DOKUZ EYLÜL UNIVERSITY
GRADUATE SCHOOL OF NATURAL AND APPLIED SCIENCES

OPTIMIZATION OF VIBRATION COMFORT OF HEAVY DUTY TRUCKS

by
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October, 2010
İZMİR

OPTIMIZATION OF VIBRATION COMFORT OF HEAVY DUTY TRUCKS

A Thesis Submitted to the
Graduate School of Natural and Applied Sciences of Dokuz Eylül University
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Mechatronics Engineering Department, Mechatronics Engineering Programme

by
Yetkin KADER

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İZMİR

M.Sc THESIS EXAMINATION RESULT FORM

We have read the thesis entitled **"OPTIMIZATION OF VIBRATION COMFORT OF HEAVY DUTY TRUCKS"** completed by **YETKİN KADER** under supervision of **ASSOCIATE PROFESSOR ZEKİ KIRAL** and we certify that in our opinion it is fully adequate, in scope and in quality, as a thesis for the degree of Master of Science.

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OPTIMIZATION OF VIBRATION COMFORT OF HEAVY DUTY TRUCKS

ABSTRACT

In this thesis, a model based approach have been applied in order to provide a better and compact design for heavy duty trucks' cabin suspensions. A 7 DOF half truck model was built in Matlab/Simulink and a multi DOF model was built in Solidworks and Matlab/Simmechanics. For the time and frequency domain the results are evaluated accoring to ISO 2631-1 standard. Then the design options have been plotted in graphics in order to provide a compact choice of solution for cabin suspension design. The differences in the results of two simulations have been observed and the reasons are discussed. A user friendly guide user interface has been developed in order to obtain quick results and easy parameter selection. After the simulations a significant performance improvement in the ride comfort has been obtained.

Keywords: Vibration analysis, vehicle dynamics, frequency analysis, model based simulation.

AĞIR TİCARİ ARAÇLARDA TİTREŞİM KONFORU OPTİMİZASYONU

ÖZ

Bu tezde, ağır ticari araçların kabin süspansiyonları için daha iyi ve kompakt bir tasarım sağlanabilmesi için model tabanlı bir yaklaşım uygulanmıştır. Matlab/Simulink ortamında 7 serbestlik dereceli bir model ve Solidworks ve Matlab/Simmechanics ortamında çok serbestlik dereceli modeller inşa edilmiştir. Zaman ve frekans domeninde elde edilen sonuçlar uluslar arası standart olan ISO 2631-1 ‘e göre değerlendirilmiştir. Daha sonra tasarım seçenekleri daha kompakt bir kabin süspansiyonu tasarımı içi çözüm seçeneği elde edebilmek için grafik olarak çizdirilmiştir. İki simülasyon arasındaki farklar incelenmiş ve aradaki farklılıkların nedenleri araştırılmıştır. Kullanıcı dostu bir kullanıcı ara yüzü tasarlanmış ve bu sayede çabuk ve hızlı bir şekilde sonuçların elde edilebilmesi sağlanmıştır. Simülasyonların sonunda seçilen yeni parametrelerle sürüş konforunda önemli derecede artış sağlanmıştır.

Anahtar Sözcükler: Titreşim analizi, taşıt dinamiği, frekans analizi, model tabanlı benzetim.

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CHAPTER ONE

INTRODUCTION

1.1 Overview of Automotive Industry and It's Needs

In today's world, reducing the manufacturing and design costs has become one of the major challenges in almost every branch of industry. Good engineering design leads to lowering the manufacturing costs and plays an important role to obtain quality in products. As a leading industry for technology development, automotive engineering studies has been increasing rapidly since last 40 years because of the increasing demand on motor vehicles. This increasing demand brought out an enormous market which has full of competition in it. Production and research times had been shortened in order compete in the market. Especially in automotive industry a short development time for the introduction of improved or new vehicle types decides about market share and economic success (Lugner&Plöchel, 2004). Thence, placing reliable model and simulation based development instead of experimental development saves a great amount of time and money in this aspect. To obtain a reliable design, an intense care should be taken of the process of model and simulation based development.

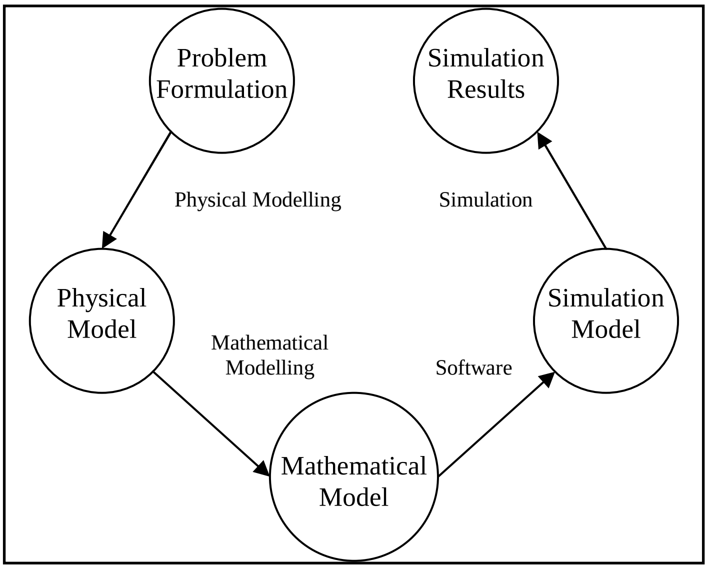

Figure 1.1 General scheme for solving a phsical problem by simulation.

One big challenge of automotive engineering is improving ride quality of motor vehicles where a great amount of worktime is spent. This thesis is primarily based on improving ride qulity of heavy duty trucks, of which ride quality is critical because of the long drive time on the road.

1.2 Definition of the Problem

With the increasing competition, the CAE tools have become more important in automotive industry. Thus, avoiding discomfort due to vibrations in both light commercial vehicles and heavy commercial vehicles is an important engineering problem today. Although semi-active and active suspension systems are performing better results in comparison with passive systems, they are still expensive. Well optimized passive suspension systems can provide a comfortable ride for long hours. Making modifications on suspension systems needs intense care. Because suspension systems have many critical tasks which might be inversly proportional to each other. Making an improvment in one, might cause a decrease in the other. Thus the tuning the suspension system in heavy duty trucks without increasing the costs is an important job. This thesis focuses on making modifications on cabin suspension system with realistic components and without avoiding the suspension system to fully fulfill other tasks.

1.3 Goal

Analyses of vibrating systems gets more difficult if the complexity of the system increases. The modelling phase depends on rigid bodies which interact each other with springs&dampers. Tuncel, Özkan&others developed a model in order to minimize the vibration transmitted cabin in 2008. The more DOF system has, the more difficult is the analytical solution is. So in investigation of the dynamic behaviour of the vehicle, some assumptions and simplifications must be done in order to simplfy solution. By making the appropriate assumptions, approximately accurate results can be achieved.

In havy duty trucks and other ground vehicle, the main reason of discomfort is the vertical excitation generated from the irregularities of the road surface. Scope of this work is decreasing the magnitude of vertical vibraitons between 4-8 Hz, in which human body is most sensitive to vertical vibrations while seated. In order to keep road handling capabilities and keeping the costs at minimum, making modifications in cabin suspension system is good choice. By investigating the time and frequency response of the vehicle for the best parameter selection of force elements (springs&dampers) in the system, a significant improvment can be achieved

CHAPTER TWO
THEORETICAL BACKGROUND

2.1 Suspension Systems

Vehicles travel at high speeds which leads to wide broadband spectrum of vibrations. These are transmitted to passengers, either by tactile, visual, or aural paths (Gillespie, 1992). The sources of vibration in vehicles depends on:

- Road irregularites,
- Tire/wheel assembly vibrations,
- Driveline vibrations,
- Engine vibrations.

The way humans percieve vibrations is described in Figure 2.1.

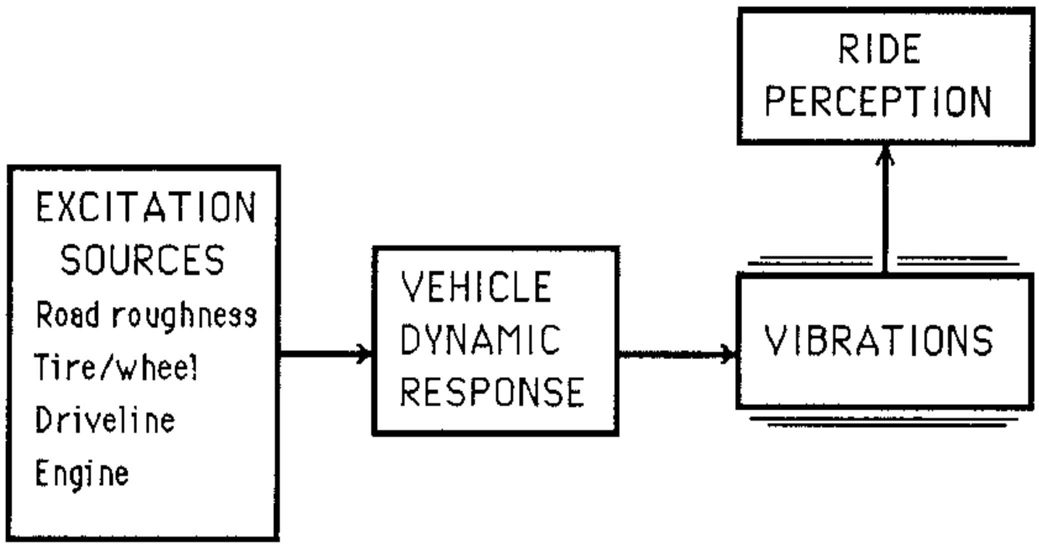

Figure 2.1 Vehicle Dynamic System (Gillespie,1992).

The major source of discomfort is the excitation due to road irregularities in most cases. Suspensions systems play a critical role in isolation of these vibrations from the body of the vehicle and the passengers. The frequency range which is considered to effect the human health, comfort and perception is between 0.5 Hz to 80 Hz according to the International Standart (ISO 2631-1, 1997). 25 Hz and above vibrations are considered as the source of noise. The full definiton of the tasks of the suspension system is made by Rajamani in 2006:

- To isolate vehicle body from road disturbances in order to provide good ride quality: Ride quality in general can be quantified by the vertical acceleration of the passenger locations. The presence of a well-designed suspension provides isolation by reducing the vibratory forces transmitted from the axle to the vehicle body. This in turns reduces vehicle body acceleration.
- To keep good road holding: The road holding performance of a vehicle can be characterized in terms of its cornering, braking and traction abilities. Improved cornering, braking and traction are obtained if the variations in normal tire loads are minimized. This is because the lateral and longitudinal forces generated by a tire depend directly on the normal tire load. Since a tire roughly behaves like a spring in response to vertical forces, variations in normal tire load can be directly related to vertical tire deflection. The road holding performance of a suspension can therefore be quantified in terms of the tire deflection performance.
- To provide good handling: The roll and pitch accelerations of a vehicle during cornering, braking and traction are measures of good handling. Half-car and full-car models can be used to study the pitch and roll performance of a vehicle. A good suspension system should ensure that roll and pitch motion are minimized.
- To support vehicle static weight: This task is performed well if the rattle space requirements in the vehicle are kept small. In the case of the quarter car model, it can be quantifiedin terms of the maximum suspension deflection undergone by the suspension.

