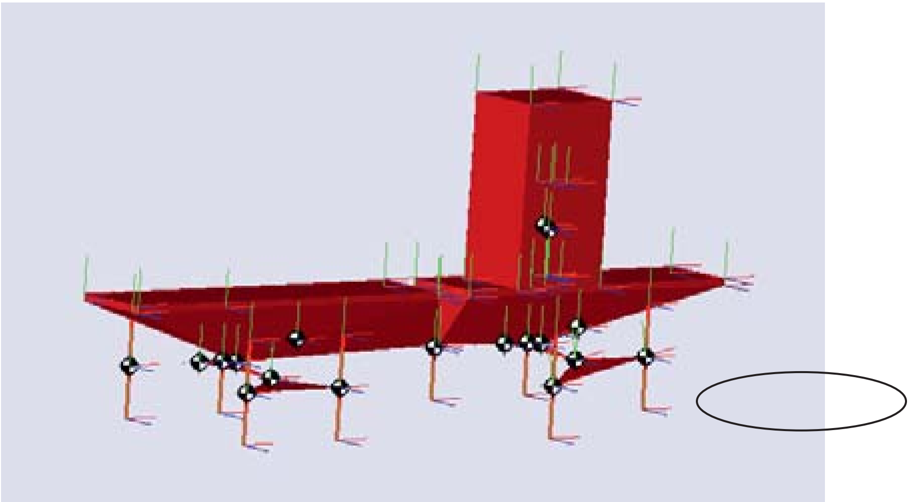


Figure 3.6.
Visualization of the whole forwarder in SimMechanics.

In figure 3.6, the front right bogie rises up towards a bump, which can be seen clearly from the circle in the visualization window. This is the advantage of SimMechanics, whose visualization displays a machine by displaying its bodies closed surfaces (convex hulls) enveloping the bodies' coordinate systems.

3.2 TERRAIN SURFACE IRREGULARITY SIMULATION

3.2.1 Test track

The measurement of a previous project was done on Skogforsk standardized test track, as shown in figure 3.7 (Skogforsk). The test track, which has two paths, is 28 m in extension. Measurement on each path of the track lasted for around 90 seconds. Specific data for the test track such as coordinates and figures are given in Appendix A Test track.

Figure 3.7.
Skogforsk standardized test ground.

3.2.2 Ground surface displacement input modeling

In the simulation, the forwarder is traveling along the test ground at a constant speed. As a result, the ground input to the forwarder is the displacement of the ground irregularity along vertical direction (Z- direction). All the eight wheels have the same ground irregularity displacement input with different timings. The time delay is dependent on the distance between different wheels on each side and the speed of the forwarder.

In order to obtain the ground surface displacement input easily, some assumptions have been drawn as follows:

- Each path is assumed as one curve, which means that the path does not have any width in Y-direction and the height of the curve is the averaged value of the original heights along Y-direction.

- The Forwarder is driven with a constant speed around 1.2 km/h (0.3 m/s). The time delay from the 1st pair wheels to the 4th pair wheels is 0s, 5s, 16.5s and 21.6s, respectively, shown in table 3.1.

- The round top (radius is about 100mm) of each obstacle is assumed to be sharp top, which is easier to model.

The flat ground between two bumps is considered as absolutely smooth and flat, while in real measurement, the flat ground may have some small rocks or other things to make it irregular.

Table 3.1.
Time delay from the 1st pair wheels to the 4th pair wheels.

Distance between wheels [mm]	Speed of the forwarder [m/s]	Time delay [s]
1 520	0.3	5
4 967	0.3	16.5
6 487	0.3	21.6

The model in Matlab of the ground surface vertical irregularity displacement with respect to time is shown as follows,

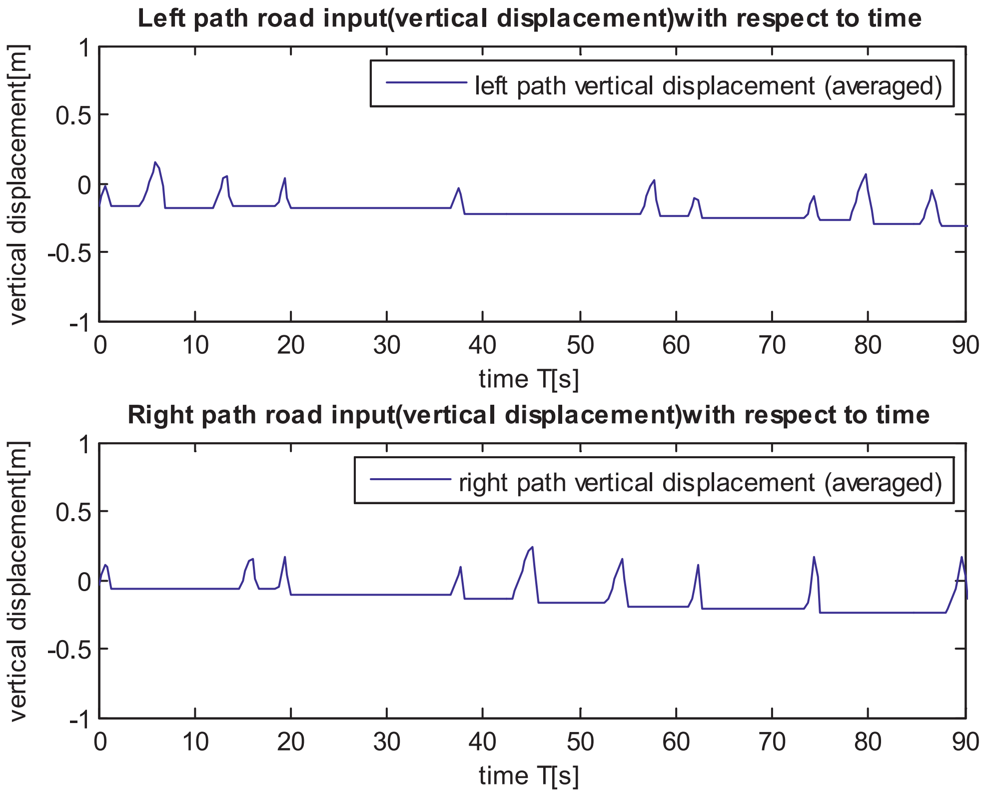


Figure 3.8.
Both path ground inputs with respect to time (1st pair wheel).

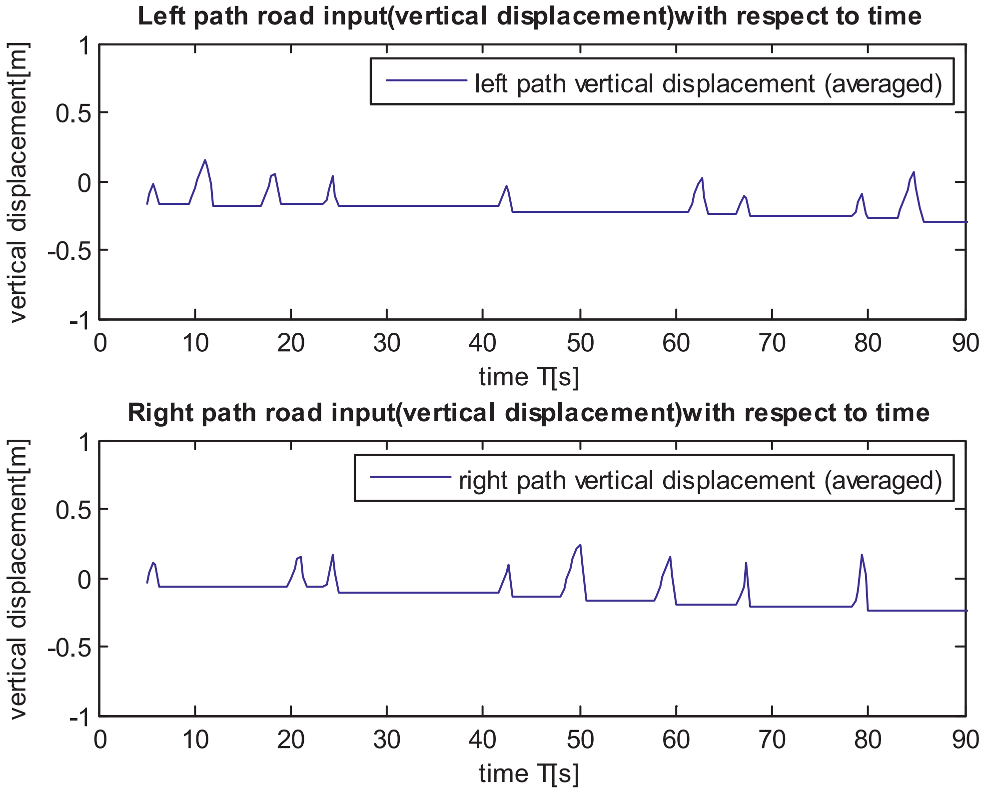


Figure 3.9.
Both path ground inputs with respect to time (2nd pair wheel)

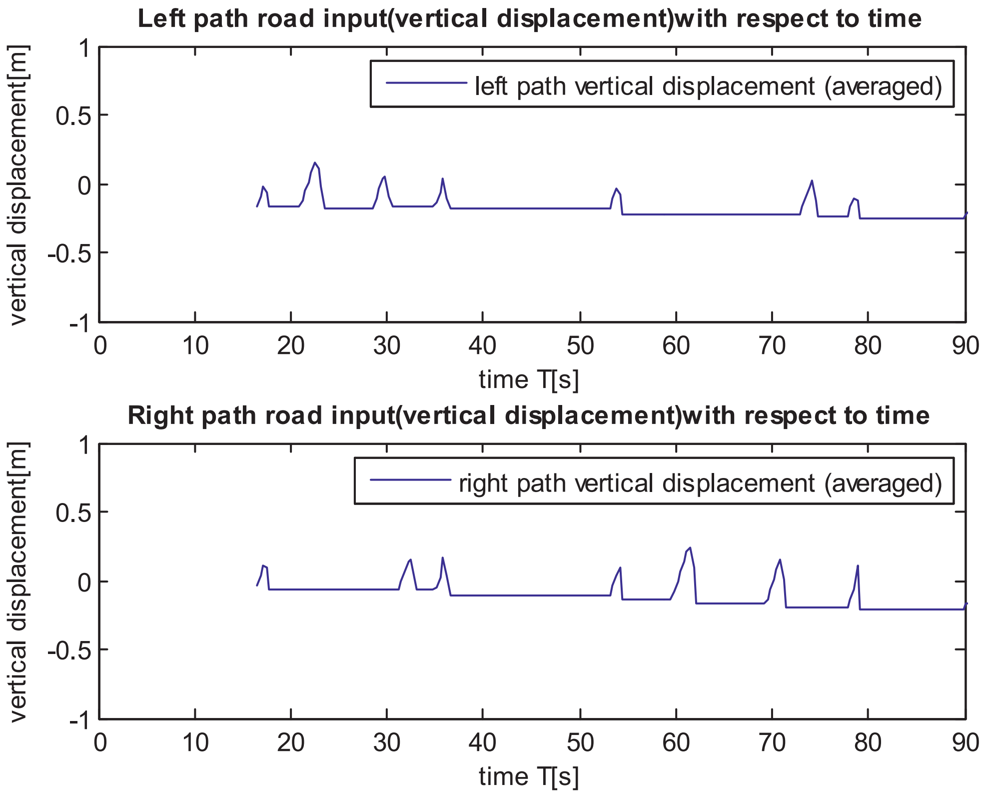


Figure 3.10.
Both path ground inputs with respect to time (3rd pair wheel).

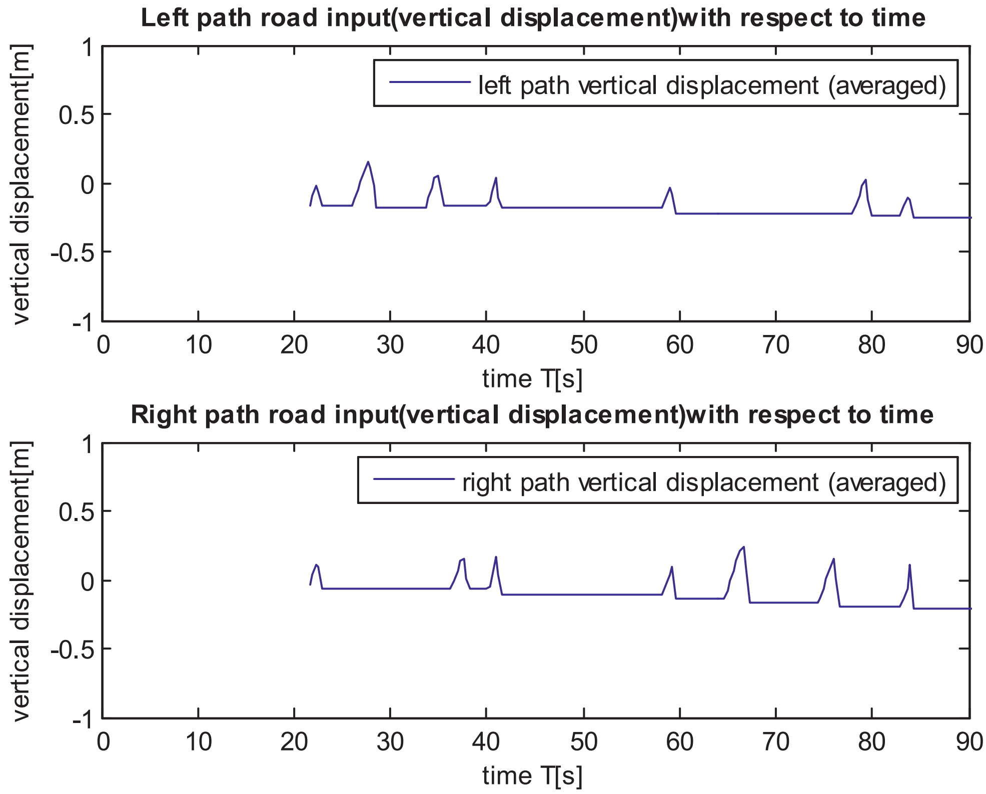


Figure 3.1.
1 Both path ground inputs with respect to time (4th pair wheel).

All the above ground input data was stored in Matlab workspace and could be loaded into Matlab/Simulink by using the block "From Workspace".

The ground profile modeled by Matlab may not be exact, which will contribute to the discrepancy between the measured and simulated results. Also, the simulation assumes a constant speed of 0.3 m/s which is a simplification. The vibration level would be influenced mostly by this assumption, while the time periods for bogies up and down will not be that consistent with those in the measurement. Some of the bogies will go up later or earlier than the time they do during the real measurement.

3.3 TIRE MODELING

All the eight wheels are assumed as simple linear spring damper systems (point contact model). Actually, a model of a tire rolling on different surfaces is very complicated. In this simulation, eight simple linear spring damper systems are applied to the whole machine system as eight tires, and the spring stiffness and damping constants are tuned according to the measurement result.

As a result, the tire model is constructed based on following assumptions:

(1) The tire always rotates with its free radius (it is 670 mm for unloaded situation in this project), which means that tire is considered as a rigid roller.

(2) The tire always has one point in contact with the ground surface during rotation.

(3) The tire elasticity is interpreted in terms of vertical stiffness and damping constant.

A number of analytical modeling techniques have been applied to tires, which can simulate the enveloping of tires on uneven grounds, and are thus well suited to ride-models (Zegallar, 1998), some of the models are illustrated in figure 3.12.

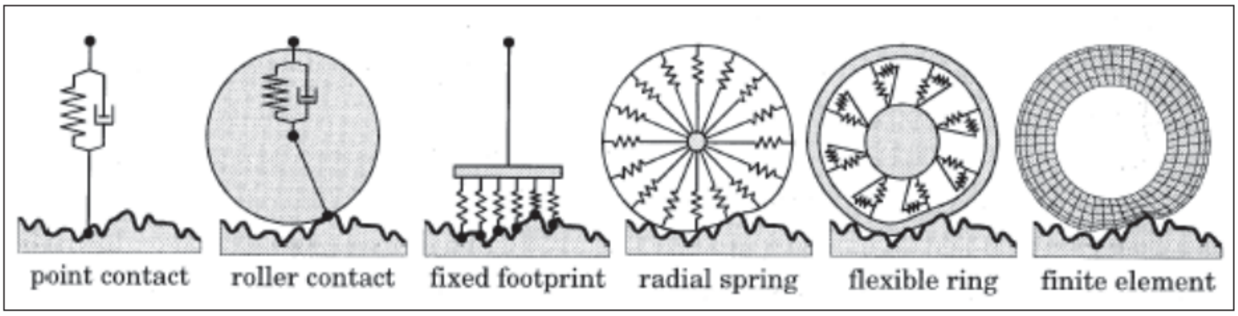


Figure 3.1.
2 Different tire models (Zegelaar,1998).

In this simulation, the point contact model has been used, which consists of a spring and a damper parallel to each other and oriented straight down from the center of the wheel tied to a point on the surface. Spring stiffness corresponds to the tire and pressurized air elasticity while damping constant corresponds to the internal losses in the rubber and the air. In this simulation, only vertical force is concerned, which would deform the wheel in vertical direction. In a real case, a horizontal force is resulting from the slope in the point contact (Löfgren, 1992), also this is disregarded in the model used here.

Compared to other models, the point contact model is the simplest one, which makes the whole complicated machine system simpler. But it also leads to a rather coarse approximation of the tire behavior. If the machine model in this simulation is static instead of movable along the surface, the ground contact length does not have to be taken into account. However, point-contact model can, to some extent, reflects the basic characteristic of a tire. In order to simplify the whole model, point-contact tire model is the most suitable.

It is, of course, a rather large simplification with such a simple-structure tire model neglecting the nonlinearity of a real tire. However, in the previous projects, the approximate ranges of the vertical tire stiffness and tire damping are obtained, and they clearly have great impact on vertical behavior of the tire. And it is evident that there is a notable interaction between the vertical behavior of the tire and the rest of the machine (Giancarlo Genta, et al., 2009).

As a result, tuning the vertical tire stiffness and tire damping according to the measurement result (bogie angles, vertical accelerations of front wagon, rear wagon, cab and seat) is a possible way to approximate the real behavior of the tire.

In the initial condition (forwarder without cabin suspension system and primary suspension system), the whole forwarder is considered as a conventional passive suspension system, which employs a spring and damper (tire) between the chassis and wheel assembly. Following is a quarter machine dual mass model (Robert Bosch GmbH, 1993) to explain how it works and the ground surface irregularity can be applied to wheels as the input of the system.

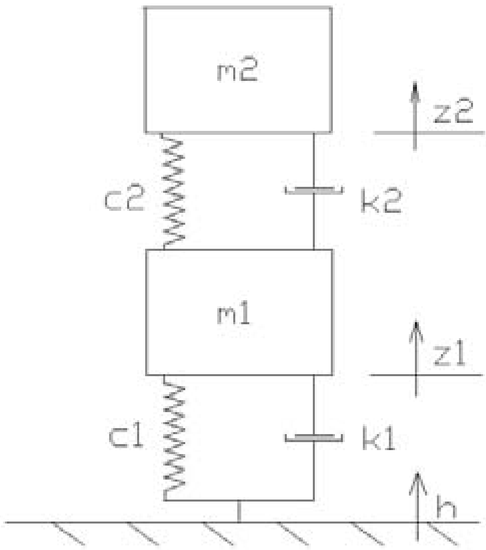


Figure 3.13.
Dual-mass quarter machine model (Robert Bosch GmbH, 1993).

In figure 3.13, m_1, m_2 are the masses of wheel assembly and chassis, k_1, k_2 are spring stiffness of tire and suspension system between chassis and wheels, c_1, c_2 are damping coefficients of tire and suspension system between chassis and wheels, h, z_1, and z_2 are ground surface irregularity (vertical displacement), vertical displacements of wheel and vertical displacement of chassis, respectively.

Here, the equation of motion of the wheel is:

$$\ddot{z}_1 = \frac{1}{m_1}[k_2(z_2 - z_1) + c_2(\dot{z}_2 - \dot{z}_1) - k_1(z_1 - h) - c_1(\dot{z}_1 - \dot{h})] \qquad (3.1)$$

So the force from ground surface irregularity through tire spring and damper system F can be concluded as,

$$F = -k_1(z_1 - h) - c_1(\dot{z}_1 - \dot{h}) \qquad (3.2)$$

Applying this force in Simulink/SimMechanics, is shown in figure 3.14,

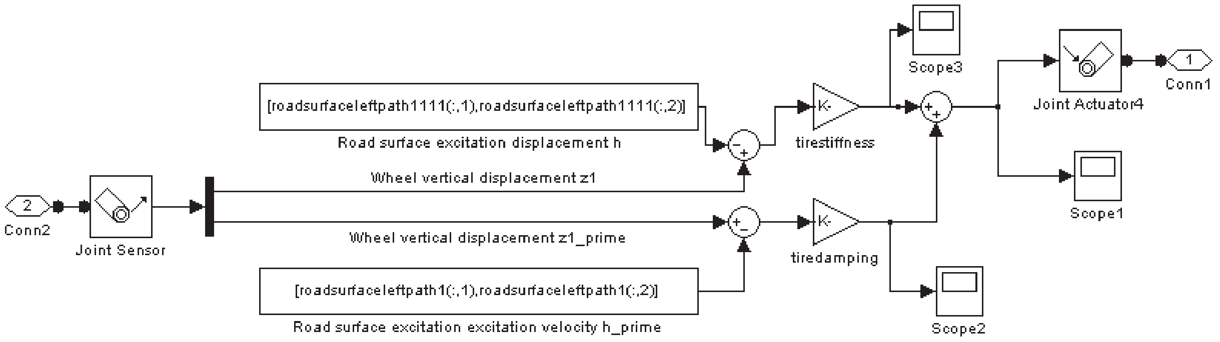


Figure 3.14.
Force from ground surface irregularity through tire spring and damper system implemented in SimMechanics.

The below figure is based on the front left tire behavior. From the block "scope", the separate tire spring and damper forces can be extracted and plotted.

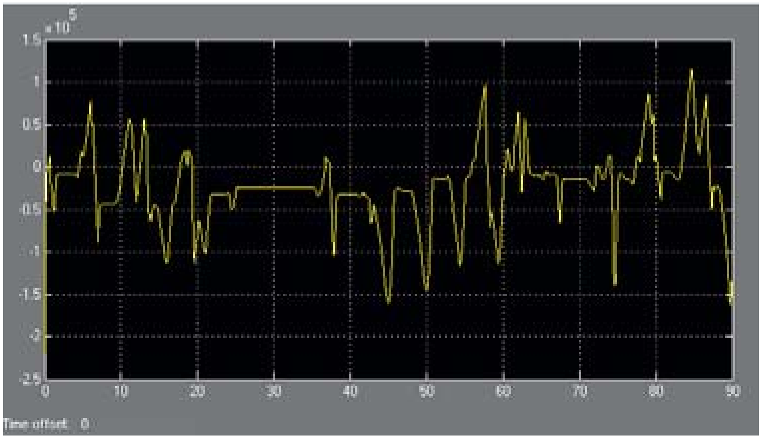


Figure 3.15.
Force that goes through 'spring' in front left tire.

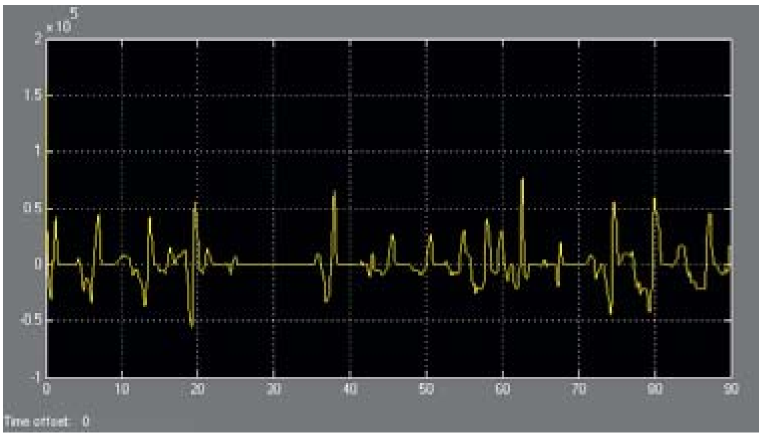


Figure 3.16.
Force that goes through 'damper' in front left tire.