In passenger cars, making modifications in suspension system might be difficult because of the reversely proportional tasks. But in heavy duty trucks, there are extra moving bodies on the frame such as the cabin and the seat. This makes the system more complex but on the other hand, gives the opportunity to improve ride quality without effecting road handling and road holding capabilities. In modern heavy duty trucks, the cabin suspension system consists of 4 spring&damper couples, which isolate the cabin from exciting vibrations from chasis.

Figure 2.2 Front cabin suspension (Mercedes-Benz, n.d.)

In order to keep the roll stability of the cabin a torsion bar is used in heavy duty trucks. The mechanism of cabin suspension may vary from truck to truck, but the main task is not changed: To provide a comfortable ride inside the cabin.

2.2 Vehicle Models

Vehicles are complex mechanical structures which have many DOFs. Thus, some simplifcations and assumptions are made while the dynamic behaviour of the vehicle is investigated. Thinking vehicle as sprung and unsprung rigid bodies over springs and damper is the main idea while modelling.

2.2.1 Quarter Vehicle Model

Quarter vehicle model is based on dividing the vehicle in four parts which leads two a 4 DOF in trucks. In some models the motor is also considered an oscillating body. In 1990, Güney&Ereke defined a quarter truck model with 5 DOF.

As seen below in Figure 2.3, the seat, the body, the cahsis, the motor and the axle are considered as translational rigid bodies each of which has 1 DOF. Each of them are connected to each other via spring&dampers. And the axle is interacted to the road suface with a tyre, which acts like spring&damper couple. The differential equations which represent the motion are given below.

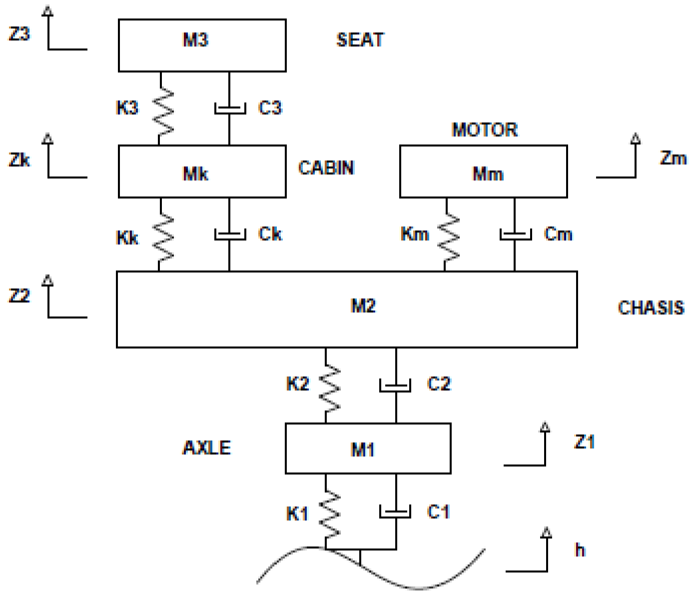

Figure 2.3 Quarter truck model (Güney&Ereke, 1990)

$$
\begin{aligned}
& m_3 \cdot \ddot{z}_3 + c_3 \cdot (\dot{z}_3 - \dot{z}_k) + k_3 \cdot (z_3 - z_k) = 0 \\
& m_k \cdot \ddot{z}_k + c_k \cdot (\dot{z}_k - \dot{z}_2) + k_k \cdot (z_k - z_2) + c_k \cdot (\dot{z}_k - \dot{z}_2) - c_3 \cdot (\dot{z}_3 - \dot{z}_k) - k_3 \cdot (z_3 - z_k) = 0 \\
& m_m \cdot \ddot{z}_m + c_m \cdot (\dot{z}_m - \dot{z}_2) + k_m \cdot (z_m - z_2) = 0 \\
& m_2 \cdot \ddot{z}_2 + c_2 \cdot (\dot{z}_2 - \dot{z}_1) + c_2 \cdot (\dot{z}_2 - \dot{z}_1) + c_k \cdot (\dot{z}_k - \dot{z}_2) - k_k \cdot (z_k - z_2) - c_k \cdot (\dot{z}_k - \dot{z}_2) - \\
& c_m \cdot (\dot{z}_m - \dot{z}_2) - k_m \cdot (z_m - z_2) = 0 \\
& m_1 \cdot \ddot{z}_1 + c_1 \cdot \dot{z}_1 + k_1 \cdot z_1 - c_2 \cdot (\dot{z}_2 - \dot{z}_1) - k_2 \cdot (z_2 - z_1) = c_1 \cdot \dot{h} + k_1 \cdot h
\end{aligned}
\tag{1}
$$

The system analyse can be made from the solution of these five equations. Although exactly correct results can not be achieved from quarter vehicle model, it is used in many applications such as semi-active and active suspension design. The model only takes the vertical vibrations into account which results a loss of data for pitch and roll motions. But it gives acceptable results for bounce motions.

2.2.2 Half Vehicle Models

Half vehicle model is good for inspecting both vertical translational and pitch motions. It cosists of five rigid bodies and 7 DOFs. 5 of them are translational and 2 of them are rotational. Because of the rotating bodies, motion equations become nonlinear and in order to solve them, some assumptions must be made.

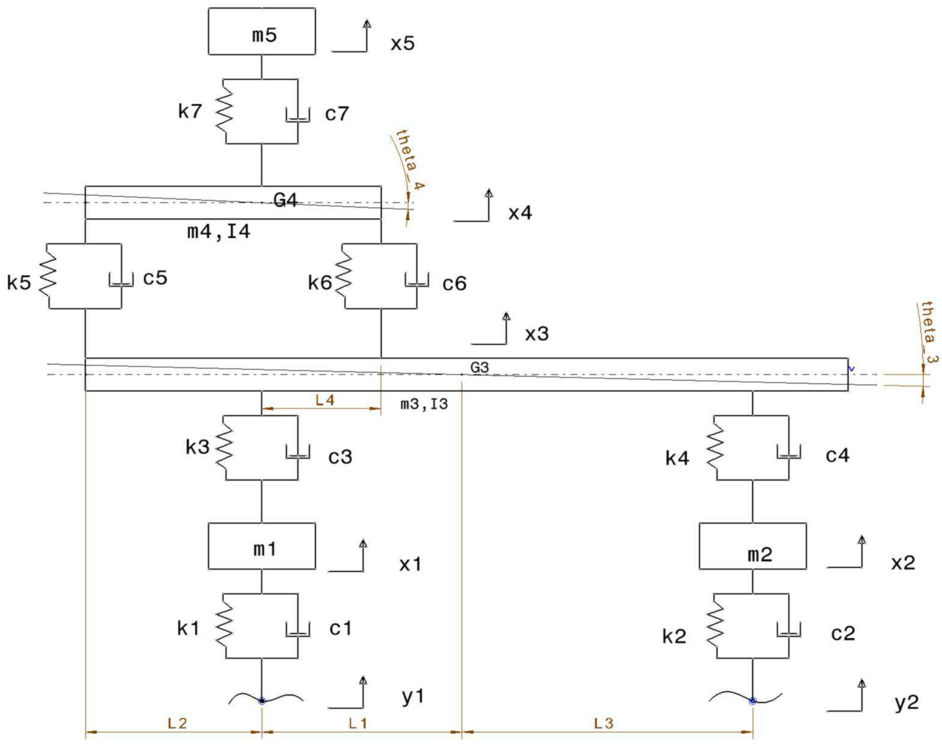

Figure 2.4 7 DOF half truck model. 5 translational and 2 rotational degree of freedom.

As seen in Figure 2.4 the representation of motion will have 7 differential equations. And the springs are assumed that they have a linear behaviour which means they have a constant stiffness. Dampers are also assumed linear. The amount of the rotational motion of cabin and the chasis are assumed very small in order to make linearization with the sine and cosine functions. The motion equations are described below.

$$m_1.\ddot{x}_1+m_2.(k_1+k_3)-k_1.y_1-k_3.x_3-k_3.L_1.\theta_3+\dot{x}_1.(c_1+c_3)$$
$$-c_1.\dot{y}_1-c_3.\dot{x}_3-c_3.L_1.\theta_3=0$$

$$m_2.\ddot{x}+x_2.(k_2+k_4)-k_2.y_2-k_4.x_3+k_4.L_3.\theta_3+\dot{x}_2(c_2+c_4)$$
$$-c_2.\dot{y}_2-c_4\dot{x}_3+c_4.L_3.\dot{\theta}_3=0$$

$$m_3.\ddot{x}_3+x_3.(k_3+k_4+k_5+k_6)+\theta_3[k_3.L_1-k_4.L_3+k_5.(L_1+L_2)$$
$$+k_6.(L_1-L_4)]-x_4.(k_5+k_6)+\theta_4.(k_6.L_4-k_5.L_2)+\dot{x}_3(c_3+c_4+c_5+c_6)$$
$$+\dot{\theta}_3.[c_3.L_1-c_4.L_3+c_5.(L_1+L_2)+c_6(L_1-L_4)]-\dot{x}_4.(c_5+c_6)$$
$$+\dot{\theta}_4.(c_6.L_4-c_5.L_2)-k_3.x_1-k_4.x_2-c_3.\dot{x}_1-c_4.\dot{x}_2=0$$

$$I_3.\ddot{\theta}_3+x_3.[L_1.k_3-L_3.k_4+(L_1+L_2).k_5+(L_1-L_4).k_6]+\theta_3[L_1^2.k_3+$$
$$L_3^2.k_4+(L_1+L_2)^2.k_5+(L_1-L_4)^2.k_6]-x_4.[(L_1+L_2).k_5+(L_1-L_4).k_6]$$
$$-\theta_4.[L_2.(L_1+L_2).k_5-L_4.(L_1-L_4).k_6]+\dot{x}_3[c_3.L_1-c_4.L_3+c_5.(L_1+L_2)$$
$$+c_6.(L_1-L_4)]+\dot{\theta}_3.[c_3.L_1^2+c_4.L_3^2+c_5.(L_1+L_2)^2+c_6.(L_1-L_4)^2]- \tag{2}$$
$$\dot{x}_4.[c_5.(L_1+L_2)+c_6.(L_1-L_4)]-\dot{\theta}_4.[c_5.(L_1+L_2).L_2-c_6.(L_1-L_4).L_4]$$
$$-L_1.k_3.x_1+L_3.k_4.x_2-c_3.L_1.\dot{x}_1+c_4.L_3.\dot{x}_2=0$$