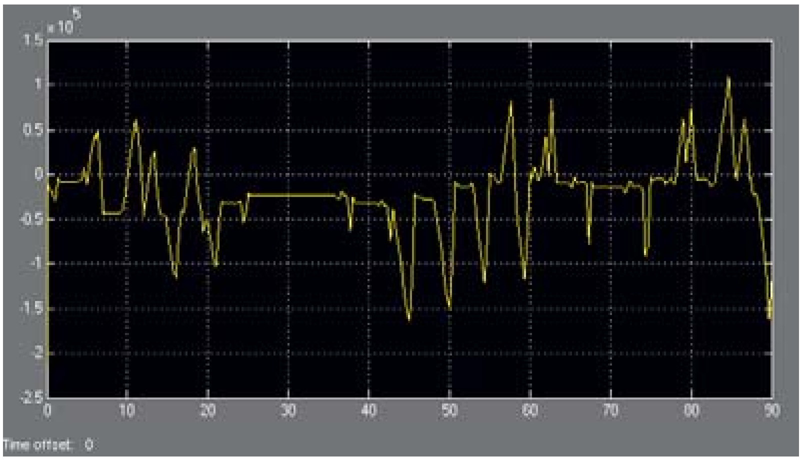


Figure 3.17.
Total force that goes through front left tire

3.4 BOGIE

On Valmet 860.4, the new features include the new refined Comfort bogie, which takes driving comfort and job satisfaction to new heights. The bogie design provides exceptional maneuverability up and down steep inclines. It also has same great maneuverability on soft and wet terrain, where the limited ground impact will astound. The new bogie improves climbing ability and maintains ride comfort when driving over obstacles (Komatsu Forest, 2010).

There are four bogies on this forwarder, and the front two comprise a pair, whose center is articulated with front bogie shaft across the forwarder in the plane perpendicular to the direction of the speed without any suspension system or attenuation system, so do the rear two bogies. As shown in figure 3.18 (Olof Karlsson, et al., 2010), two wheels are placed symmetrically with the bogie in the same manner. Bogies and bogie shafts serve some purposes such as supporting the forwarder body and minimizing the generation of track irregularities. But no stiffness and damping between bogie and wheels contributes to the mainframe without suspension system, which will cause big vibration problems. The behaviour of bogie angles for 4 bogies will be compared with those from measurement in Chapter 4.2.3.

Figure 3.18.
Bogie.

4 Validation of simulation model

4.1 DATA PROCESSING

All the data from measurement and simulation are processed in Matlab. Data processing consists of low pass filtering and Fourier Transform.

4.1.1 Low pass filtering

Since the most serious resonant frequencies for Z-direction (vertical) vibration lie between 0.5 and 8 Hz, while the most serious resonant frequency range for X- and Y- direction vibration is between 0.5 and 2 Hz. So in the processing, the cut-off frequency for low pass filtering is chosen as 10 Hz, which eliminates all the irrelevant frequencies above 10 Hz. The sampling frequency for the measurement is set as 2000 Hz. However, the sampling for simulation in SimMechanics is not regular, so irregular sampling frequency should be considered. In the processing, a 1-D interpolation method (see Matlab code below) has been used to make the simulation data with the same time interval Δt, and change the irregular sampling into an equivalent regular sampling, which is better and easier for data processing. Δt for simulation is set as the same as that in the measurement $\Delta t = 0.0005$ s, so the sampling frequency $F_s = \frac{1}{\Delta t} = \frac{1}{0.0005} = 2000$ Hz.

```
x_initial = tout; % time for simulation

y_s = seat_acc_initial.signals.values(:,2); %
acceleration data from simulation

xi = 0:0.0005:90;% set new time interval as 0.0005s

y_s_new = interp1(x_initial,y_s,xi);% simulation
data after interpolation by using 1-D interpolation
(interp1) in matlab
```

A Butterworth filter is used as the low pass filter, and the low pass filtering process is shown below:

```
Fs = 2000;% Sampling frequency

fNorm = 10/(Fs/2);% Normalized cutoff frequency for
Butterworth filter

[b,a] = butter(5,fNorm,'low')% Butterworth filter

fy_s = filter(b,a,y_s);% low pass filtering
```

4.1.2 Fourier Transform

Acceleration representation in the frequency domain intrigues more interest than that in time domain for the particular purpose, so fast Fourier Transform has been applied to transform the simulation results from time domain to frequency domain. The fast Fourier Transform process in terms of Matlab code is shown below. Here, a hamming window is used to reduce the leak. And variable *y_fft* is the final result after Fourier transformation and adding hamming window.

```
Fs=2000;% Sampling frequency in Hz
hamming=0.5; %Low hamming in Hz
L=size(In,1);
NFFT = 2^nextpow2(L);
hammingwindow=((2*hamming)/Fs);
Ham=tukeywin(NFFT, hammingwindow);
y = (fft(In,NFFT))/L;
y=Ham.*y;
x = (Fs/2)*linspace(0,1,NFFT/2+1);
y_fft =2*abs(y(1:0.5*NFFT+1));
```

An example result of seat vertical acceleration from measurement after low pass filtering and Fourier transform is shown in figure 4.1,

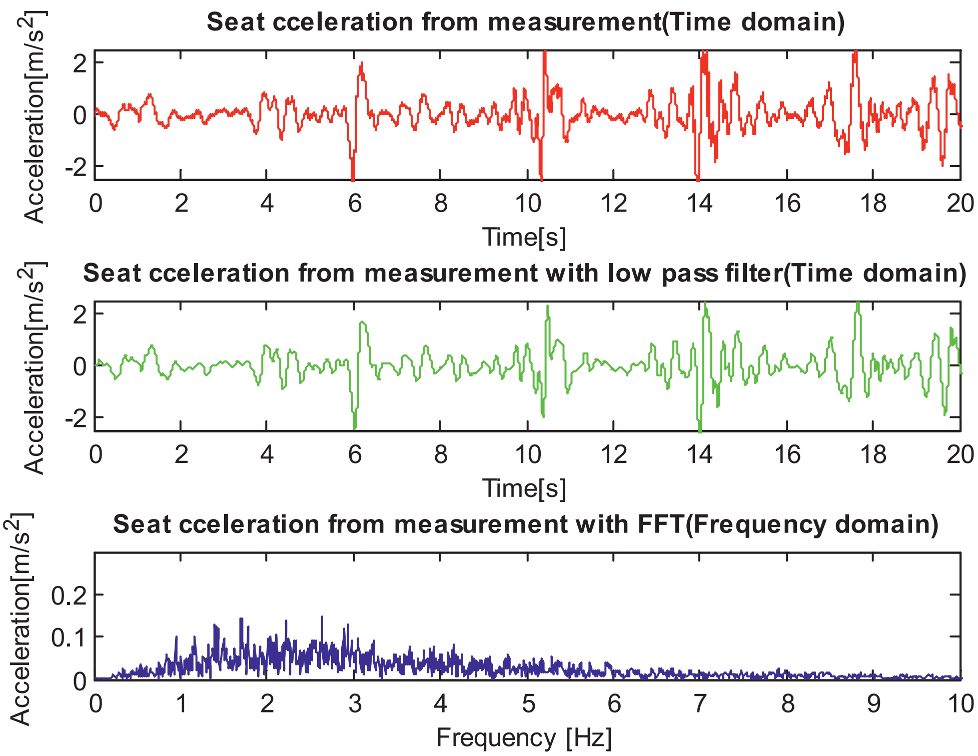


Figure 4.1.
Seat vertical acceleration processing.

4.1.3 Power spectral density (PSD)

The power spectral density describes how the energy of a signal is distributed along the frequency. Figure 4.2 shows the PSD comparison of seat vertical accelerations in the simulation and in the measurement, respectively. We can see that the frequency for peak value of PSD in the measurement is above 4 Hz, which for vertical vibrations lies in the most sensitive frequency for a human, and means that the energy is concentrated close to 4 Hz, which is not good for ride comfort. Because of the interpolation in data processing and the discrepancy of the simulation, the simulation result and the measurement result are not perfectly matched.

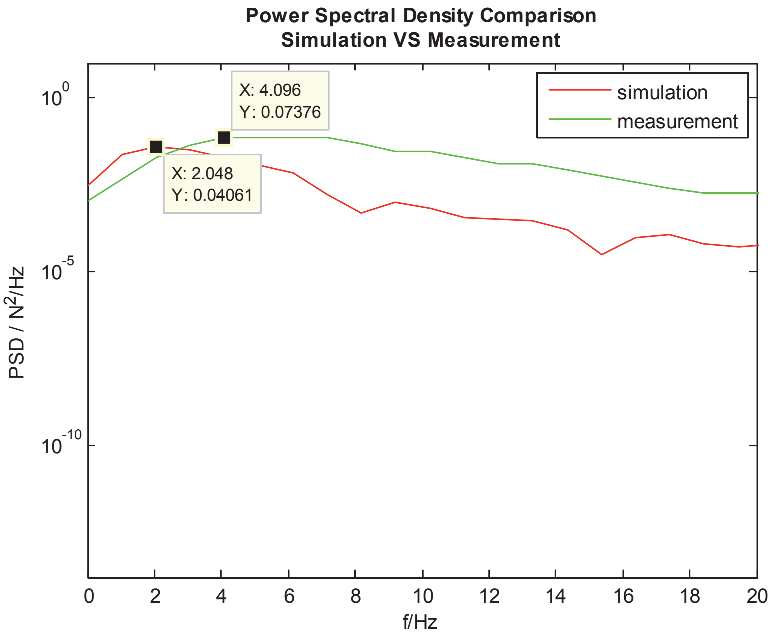


Figure 4.2.
Power spectral density comparison for seat vertical acceleration (simulation VS measurement)

4.2 VALIDATION OF SIMULATION MODEL

Firstly, in order to validate the simulation model, it is critical that the ground irregularities imported to the model are close to those of the measurement. However, it is difficult to determine whether the modeled ground irregularities are correct or not. Because of 4 main assumptions in Chapter 3.2.2.2 and unknown starting and stopping place of the forwarder, modeled ground irregularities may differ from those in the measurement. Moreover, two paths of the ground has different numbers, layouts and shapes of bumps, so it is hard to tell when there is a bump just according to the relationship between the acceleration and time.

4.2.1 Evaluation method

4.2.1.1 Three basic methods

As mentioned in Chapter 2.3, in ISO 2631-1:1997, some evaluation methods are mentioned to evaluate the effects of vibration on the health. If ratio of peak acceleration to RMS value (crest factor) is less than 9, frequency weighted RMS value can be used as the basic evaluation method. Otherwise, additional evaluation method such as VDV (vibration dose value) is the alternative. BS 6841-1987 and the European Union Directive 2002/44/EC also accept RMS value and VDV as the basic evaluation methods. Moreover, the European Union Directive and AFS vibrations 2005:15 also regulate the amount of vibration that an operator may be exposed to during 8 hours in a working day, which is called A (8) value. So in this report, weighted RMS value, VDV and A (8) are the three evaluation methods used.

According to ISO 2631-1:1997, the weighted RMS value is defined as:

$$a_w = [\frac{1}{T}\int_0^T a_w^2(t)dt]^{\frac{1}{2}} \qquad (4.1)$$

Where, $a_w(t)$ is the weighted acceleration (translational or rotational) as a function of time (time history), in m/s^2 or rad/s^2, respectively; T is the duration of the measurement, in seconds.

The fourth power vibration dose value (VDV) (ISO 2631-1:1997) is defined as:

$$VDV = \left\{\int_0^T [a_w(t)]^4 dt\right\}^{\frac{1}{4}} \qquad (4.2)$$

Where, $a_w(t)$ is the instantaneous frequency-weighted acceleration; T is the duration of measurement.

A (8) value is defined as (J.C. Kirstein, 2005),

$$\mathrm{A(8)} = \sqrt{\frac{1}{\mathrm{f_s T_8}} \sum_{\mathrm{i=0}}^{\mathrm{N-1}} \mathrm{a_w^2(i)}} \qquad (4.3)$$

Where, $\mathrm{a_w(i)}$ is $\mathrm{i^{th}}$ sample of the weighted acceleration, N is the total number of samples, $\mathrm{f_s}$ is the sampling rate, $\mathrm{T_8}$ is eight hours (28800s).

4.2.1.2 Frequency weighting

Vibration frequency content determines different effects of vibration to health, comfort and perception. As a result, different axes of vibration require different frequency weightings. According to ISO 2631-1:1997, two principal frequency weightings are given in figure 4.3,

$\mathrm{W_k}$ is used for the Z-direction and for vertical recumbent direction (except head);

$\mathrm{W_d}$ is used for the x and y directions and for horizontal recumbent direction.

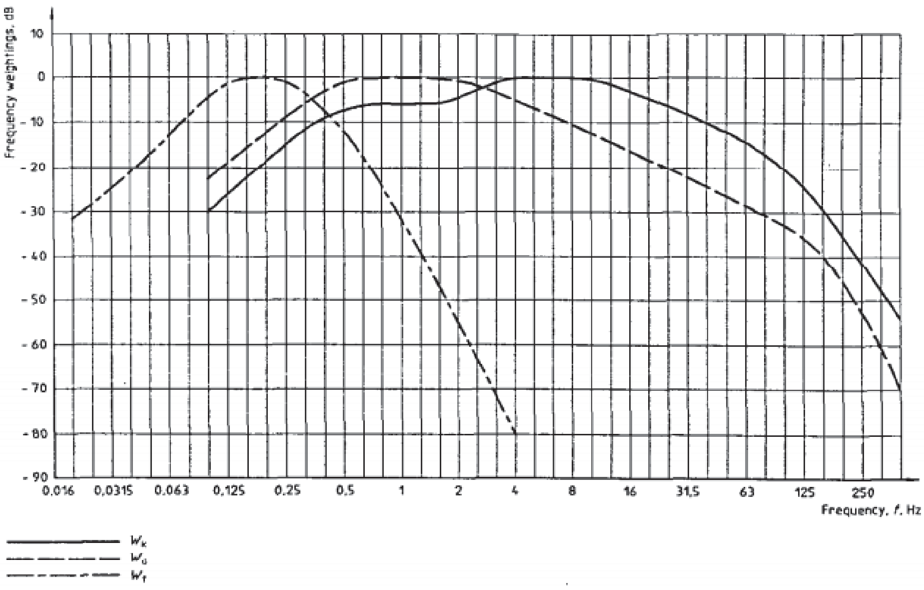


Figure 4.3.
Frequency weighting curves for principle weightings (ISO 2631-1:1997).

In this project, only the vertical direction is focused on, so frequency weighting $W_k = 1.0$ according to ISO 2631-1:1997.

Based on the model established in SimMechanics, some results can be developed for the validation. The validation of simulation model includes various parts, such as bogie angle validation, front wagon vertical acceleration, rear wagon vertical acceleration, cabin vertical acceleration and seat vertical acceleration validations.

4.2.2 Validation of vertical accelerations

Figure 4.4-figure 4.7 show the vertical accelerations of seat, cabin, front wagon and rear wagon in frequency domain, respectively. Red lines represent the data from simulation, while the green lines represent the measurement data.

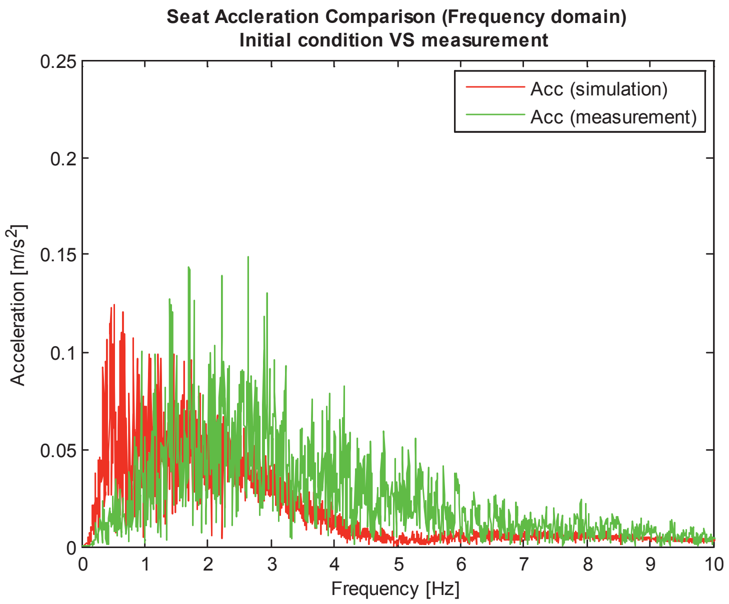


Figure 4.4.
Seat acceleration comparison (frequency domain).

Initial condition VS measurement

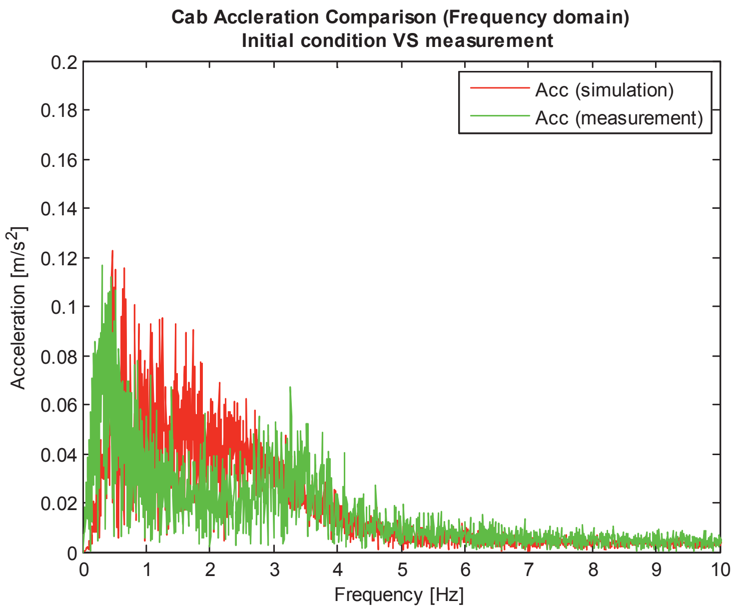


Figure 4.5.
Cabin acceleration comparison (frequency domain). Initial condition VS measurement.

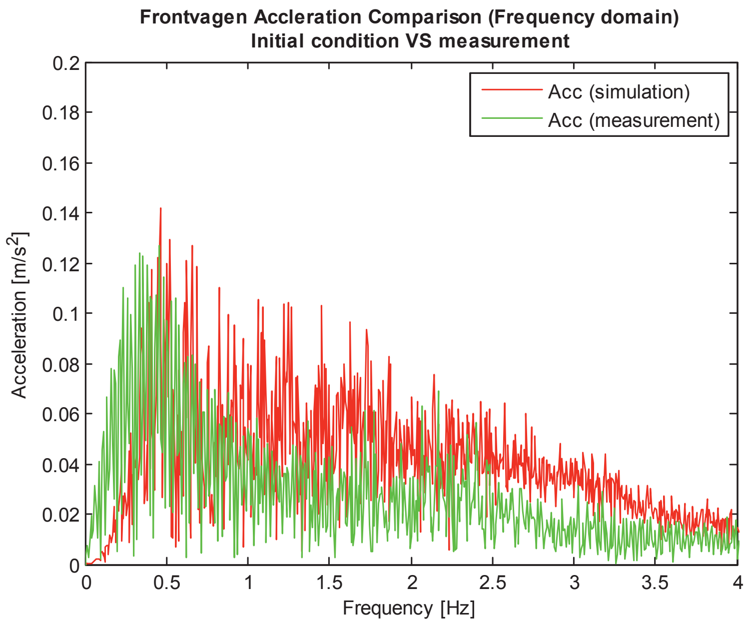


Figure 4.6.
Front wagon acceleration comparison (frequency domain).

Initial condition VS measurement

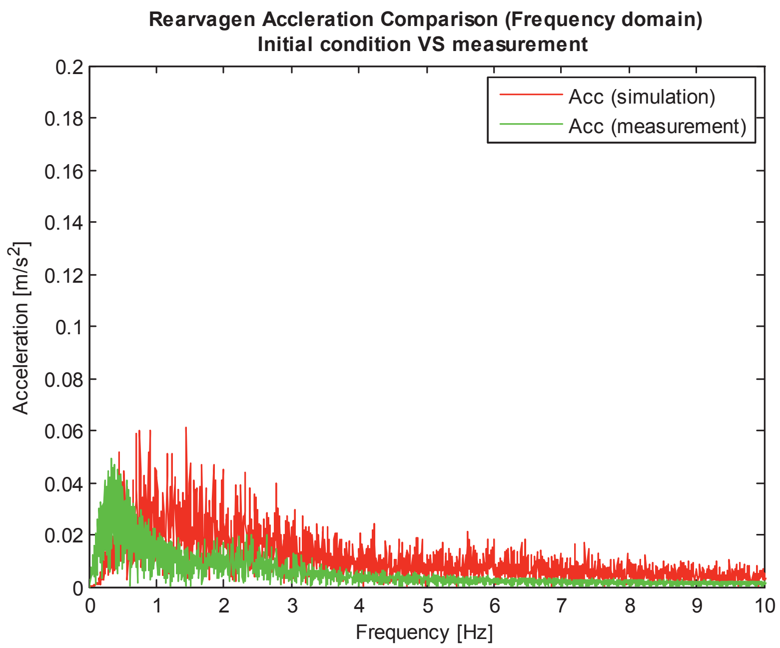


Figure 4.7.
Rear wagon acceleration comparison (frequency domain). Initial condition VS measurement.

From the above 4 figures, it is easy to tell that the vertical acceleration of front wagon is the biggest because of the complex configuration for front suspended mass. Compared to front wagon, rear wagon does not have that high acceleration due to empty load. And for the seat vertical acceleration, the biggest amplitude in the simulation and the biggest amplitude in the measurement do not appear in the same frequency range. In the measurement, the frequency is around 3 Hz, which is in the most sensitive range for vertical acceleration, while in the simulation, the frequency is around 1 Hz.

Table 4.1.
RMS, VDV and A (8) validation.

	Seat		Cabin		Front wagon		Rear wagon	
	Measurement	Simulation	Measurement	Simulation	Measurement	Simulation	Measurement	Simulation
RMS $[m/s^2]$	0.7439	0.6442	0.6947	0.6201	1.0005	0.6832	0.6975	0.3210
VDV $[m/s^{1.75}]$	1.5119	2.1628	1.0775	2.1462	1.0038	2.4155	1.0729	0.9715
A(8) $[m/s^2]$	0.0416	0.0360	0.0389	0.0347	0.0561	0.0382	0.0391	0.0179

RMS, VDV and A (8) values for comparison between simulation and measurement are shown in table 4.1. Because rear wagon is empty-loaded and all the separate components such as crane, bunks and grind are not modeled in the simulation, the vibration level for rear wagon is not as high as the measurement data, even around half of the value in measurement. However, it is OK to see that the seat acceleration and cabin acceleration in the simulation are almost close to those in the measurement. The front wagon supports more components than rear wagon both in the measurement and in the simulation, so it has the biggest vibration level, which means that more vibration will be transmitted into the cabin and the seat. The seat suspension under the seat is unknown in this project, but the seat vertical acceleration in the measurement is bigger than cabin floor vertical acceleration. This indicates that the seat suspension could be improved.

Displacements of each component are recorded in figure 4.8.

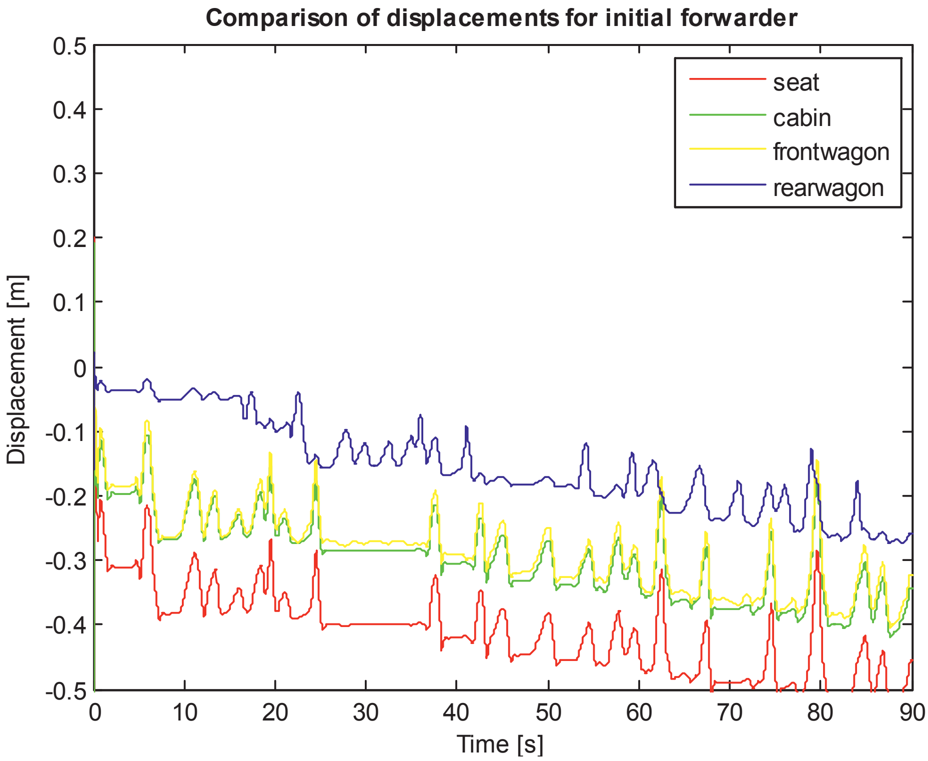


Figure 4.8.
Comparison of displacements for initial forwarder.

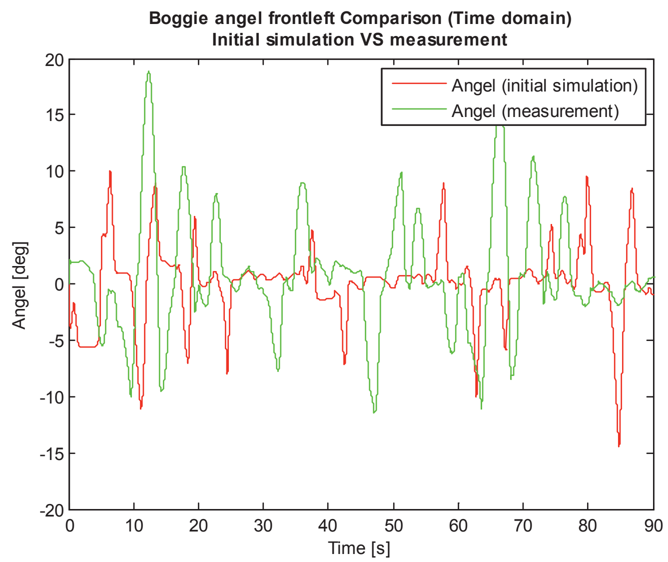
4.2.3 Validation of bogie angles

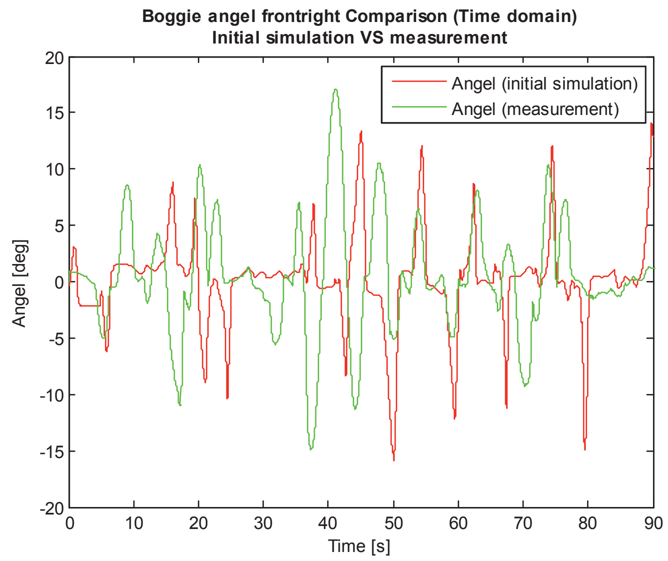
Left and right path of the ground have different layouts, numbers and shapes of bumps, so with a constant speed of 0.3 m/s, the approximate time for going through each bump can be calculated, shown in table 4.2. Different types of bumps are shown in Appendix A.

Table 4.2.
Time for passing bumps on both paths.

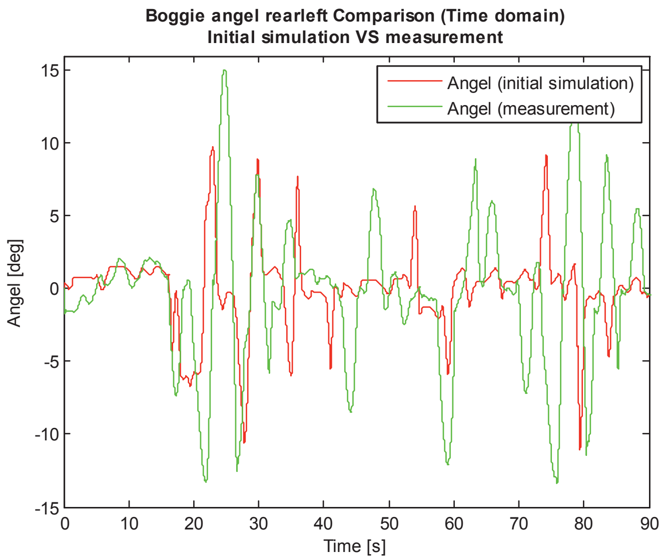
Number of bumps	Left Path		Right path	
	Time[s]	Obstacle type	Time[s]	Obstacle type
1	0-1.22	Type 1	0-1.22	Type 1
2	4.50-6.98	Type 3	14.70-16.55	Type 2
3	12.00-13.85	Type 2	18.57-19.79	Type 1
4	18.57-19.79	Type 1	36.67-37.89	Type 1
5	36.67-37.89	Type 1	43.07-45.54	Type 3
6	56.40-58.25	Type 2	52.83-54.69	Type 2
7	61.37-62.59	Type 1	61.37-52.59	Type 1
8	73.50-74.42	Type 1	73.50-74.72	Type 1
9	78.03-80.51	Type 3	88.20-90.05	Type 2
10	85.5-87.35	Type 2	92.06-93.29	Type 1
11	92.07-93.29	Type 1	–	–

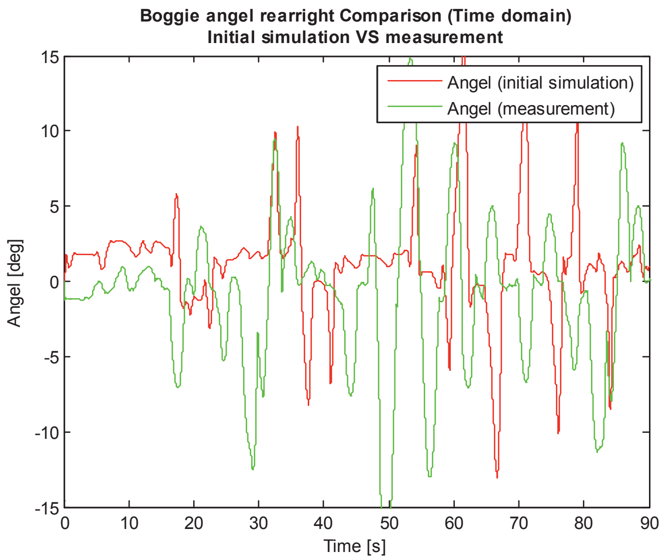
Angles of 4 bogies are recorded with respect to time, shown in figure 4.9. Red lines represent angles in the simulation, while green lines represent angles in the measurement. It is obvious to see that the shapes of the bogie angle look similar. Constant driving velocity 0.3 m/s is assumed for the simulation, which is hard to control in the practical situation, so we can see in figure 4.9, there are time differences between red lines and green lines. However, during some ranges such as between 50 s-80 s for front right bogie, between 0–40 s for rear left bogie and 0-18 s for rear right bogie, they match really well. Moreover, in real situation, the flat ground is relatively flat with small rocks, while the flat ground in the simulation is absolutely smooth, which does not provide any irregularity to the system. Because of these assumptions and simplifications, following results are good enough to verify the correct configuration of the simulation model.





a) Bogie angle (front left) b) Bogie angle (front right)





c) Bogie angle (rear left)   d) Bogie angle (rear right)

Figure 4.9.
Validation of bogie angles

4.3 CASE STUDY

In order to figure out the structure of the simulation model more clearly, a case established, in which the forwarder is static with the first front right wheel going over a bump with the following shape (all data are in mm):

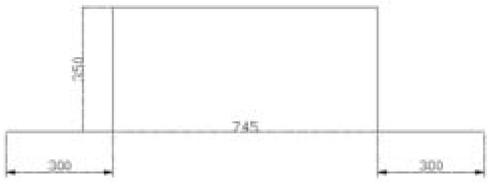


Figure 4.10.
Bump dimensions in case study.

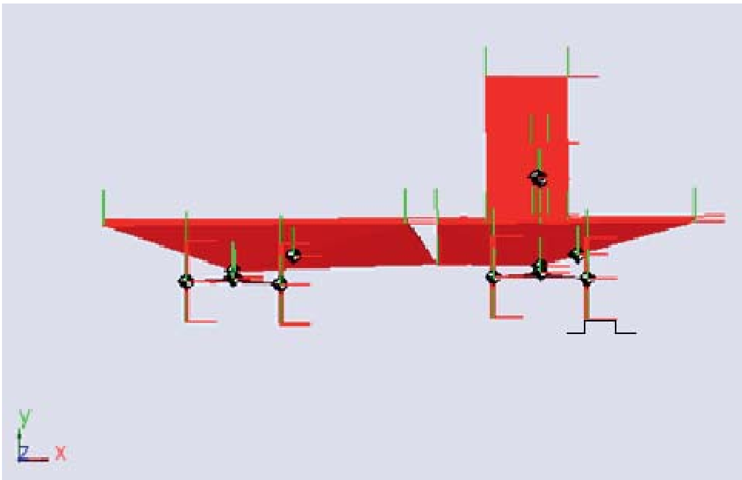


Figure 4.11.
Sketch of the case study.

Figure 4.11 shows the sketch of the case study. Running the simulation for only 5 seconds, we can get the roll of the seat as shown in figure 4.12.

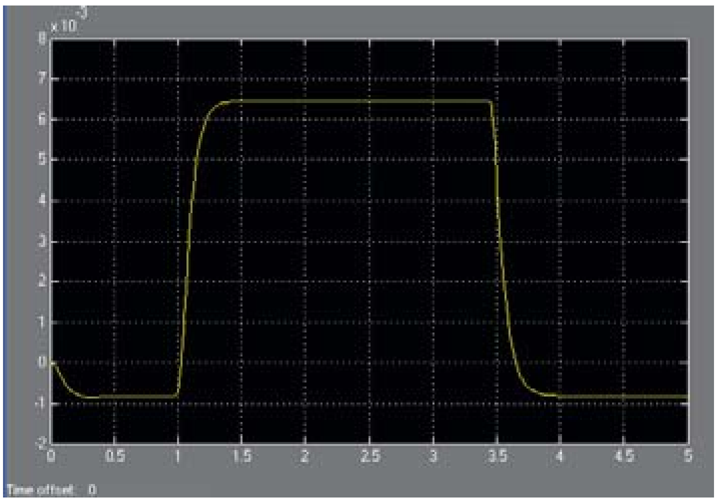


Figure 4.12.
Roll of the seat in case study.

In case study, we focus on the displacements of different components to see the relationship of them.

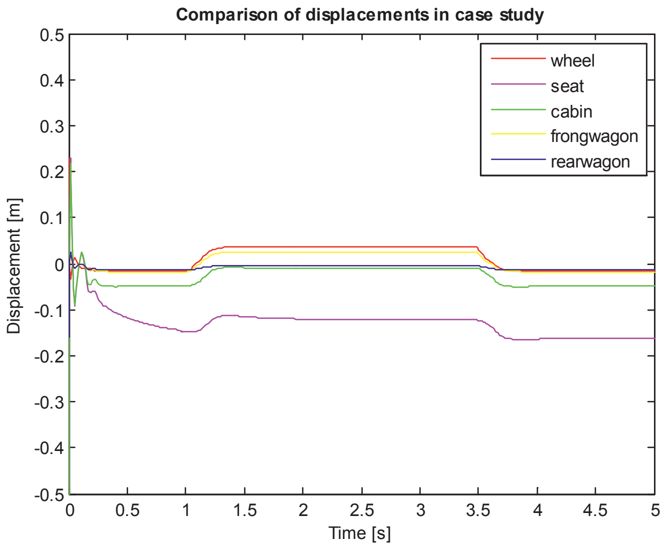


Figure 4.13.
Comparison of displacements in case study.

From figure 4.13, we can see the comparison of displacements of each component relative to the origin of the system in case study. Between 0 and 1 second, the simulation may not be stable at the beginning, and from 1 second, it starts to move from the first bump. The displacement of the seat relative to the origin is the biggest. The displacement of the front wagon and that of the cabin are in the opposite directions. Compare the front wagon with the rear wagon, because the forwarder is empty-loaded, so the displacement of the rear wagon is much smaller than that of the front wagon. Generally, the displacements of components change with the shape change of the bump.

In Chapter 6, after inserting the suspension system, the effect of the suspension system to this case will be shown.

4.4 SIMMECHANICS KNOWLEDGE IN MODELING

The toolbox of Matlab/Simulink-SimMechanics is used as the main tool to model the whole forwarder in this project. It can be used to model three-dimensional mechanical systems within the Simulink environment. Instead of deriving and programming equations, you can use this multibody simulation tool to build a model composed of bodies, joints, constraints, and force elements that reflects the structure of the system (MathWorks, 2010). For vehicle systems, it can easily model the suspension system and other components. And it can also combine other MathWorks physical modeling products to model complex interactions in multidomain physical systems (MathWorks, 2010).

The advantages and disadvantages of SimMechanics in modeling are listed below:

Advantages:

1. Construct model directly instead of deriving and programming equations;

   Unlike Simuilnk, in SimMechancis the user does not need to derive and program the equations of the motion for each component in the system, just build the model with components, joints, constraints and force elements directly. So in this project, to construct the whole forwarder, the most important thing is to figure out the mass, the moment of inertia and the coordinates of each component, and then make the connection between two subsystems clear. For example, the connection between the wheel and the bogie is the revolute joint around Y-axis between them, so we can simply use the block “revolute joint” to connect the component “wheel” and the component “bogie”. As a result, if the model is very complicated, like the Valmet 860.4 in this project, the critical problem for modeling is to make sure the physical properties of the components, and the connection between different components are accurate.

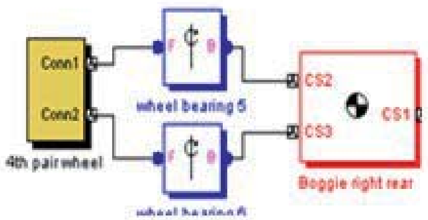


Figure 4.14.
Revolute joint between bogie and wheel.

2. It is easy to make use of other modeling products within Simulink environment;

   There are various toolboxes under Simulink, and because they share the same environment, it is very convenient to use them together to achieve the same goal. For example, in this project, I modeled the whole forwarder in SimMechanics and ground surface irregularity in Matlab, and also established different suspension models in Simulink, and finally did the data processing in Matlab, and I found out that those combinations and transfer are done without any problems. In another way, if we use other modeling software such as ADAMS, we have to output all the data into Matlab and this process will take a lot of efforts and time.

3. Visualization and animation of mechanical system dynamics;

   In this project, a visualization and animation of the whole forwarder have been made to make others to understand the motion of the forwarder directly, as shown in figure 3.5.

4. It is convenient to “measure” physical values in the simulation;

   In this project, we need to capture bogie angle, the vertical acceleration and displacement of each component, spring and damping force, etc. It is very convenient to use the sensor blocks “body sensor” or “joint sensor” to get the data you want. And then store those data we get from the sensors in Matlab Workspace for doing the data processing.

Some aspects of using SimMechanics in this project:

1. The shape of components is not that exact;

   The components in SimMechancis are defined by mass, center of gravity, moment of inertia and the coordinate system. If one of the parameters is not exact, the shape of the component is not exact accordingly. It is seen from the visualization of the whole forwarder, the shape of front wagon and rear wagon as the real one;

2. Tire model in terms of simple spring and damper system is not exact;

   In this project, only simple spring and damper system with constant spring stiffness and damping coefficient is used. Only linearity is displayed instead of nonlinearity. Actually, in SimMechanics, nonlinear springs can be modeled by using Simulink look-up tables;

3. Variable step in SimMechanics causes problem for data processing;

   The primary step option in SimMechanics is variable instead of fixed step in a normal case, which may cause some problem for data processing. Variable step results in variable time interval and sampling frequency along the time. In data processing, I have to use interpolation to change variable time interval into constant time interval so as to get a constant sampling frequency, making data processing much easier in Matlab. However, to some extent, this processing is another element contributing to the inexact validation of the model.

4. It is difficult to identify and resolve modeling flaws that may appear in the simulation;

   When modeling the whole forwarder in SimMechanics, the connecting degrees of freedom between two subsystems should be clear enough to define the connection, otherwise simulation will stop because of the redundant constraints, which means that some constraints can restrict what another constraint is already restricting.

Suggestions:

1) CAD models in SolidWorks, Autodesk Inventor or Pro/Engineer are needed to do better animations;
2) Make sure more accurate physical properties of each component of the simulated system are obtained;
3) Carefully figure out that the connecting degrees of freedom between any two subsystems are correct;
4) Be familiar with SimMechanics and Simulink;
5) Try to model the system starting from the easiest and most basic components to make sure the system model is correct step by step, e.g. from the interaction between the ground surface and the tire, then from the wheel to the bogie, then from the bogie to the front frame and rear frame, then from front frame to the cabin, finally from the cabin to the seat and the operator.

5 Suspension systems

In this chapter, different levels and types of suspension systems have been modeled and inserted into the whole forwarder. Because of the limited time, those suspension systems are cited directly from some published papers, and they are modeled in Simulink, and then inserted into SimMechanics to do the simulation.

5.1 CABIN SUSPENSION SYSTEM

Four bushings under the cabin are not enough to reduce the vibration from frame to the cabin, and a semi-active hydro pneumatic cabin suspension has been modeled according to the paper "Improvement of vibrational comfort on agricultural machines by passive and semi-active cabin suspensions". This type of suspension can be used to improve operator comfort inside the cabin of a combine harvester.

5.1.1 Structure of cabin suspension system

In our case, the similar suspension system has been applied to Forwarder Valmet 860.4 to reduce the vibration from main frame and the uneven ground. The structure of the suspension system is shown in figure 5.1. It is consisted of a hydraulic cylinder, a connecting rod, a piston, two valves with variable volume and two nitrogen bulbs.

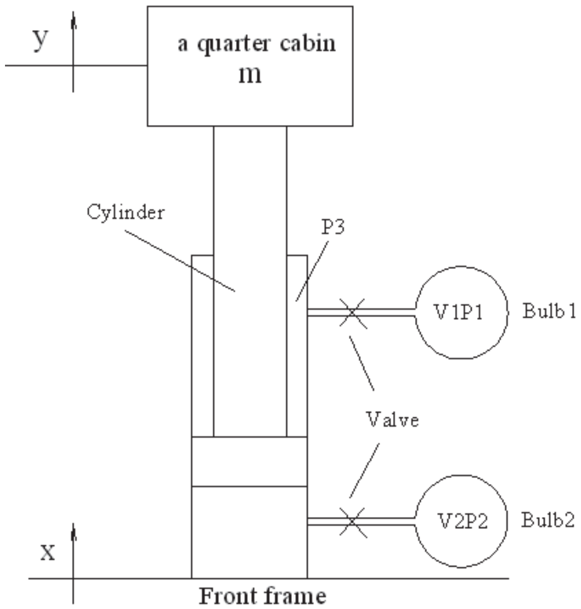


Figure 5.1.
Structure of the cabin suspension (K. Deprez et al., 2005).

The effect of variable damping is achieved by the motion of the piston, the compress of the gas and the opening of the valve. To obtain a low damping rate, the motion of the piston compresses a large volume of gas in two separate bulbs 1 and 2. Meanwhile, a high damping rate can be achieved by closing the valve, and a small gas volume is compressed into bulb 2.

The parameters for this system are listed as below:

Table 5.1.
Parameters for cabin hydro pneumatic suspension system.

Components	Parameters	Value	Unit
Quarter Cabin	Mass (m)	275	Kg
Cylinder	Diameter	32	mm
	Area (A_{cyl})	8.0425×10^{-4}	m
	Pressure (p^3)	3.5	MPa
Rod	Diameter	18	mm
	Area (A_{rod})	2.5447×10^{-4}	m
Valve	Diameter (d)	10	mm
Bulb	Volume	1.5	L
	Pressure (p^1)	1.624	MPa
	Pressure (p^2)	1.624	MPa
Others	Fs	200	N
	Fc	100	N
	Fv	2	Ns/m
	$\dot{x}_s$	0.003	m/s

The equation of motion for a quarter of cabin m is shown in following equation (K. Deprez et al., 2005).

$$m\ddot{y} = P_2 A_{cyl} - mg - P_3\left(A_{cyl} - A_{rod}\right) - F_w \tag{5.1}$$

Where, $P_2, P_3, A_{cyl}, A_{rod}$ and m are shown in the table 5.1, $\ddot{y}$ is vertical acceleration of the front frame, F_w is the friction force acting between piston and cylinder.

Then nitrogen bulbs compress and expand according to the ideal gas law in engineering applications.

$$PV = RT \tag{5.2}$$

Where, R is the universal gas constant, $R = 8.314472\ J \cdot K^{-1} \cdot mol^{-1}$

T is the absolute temperature, T=20+273=293K

There is a relationship between P_1 and P_3 by using following equation (K. Deprez et al., 2005).

$$\left(A_{cyl} - A_{rod}\right)(\dot{y} - \dot{x}) \cong \frac{\pi d^2}{4}\sqrt{p_1 - p_3} \tag{5.3}$$

And the friction force F_w satisfies following equation (K. Deprez et al., 2005).

$$F_w = F_c + (F_s - F_c)e^{-\left(\dot{y} - \frac{\dot{x}}{\dot{x}_s}\right)^2} + F_v(\dot{y} - \dot{x}) \tag{5.4}$$

Where, F_c is the minimum Coulomb friction, F_s is the level of static friction, F_v is the viscous friction term, and $\dot{x}_s$ is an emprical paramter to regulate the transition between static friction and Coulomb friction.

This semi-active suspension system is controlled by semi-active control law, which is based on skyhook control principle. In this law, the semi-active part of the damping force is proportional to the relative velocity between input and output and proportional to the absolute output velocity (K. Deprez et al., 2005).

Thus, the damping force is given by (K. Deprez et al., 2005).

$$F_D = \begin{cases} -\beta(\dot{y} - \dot{x}) - \beta_{SA}|\dot{y}|(\dot{y} - \dot{x}) & \text{if } \dot{y}(\dot{y} - \dot{x}) > 0, \\ -\beta(\dot{y} - \dot{x}) & \text{if } \dot{y}(\dot{y} - \dot{x}) < 0, \end{cases} \tag{5.5}$$

Where, F_D is the damping force, β is the damping coefficient of the passive part of the damping force and β_{SA} is the damping coefficient of the semi-active part of the damping force. In this report, $\beta = 11000\ \mathrm{N \cdot s/m}$, $\beta_{SA} = 5000\ \mathrm{N \cdot s/m}$.