$$m_4.\ddot{x}_4+x_4.(k_5+k_6+k_7)+\theta_4.(k_5.L_2-k_6.L_4)-\theta_3.[k_5.(L_1+L_2)+$$
$$k_6.(L_1-L_4)]-x_3.(k_5+k_6)-k_7.x_5+\dot{x}_4.(c_5+c_6+c_7)+\dot{\theta}_4.(c_5.L_2-c_6.L_4)$$
$$-\dot{\theta}_3.[c_5.(L_1+L_2)+c_6.(L_1-L_4)]-\dot{x}_3.(c_5+c_6)-c_7.\dot{x}_5=0$$

$$I_4.\ddot{\theta}_4+x_4.(k_5-k_6.L_4)+\theta_4.(k_5.L_2^2+k_6.L_4^2)-x_3.(k_5.L_2-k_6.L_4)-$$
$$\theta_3.[k_5.L_2.(L_1+L_2)-k_6.L_4.(L_1-L_4)]+\dot{x}_4.(c_5.L_2-c_6.L_4)+$$
$$\dot{\theta}_4.(c_5.L_2-c_6.L_4^2)-\dot{x}_3.(c_5.L_2-c_6.L_4)-\dot{\theta}_3.[c_5.L_2.(L_1+L_2)-$$
$$c_6.L_4.(L_1-L_4)]$$

$$m_5.\ddot{x}_5+k_7.(x_5-x_4)+c_7.(\dot{x}_5-\dot{x}_4)=0$$

Seven differential equations can represent the vertical translational motion and the pitch motion of both the cabin and the chasis. The characteristic equation of the given system is stated below by the Equaiton 3. By solution of the characteristic equation we can achive the eigenvalues of the system which leads us to natural frequencies of the system.

$$[m].\{\ddot{x}\}+[c].\{\dot{x}\}+[k].\{x\}=\{u\} \tag{3}$$

Where [m] is the mass matrix, [c] is the damping matrix and [k] is the stiffness matrix. Rearranging the equations we can obtain the mass matrix as;

$$\begin{bmatrix} m_1 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & m_2 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & m_3 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & I_3 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & m_4 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & I_4 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & m_5 \end{bmatrix} \begin{pmatrix} \ddot{x}_1 \\ \ddot{x}_2 \\ \ddot{x}_3 \\ \ddot{\theta}_3 \\ \ddot{x}_4 \\ \ddot{\theta}_4 \\ \ddot{x}_5 \end{pmatrix} \tag{4}$$

By the same way, making the appropriate arrangements in the equations, the damping matrix can be obtained as;

$$\begin{bmatrix} (c_1+c_3) & 0 & -c_3 & -c_3.L_1 & 0 & 0 & 0 \\ 0 & (c_2+c_4) & -c_4 & c_4.L_3 & 0 & 0 & 0 \\ -c_3 & -c_4 & (c_3+c_4+c_5+c_6) & \begin{bmatrix} c_3.L_1-c_4.L_3+ \\ c_5.(L_1+L_2)+c_6.(L_1-L_4) \end{bmatrix} & -(c_5+c_6) & [c_6.L_4-c_5.L_2] & 0 \\ -c_3.L_1 & c_4.L_3 & \begin{bmatrix} c_3.L_1-c_4.L_3+ \\ c_5.(L_1+L_2)+c_6.(L_1-L_4) \end{bmatrix} & \begin{bmatrix} c_3.L_1^2+c_4.L_3^2+ \\ c_5.(L_1+L_2)^2+c_6.(L_1-L_4)^2 \end{bmatrix} & -\begin{bmatrix} c_5.(L_1+L_2)+ \\ c_6.(L_1-L_4) \end{bmatrix} & -\begin{bmatrix} c_5.(L_1+L_2).L_2 \\ +c_6.(L_1-L_4).L_4 \end{bmatrix} & 0 \\ 0 & 0 & -(c_5+c_6) & -\begin{bmatrix} c_5.(L_1+L_2)+ \\ c_6.(L_1-L_4) \end{bmatrix} & (c_5+c_6+c_7) & [c_5.L_2-c_6.L_4] & -c_7 \\ 0 & 0 & [c_6.L_4-c_5.L_2] & -\begin{bmatrix} c_5.(L_1+L_2).L_2 \\ +c_6.(L_1-L_4).L_4 \end{bmatrix} & [c_5.L_2-c_6.L_4] & [c_5.L_2^2+c_6.L_4^2] & 0 \\ 0 & 0 & 0 & 0 & -c_7 & 0 & c_7 \end{bmatrix} \begin{bmatrix} \dot{x}_1 \\ \dot{x}_2 \\ \dot{x}_3 \\ \dot{\theta}_3 \\ \dot{x}_4 \\ \dot{\theta}_4 \\ \dot{x}_5 \end{bmatrix} \tag{5}$$

As seen above, the matrix with Equation number 5, is simetric along it’s diagonal which is the proof of calculations in first sight. The stiffness matrix can be formed again by making the appropriate arrangement in the motion equations. It is also simetric along it’s diogonal. The stiffness matrix is given below.

$$\begin{bmatrix} (k_1+k_3) & 0 & -k_3 & -k_3.L_1 & 0 & 0 & 0 \\ 0 & (k_2+k_4) & -k_4 & k_4.L_3 & 0 & 0 & 0 \\ -k_3 & -k_4 & (k_3+k_4+k_5+k_6) & \begin{bmatrix} k_3.L_1-k_4.L_3+ \\ k_5.(L_1+L_2)+k_6.(L_1-L_4) \end{bmatrix} & -(k_5+k_6) & [k_6.L_4-k_5.L_2] & 0 \\ -k_3.L_1 & k_4.L_3 & \begin{bmatrix} k_3.L_1-c_4.L_3+ \\ k_5.(L_1+L_2)+k_6.(L_1-L_4) \end{bmatrix} & \begin{bmatrix} k_3.L_1^2+k_4.L_3^2+ \\ k_5.(L_1+L_2)^2+k_6.(L_1-L_4)^2 \end{bmatrix} & -\begin{bmatrix} k_5.(L_1+L_2)+ \\ k_6.(L_1-L_4) \end{bmatrix} & -\begin{bmatrix} k_5.(L_1+L_2).L_2 \\ +k_6.(L_1-L_4).L_4 \end{bmatrix} & 0 \\ 0 & 0 & -(k_5+k_6) & -\begin{bmatrix} k_5.(L_1+L_2)+ \\ k_6.(L_1-L_4) \end{bmatrix} & (k_5+k_6+k_7) & \left[k_5.L_2-k_6.L_4\right] & -k_7 \\ 0 & 0 & [k_6.L_4-k_5.L_2] & -\begin{bmatrix} k_5.(L_1+L_2).L_2 \\ +k_6.(L_1-L_4).L_4 \end{bmatrix} & \left[k_5.L_2-k_6.L_4\right] & \left[k_5.L_2^2+k_6.L_4^2\right] & 0 \\ 0 & 0 & 0 & 0 & -k_7 & 0 & k_7 \end{bmatrix} \begin{bmatrix} \dot{x}_1 \\ \dot{x}_2 \\ \dot{x}_3 \\ \dot{\theta}_3 \\ \dot{x}_4 \\ \dot{\theta}_4 \\ \dot{x}_5 \end{bmatrix} \quad (6)$$

By taking the determinant of the charateristic polynomial, we can achive the damped natural frequencies. Here is the example Matlab code to do this job. The stiffness, damping and mass matrices which are a part of the code are not given here.

```
clc
clear

% Masses and moment of inertias %
m1=450/2;           I3=2993/2;
m2=450/2;           I4=386/2;
m3=2089/2;
m4=1000/2;
m5=125;

% Spring and damper constants %
k1=900000;          c1=2000;
k2=900000;          c2=2000;
k3=250000;          c3=7500;
k4=250000;          c4=7500;
k5=180000;          c5=6000;
k6=60000;                 c6=2000;
k7=10000;                 c7=100;

% Geometrical dimensions %
L1=2.323;
L2=1.160;
L3=1.875;
L4=0.402;

syms s;

kpol=M*s^2+C*s+K;

kpoldet=det(kpol);

p=solve(kpoldet)/(2*pi)

vpa(p,12)
```

With the parameters given above the natural frequency of the seat mass is found around 1.25 Hz. The effect of parameters on damped natural frequency can be shown clearly with the aid of this code. The effect of natural frequency on frequency response will be mentioned in chapter 4.

2.2.3. Full Vehicle Model

Full vehicle model consists of many joints and bodies. Full vehicle models are usually used for cars. In trucks the number of equations and system complexity increase grealty and system becomes very difficult to solve analytically. In this thesis a full truck model devoloped in Matlab/Simmechanics will be investigated in chapter seven.

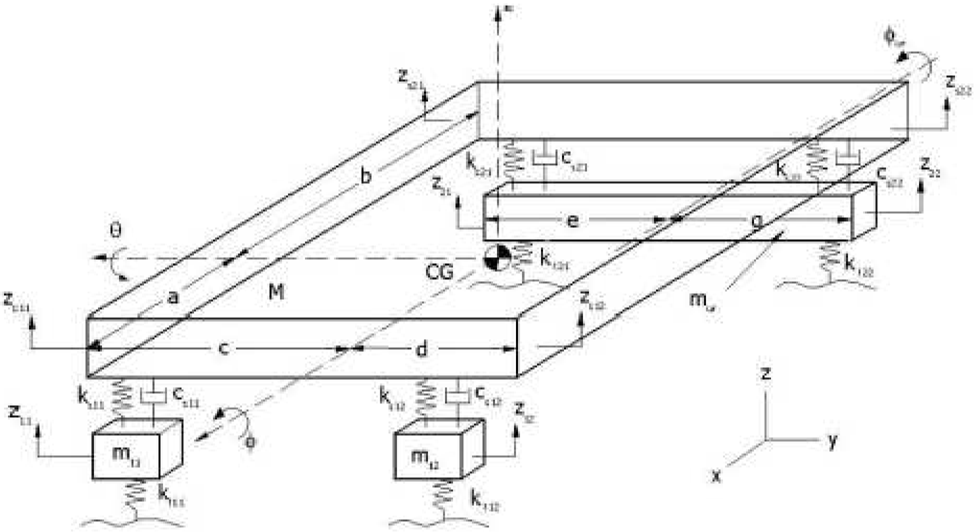

Figure 2.5 A 7 DOF full car model (Emekli, 2008).

In Figure 2.5, a full car model is shown. It has 3 rotational and 4 translational degrees of freedom. The model is devolep for building a semi-active suspension control system for a light transport vehicle. With the aid of this model, roll and pitch motions of the car body can be monitored and controlled.

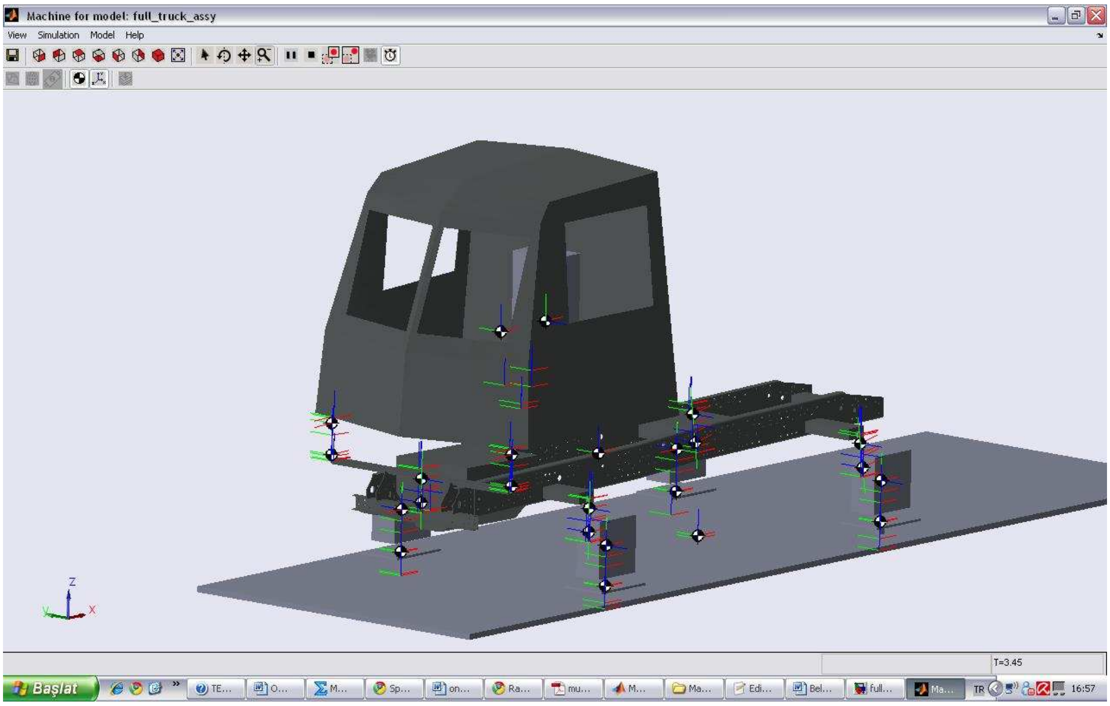

Figure 2.6 A full truck model developed in Solidworks and Matlab/SimMechanics.

In Figure 2.6, a full truck model which consists of many DOFs. It’s designed in Solidworks and translated to Matlab environment through CAD Translator software. With the aid of this model, complete vibration motions of the vehicle can be observe, if desired a control system can be implemented to simulation.

2.3 Road Modelling

Defining the road is a very difficult concept since the real road profiles itself have a stochastic structure. Road profile consists of local irregularites, potholes, bumps and roughness on the road surface. Therefore the precise modelling of the road is imposible. Besides single obstacles or track grooves the irregularities of a road are of stochastic nature. A vehicle driving over a random road prodile mainly performs hub, pitch and roll motions. The local inclination of the road profile also induces longitudinal and lateral motions as well as yaw motions. On normal roads the latter motions have less influence on ride comfort and ride safety. To limit the effort of the stochastic description usually simpler road models are used (Rill, 2005). We can form the road models in two ways: Deterministic and random profiles.

2.3.1 Deterministic Profiles

Deterministic profiles are usuallay based on making estimations, assumptions and simplifications on road profiles.

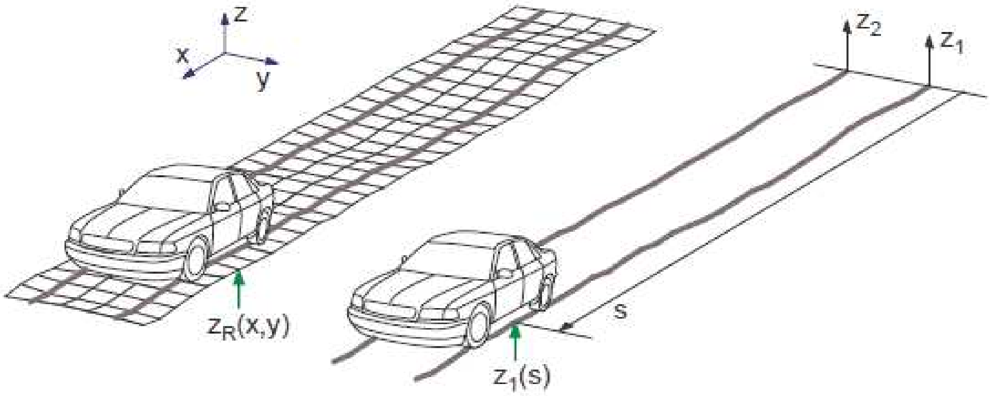

Figure 2.7 Parallel track road model (Rill, 2005).

For example in Figure 2.7 shown above, the two tracks of the car are assumed to be identical to each other, naturally the input is degraded to one. And also the local inclinations are not taken into account if not stated. These simplification has to be done in order to form an input output relation between the road surface and the vehicle.

When the potholes and bumps are considered which have nearly arbitrary shape, they are single obstacle on the road surface. Already with simple rectangular cleats the dynamic reaction of a vehicle or a single tire to a sudden impact can be investigated. If the shape of the obstacle is approximated by a smooth function, like a cosine wave, then, discontinuities will be avoided (Rill, 2005).

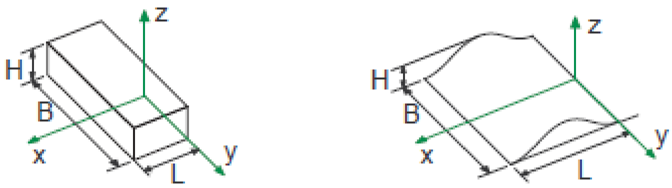

Figure 2.8 Cleat and cosine shaped bump (Rill, 2005)

Above in Figure 2.8, the representation of a cleat and a cosine shaped bump can be seen. Given the dimensions in the Figure, mathematical functions can be obtained. The function of a cleat can be written as;

$$z(x,y)=\begin{cases} H & if \quad 0<x<L \quad and \quad -\frac{1}{2}\cdot B<y<\frac{1}{2}\cdot B \\ 0 & else \end{cases} \tag{7}$$

The representation of a bump on the other hand is obtained by a similar way. It asumes the bump as it is cosine shaped. The mathematical function of the bump can be written as;

$$z(x,y)=\begin{cases} \frac{1}{2}\cdot H\cdot\left(1-\cos\left(2\pi\frac{x}{L}\right)\right) & if \quad 0<x<L \quad and \quad -\frac{1}{2}B<y<\frac{1}{2}B \\ 0 & else \end{cases} \tag{8}$$

where H, B and L denote height, width and length of the obstacle. Potholes are the opposite of bump and cleats. The negative values of the height (H) is used.
For hormonic motion representation, sine waves are appropriate to use. A general representation for sinewave can be expressed as;

$$z(s)=A.\sin\left(\Omega.s\right) \tag{9}$$

where s is the path variable, A represents the path amplitude and Ω is the wave number. If there is a phase difference, the equation (9) becomes;

$$z(s)=A.\sin\left(\Omega.s-\Psi\right) \tag{10}$$

where Ψ is the phase lag. The harmonic excitation is needed when the frequency behaviour of the vehicle is to be inspected. By using the deterministic road models, the time and the frequency behaviours of vehicle can be investigated and different design secenarios can be compared to each other.

2.3.2 Random Profiles

Random profiles are generated by statistical approaches. Rill (2005) stated that the road profiles fit the category of stationary Gaussian random processes. Hence the irregularities of a road can be described either by profile itself $z_R = z_R$ (s).

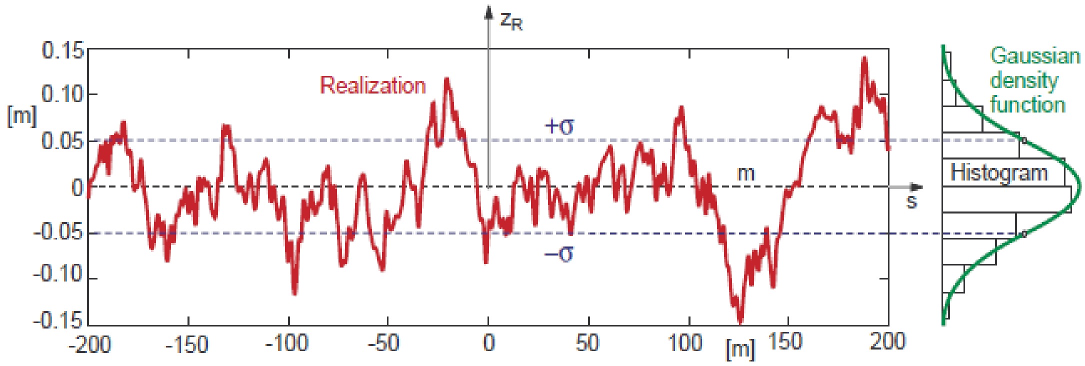

Figure 2.9 Road profile and statistical properties (Rill, 2005).

In this thesis deterministic profiles will be used to investigate the frequency and time response of the vehicle.

2.4 Time and Frequency Response Analyses

Investigation of vehicle dynamics mainly depends on two analyses. One of them is time domain analysis which represents the vehicle's response to specific input with specific configuration in a limited time interval. This type of analyses are very useful to observe the response of the dynamic system to a specific input and to compare various vehicle suspension configurations. In Figure 2.10, a typical time response of truck for a step input is shown. The settling time, the overshoot can be observed clearly on time response.