5.1.2 Cabin suspension system in Simulink

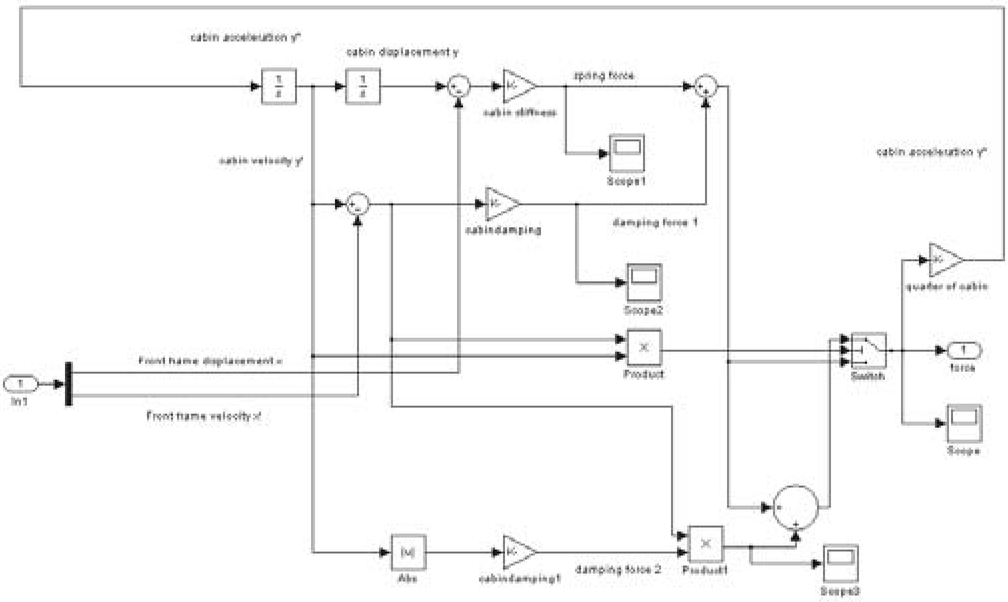
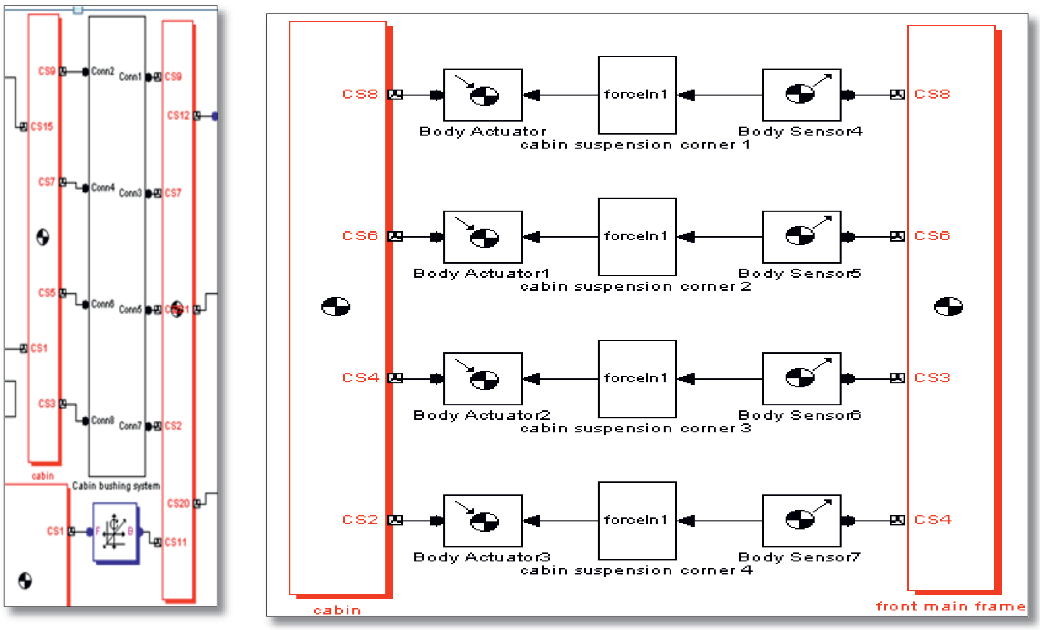


Figure 5.2.
Cabin suspension in Simulink.

This suspension shown in figure 5.2 is only a quarter part of the whole cabin suspension system, so four suspension like this are installed at the coordinates [792.318mm,722.959mm, -531.937mm], [2136.36mm, 722.959mm, – 531.937mm], [792.318mm, 722.959mm, 531.937mm], [2136.36mm, 722.959mm, 531.937mm] symmetrically, shown in figure 5.3. Insert this hydro pneumatic cabin suspension system into the real forwarder, input of the system is the vertical displacement (x) and velocity of front main frame (x') while the output is the vertical suspension force applied to the cabin.

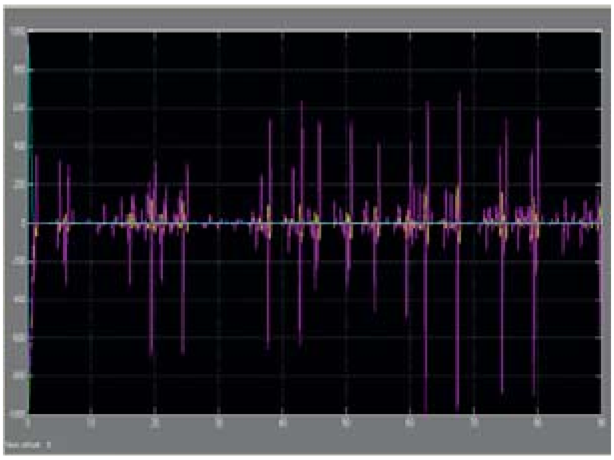


a) Previous cabin model in SimMechancis.

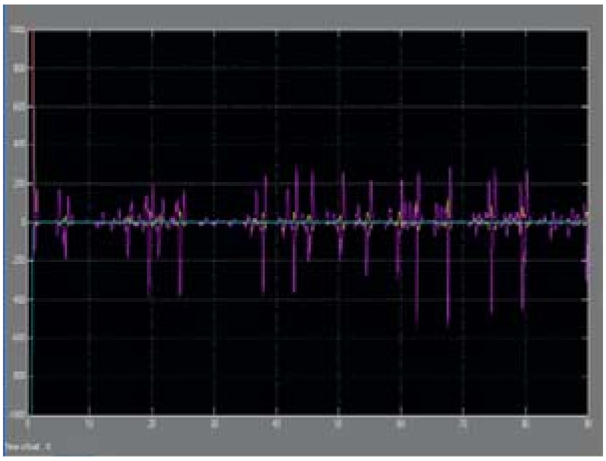
b) Cabin simulink model in SimMechanics.

Figure 5.3.
Cabin suspension model in the whole forwarder in SimMechancis.

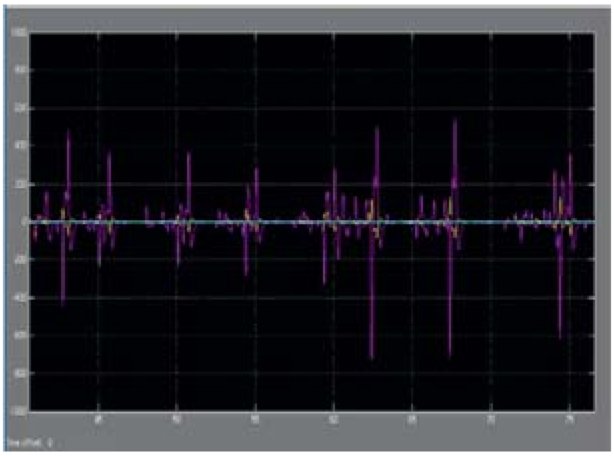
In the simulation, the total force from the spring and damper, the spring force, the passive part of the damping force and the semi-active part of the damping force are shown in figure 5.4. Magenta lines represent forces in vertical direction.



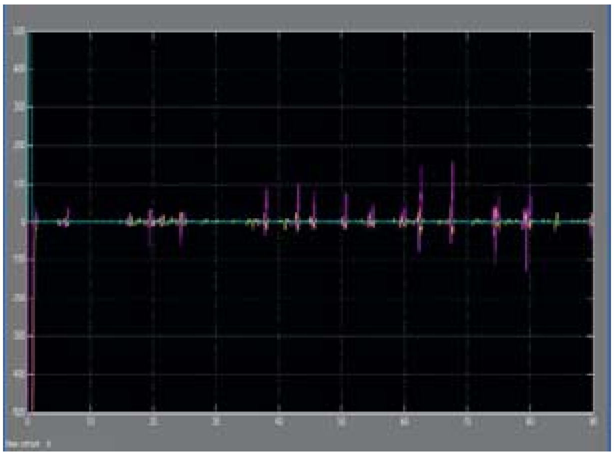
a) Total force from spring and damper.



b) Spring force.



c) Damping force (passive part)



d) Damping force (semi-active part)

Figure 5.4.
Cabin suspension force.

The comparison result between initial forwarder and forwarder with semi-active cabin suspension system will be shown in Chapter 6.1.

5.2 SEAT SUSPENSION SYSTEM

Different types of seat suspension have been applied to off-ground machines, such as compact mechanical suspension, non-compact mechanical suspension, pneumatic suspension, electro-pneumatic active suspension, and electro-hydraulic active suspension (P.Donati, 2002).

5.2.1 Structure of seat suspension system

Right now, no specific information about seat suspension can be provided to us, so just according to the space between seat and cabin floor, a scissor seat suspension with an air spring has been modeled.

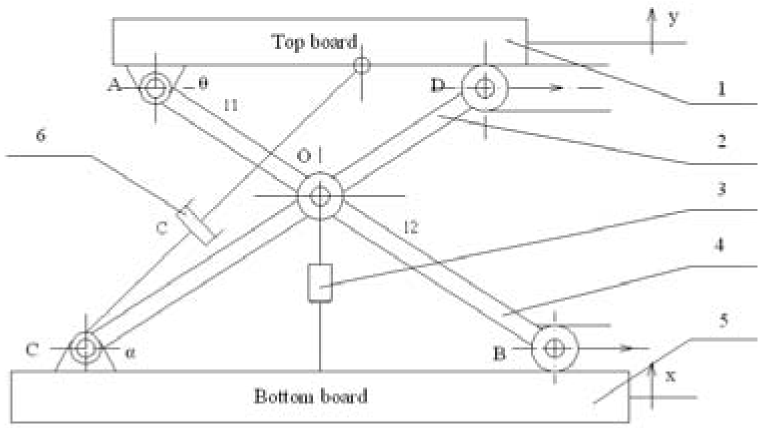


Figure 5.5.
Schematic drawings of seat suspension system (Minna Wang, 2008). 1; seat suspension top board. 2, 4; scissor linkage. 3; air spring, 6; seat suspension bottom boarddamper

Figure 5.5 (Minna Wang, 2008) is the schematic structure of seat suspension system. The top board 1 and bottom board 5 are connected through scissor 2 and 4 which are hinged at O. Scissor 2 and the bottom board are hinged at point C, and point D can slide in horizontal direction in the top board; In the same way, scissor 4 and the top board are hinged at point A, and point B can slide horizontally in the bottom board. The top and bottom of air spring 3 are connected with the point O and bottom board, respectively. This linkage system provides necessary height adjustment for the seat suspension, and the limiting factor is available room. This is a conventional scissor linkage, which still remains the popular one for off-ground machine seats (P.E. Boileau, et al., 1990). The characteristic of the air spring is nonlinear, whose stiffness characteristic is shown in figure 5.6. The damper is installed between the top board and bottom board, with a damping coefficient c. Specific parameters of this seat suspension system are shown in table 5.2.

Table 5.2.
Parameters of seat suspension system.

Parameters	Value	Explanation
m	98	75% of operator mass + top board mass
l_1[m]	0.102	Length of scissor linkage 4
l_2[m]	0.143	Length of scissor linkage 2
L [m]	0.245	Total length of scissor linkage
f_d [/]	0.02	Sliding friction coefficient
θ [i]	21.6	Shown in figure 5.5
α [.]	11.2	Shown in figure 5.5
c [kN·s/m]	8.5	Damping coefficient
c_s [kN·s/m]	5.7385	Equivalent damping coefficient

The selection of air spring is determined by the mass of seat and its position in the seat suspension. In this case, the equivalent suspended mass is equal to 75% of driver mass plus top board mass, which is approximately 98 kg. Some theoretical results imply that force applied to the air spring should be twice of equivalent mass, and air spring can afford a load of more than 190 kg.

According to the special requirements of air spring, ASNC2-1-1 Sleeve type air spring from Numatics has been chosen as the air spring in this seat suspension system. Because a constant volume air spring is not acceptable for seats, since it cannot achieve the satisfactory characteristics of stroke and natural frequency (P.E. Boileau, et al., 1990), ASNC2-1-1 has a variable volume, which can provide variable spring stiffness and natural frequency. The relationship between spring height and load is shown in figure 5.6. X-axis represents the spring height, left Y-axis represents variable volumes, and right Y-axis represents load force. The solid line represents the load changes with respect to different spring height, and the dash line represents the variable volumes changes with respect to different spring height.

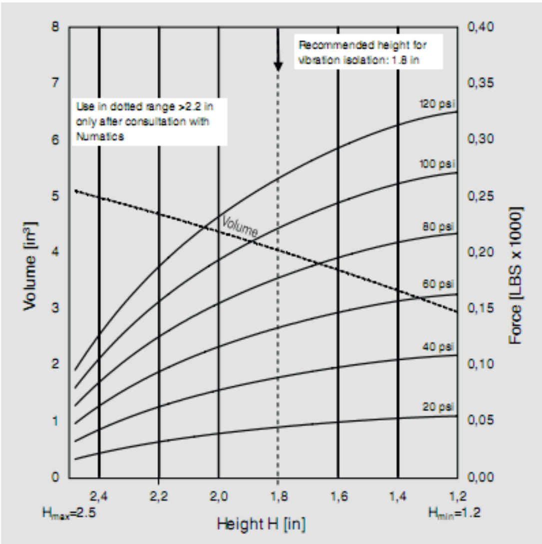


Figure 5.6.
Force-displacement curve for ASNC2-1-1 (Numatics).

Dynamic characteristic value for Sleeve type air spring ASNC 2-1-1 for volume is 1 048 mm^3 is shown in table 5.3,

Table 5.3.
Dynamic characteristic value for Sleeve type air spring ASNC2-1-1 (Numatics)

Pressure[MPa]	0.2758	0.4137	0.5116	0.6895	0.8274
Load [N]	391	587	783	979	1175
Spring rate [N/m]	19250	26950	34125	40425	47600
Natural frequency[Hz]	3.5	3.4	3.3	3.2	3.2

5.2.2 Seat suspension system in Simulink

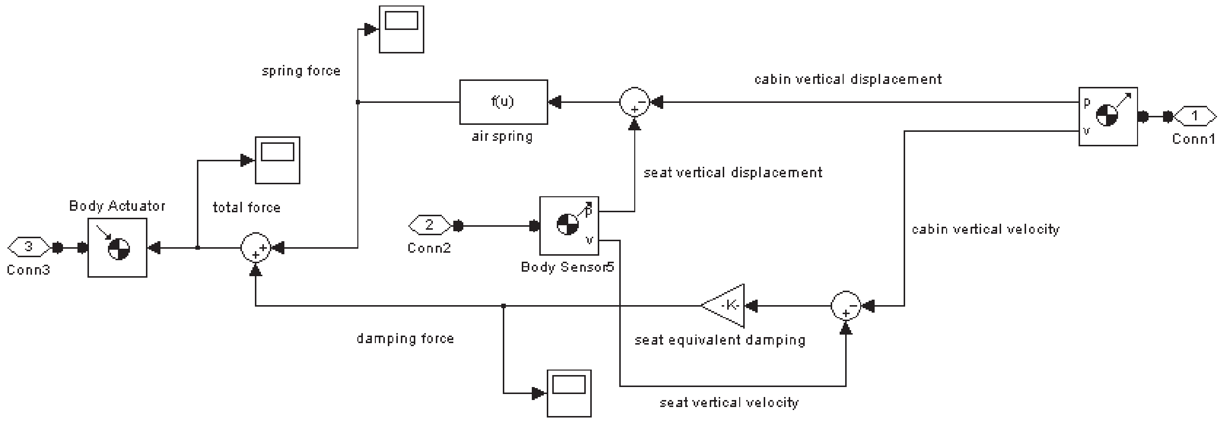


Figure 5.7.
Seat suspension in Simulink.

Then this model is inserted into the Simulink/SimMechanics model for whole machine system. The input of the system is the vertical displacement and velocity of the cabin while the output is the suspension force applied to the seat.

Meanwhile, this model can be also modeled in SimMechanics with components and joints, shown in figure 5.8.

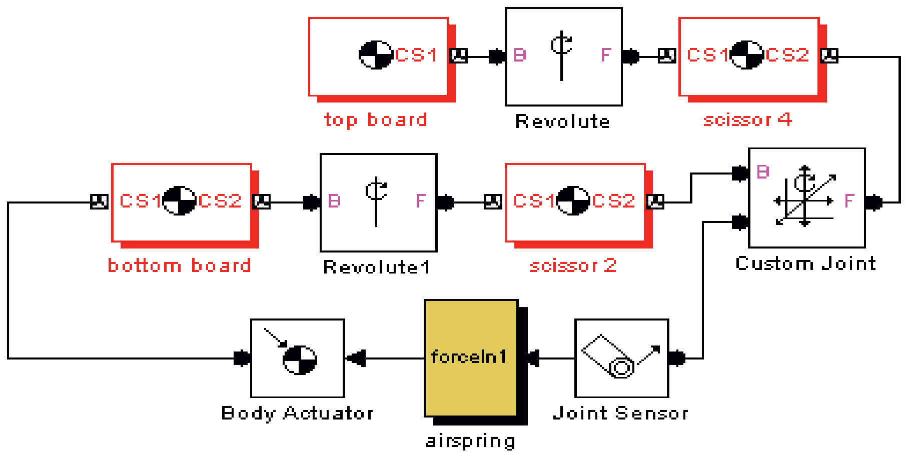
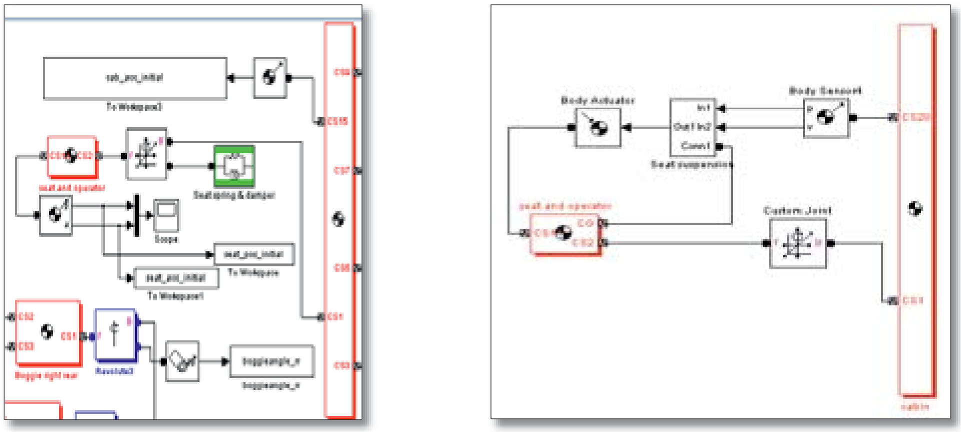


Figure 5.8.
Seat suspension in SimMechanics.

In this project, seat suspension model in Simulink was inserted into the whole forwarder (shown in figure 5.9) instead of seat suspension in SimMechanics, because if seat suspension in SimMechanics is inserted into the whole forwarder in SimMechanics, there will be more components and more connecting degrees of freedom, which may cause the problem of redundant constraints.



a) Previous seat model in SimMechancis.

b) Seat simulink model in SimMechanics.

Figure 5.9.
Seat suspension model in the whole forwarder in SimMechancis.

The comparison result between initial forwarder and forwarder with seat cabin suspension system will be shown in Chapter 6.1.

5.3 PRIMARY SUSPENSION SYSTEM AT FRONT AND REAR AXLES

A primary suspension at the axels can significantly improve the ride comfort by increasing the ratio of sprung mass to unsprung mass of the machine (P.E. Boileau, et al., 1990). However, for most forestry machines such as forwarders, harvesters, and skidders, they don't have primary suspension, even if they have, just passive suspension system, which does not contribute that much to the vibration isolation and reduction. And due to high cost of active suspensions, they only appear in cars instead of off-ground machines. However, either without any suspension system or with simple passive suspension system is not sufficient for the vibration reduction. But the active suspension is not the money-cost choice for manufacturers, so semi-active suspension system is the best alternative for forestry machines, with more energy than passive suspension and less expensive than active suspension. In this report, three passive suspensions with different spring stiffness and damping coefficients, and three different semi-active suspensions are modeled in Simulink, and then inserted into the whole forwarder to run for the simulation. The simulation results and comparisons between them will be shown in Chapter 6.2.

5.3.1 PASSIVE SUSPENSION

5.3.1.1 Passive suspension structure

Passive suspension which is consisted of a spring and damper, has been applied to most machines in the world, and also popular in the forestry machines. Primary suspension system with different front and rear spring stiffness and damping coefficients has been studied. The values of stiffness and damping coefficients are cited directly from previous project "Vibrations-dämpning av skotare" by Jaoquin Baes. The passive suspension system in Simulink is shown in figure 5.10 below,

5.3.1.2 Passive suspension in simulink

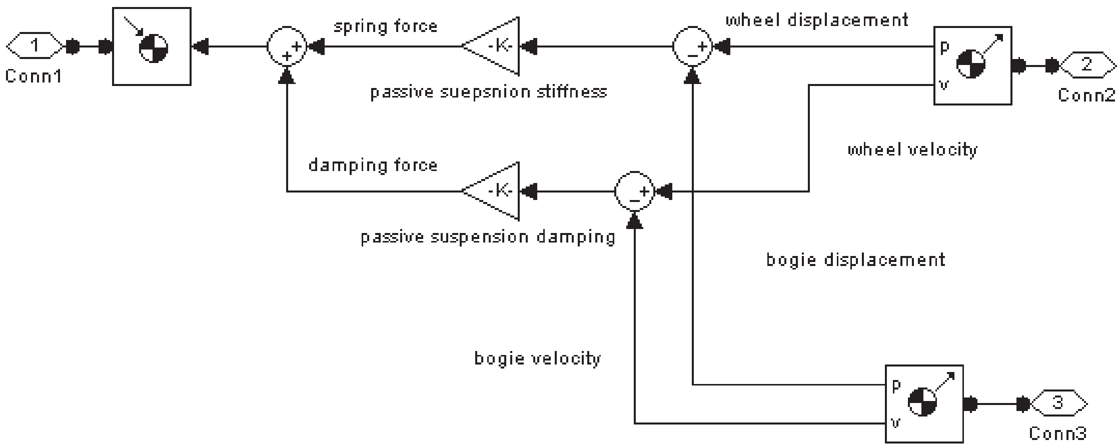
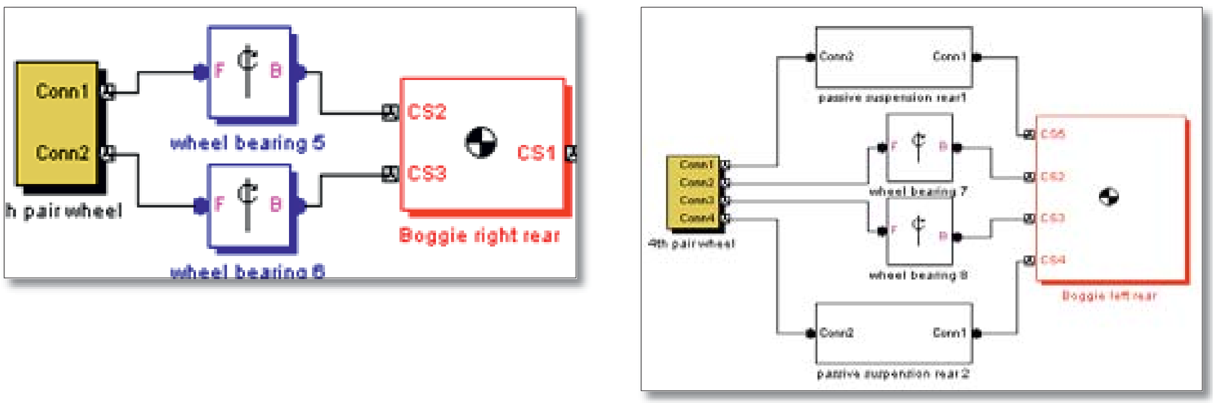


Figure 5.10.
Passive primary suspension in Simulink.

The input of this system is the vertical displacement and velocity of the wheel while the output is the suspension force applied to the bogie. In our simulation, the forwarder is unloaded, but if it is a fully loaded machine, it is necessary to consider the role of rear frame movement, which contributes a lot to the machine vibration. The passive suspension is installed between the bogie and wheels, and the spring stiffness and damping coefficient can be regarded as the twist stiffness and damping for the "revolute" joint between bogies and wheels. Figure 5.11 shows the passive suspension Simulink model in SimMechanics and table 5.4 shows parameters for different passive primary suspensions.



a). Previous passive suspension model in SimMechancis.

b) Passive suspension simulink model in SimMechanics.