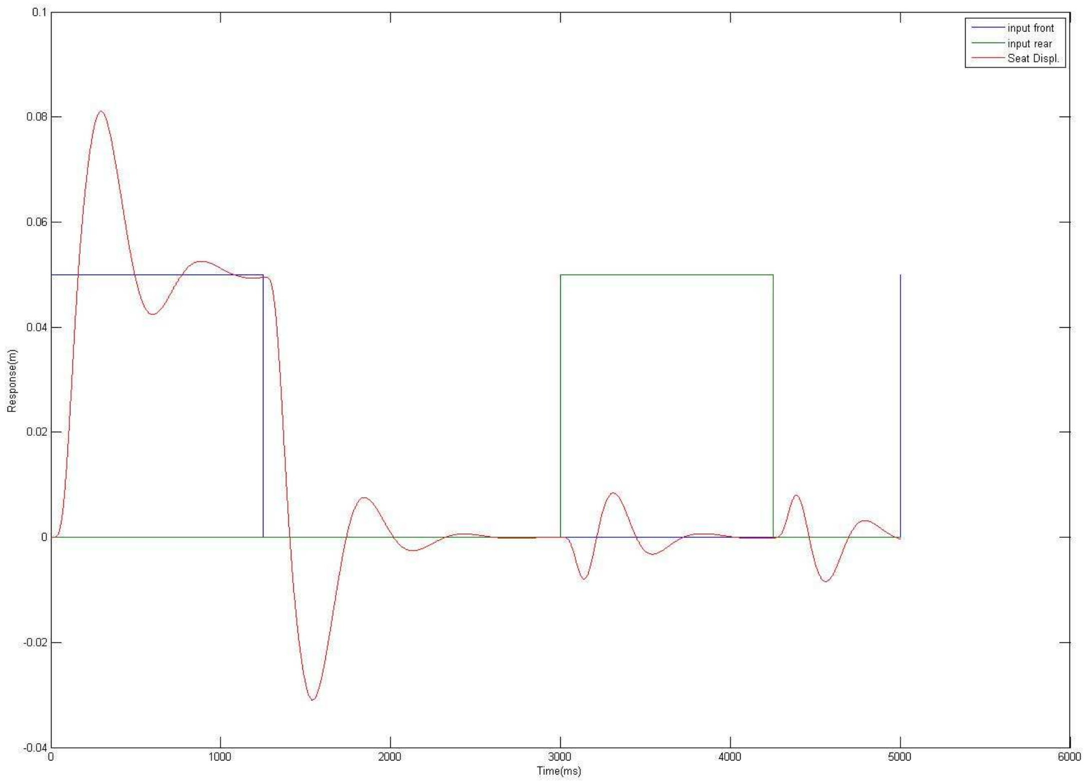

Figure 2.10 Time response of truck-seat for a step input of front axle and rear axle respectively.

The input to the front axle creates displacement which is much bigger than the displacement that is created by the rear axle input. The detailed investigation of the time response of a truck will be investigated in chapter four.

The investigation of vehicle dyamics for harmonic inputs which mainly consists of vibration in real life; depends on frequency response analyze tecniques. Frequency domain is independent from time, calculates the response of dynamic system to harmonic inputs. There are analytical ways of calculating frequency response of a linear system such as Bode plots and Nyquist diagrams. But the nature of dynamic systems are highly non-linear. Therfore numerical approaces are usually a more widespread technique for calculating frequency response. Modelling of the dynamic system with a high complexity are now simpler thanks to CAE tools developed today. In Figure 2.11, there is a typical frequency response of a truck shown.

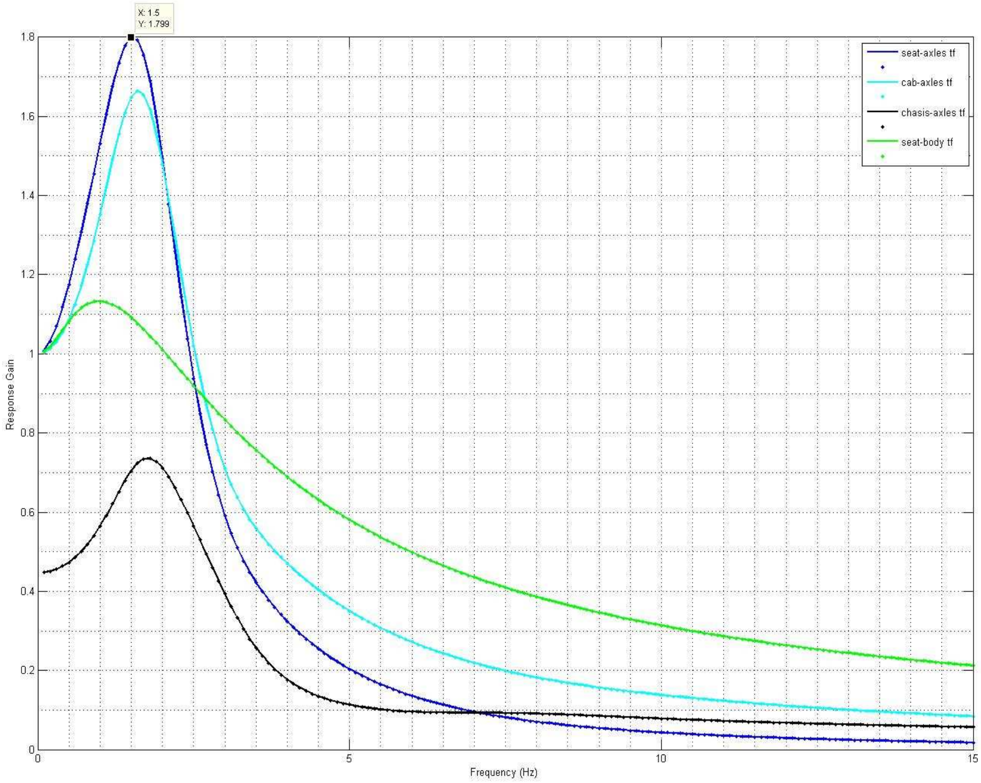

Figure 2.11 Typical frequency response of a truck for an excitation spectrum of 0-15 Hz.

The horizontal axis shows the frequency excitation scale and the vertical axis shows the gain of the response for the input. The driver feels the vibrations higher when the response gain is over 1. And the amplitudes of vibrations are decreased in the value of gains lower than 1. This representation is also known as transfer function.

CHAPTER THREE
MODELLING OF A HEAVY DUTY TRUCK

3.1 Concept of Modelling

In cahpter 2, the motion equations of a half truck model were stated. The modelling study in this thesis will be carried out over these half truck equations. Matlab/Simulink is a powerful tool for modelling dynamic systems. It can be used for solving differential equations which are highly needed for modelling of dynamical systems. In Figure 3.1, a spring mass damper system is shown.

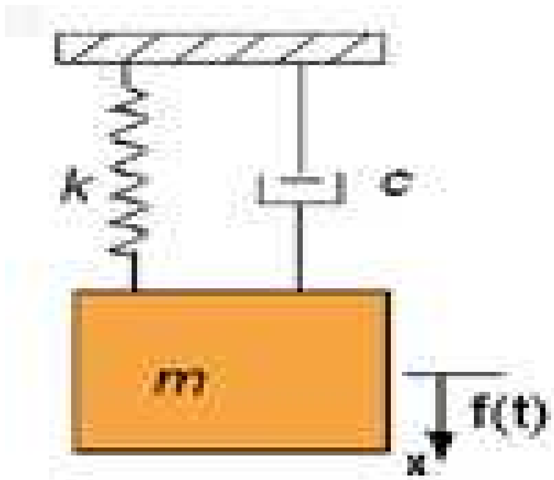

Figure 3.1 Spring-mass-damper system

The differential equation of the system is shown below in Equation (11).

$$m.\ddot{x} + c\dot{x} + kx = f(t) \tag{11}$$

where m is the mass, c is the damping coefficient and k is the spring stiffness. The system can be modelled in Simulink easily.

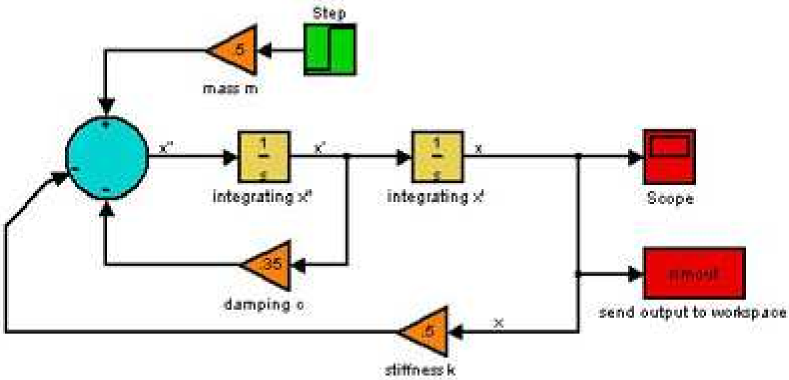

Figure 3.2 The simulink block diagram of mass-sprin-damper system

In Figure 3.2, the block diagram of the mass-spring-damper system can be observed. The modelling in simulink mainly depends of integration of acceleration variable through velocity and postion. Then velocity and position are fedback to summation point after they are multiplyed by the spring and damper coefficients. The variable which is to be observed can be viewed throgh a scope and transfered to the Matlab workspace for data processing.

3.2 Building a 7 DOF Model in Matlab/Simulink Environment

3.2.1 General Overview of the Model

The 7 DOF model cosists of seven motion equations which was mentioned in cahpter two. The schematic and motion equations were clearly stated. According to the schematic and motion equations the construction of the model are shown below in Figures.

Building the model in simulink is based on block diagrams which performs the necessary mathematical functions and operations. The more system gets complex, the more system gets harder to model it in Simulink. In this aspect creating subsytems is very useful. Below in Figure 3.3, the model developed in Simulink has two inputs consists of 7 subsystems giving out 7 outputs.

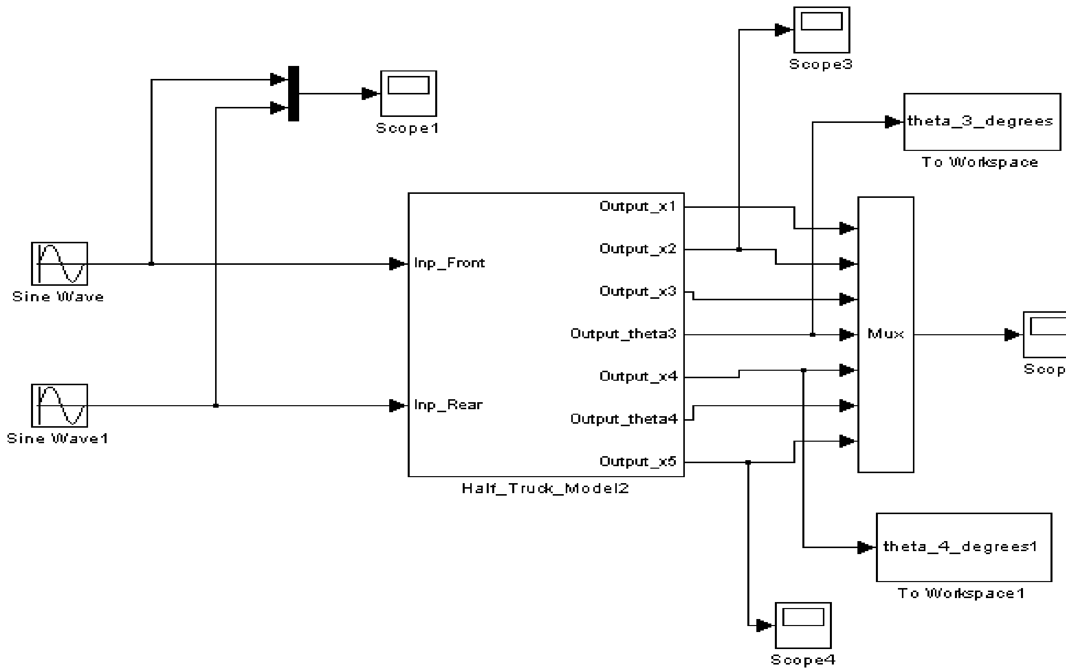

Figure 3.3 The 7 DOF model constructed in Simulink.

The outputs are displacements of the translational masses which are; seat, body, chasis and axles. And the others are rotations of the body and the chassis. Velocity and acceleration informations can be obtained easily by using derivation operators. User defined functions or builtin Matlab functions can be used as inputs.

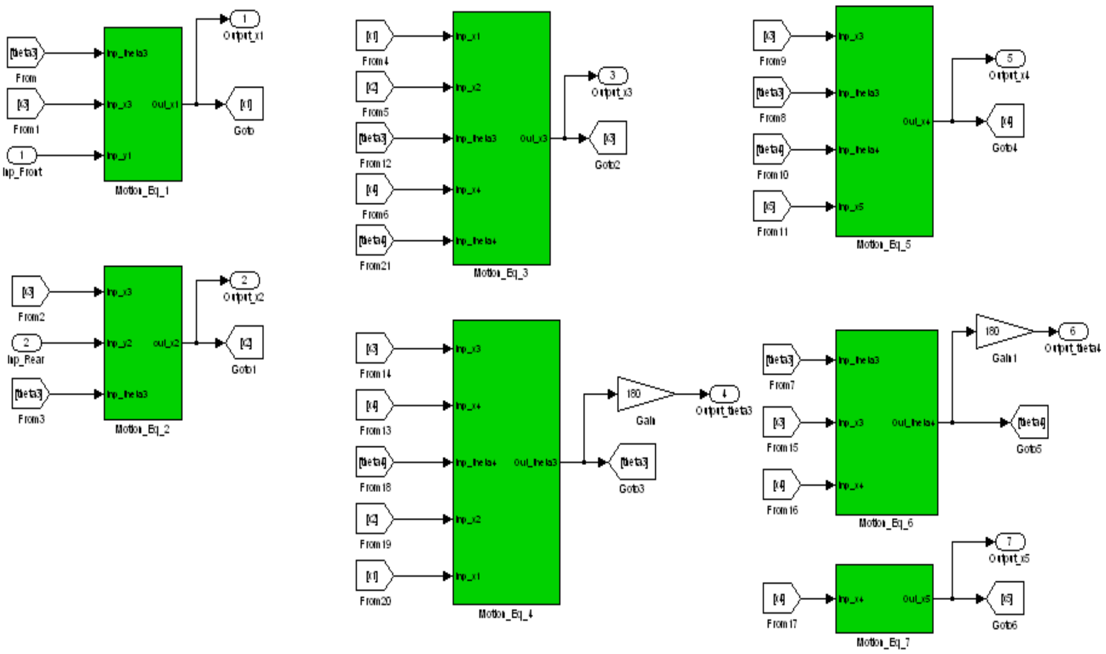

Figure 3.4 Seven motion equations in seven subsystem blocks. The subsystems are connected to each other. Each subsytem has inputs and outputs.

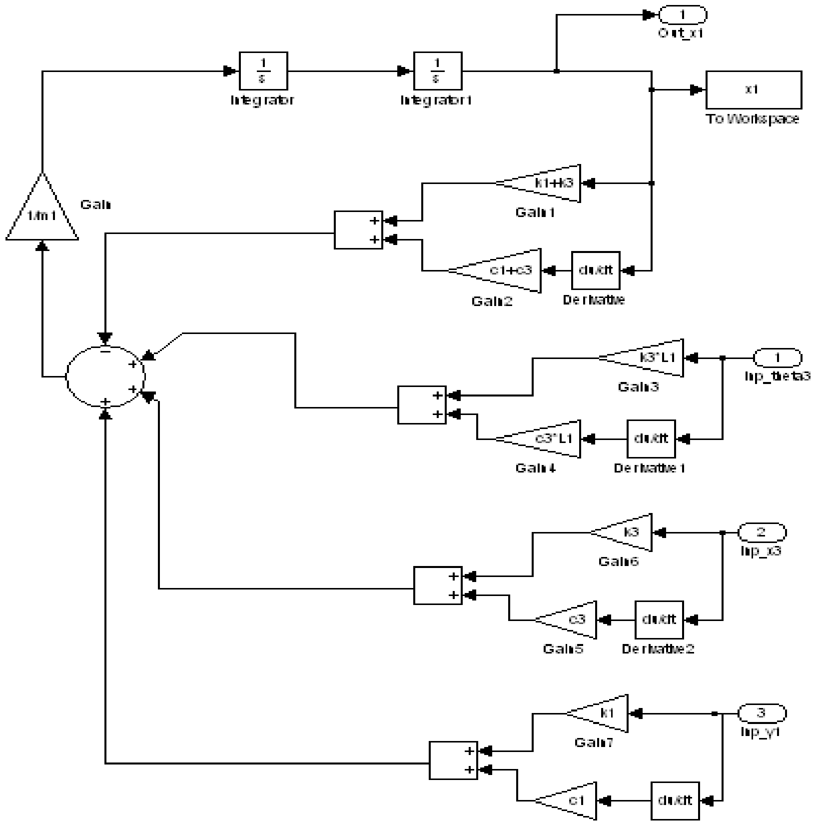

Figure 3.5 Motion equation for front axle modelled in simulink.

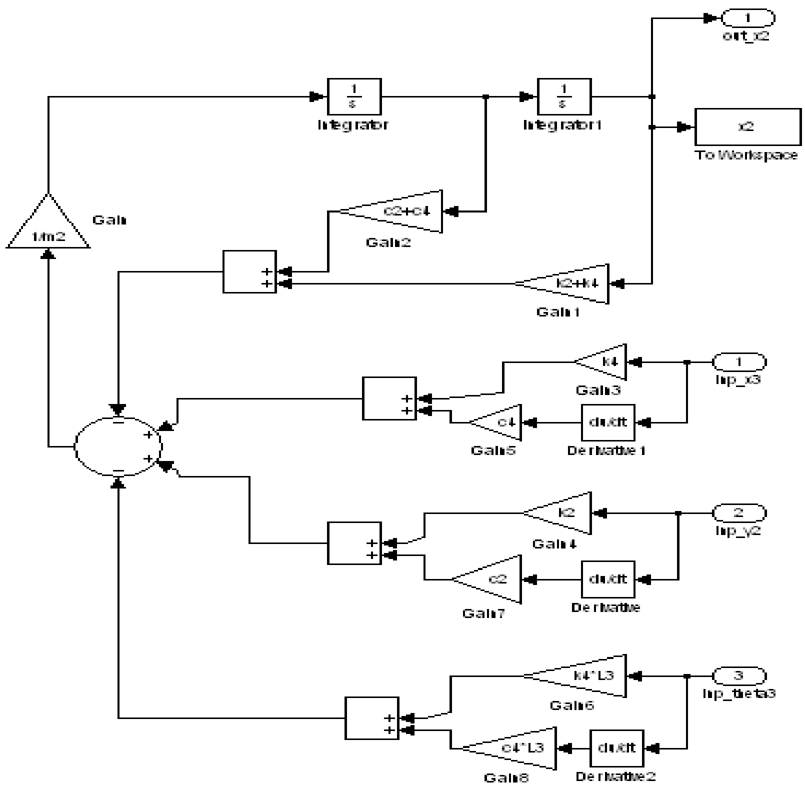

Figure 3.6 Motion equation for rear axle modelled in simulink.

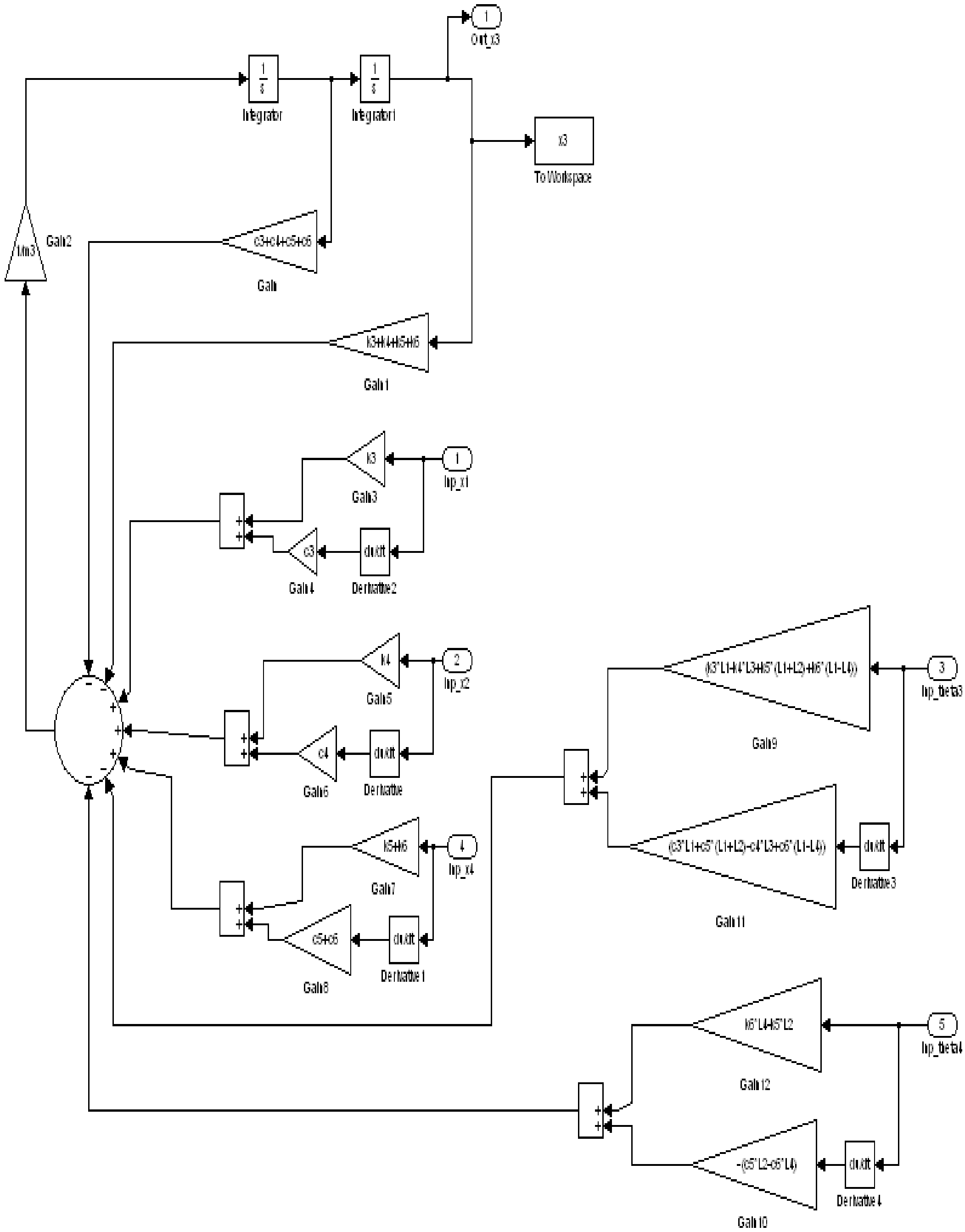

Figure 3.7 Motion equation for chasis in translational mode modelled in simulink.