Figure 5.11.
Seat suspension model in the whole forwarder in SimMechancis.

Table 5.4.
Passive primary suspension parameters.

Condition	Front stiffness [N/m]	Front damping $[\mathrm{N} \cdot \mathrm{s/m}]$	Rear stiffness [N/m]	Rear damping $[\mathrm{N} \cdot \mathrm{s/m}]$
Condition 1	500 000	50 000	3 300 000	330 000
Condition 2	400 000	40 000	2 300 000	230 000
Condition 3	350 000	35 000	1 900 000	190 000

Finding the optimum values of spring stiffness for a passive system is an important compromise between comfort and handling. Softer springs typically yield lower acceleration levels but less stability against the ground surface. It is also important that stability is not affected so much of the load changes (Jaoquin Baes, 2010). Since in real situation, the forwarder is loaded with a maximum weight at the rear part, so rear axle needs higher spring stiffness than that at the front. In a real case, the springs have nonlinear characteristics, which means that the spring stiffness increases more than proportionately with compression spring or that increases with the load (Jaoquin Baes, 2010). In this project, all the springs and dampers are assumed to be linear, and the damping coefficients are approximately set to 10 % of the spring stiffness.

In order to find the correct choice of spring constants, investigation of pitch angle in steady state was investigated. In the measurement of the loaded forwarder runs over a bump with 30 cm height, the front spring stiffness is set to 400 KN/m, while the rear spring stiffness are varied. When change the rear spring stiffness to 2300 KN/m, the forwarder is almost in a steady state and has its equilibrium, and the “nod” angle is around 0. As a result, when the spring stiffness at rear is larger than 2300 KN/m, the forwarder starts to lean forward, while lean backward when rear spring stiffness is smaller than 2300 KN/m.

5.3.2 SEMI-ACTIVE SUSPENSION

5.3.2.1 Semi-active suspension structure

Semi-active suspension is based on paper “Semi-active vibration isolation system with variable stiffness and damping control” by Yanqing Liu et al. Take a simple mechanism for a quarter of a machine as shown in figure 5.12, and this semi-active suspension system has two springs and two controllable dampers.

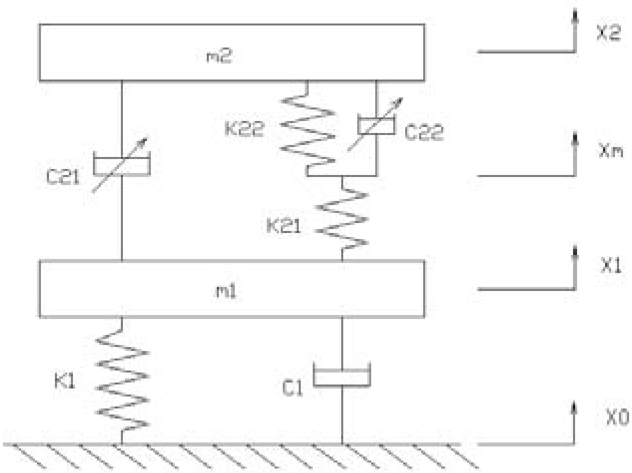


Figure 5.12.
Quarter car model.

Where, m_1 and m_2 represent wheel mass and a quarter chassis mass, k_1 and c_1 represent wheel stiffness and damping, k_{21}, k_{22}, $c_{21}$and c_{22} comprise the semi-active part of the suspension, x_0, x_1, x_m, and x_2 represent ground irregularity, wheel vertical displacement, relative displacement between two springs and chassis vertical displacement, respectively.

The equations of motion of this system are (Yanqing Liu, 2008).

$$m_1\ddot{x_1} + k_1(x_1 - x_0) + c_1(\dot{x_1} - \dot{x_0}) = c_{21}(\dot{x_2} - \dot{x_1}) + k_{21}(x_m - x_1) \tag{5.6}$$

$$m_2\ddot{x_2} + k_{22}(x_2 - x_m) + c_{22}(\dot{x_2} - \dot{x_m}) + c_{21}(\dot{x_2} - \dot{x_1}) = 0 \tag{5.7}$$

For the connecting point of two springs, the equation of force balance is (Yanqing Liu, 2008).

$$c_{22}(\dot{x_2} - \dot{x_m}) + k_{22}(x_2 - x_m) - k_{21}(x_m - x_1) = 0 \tag{5.8}$$

In this system, k_1, k_{21} and k_{22} are constant. The whole stiffness of this suspension can be adjusted by changing the value of c_{22}. If c_{22} has a small damping, the equivalent total stiffness of the system is almost equal to the joint effort of k_{21} and k_{22}, namely, $\frac{k_{21}k_{22}}{k_{21}+k_{22}}$. If c_{22} has a high damping, the equivalent total stiffness of the model is approximately equal to k_{21}. As a result, adjusting c_{22} can provide variable damping for the system, which achieves variable adjustment of stiffness and damping (Yanqing Liu, 2008).

Combining Skyhook control and on-off control in Chapter 2.6, the damping force F_{d21} and F_{d22} in the model should satisfy (Yanqing Liu, 2008).

$$F_{d21} = \begin{cases} -c_{21on}(\dot{x}_2 - \dot{x}_1) & \text{if } \dot{x}_2(\dot{x}_2 - \dot{x}_1) > 0 \\ -c_{21off}(\dot{x}_2 - \dot{x}_1) & \text{if } \dot{x}_2(\dot{x}_2 - \dot{x}_1) \leq 0 \end{cases} \quad (5.9)$$

$$F_{d22} = \begin{cases} -c_{22on}(\dot{x}_2 - \dot{x}_m) & \text{if } \dot{x}_2(\dot{x}_2 - \dot{x}_1) > 0 \\ -c_{22off}(\dot{x}_2 - \dot{x}_m) & \text{if } \dot{x}_2(\dot{x}_2 - \dot{x}_1) \leq 0 \end{cases} \quad (5.10)$$

Where, c_{21on} and c_{22on} represent on-state (higher) damping, while c_{21off} and c_{22off} represent off-state (lower) damping.

Three conditions of semi-active suspension can be achieved through the above system, which are shown in table 5.5,

Table 5.5.
Three conditions of semi-active suspension (Yanqing Liu, 2008)

Condition	Name	Damping c_{21}	Damping c_{22}	Equivalent k	Explanation
1	Damping on-off	On-off	On	k_{21}	Constant stiffness and variable damping
2	Stiffness on-off	On	On-off	On-off	Constant damping and variable stiffness
3	Stiffness+damping on-off	On-off	On-off	On-off	Variable damping and stiffness

In condition 1, c_{21} is in “on-off” state while c_{22} is in “on” state, which is called damping on-off control, and this type can be regarded as semi-active suspension with constant stiffness and variable damping; In condition 2, c_{21} is in “on” state while c_{22} is in “on-off” state, which is called stiffness on-off control, and this type can be categorized as semi-active suspension with constant damping and variable stiffness; In condition 3, c_{21} and c_{22} are both in “on-off” state, and this type is a semi-active suspension with variable stiffness and damping. As a result, by using this “on-off” control, three different semi-active suspensions are realized.

For practical applications, the stiffness ratio $\frac{K_{22}}{K_{21}}$ should be small in order to achieve a large variation of stiffness by changing damper 2. Specific k and c for the semi-active suspension are shown in table 5.6.

Table 5.6.
K and C for semi-active suspension

Parameters	Front wagon	Rear wagon
K_{21}[N/m]	400 000	1 500 000
K_{22}[N/m]	80 000	360 000
C_{21on}[N1on]]	3 000	18 000
C_{21off}[N1off]	1 600	11 000
C_{22on}[N2onf]	40 000	150 000
C_{22off}[N2off]	5 000	40 000

5.3.2.1 Semi-active suspension in Simulink

Three conditions of primary semi-active suspension systems are modeled in Simulink, shown in figure 5.13-figure 5.15. The semi-active suspension is installed in the same position as passive suspension shown in Chapter 5.3.1. The input of the system is wheel vertical displacement and velocity, and bogie vertical displacement and velocity, while the output is suspension force applied to the bogie.

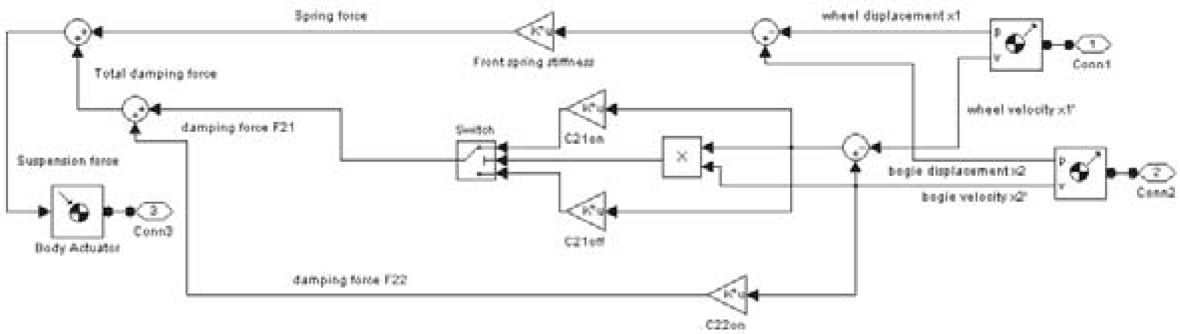


Figure 5.13.
Semi-active primary suspension condition 1.

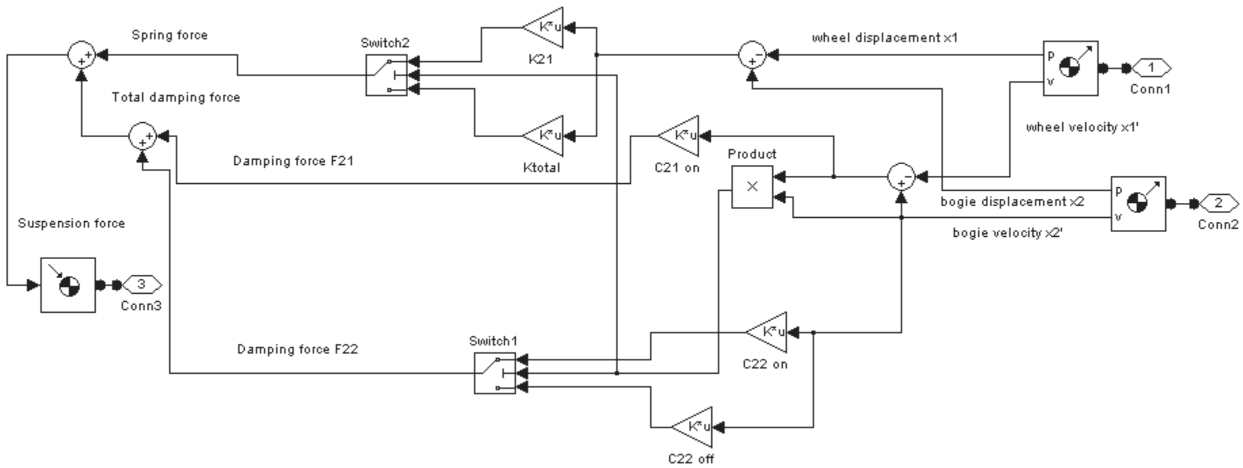


Figure 5.14.
Semi-active primary suspension condition 2.

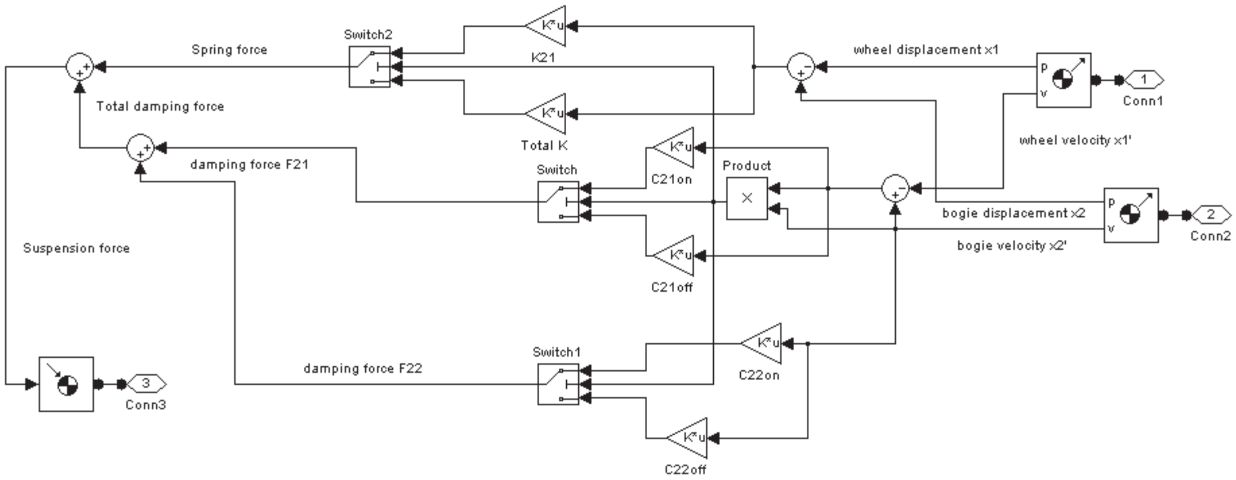


Figure 5.15.
Semi-active primary suspension condition 3.

The comparison result of three conditions will be shown in Chapter 6.2.

6 Results and analysis

6.1 DIFFERENT LEVELS OF SUSPENSION SYSTEM COMPARISON

6.1.1 Comparison

Three different levels of suspension systems, namely, cabin suspension, seat suspension and primary suspension have been modeled and inserted into the whole forwarder. In the following figures, magenta lines, green lines, blue lines and yellow lines represent accelerations for initial condition, condition with seat suspension, condition with cabin suspension, and condition with primary passive suspension, respectively.

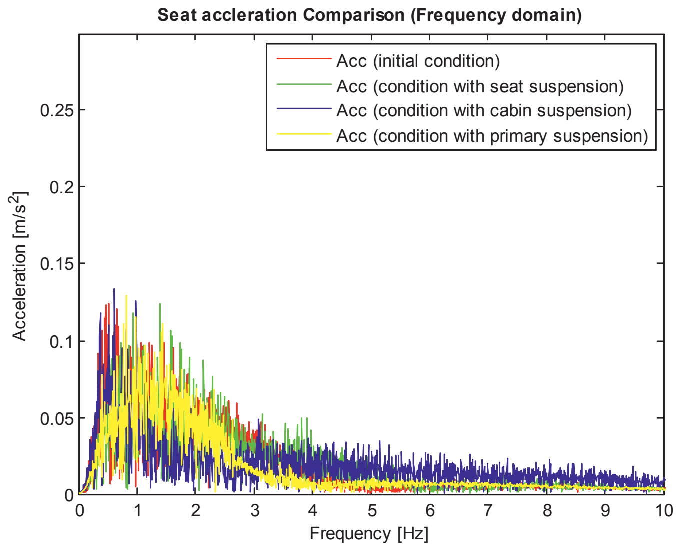


Figure 6.1.
Seat acceleration comparison (different levels of suspensions).

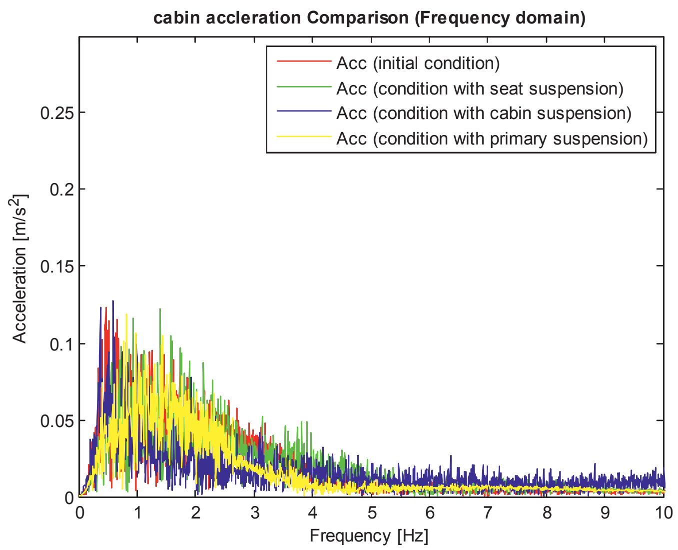


Figure 6.2.
Cabin acceleration comparison (different levels of suspensions).

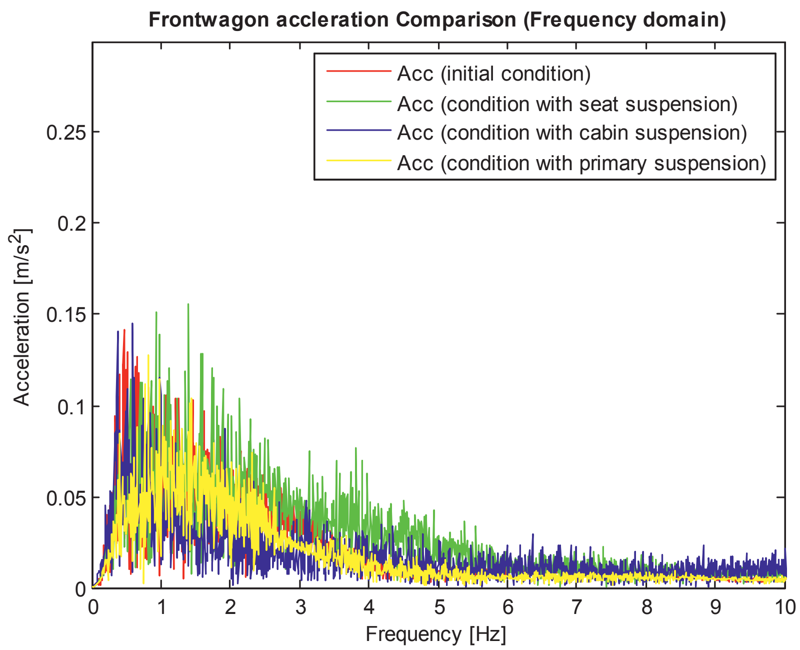


Figure 6.3.
Front wagon acceleration comparison (different levels of suspensions).

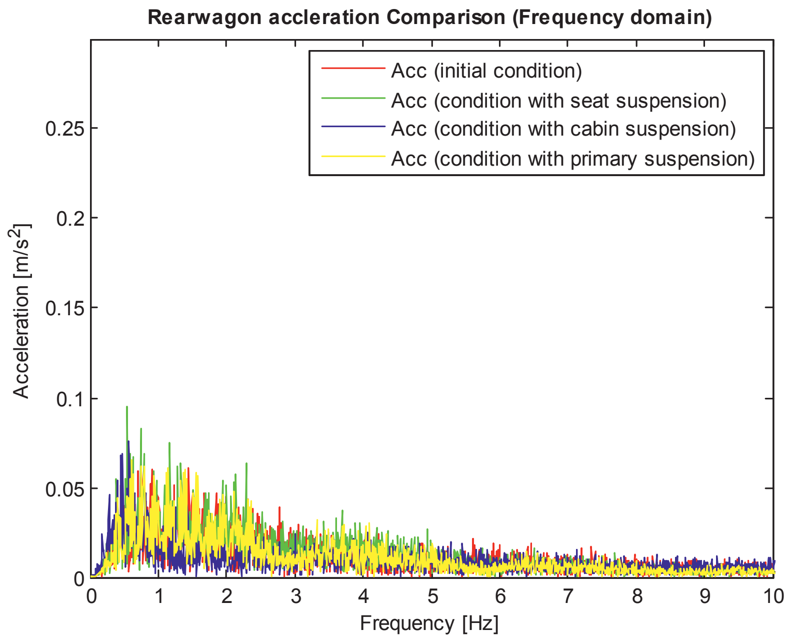


Figure 6.4.
Rear wagon acceleration comparison (different levels of suspensions).

It is easy from the above figures to see, the acceleration level of the initial condition has been reduced by inserting those suspension system. In order to evaluate the detailed information about vibration reduction, RMS, VDV and A (8) value have been recorded in table 6.1, and the Effect % of different components is also recorded in table 6.2.

Table 6.1.
RMS, VDV, A8 Comparison (Initial condition VS conditions with different levels of suspensions).

Conditions	Parts	RMS $[m/s^2]$	VDV $[m/s^{1.75}]$	A(8) $[m/s^2]$
Initial Condition	Seat	0.6442	2.1628	0.0360
	Cabin	0.6201	2.1462	0.0347
	Front wagon	0.6832	2.4155	0.0382
	Rear wagon	0.3210	0.9715	0.0179
Condition with cabin suspension	Seat	0.5901	1.6551	0.0330
	Cabin	0.5721	1.5839	0.0320
	Front wagon	0.6453	1.7462	0.0361
	Rear wagon	0.3273	0.9137	0.0183
Condition with Seat suspension	Seat	0.5803	1.8576	0.0326
	Cabin	0.5934	1.8621	0.0332
	Front wagon	0.6813	2.2464	0.0359
	Rear wagon	0.3329	1.0085	0.0180
Condition with Primary suspension	Seat	0.5856	2.1296	0.0327
	Cabin	0.5566	2.0739	0.0311
	Front wagon	0.5834	2.2818	0.0326
	Rear wagon	0.3355	0.9084	0.0188

Table 6.2.
Effect % of on different components (with different levels of suspensions)

Conditions	Parts	RMS, %	VDV, %	A(8), %
Condition with cabin suspension	Seat	–8.39	–23.5	–9.09
	Cabin	–7.74	–26.2	–7.78
	Front wagon	–5.55	–27.7	–5.50
	Rear wagon	+1.96	–5.94	+2.23
Condition with Seat suspension	Seat	–9.91	–14.1	–9.44
	Cabin	–4.30	–13.2	–4.32
	Front wagon	–0.28	–7.00	–6.02
	Rear wagon	–3.71	+3.80	+0.56
Condition with Primary suspension	Seat	–9.09	–1.52	–9.174
	Cabin	–10.24	–3.43	–10.4
	Front wagon	–14.6	–5.54	–14.7
	Rear wagon	–4.52	–6.50	+5.03

Seat vibration level is reduced most by condition with seat suspension with 9.91%, Cabin vibration level is reduced most by condition with cabin suspension with 10.24 %, and front wagon and rear wagon vibration level are reduced most by condition with primary suspension with 14.6 % and 4.52 %, respectively. We can conclude that those inserted suspension system contribute to the vibration reduction for each component of the initial forwarder, and the primary suspension system determines the most important part reducing whole body vibration level while secondary suspension system can help a lot.

6.1.2 SEAT evaluation

Seat suspension system is the final suspension system in the forwarder, in order to check whether the inserted seat suspension (air spring) can really help reduce the vibration level from cabin floor to seat or not, SEAT % is used to evaluate the seat isolation efficiency.

Seat Effective Amplitude Transmissibility (SEAT) is define as.

$$\mathrm{SEAT\%} = \left(\frac{\int G_{ss}(f) W_i^2(f) df}{\int G_{ff}(f) W_i^2(f) df}\right)^{\frac{1}{2}} \times 100\% \tag{6.1}$$

Where,

$G_{ss}(f)$ is seat power spectral density [W/Hz],

$G_{ff}(f)$ is floor power spectral density [W/Hz],

$W_i(f)$ is frequency weighting for the human response to vibration.

In other ways, SEAT, values can be calculated from either RMS or VDV of the frequency-weighted acceleration (G. S. Paddan, 2002) as follows.

$$\mathrm{SEAT}_{r.m.s} = \frac{r.m.s_{seat}}{r.m.s_{cabin\ floor}} \times 100\% \tag{6.2}$$

$$\mathrm{SEAT}_{VDV} = \frac{VDV_{seat}}{VDV_{cabin\ floor}} \times 100\% \tag{6.3}$$

$\mathrm{SEAT}_{r.m.s}$ is the ratio of the frequency-weighted acceleration on the seat $r.m.s_{seat}$ to the frequency-weighted acceleration on the cabin floor $r.m.s_{cabin\ floor}$; In the same way, SEAT_{VDV} is the ratio of the frequency-weighted vibration dose value on the seat VDV_{seat} to the frequency-weighted acceleration on the cabin floor $VDV_{cabin\ floor}$ (G. S. Paddan, 2002).

As a result, $\mathrm{SEAT}_{r.m.s}$ and SEAT_{VDV} can be compared between the initial condition and the condition with seat suspension.

Table 6.3.
Comparison of $\mathrm{SEAT}_{r.m.s}$ and SEAT_{VDV} (Initial condition VS condition with seat suspension).

Conditions	$\mathrm{SEAT}_{r.m.s}$, %	SEAT_{VDV}, %
Initial condition	103.8	100.8
Condition with seat suspension	97.7	99.8

As shown in table 6.3, after the seat suspension (with air spring) has been used, $\mathrm{SEAT}_{r.m.s}$ is reduced from 103.8% in the initial condition to 97.7%, which is smaller 100%, indicating that an overall improvement in ride comfort. However, it is obvious for us to see that for the initial forwarder, the ride comfort is bad because SEAT% is larger than 100%.

6.2 DIFFERENT TYPES OF PRIMARY SUSPENSION SYSTEM COMPARISON

Different kinds of primary suspension systems shown in Chapter 5.3 have been modeled and inserted into the whole forwarder, the results are presented as follows.

6.2.1 Different passive suspensions

Fig 6.5 – figure 6.8 show the seat, cabin, front wagon, rear wagon acceleration between initial condition and conditions with different passive primary suspension. Red lines, blue lines, green lines and yellow lines represent accelerations for initial condition, passive primary suspension condition 1, passive primary suspension condition 2, and passive primary suspension condition 3, respectively.

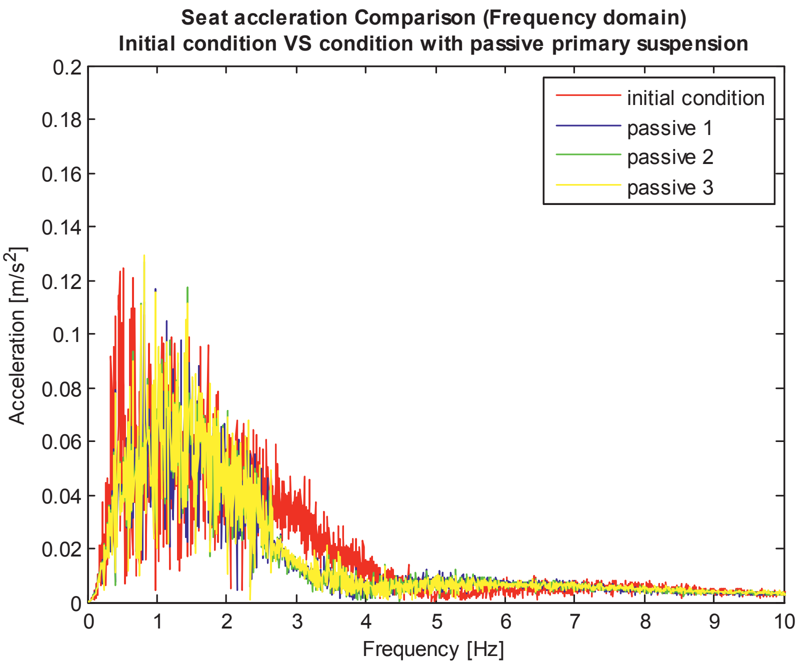


Figure 6.5.
Seat acceleration comparison (different primary passive suspensions).

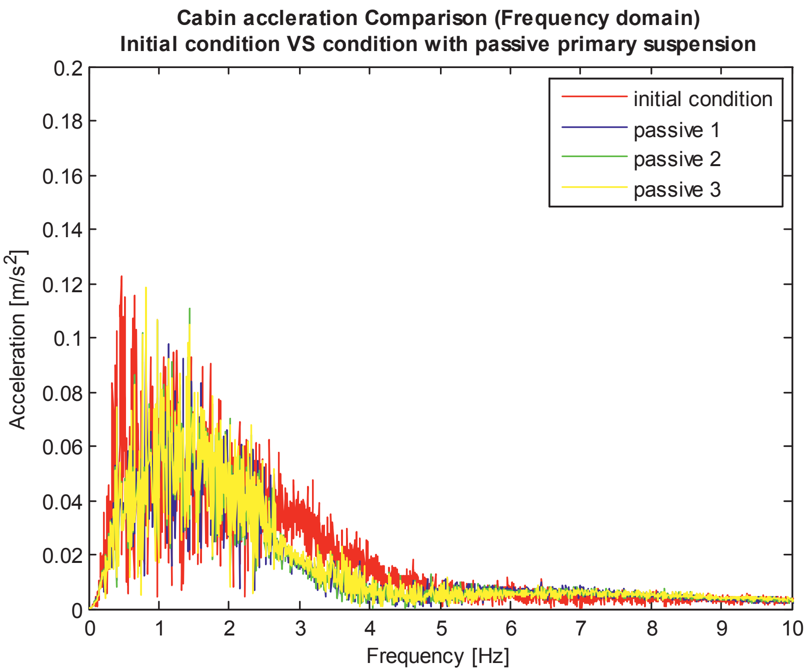


Figure 6.6.
Cabin acceleration comparison (different primary passive suspensions).

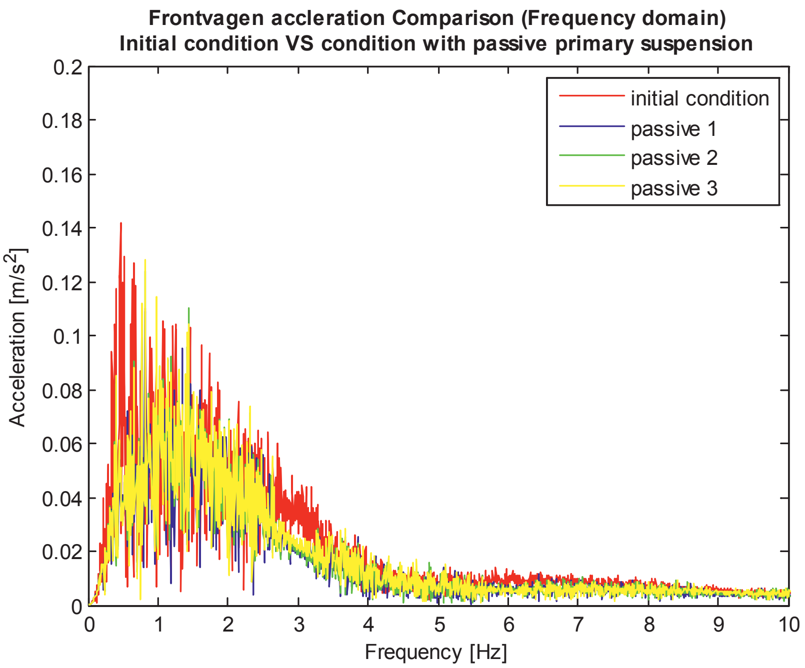


Figure 6.7.
Front wagon acceleration comparison (different primary passive suspensions).

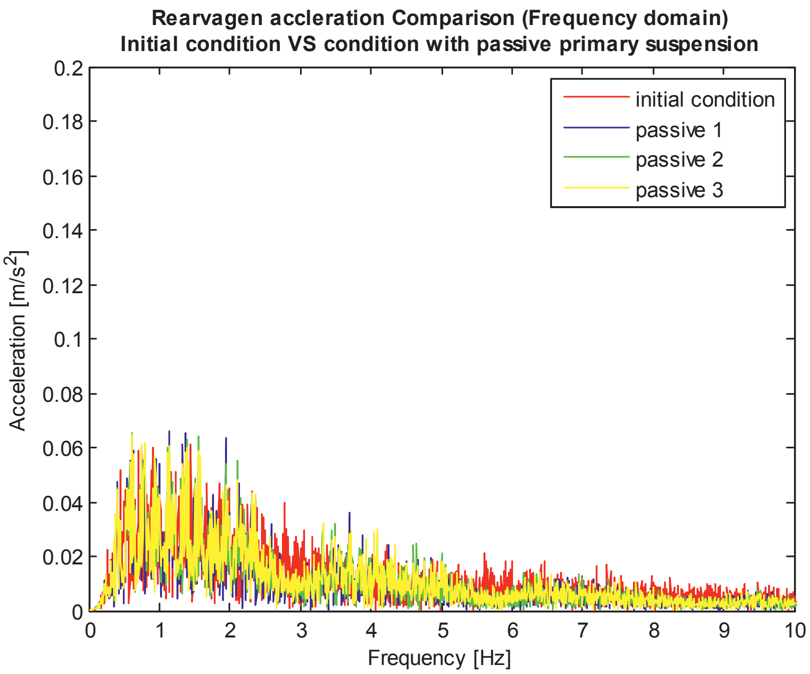


Figure 6.8.
Rear wagon acceleration comparison (different primary passive suspensions).

Also, to evaluate the detailed information about vibration reduction, RMS, VDV and A (8) value have been recorded in table 6.4, and the Effect % of different components is also recorded in table 6.5.

Table 6.4.
RMS, VDV, A8 Comparison (Initial condition VS conditions with different primary passive suspensions).

Conditions2	**Parts**	**RMS** $[m/s^2]$	**VDV** $[m/s^{1.75}]$	**A(8)** $[m/s^2]$
Initial Condition	Seat	0.6442	2.1628	0.0360
	Cabin	0.6201	2.1462	0.0347
	Front wagon	0.6832	2.4155	0.0382
	Rear wagon	0.3210	0.9715	0.0179
Condition 1	Seat	0.5740	2.1387	0.0321
	Cabin	0.5435	2.0693	0.0304
	Front wagon	0.5560	2.1822	0.0311
	Rear wagon	0.3125	0.7508	0.0175
Condition 2	Seat	0.5855	2.1445	0.0327
	Cabin	0.5557	2.0834	0.0311
	Front wagon	0.5772	2.2563	0.0323
	Rear wagon	0.3290	0.8588	0.0184
Condition 3	Seat	0.5856	2.1296	0.0327
	Cabin	0.5566	2.0739	0.0311
	Front wagon	0.5834	2.2818	0.0326
	Rear wagon	0.3355	0.9084	0.0188

Table 6.5.
Effect % of on different components (with different primary passive suspensions).

Conditions	Parts	RMS, %	VDV, %	A(8), %
Condition 1	Seat	–10.9	–1.11	–10.8
	Cabin	–12.4	–3.58	–12.4
	Front wagon	–18.6	–9.65	–18.6
	Rear wagon	–2.65	–22.7	–2.23
Condition 2	Seat	–9.11	–0.85	–9.17
	Cabin	–10.4	–2.93	–10.4
	Front wagon	–15.5	–6.59	–15.4
	Rear wagon	+2.49	–11.6	+2.79
Condition 3	Seat	–9.09	–1.52	–9.17
	Cabin	–10.24	–3.43	–10.4
	Front wagon	–14.6	–5.54	–14.7
	Rear wagon	+4.52	–6.50	+5.03

Condition 1 has better vibration reduction result than that for condition 2 and 3, with higher spring stiffness and damping coefficients, which can reduce the initial vertical vibration level as 10.9 % for seat, 12.4% for cabin, 18.6% for front wagon and 2.65% for rear wagon. The most important part is for front wagon, which support the whole cabin, seat and driver. Thus, in empty-loaded case, front wagon is a key component to reduce the vibration level.

6.2.2 Different semi-active suspensions

Fig 6.9 - figure 6.12 show the seat, cabin, front wagon, rear wagon acceleration between initial condition and conditions with different passive primary suspension. Red lines, green lines, deep blue lines and light blue lines represent accelerations for initial condition, semi-active primary suspension condition 1, semi-active primary suspension condition 2, and semi-active primary suspension condition 3, respectively.

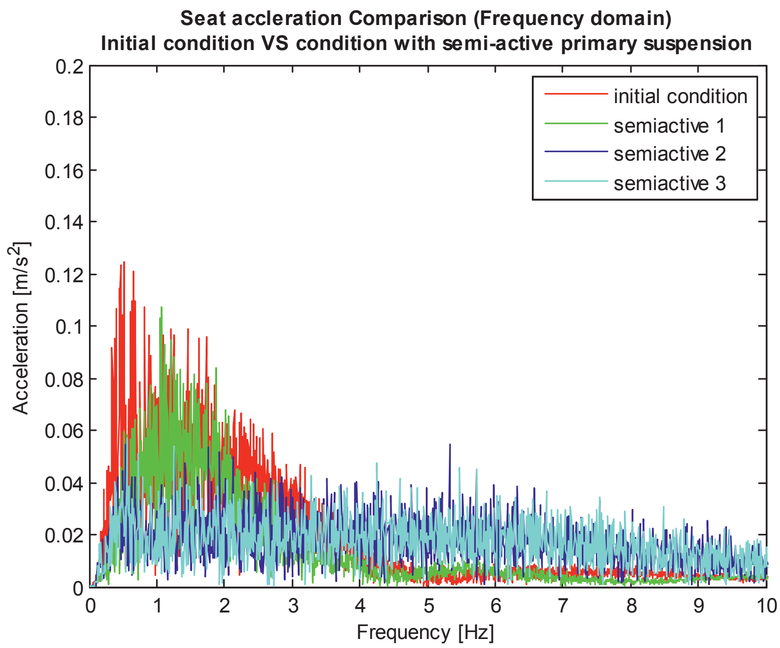


Figure 6.9.
Seat acceleration comparison (different primary semi-active suspensions).

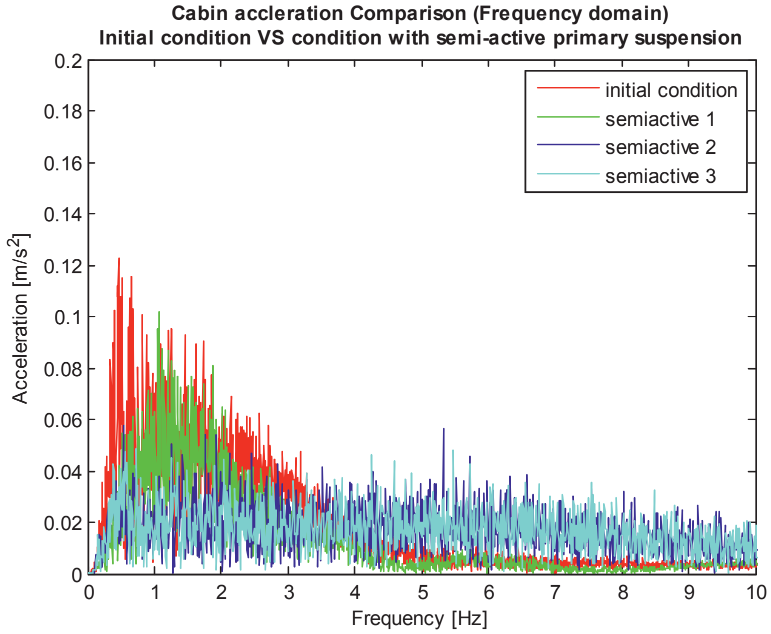


Figure 6.10.
Cabin acceleration comparison (different primary semi-active suspensions).

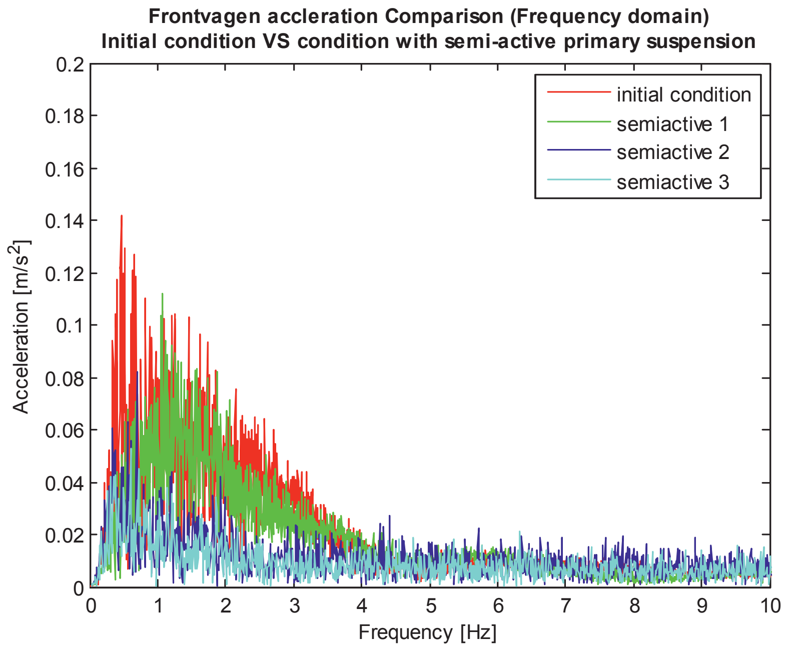


Figure 6.11.
Front wagon acceleration comparison (different primary semi-active suspensions).

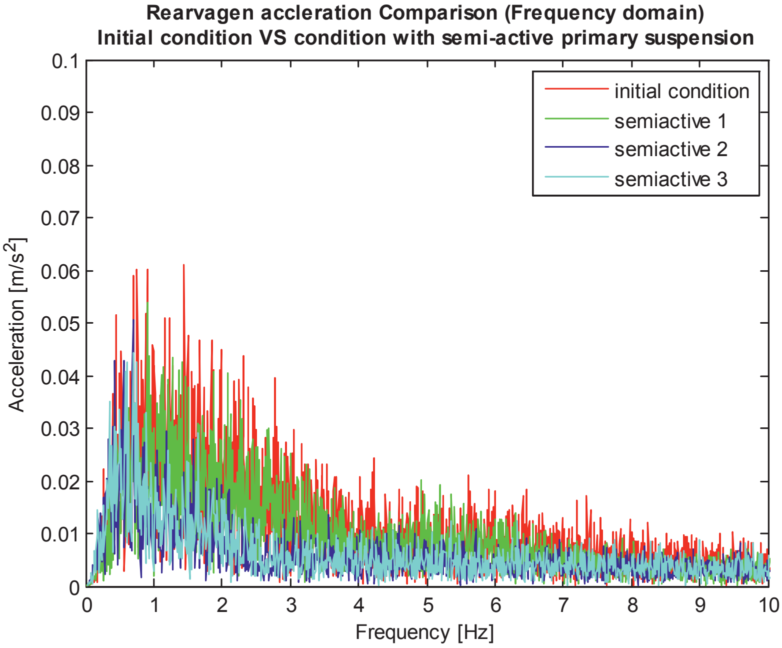


Figure 6.12.
Rear wagon acceleration comparison (different primary semi-active suspensions).

Also, to evaluate the detailed information about vibration reduction, RMS, VDV and A (8) value have been recorded in table 6.6, and the Effect % of different components is also recorded in table 6.7.

Table 6.6.
RMS, VDV, A8 Comparison (Initial condition VS conditions with different primary semi-active suspensions).

Conditions	Parts	RMS $[m/s^2]$	VDV $[m/s^{1.75}]$	A(8) $[m/s^2]$
Initial Condition	Seat	0.6442	2.1628	0.0360
	Cabin	0.6201	2.1462	0.0347
	Front wagon	0.6832	2.4155	0.0382
	Rear wagon	0.3210	0.9715	0.0179
Condition 1	Seat	0.4935	1.6307	0.0276
	Cabin	0.4695	1.6289	0.0262
	Front wagon	0.5613	2.3320	0.0314
	Rear wagon	0.2656	0.9987	0.0148
Condition 2	Seat	0.4612	1.5910	0.0258
	Cabin	0.4601	1.5982	0.0257
	Front wagon	0.5735	1.8180	0.0321
	Rear wagon	0.2747	0.4550	0.0154
Condition 3	Seat	0.4550	1.5753	0.0254
	Cabin	0.4553	1.5815	0.0255
	Front wagon	0.4563	1.6348	0.0255
	Rear wagon	0.2826	0.4600	0.0158

Table 6.7.
Effect % of on different components (with different primary semi-active suspensions).

Conditions	Parts	RMS, %	VDV, %	A(8), %
Condition 1	Seat	–23.4	–24.6	–23.3
	Cabin	–24.3	–24.1	–24.5
	Front wagon	–17.8	–3.46	–17.8
	Rear wagon	–17.3	+2.79	–17.3
Condition 2	Seat	–28.4	–26.4	–28.3
	Cabin	–25.8	–25.5	–25.9
	Front wagon	–16.1	–24.7	–16.0
	Rear wagon	–14.4	–53.1	–14.0
Condition 3	Seat	–29.3	–27.2	–29.4
	Cabin	–26.6	–26.3	–26.5
	Front wagon	–33.2	–32.3	–33.2
	Rear wagon	–12.0	–52.7	–11.7

After inserting semi-active suspension system, because either the spring stiffness or damping coefficient is not constant and changes with the input from the ground irregularity, the ride comfort becomes much better. Compared to passive primary suspension, semi-active suspension is more efficient in reducing vertical vibration, and even the worst condition 1 can reduce the seat vertical acceleration by 23.4 %, not to say the condition 3 can reduce front wagon vertical acceleration by 33.2 %. Compare condition 1 and condition 2, we can see that variable spring stiffness is more efficient than variable damping coefficient by around 4 % for RMS value reduction. Condition 2 and condition 3 are not regular semi-active suspension, which only allows for variable damping. As a result, in the future work, more attention may need to be paid to semi-active suspension with variable spring stiffness.

6.2.3 Combined situation

Based on the above analysis, combined situation comes out in terms of seat suspension (air spring) + cabin hydro pneumatic suspension + primary semi-active suspension (condition 3). In this way, the forwarder is fully suspended with both primary suspension and secondary suspension. The simulation results for combined situation model are shown below,

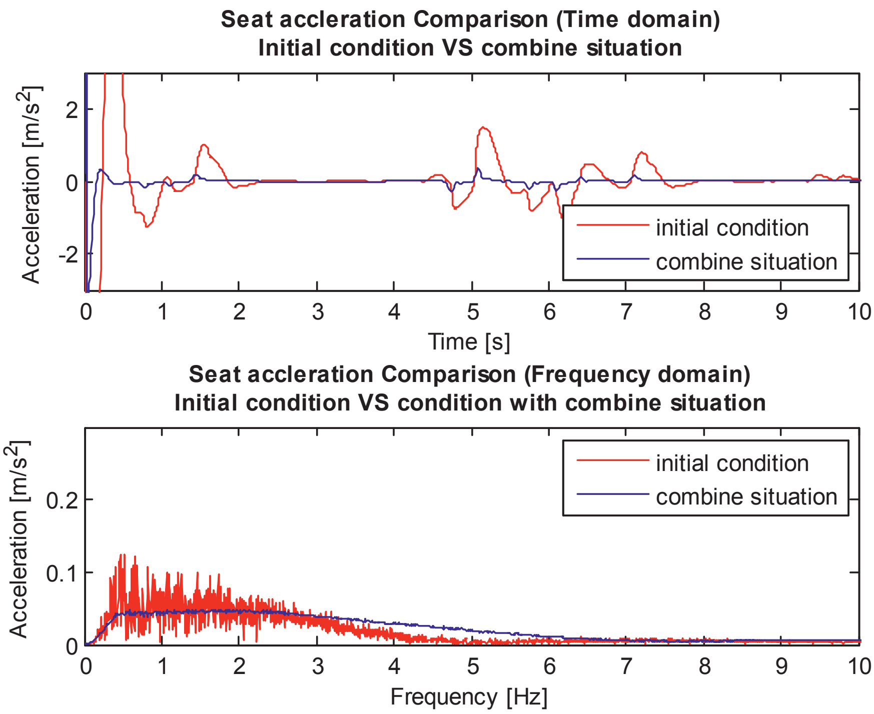


Figure 6.13.
Seat acceleration comparison (Initial condition VS Combined condition)

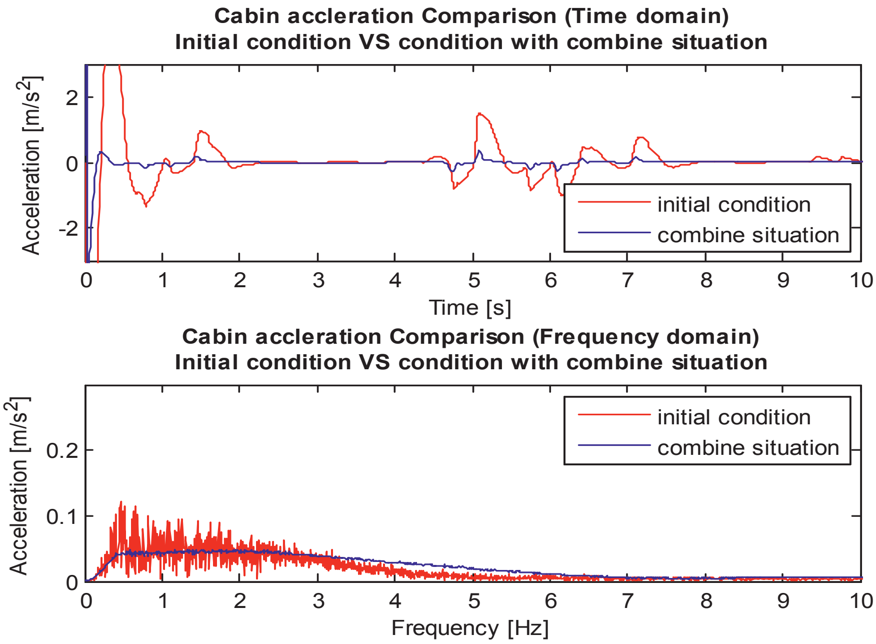


Figure 6.14.
Cabin acceleration comparison (Initial condition VS Combined condition).