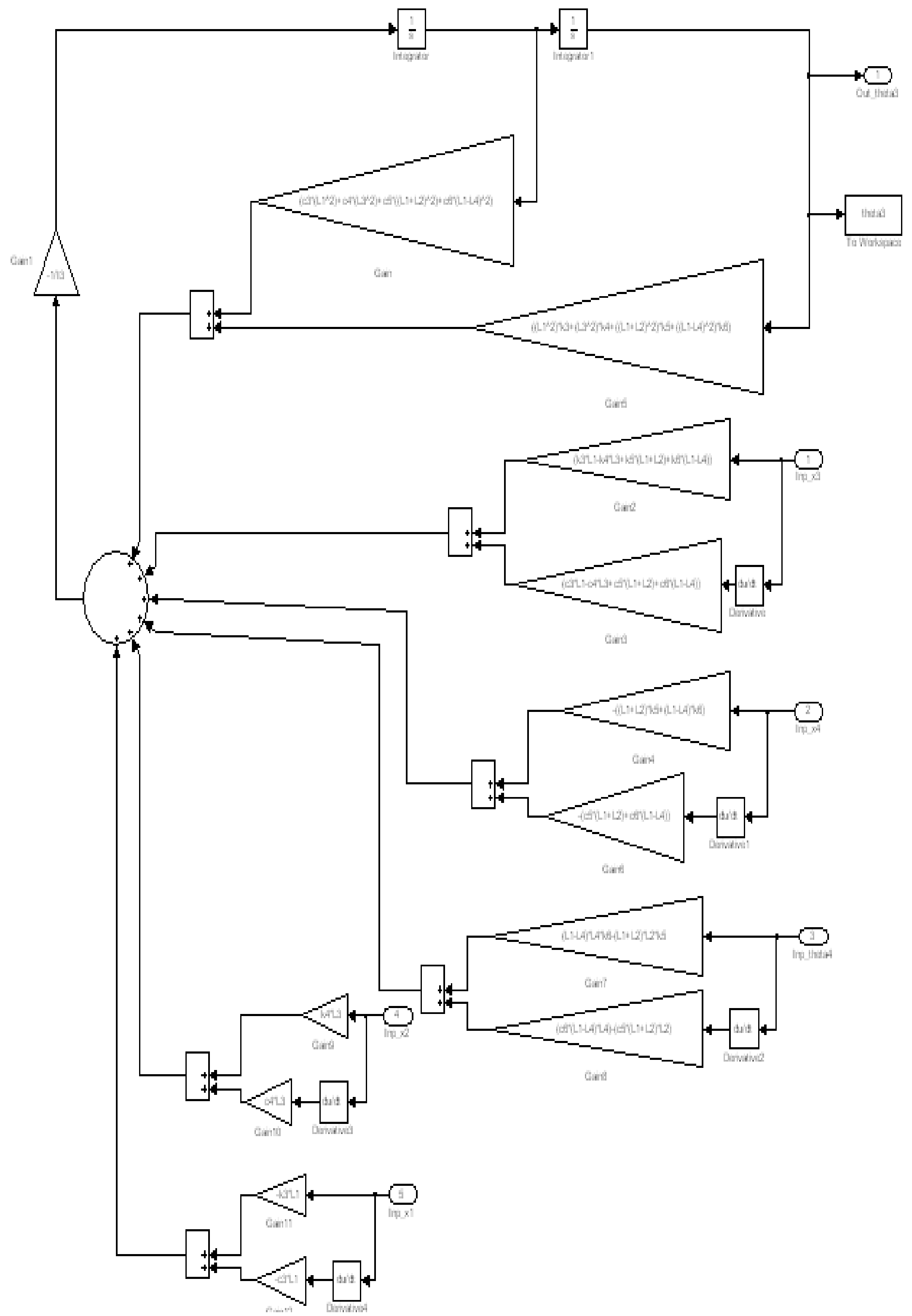

Figure 3.8 Motion equation for chasis in rotational mode modelled in simulink.

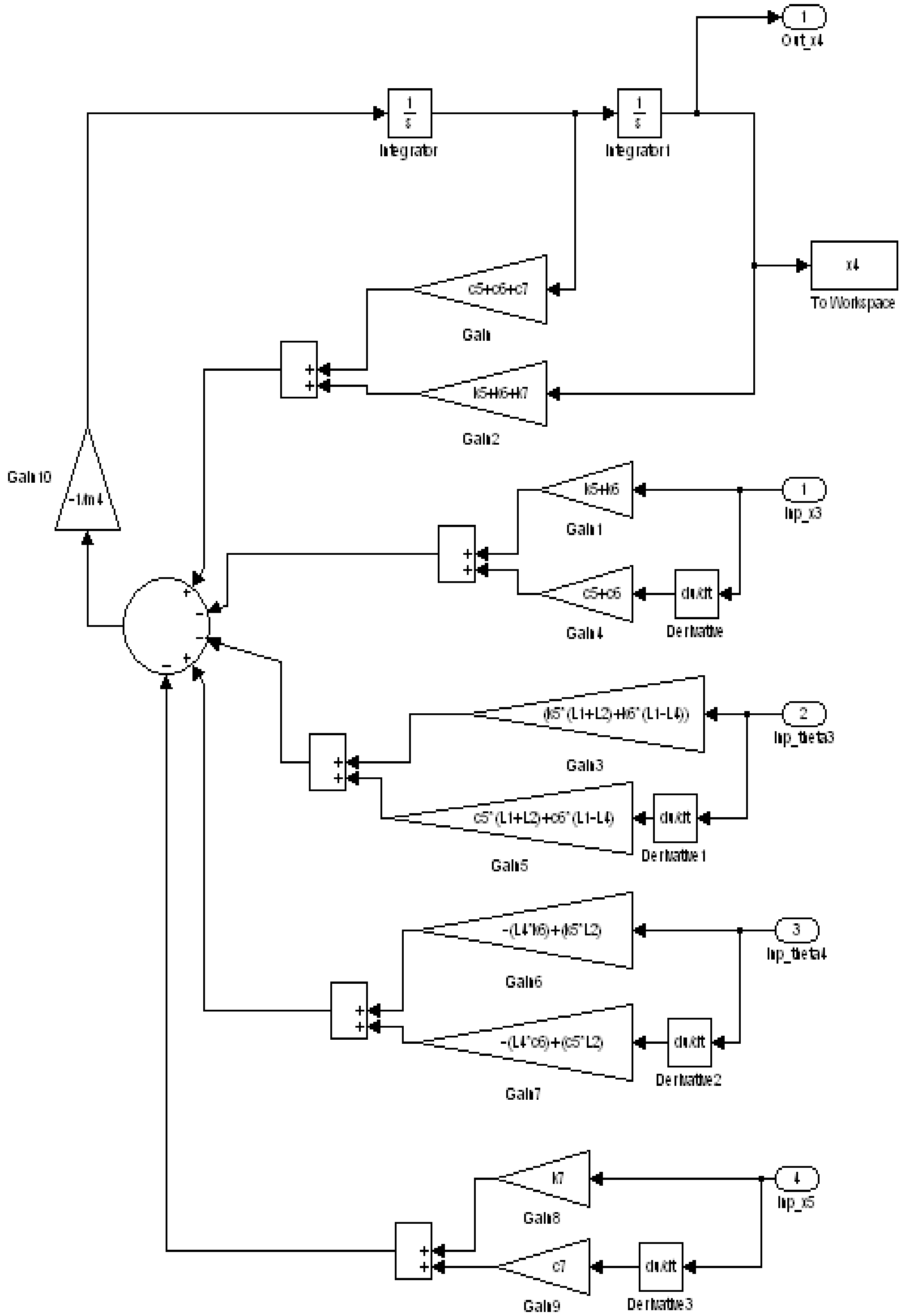

Figure 3.9 Motion equation for cabin in translational mode modelled in simulink.