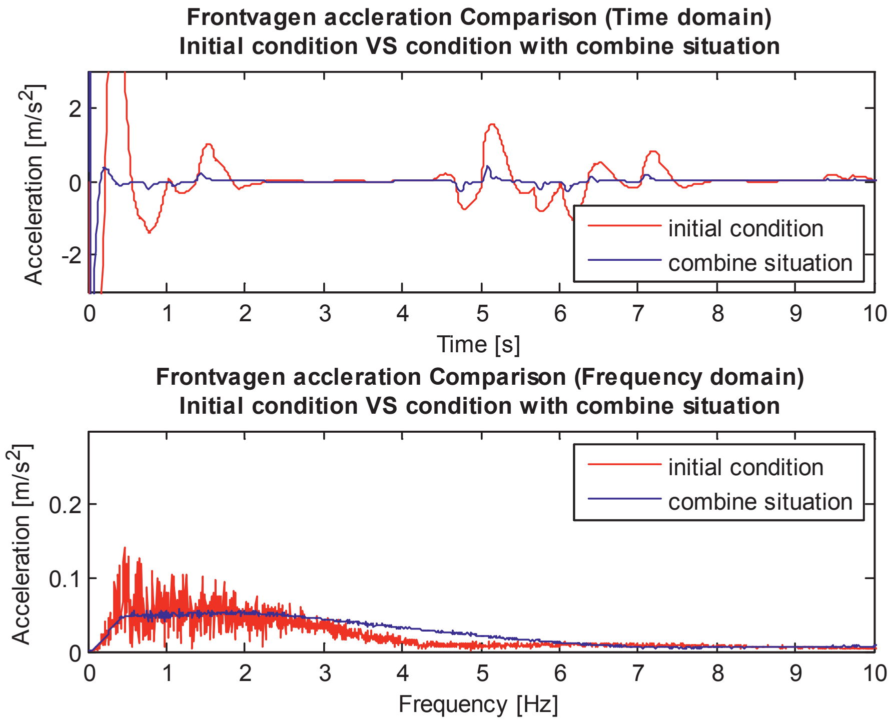


Figure 6.15.
Front wagon acceleration comparison (Initial condition VS Combined condition).

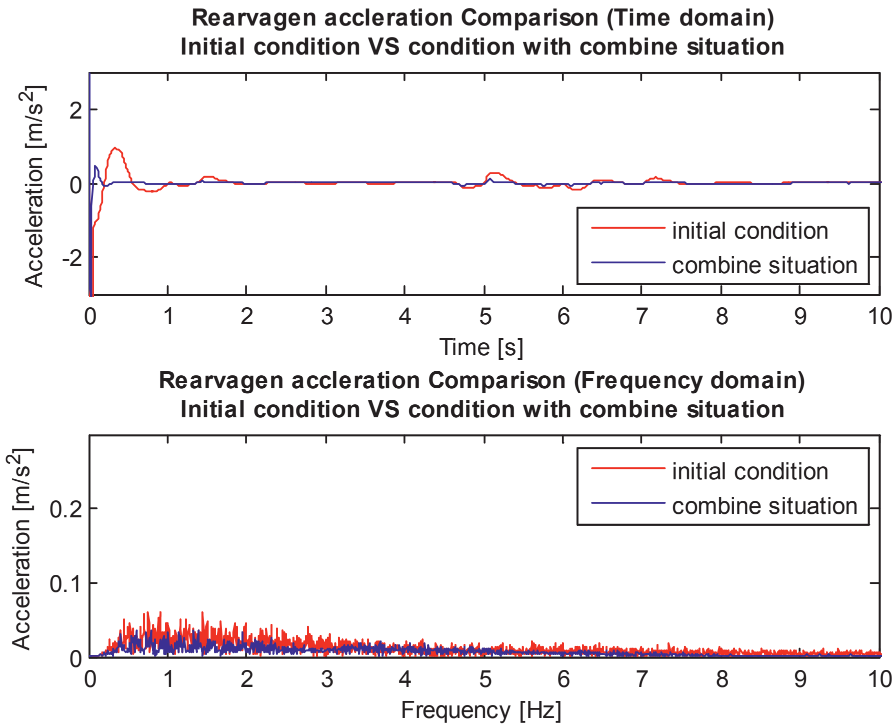


Figure 6.16.
Rear wagon acceleration comparison (Initial condition VS Combined condition).

Also, to evaluate the detailed information about vibration reduction, RMS, VDV and A (8) value have been recorded in table 6.8, and the Effect % of different components is also recorded.

Table 6.8.
RMS, VDV, A (8) and Effect % comparison (Initial condition VS combined condition).

Conditions	Parts	RMS $[m/s^2]$	VDV $[m/s^{1.75}]$	A(8) $[m/s^2]$
Initial Condition	Seat	0.6442	2.1628	0.0360
	Cabin	0.6201	2.1462	0.0347
	Front wagon	0.6832	2.4155	0.0382
	Rear wagon	0.3210	0.9715	0.0179
Combined Condition	Seat	0.4713	1.3535	0.0216
	Cabin	0.4679	1.3492	0.0214
	Front wagon	0.5127	1.5085	0.0304
	Rear wagon	0.2595	0.4594	0.0137
Effect %	Seat	–26.8	–37.4	–40.0
	Cabin	–24.5	–37.1	–38.3
	Front wagon	–25.0	–37.5	–20.4
	Rear wagon	–19.2	–52.7	–23.5

This combined situation is the best improvement for the initial forwarder compared to other forms in this report. Seat vertical acceleration level is reduced by 25.8%, and even the rear wagon vibration level is reduced by 19.2 %, which is small for an empty-loaded forwarder.

Suppose that the acceleration level for seat is the same as the vibration exposure level for the driver, and according to ISO 2631-1:1997, the comfort reactions to vibration environment can be compared between the initial condition and the combined condition.

Table 6.9.
Comfort reactions to vibration environments (ISO 2631-1:1997).

Acceleration range	Comfort reactions
Less than 0.315 m/s^2	Not uncomfortable
0.315m/s^2 to 0.63 m/s^2	A little uncomfortable
0.5 m/s^2 to 1 m/s^2	Fairly uncomfortable
0.8 m/s^2 to 1.6 m/s^2	Uncomfortable
1.25 m/s^2 to 2.5 m/s^2	Very uncomfortable
Greater than 2 m/s^2	Extremely uncomfortable

Thus, in the initial condition, 0.6442 m/s^2 is in the level of fairly uncomfortable, while in combined condition, the acceleration level at the seat was reduced to 0.4713 m/s^2, which is much better, and at this time, the driver only feels a little uncomfortable. However, the driver usually has to work around 8 hours each day, so he will still get pain for shoulder and back if he works for a long period consistently. As a result, the best way may be applying active primary suspension into the forwarder, or improving seat suspension and cabin suspension.

RMS value comparisons of each component in different models are shown in figure 6.17, the combined situation may not be the best option for the forwarder to reduce only seat vertical acceleration, but it will be the optimal alternative to reduce the whole body vibration for the whole forwarder.

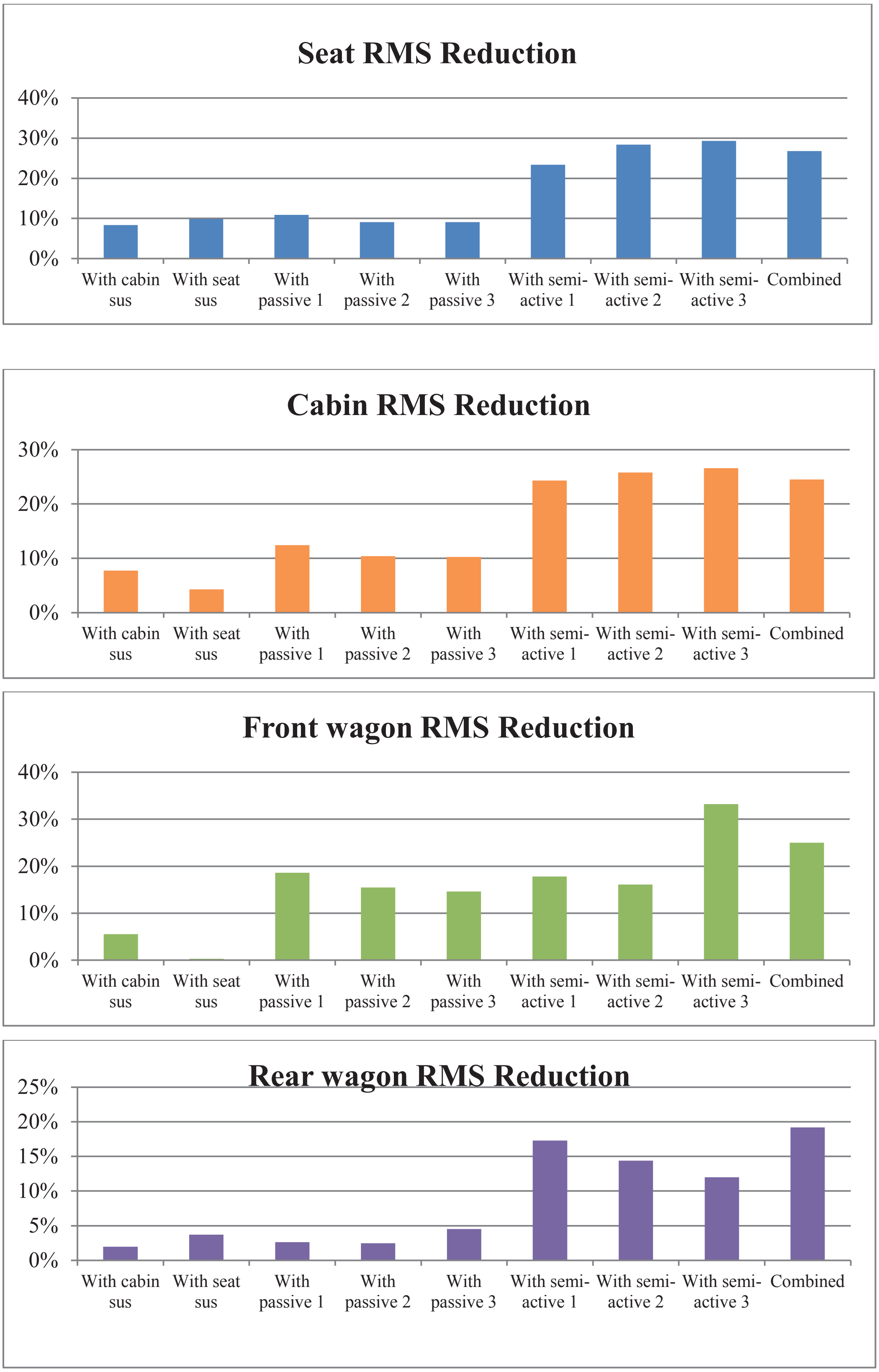


Figure 6.17.
RMS comparison in different models.

From the above results, we can make a grading of that the performance of reducing vertical acceleration of different components for those suspension systems.

Semi-active suspension condition 3> semi-active suspension condition 2> semi-active suspension condition 1> passive suspension condition 1> passive suspension condition 2> passive suspension condition 3>cabin suspension > seat suspension.

6.2.4 Increasing the forwarder speed

Actually, the most critical aspect that manufacturers care about is how to increase the productivity of the forwarder, and one way of achieving this is to increase the speed of the forwarder. In this section, the speed of the forwarder in the simulation is increased from 0.3 m/s to 0.6 m/s, and the new speed is applied to the combined situation in Chapter 6.2.3. And the comparison of different effects for each component with different speeds are shown in figure 6.18-figure 6.21.

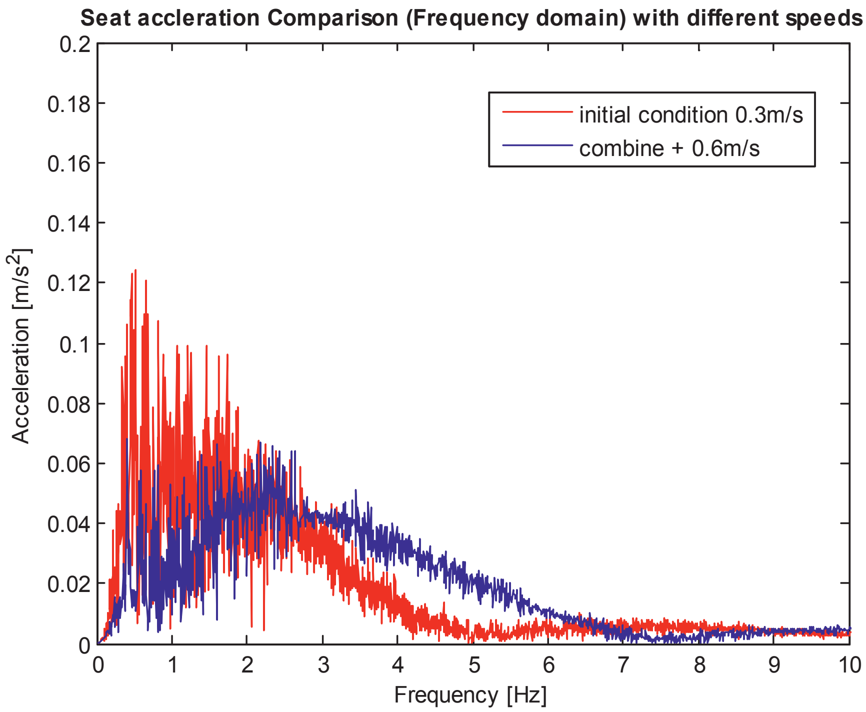


Figure 6.18.
Seat acceleration comparison (0.3 m/s VS 0.6 m/s).

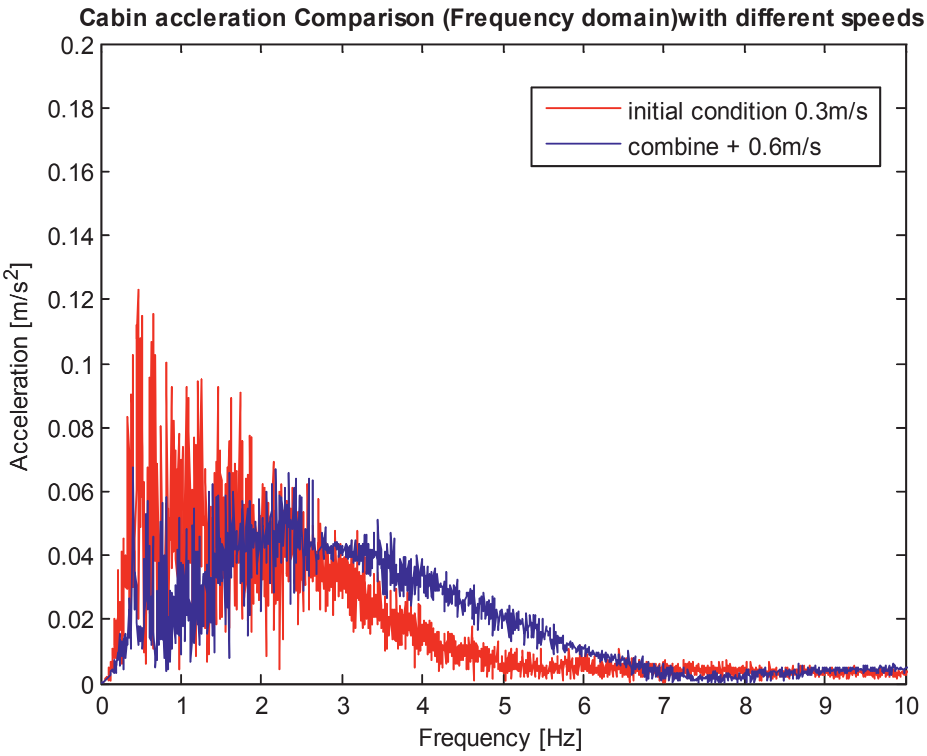


Figure 6.19.
Cabin acceleration comparison (0.3 m/s VS 0.6 m/s).

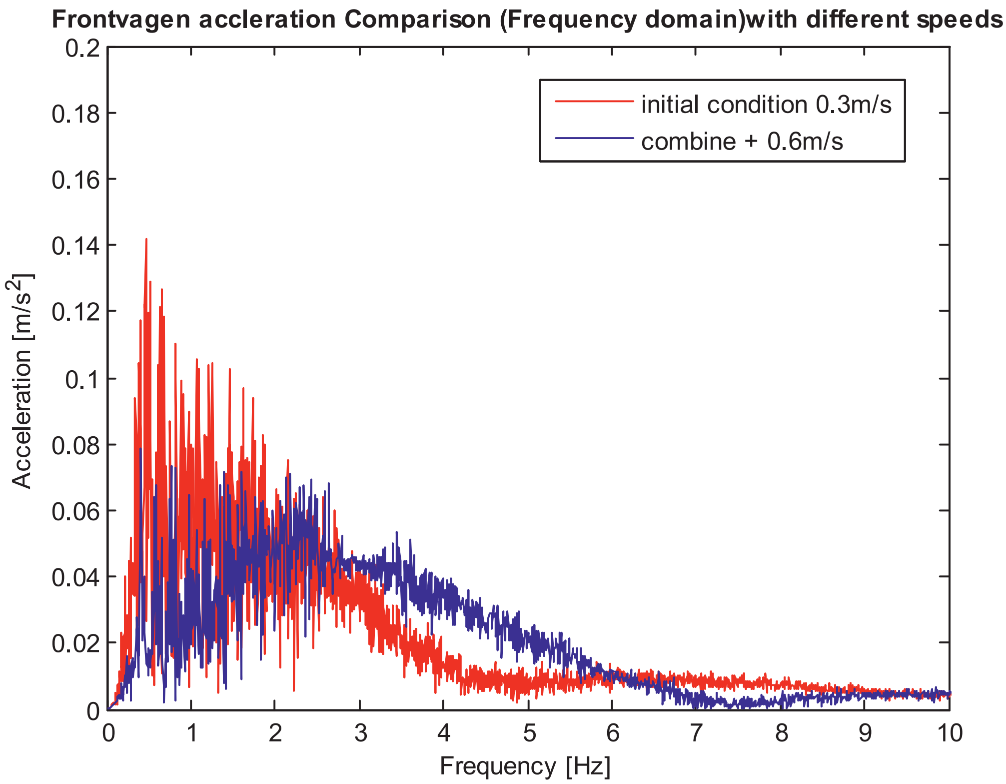


Figure 6.20.
Front wagon acceleration comparison (0.3 m/s VS 0.6 m/s).

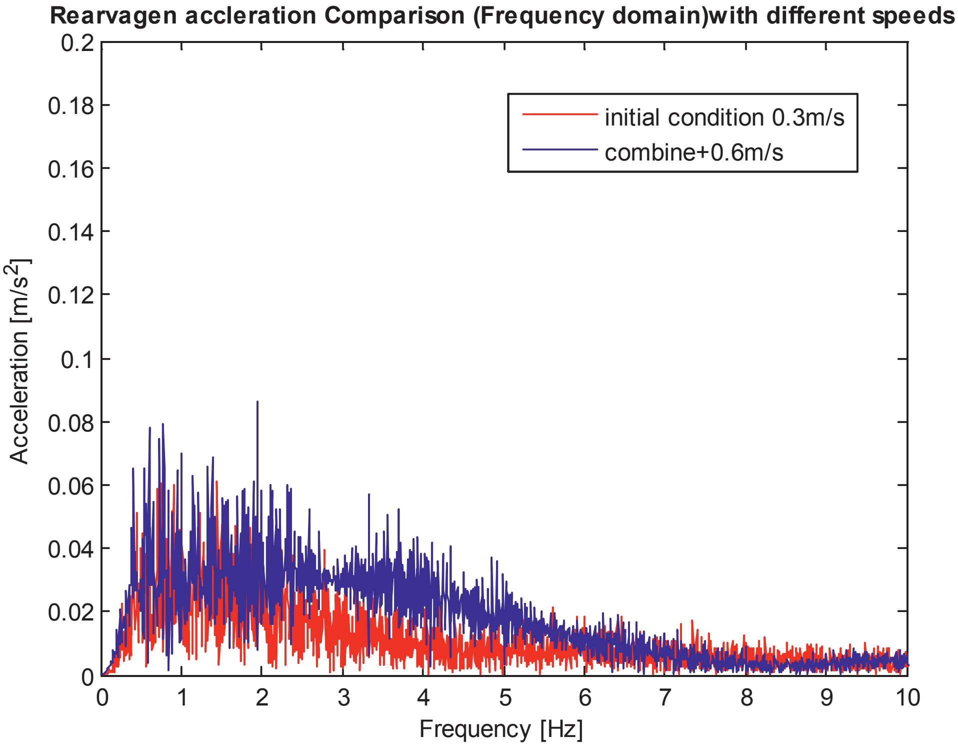


Figure 6.21.
Rear wagon acceleration comparison (0.3 m/s VS 0.6 m/s).

It is easy to see that the vibration level can still be reduced with the combine situation when the speed of the forwarder is doubled, which means that the suspension system can contribute to the vibration reduction as well as higher productivity. In this way, manufacturers will be very pleased to see this result. And with the combined situation, the frequency range for the peak value can be increased around 2-3 Hz, will be far away from the sensitive range for motion sickness.

7 Conclusions and recommendations

7.1 CONCLUSIONS

Related literature study for off-ground machines has been performed, and the previous research is always based on ADAMS simulation and measurement result. In this report, new software SimMechanics has been used instead of ADAMS. As a result, some specific ways about how to simulate the real forwarder needed to be figured out. Measurement results come from previous projects, which may have different test environment, which will contribute to the deflection of the simulation result.

A literature study of modeling techniques was completed to determine which assumptions may be made to simplify the model of a machine. However, different types of machines have different simplifications. According to the structure of Valmet 860.4 and the goal of the project, the forwarder was simplified into a 47 degree-of-freedom rigid system.

The three major components of a simulation model is the ground irregularity, the forwarder and the operator. The ground irregularities are modeled in Matlab in terms of ground surface vertical displacement and vertical velocity according to the structure of the test ground in Skogforsk, which will be applied to eight wheels as the input of the whole system. The ground surface input is based on some simplifications shown in Chapter 3.2 as well; the forwarder is modeled according to the real structure of the forwarder, and the mass, the centers of gravity and the moment of inertia are from previous research. Especially, only vertical direction motion are recorded in the simulation, which plays the most important role in machine vibration; the driver and the seat was modeled with a simple 1-DOF lumped parameter model.

Evaluate the simulation result with measurement result, and compare accelerations of seat, cabin, front wagon and rear wagon both in time domain and frequency domain. Due to some simplifications and assumptions for the whole forwarder and ground irregularity, the simulation result and measurement result did not match very well. However, in order to further the development of SimMechanics model, more effort should be devoted to familiarity with Simulink and SimMechanics to obtain more consistent comparison result.

Cabin suspension, seat suspension, primary passive suspension and primary semi-active suspension have been modeled according to literature and inserted into the whole forwarder. Passive suspension has 3 different conditions with different spring stiffness and damping coefficients while semi-active suspension also has 3 different conditions which are controlled by on-off control principle.

Finally, a combined situation comes out in terms of the initial forwarder with cabin hydro pneumatic suspension, seat suspension (air spring) and semi-active suspension (condition 3), which turns out to be the most effective form to reduce the vibration from the ground compared to other forms in the report.

As a result, the main conclusions that may be derived from this report are as follows,

1. A simplified simulation model based on some assumptions in SimMechanics can be used for research to reduce vibration of a forwarder. However, more specific study and development needed to combine CAD models with SimMechanics directly;

   In chapter 4, the validation of simulation model shows that although the simulation result and the measurement result do not match very well, the simulation can reflect the basic characteristics of the real model based on those assumptions and simplifications. More specific information about the model should be obtained such as the dimension of the shape, exact mass, centers of gravity and moments of inertia, which are the basic guarantee for the correctness of the simulation model structure.

2. Tire model should be more specified to display its nonlinear characteristics;

   Wheel model for Valmet 860.4 is Nokian 710/45 26.5, and the front wheels have different dimensions with the rear wheels, which is ignored in the simulation. Meanwhile, tire model is assumed as a simple spring damper system with linear spring stiffness and damping coefficient, is totally different from the real situation. More specific information about how to model nonlinearity of the tire in Simulink or Matlab should be focused. And this also has a close relationship with the exactness of the simulation model.

3. It is possible to reduce the vibration levels of the forwarder by changing the parameters of the passive suspension system;

   In Chapter 6.2, 3 different passive primary suspension systems are modeled with different spring stiffness and damping coefficients, and they result in various vibration levels for each component. For example, the RMS value for the seat can be reduced by 10.9 %, 9.11 % and 9.09 % by using condition 1, condition 2 and condition 3, respectively. The difference of the effects is small, but it verifies that by only changing the passive suspension system's parameters, the exposure levels of the driver can be improved. It is also shown that the improvement of the vibration levels can be achieved without changing the operation, structure dimensions or the layout of the suspension system.

4. The largest improvement was obtained by on-off controlling of the semi-active suspension at front and rear axle;

   From Chapter 6, for the acceleration level of seat, cabin, front wagon or rear wagon, semi-active suspension has the biggest effect to reduce the vibration. However, other controlling principles for semi-active suspension exist and need more research and development, it can be seen that the semi-active suspension are more efficient than passive primary suspension, cabin suspension and seat suspension. No matter how good performance that the secondary suspension can achieve, the best concentration should be on primary suspension system. The semi-active suspension condition 3 can reduce the seat vertical acceleration by 29.3%, which is really more comfortable for the operator who should sit in the cabin all the day.

5. The suspension system can contribute to the vibration reduction with higher productivity;

   As can be seen in Chapter 6.2.4, when the speed of the forwarder is doubled from 0.3 m/s to 0.6 m/s, the vibration of each component can still be reduced with the combined situation (all the suspension system are applied). As a result, this combined situation is valid and effective to reduce the vertical vibration of the forwarder, even with a higher productivity.

7.2 RECOMMENDATIONS

However, the present work with SimMechanics is not as precise as the real forwarder, further work, time and effort should be devoted to vibration reduction of off-ground machines. Several proposals for future work have been raised as follows,

1. Precise CAD models are the critical element for simulation;

   If it is preferred to use SimMechanics, CAD models in SolidWorks, Autodesk Inventor or Pro/Engineer are possible to be translated directly by using SolidWorks-To-SimMechanics translator, which allows you to save CAD assemblies in the Physical Modeling XML file format, and then SimMechanics can read these files to generate Simulink models. In this way, it is easy to realize the simulation with automatic geometry, center of gravity and moment of inertia, visualization and animation. The shape and structure of the simulation model is more close to that of the real forwarder. Opposite to this, more time and effort will be wasted to figure out the geometry, center of gravity and moment of inertia for each part of the model, which is as what has been done in this report and not precise enough as well.

2. Vibration reduction in other directions other than vertical direction should be investigated;

   Reducing the vibration exposure of lateral and longitudinal vibrations is as important as reducing the vibration exposure in the vertical direction and should therefore be investigated (J.C. Kirstein, 2005). Actually, in the real situation, especially when a machine is driving on uneven ground, the driver can feel the almost the similarly strong vibration from vertical direction and lateral direction. In the measurement, when a van goes through a bump with 60 cm length and 6 cm height, the lateral vibration peak value is as high as vertical vibration peak value, which means that human is more sensitive to lateral vibration. A lot of previous research indicates that the roll and lateral vibration of machines cannot be ignored.

3. The validity of active suspension system should be investigated and related development should be continued;

   Passive suspensions are not available for all the ground conditions, so the semi-active and active suspension should be developed because of their variable spring and damping characteristics. Although with high expense, active suspension system is still the main trend to improve suspension system. Moreover, further development of controlling principles should be deepened such as fuzzy logic control and neural network control instead of simple skyhook control or on-off control.

4. Different tire models should be modeled;

   Tire is one of the most critical components to reduce vibration and noise from uneven ground. As shown in Chapter 2, there are more precise tire models than point contact tire model. And in the report, the tire is too simple to be precise. Only a simple spring and damper system is not enough for the simulation result. Tire has nonlinear characteristics which is totally different from linearity.

5. Difference between fully loaded and empty loaded situations should be taken into consideration;

   The forwarder has a working cycle, and each cycle comprises 4 steps as follows,

   1) Driving across field to log/timber loading area (empty);
   2) Loading (empty to fully laden);
   3) Driving across field to collection site (fully laden);
   4) Unloading (fully laden to empty).

   Different steps have various load conditions, for example, in step 1) the forwarder is empty loaded while in step 3) it is almost fully loaded. In the future work, more attention should be focused on fully loaded condition, which means that the mass, center of gravity and moment of inertia of rear wagon will be changed, so the vibration level of rear wagon will be increased. In the measurement and simulation in the report, the forwarder is empty loaded, so the vibration level for each component is not as high as expected.

8 References

AFS 2005:15. "Vibrationer". Arbetarskyddsstyrelsens föreskrifter om belastnings-ergonomi samt styrelsens allmänna råd om tillämpningen av föreskrifterna, 2005.

BS 6841:1987. Guide to Measurement and evaluation of human exposure to whole-body mechanical vibration and repeated shock, 1987.

Giancarlo, G. & Lorenzo, M. 2009. Automotive chassis: Components design.

Gongyu, P., Qingqing, Z. & Lifu, C. 2009. "Model establishment and performance simulation of semi-active suspension with variable rigidity and damping", Design and Calculation, Vol 40, 2009, pp 46-49.

Paddan, G. S. & Griffin, M. J. 2002. "Effect of Seating on Exposures to Whole-body Vibration in Machines". Journal of Sound and Vibration, 253 (1), 2002, p 215–241.

ISO 2631-1:1997(E). Mechanical vibration and shock. Evaluation of human exposure to whole-body vibration. Part 1: General requirements, 1997.

Baes, J. 2008. Vibrationsdämpning av skotare.

Kirstein, J. C. 2005. "Suspension System Optimisation to Reduce Whole Body Vibration Exposure on an Articulated Dump Truck", MscEng (Mech) Thesis, Stellenbosch University, 2005.

Koen, D., Dimitrios, M., Jan, A., Josse, De. B. & Herman R. 2005. "Improvement of vibrational comfort on agricultural machines by passive and semi-active cabin suspensions", Computers and Electronics in Agriculture, Vol.49, 2005, pp. 431–440.

Komatsu Forest. 2010. http://www.komatsuforest.com/default.aspx?id=2513

Löfgren, B. 1992. "Däck–markmodeller" Forskningsstiftelsen Skogsarbeten, Kista.

MATLAB R2010a. Copyright 1984–2010, The MathWorks Inc.

Mehdi, A. & James, J. V. 2000. "*An Experimental Evaluation of Noise and Vibrations in Modern Locomotive Cabs*", Proceedings of 2000 IEEE/ASME Joint Rail Conference, Newark, New Jersey.

Karlsson, O. & Nisserud, F. 2010. "Utveckling av en dynamisk helfordonsmodell för skotare", Skogforsk.

Donati, P. 2002. "Survey of Technical Preventative Measures to Reduce Whole-body Vibration Effects When Designing Mobile Machinery", Journal of Sound and Vibration, 253(1), 2002, pp169–183.

Boileau, P.E. & Rakheja, S. 1990. "Vibration Attenuation Performance of Suspension Seats for Off-ground Forestry Machines", International Journal of Industrial Ergonomics, Vol.5, 1990, pp 275–291.

Jönsson, P. 2010. "Vibration measurement", Skogforsk.

Bosch, R. 1993. GmbH, Automotive Handbook.

The European Vibration Directive 2002/44/EC, 2002.

Wang, M. 2008."Research on vibration characteristic of scissors linkage seat with suspension of the air spring with auxiliary chamber", Nanjing Agricultural University, 2008.

Zegelaar, P.W.A., 1998. The dynamic response of tyres to brake torque unevennesses, PhD Thesis, Delft University of Technology.

Yanqing, L. Hiroshi, M. & Hideo, U. 2008. " Semi-active vibration isolation system with variable stiffness and damping control ", Journal of Sound and Vibration, 2008, pp16–28.

Appendix 1

Test track

The general profile for test track is shown in figure A1 (Skogforsk, 2010). The standardized test ground is made up of two paths, each path has different numbers of obstacles in three different sizes.

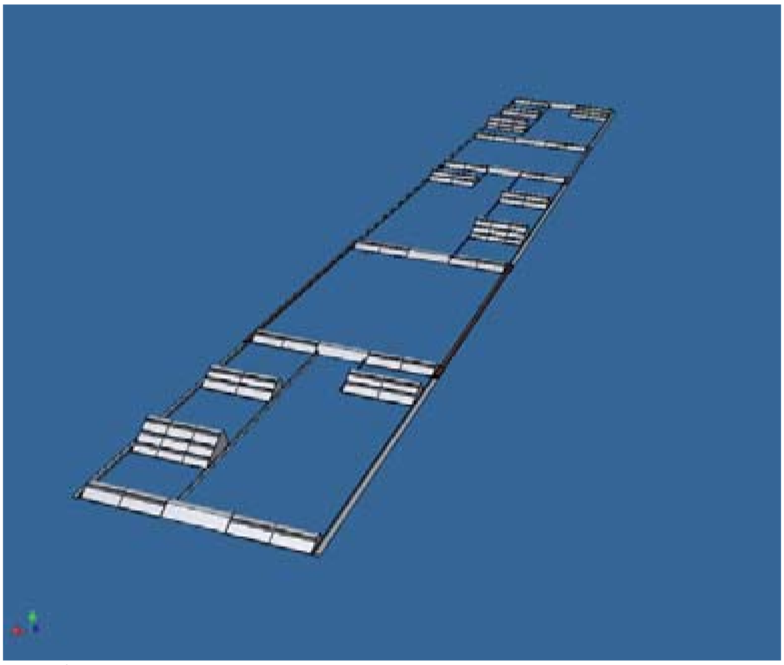


Figure 1.
General profile of test ground in Skogforsk.

Different shapes and sizes for obstacles are shown in figure A2 (Petrus Jönsson, 2010). Three different obstacles compose the track. All the dimensions in the figure are in mm. From figure A1, it is clear to tell that for both paths, they start and end with the obstacle with the height of 150 mm. The left path seems more 'irregular' than the right path, since the left path has 11 obstacles while the right path has 10 obstacles in total.

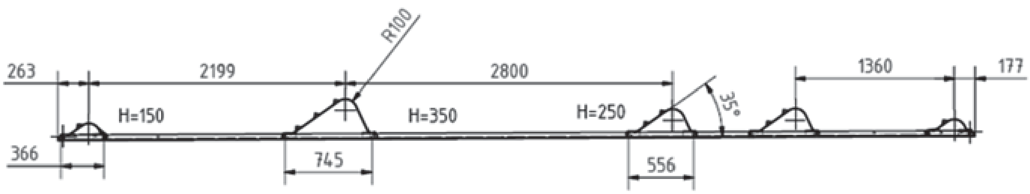


Figure 2.
Shapes and sizes for obstacles

In the measurement, the origin of the test ground is set at the zero point of X-coordinate and Z-coordinate of the right path, which means that at the beginning, the forwarder goes through the first obstacle of the right path. Table A1 (Olof Karlsson, et al., 2010) shows the coordinates of X and Z directions for both paths, all the data are recorded in mm. The color of white, green and red means the obstacle with the height of 150 mm, 250 mm and 350 mm, respectively.

Table 1.
Coordinates (X- and Z- direction) for both paths.

Left path		Right Path		Data in mm
Z-left side	Z-right side	Z-left side	Z-right side	X-coordinate
–329	–300	–269	–207	27 620
–	–	–256	–216	26 460
–308	–281	–	–	25 650
–271	–271	–	–	23 410
–267	–254	–223	–186	22 050
–249	–233	–207	–182	18 410
–230	–220	–	-	16 920
–	–	–188	–156	15 850
–	–	–141	–123	12 920
–201	–152	–114	–98	11 000
–186	–133	–95	–38	5 570
–	–	–91	–41	4 410
–198	–153	–	–	3 600
–200	–123	–	–	1 350
–187	–134	–76	0	0

Appendix 2

Component parameters

Each component properties such as mass, moment of inertia are shown in table B1.

Table 1.
Component mass properties.

Component	Number	Mass[Kg]	Moment of inertia[$\mathbf{kg \cdot m^2}$]					
			I_{xx}	I_{yy}	I_{zz}	I_{xy}	I_{xz}	I_{yz}
Front frame	1	1154.5	220.20	2034.10	3434.80	3.15	44.60	31.01
Rear frame	1	1398	129.46	3500.78	3474.46	0.85	–153.15	0.09
Cabin	1	1100	717.32	873.95	625.44	–0.40	–5.58	–3.71
Driver and seat	1	173.76	1.96	21.83	21.83	0.00	0.00	0.00
Bogie	4	953.939	51.06	355.44	365.62	0.00	–0.001	–11.40
Wheel	8	290	290.34	425.71	290.34	0.00	0.00	0.00
Bogie axle	2	392.122	29.11	12.49	32.17	0.02	0.00	0.00
Engine hood	1	90	18.39	30.18	33.62	–1.15	–4.24	–0.19
Engine installation	1	85	9.84	9.86	0.42	–0.01	0.07	–0.04
Hydraulic tank	1	549.3	94.66	53.96	65.09	–0.01	0.07	–0.01
Grind	1	670	700.71	178.08	548.73	–0.05	3.33	0.26
Bunk	3	436	595.49	189.00	410.55	0.06	–0.30	0.03
Crane	1	2216	971.41	4085.20	3236.46	0.20	–434.45	–1.75
Drivetrain	1	80	0.043	64.09	64.09	0.00	0.00	0.00

Coordinates of center of mass for each component in the simulation model are shown in table B2.

Table 2.
Coordinates of center of mass for each component.

Component	Coordinates		
	X-coordinate [mm]	Y-coordinate [mm]	Z-coordinate [mm]
Front frame	2268.9	9	145.3
Rear frame	–2321.5	0.018	160.8
Cabin	1629.32	36.21	1463.73
Driver and seat	1680.8	0	1527.7
Bogie FR	1670	–698.78	–146.3
Bogie FL	1670	698.78	–146.3
Bogie RR	–3297	–698.78	–199.8
Bogie RL	–3297	698.78	–199.8
Wheel FL1	2430	1156.78	–229.9
Wheel FL2	910	1156.78	–229.9
Wheel FR1	2430	–1156.78	–229.9
Wheel FR2	910	–1156.78	–229.9
Wheel RL1	–2537	1156.78	–283.4
Wheel RL2	–4057	1156.78	–283.4
Wheel RR1	–2537	–1156.78	–283.4
Wheel RR2	–4057	–1156.78	–283.4
Bogie shaft front	1670	0	–38.834
Bogie shaft rear	–3297	0	–92.334
Engine hood	3480.46	23.51	813.69
Engine installation	2760.9	–551	1511.6
Hydraulic tank	488.69	4.6609	858.37
Grind	–846.448	0.693	1377.669
Bunk 1	–4799	0.4	1077.1
Bunk 2	–5231	0.4	1077.1
Bunk 3	–846.448	0.693	1377.669
Crane	–1584.8	–1.2	1371.6
Drivetrain	–1550	0	0

Appendix 3

Spring and damper

Spring and damper systems for the initial forwarder are shown in table C1. The spring stiffness and damping coefficients are tuned in order to get a good match between the simulation result and measurement result.

Table 1.
pring and damper systems for initial forwarder.

Components	Spring stiffness [N/m]			Damping coefficient $[\mathrm{N \cdot s/m}]$		
	X	Y	Z	X	Y	Z
Seat suspension	–	–	15 000	–	–	7 540
Cabin bushing	440 000	1 170 000	840 000	44 000	117 000	84 000
tire	–	–	1 370 000	–	–	137 000

Arbetsrapporter från Skogforsk fr.o.m. 2010

2010	
Nr 700	Hannerz, M. & Cedergren, J. 2010. Attityder och kunskapsbehov – förädlat skogsodlingsmaterial. 56 s.
Nr 701	Rytter, R.M. 2010. Detektion av röta i bokved – resultat av mäthöjd, riktning och tidpunkt. 10 s.
NR 702	Rosvall, O. & Lundström, A. 2010. Förädlingseffekter i Sveriges skogar - kompletterande scenarier till SKA-VB 08. 31 s.
Nr 703	von Hofsten, H. 2010. Skörd av stubbar – nuläge och utvecklingsbehov. 18 s.
Nr 704	Karlsson, O. & Nisserud, F. 2010. Utveckling av en dynamisk helfordonsmodell för skotare. 73 s.
Nr 705	Eliasson, L. & Johannesson, T. 2010. Förröjningens påverkan på grotskotning – En studie av produktivitet, ekonomi, grotkvalitet hos SCA skog. 9 s.
Nr 706	Rytter, L. & Stener L.G. 2010. Uthållig produktion av hybridasp efter skörd – Slutrapport 2010 för Energimyndighetens projekt 30346. 23 s.
Nr 707	Bergkvist, I. 2010. Utvärdering av radförbandsförsök anlagda mellan 1982-1984. 16 s.
Nr 708	Hannrup, B. & Jönsson, P. 2010. Utvärdering av sågmotorn F11-iP med avseende på uppkomsten av kapsprickor – en jämförande studie. 28 s.
Nr 709	Iwarsson Wide, M., Belbo, H. 2010. Jämförande studie av olika tekniker för skogsbränsleuttag i mycket klen skog Skogsbränsleuttag med Naarva-Gripen 1500-40E och Log Max 4000, Mellanskog, Simeå 28 s.
Nr 710	Englund, M., Löfroth, C. & Jönsson, P. 2010. Inblandning av rött ljus i LED-lampor – Laboratoriestudier av hur människor uppfattar tre olika ljusblandningar. 7 s.
Nr 711	Mullin, T.J., Hallander, J., Rosvall, O. & Andersson, B. 2010. Using simulation to optimise tree breeding programmes in Europe: an introduction to POPSIM™. 28 s.
Nr 712	Jönsson, P. 2010. Hydrauliskt dämpad hytt – ett lyft för arbetsmiljön? 14 s.
Nr 713	Eriksson, B. & Sonesson, J. 2010. Tredje generationen skogsbruksplaner – Slutrapport DElproj 4 – Arbetsgång vid planläggning. 23 s.
Nr 714	Sonesson, J. 2010. Nya arbetssätt i skogsbruksplanläggning. 20 s.
Nr 715	Eliasson, L. 2010. Huggbilar med lastväxlarsystem. 13 s.
Nr 716	Eliasson, L. & Granlund P. 2010. Krossning av skogsbränsle med en stor kross – En studie av CBI 8400 hos Skellefteå Kraft. 6 s.
Nr 717	Stener, L.G. 2010. Tillväxt, vitalitet och densitet för kloner av hybridasp och poppel i sydsvenska försök. 46 s.
N 718	Palmquist, C. & Sandberg, J. & Vibrationskomfort och ergonomi på förarstolar i skotare. 100 s.
Nr 719	Thor, M. 2010. Avverkning och hantering av virke och avverkningsrester vid angrepp av tallvedsnematoder i svensk skog. 42 s.
Nr 720	Fogdestam, N. 2010. Studier av Biotassu Griptilt S35 i gallring. 11 s.
Nr 721	Brunberg, T. 2010. Bränsleförbrukningen i skogsbruket. 12 s.
Nr 722	Brunberg, T. 2010. Rätt begrepp. 25 s.
Nr 723	Löfroth, C. & Svenson, G. 2010. ETT – modulsystem för skogstransporter – Delrapport för de två första åren. 130 s.
Nr 724	Rytter, L. & Lundmark, T. 2010. Slutrapport för Energimyndighetens projekt 30658. Trädslagsförsök med inriktning på massaproduktion. – Tree species trial with emphasis on biomass production. 24 s.
Nr 725	Rytter, R.M. & Högbom, L. 2010. Slutrapport för Energimyndighetens Projekt 30659. Markkemi och fastläggning av C och N i produktionsinriktade bestånd med snabbväxande trädslag – Soil chemistry and C and N sequestration in plantations with fast-growing tree species. 64 s.

Nr 726	Brunberg, T., Eliasson, L. & Lundström, H. 2010. Skotning av färsk och hyggestorkad grot. 15 s.
Nr 727	Enström, J. 2010. Inlandsbanans potential i Sveriges skogsbränsleförsörjning. 34 s.
Nr 728	Häggström, C. & Thor, M. 2010. Human factors in forest harvester operation. 25 s.
Nr 729	Westlund, K. 2010. WP-5100 Alternative logistics concepts fitting different wood supply situations and markets. 50 s.
Nr 730	von Hofsten, H. Jämförelse mellan CeDe stubbrytare och Pallari 140. 9 s.
Nr 731	Berg, R., Bergkvist, I., Lindén, M., Lomander, A., Ring, E. & Simonsson, P. Förslag till en gemensam policy angående körskador på skogsmark för svenskt skogsbruk 18 s.
Nr 732	Jönsson, P. 2010. Stolar och armstöd – Ergonomisk granskning enligt European ergonomic and safety guidelines for forest machines. 37 s.
2011	
Nr 733	Rytter, L., Johansson, T., Karačić, A., Weih, M. m.fl. 2011. Orienterande studie om ett svenskt forskningsprogram för poppel. 210 s.
Nr 734	Hannerz, M. & Fries, C. 2011. Användningen av webbtjänsterna Kunskap Direkt och Skogsskötselserien. – En enkätundersökning bland skogsbrukets fältpersonal. 48 s.
Nr 735	Andersson, M. & Berglund, A. 2011. Test av pekskärmsmobiler. 22 s.
Nr 736	Löfgren, B., Englund, M., Forsberg, N., Jönsson, P., Lundström. L. & Wästerlund, I. 2011. Spårdjup och vibrationer för banddrivna skotare Lightlogg C och ProSilva. 32 s.
Nr 737	Brunberg, T. 2011. Studie av flerträdshantering i slutavverkning med John Deere 1470D hos SCA Skog hösten 2010.
Nr 738	Fogdestam, N. & Lundström, H. 2011. Studier av Offset Crane Concept, OCC hos Kjellbergs Logistik & Teknik i Hällefors. 15. S.
Nr 739	Bhuiyan, N., Arlinger, J. & Möller J.J. 2011. Utveckling och utvärdering av en standardiserad metod för volymbestämning och stamräkning vid avverkning med flerträdshanterande skördaraggregat. 30 s.
Nr 740	Iwarsson Wide, M. & Fogdestam, N. 2011. Jämförande studie av olika uttags-metoder av massaved och skogsbränsle i klen gallring. – Energived- och massavedsuttag med LOG MAX 4000B, Stora Enso Skog, Dalarna. 45 s.
Nr 741	Brunberg, T. 2011. Uppföljning av utbildningseffekten hos maskinlag hos SCA Skog AB 2010. 8 s.
Nr 742	Hannrup, B., Andersson, M., Bhuiyan, N., Wikgren, E., Simu, J., Skog, J. 2011. Vinnova_Slutrapport_P34138-1_101221. – Slutrapport för projekt ”Beröringsfri diametermätning i skördare – utveckling av mätsystem och tester i produktions-miljö”. 84 s.
Nr 743	Åström, Hannes. 2011. Förbättring av arbetsförhållande i skördare. 126 s.
Nr 744	Cheng. C. 2011. Modeling the Ride Comfort of a Forwarder. Modellering av åkkomforten i en skotare. 93 s.
Nr 745	Jonsson, J. 2011. Dynamisk däcksmodellering och markinteraktion för skogsmaskiner. Dynamic tire modeling and soil interaction regarding forestry machines. 52 s.
Nr 746	Grönqvist, D. 2011. Konceptutveckling av hybriddrivlina för skogsmaskiner. Concept development of a hybrid powertrain for forest machines. 180 s.