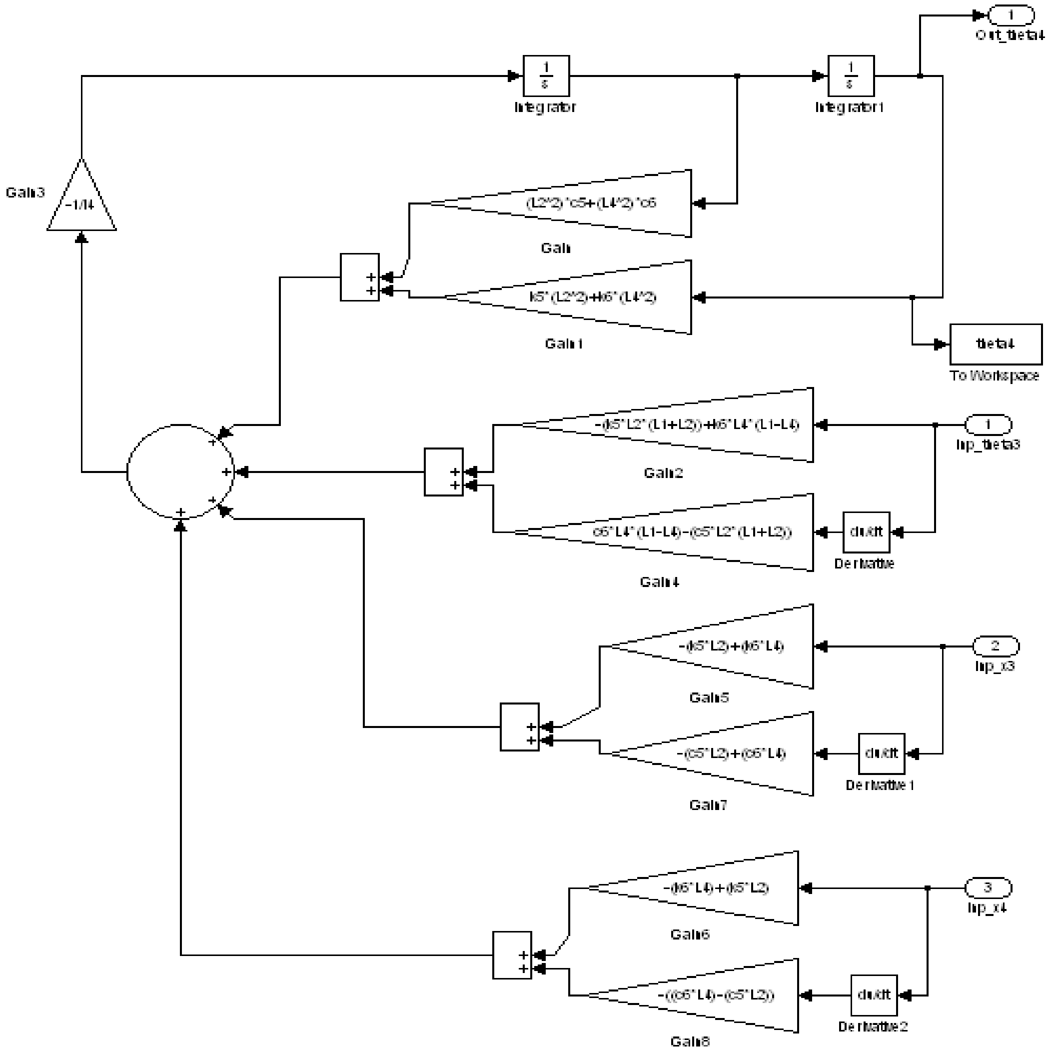

Figure 3.10 Motion equation for cabin in rotational mode modelled in simulink.

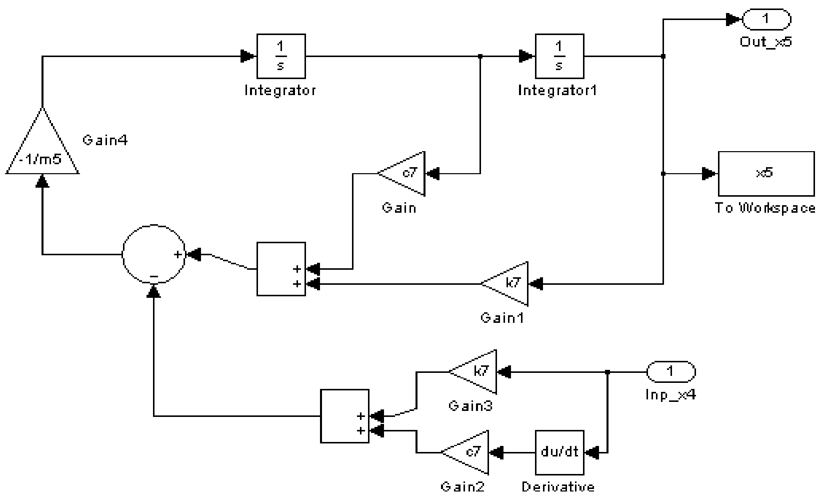

Figure 3.11 Motion equation for cabin in translational mode modelled in simulink.

As seen in the figures above, the models are in a input-output relation to each other. Solving 7 equations together provides the outputs that give valuable information of the vibration and the ride characteristics of the vehicle.

3.2.2 Necessary Assumptions and Realizations to Build the Model

The vehicle itself is comlex dynamic system which have nonlinear differential equations that are difficult to solve. In order to solve the differential equations, some linearizations are made. The rotation motion around the center of gravity chasis are assumed to be small.

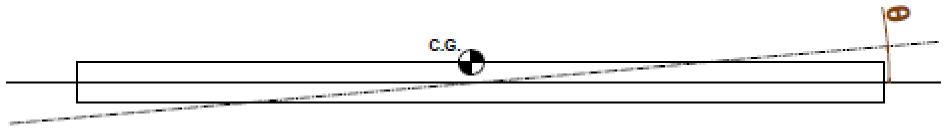

Figure 3.12 The rotation of rigid body around the center of gravity.

It is assumed small enough to ensure the equation;

$$\sin\theta \cong \theta \tag{12}$$

In the motion equations of rotation of cabin and chasis, this assumption had been made to solve the equations.

Another assumption is the to limit the vehicle's motion in vertical mode. The bodies in this model are assumed to be rigid and move in only vertical direction.

CHAPTER FOUR
DESIGN SCENARIOS AND SIMULATION

4.1 Parameter Selection for Design Scenarios

Driving comfortably in heavy trucks is very important due to the long drive times. A comfortable ride quality makes the truck more economical becasuse the driver could use the truck longer which obtains quicker transportation time. Therefore decreasing the acceleration on truck cabin and seat is highly important for commercial heavy trucks.

The tires are the interaction point between the road and the vehicle. So making modifications in suspension system for improving ride comfor may cause road holding problems. Therefore making modifications on the cabin suspension system is the way to improve comfort without decreasing the road holding capabilities.

The cabin suspension system consists of 4 springs and 4 dampers in modern heavy commercial vehicles. The selection of the spring stiffness and damper ratings effects to natural frequency of cabin directly. The aim here is to reduce accelerations on the cabin with the correct combination of spring&damper combinations.

4.2 Realization of Parameters

Selection of the softest spring might provide the best ride but constructional constraints are a huge barrier in front of this. The displacement of the rigid bodies have to be limited in order to avoid collusions. In Table 4.1 the cabin suspension stiffnesses of some commercial vehicles are seen.

Table 4.1 Cabin suspension stiffnesses of some heavy commercial vehicles.

Vehicle Model	Front Cabin Suspension Stiffness(N/m)	Rear Cabin Suspension Stiffness(N/m)
Type-1	180000	60000
Type-2	42000	40000
Type-3	22000	23000

Generally implementing softer springs improves the ride quality, but too soft spring may cause motion sickness in some cases and maight cause collusions in oscillating components. The oscillations under 1 Hz causes motion sickness. The dynamic response for several parameters for spring and dampers and thier effects to ride will be investigated in next section

.

4.3 Time Response of the Model

In this section several design parameters will be tried on simulation in order to find optiumum ride comfort. For time domain analyses, the response of the vehicle to bumps and harmonic inputs will be investigated. For the selection of parameters and comparison of performances, several design secneraios have been prepared. The change in different design scenarios will only be in cabin suspension parameters. So the existing parameters Type-1 which is heavy commercial vehicle produced in Turkey are given in Table 4.2.

Table 4.2 Existing parameters of a heavy commercial vehicle.

Parameter	Value
Equivalent mass of front axle-m1	225 kg
Equivalent mass of rear axle-m2	225 kg
Equivalent mass of chasis-m3	1044 kg
Equivalent mass of cabin-m4	500 kg
Equivalent mass of seat+driver-m5	100 kg
Moment of inertia of chasis around y axis-I3	1496 $kg.m^2$
Moment of inertia of cabin around y axis-I4	193 $kg.m^2$

Stiffness of front tyre-k1	900.000 N/m
Stiffness of rear tyre-k2	900.000 N/m
Stiffness of front suspension-k3	170.000 N/m
Stiffness of rear suspension-k4	250.000 N/m
Stiffness of front cabin suspension-k5	180.000 N/m
Stiffness of rear cabin suspensions-k6	60.000 N/m
Stiffness of seat spring-k7	10.000 N/m
Damping rating of front tyre-c1	2000 N.s/m
Damping rating of rear tyre-c2	2000 N.s/m
Damping rating of front suspensions-c3	7500 N.s/m
Damping rating of rear suspension-c4	7500 N.s/m
Damping rating of front cabin suspension-c5	6000 N.s/m
Damping rating of rear cabin suspension -c6	2000 N.s/m
Damping rating of seat damper-c7	2000 N.s/m

The design scenarios to be applied in simulation are below in Table 4.3.

Table 4.3 Design scenarios

	c_5 (N.s/m)	**c_6 (N.s/m)**	**k_5 (N/m)**	**k_6 (N/m)**
Design scenario-1 (Existing)	6000	2000	180000	60000
Design scenario-2	4000	4000	90000	60000
Design scenario-3	4000	4000	60000	60000
Design scenario-4	3000	3000	60000	60000
Design scenario-5	2500	2500	60000	60000
Design scenario-6	2000	2000	60000	6000
Design scenario-7	2000	2000	40000	40000
Design scenario-8	1500	1500	40000	40000
Design scenario-9	1500	1500	30000	30000
Design scenario-10	1250	1250	30000	30000
Design scenario-11	2000	2000	25000	25000
Design scenario-12	1500	1500	25000	25000

4.3.1 Time Response for Bumps

In this section, the response of the half-truck to a bump will be investigated for different design scenarios will be investigated. The mathematical representetion of a bump was given in section 2.3.1. Acoording to the formula, the simulink model is obtained which is shown in Figure 4.1.

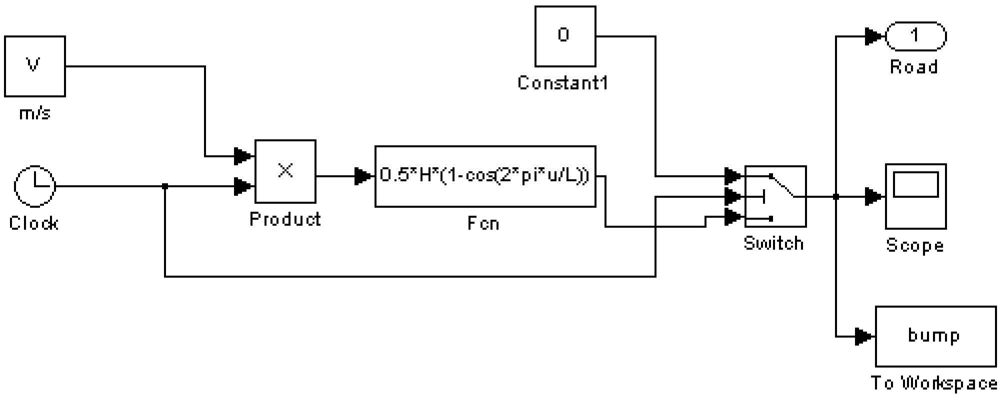

Figure 4.1 Simulink block diagram representation of a bump.

The position input here depends on the vehicle speed, bump length and bump height. In the simulations a speed of 10 km/h is tried. The results are given below for design scenarios.

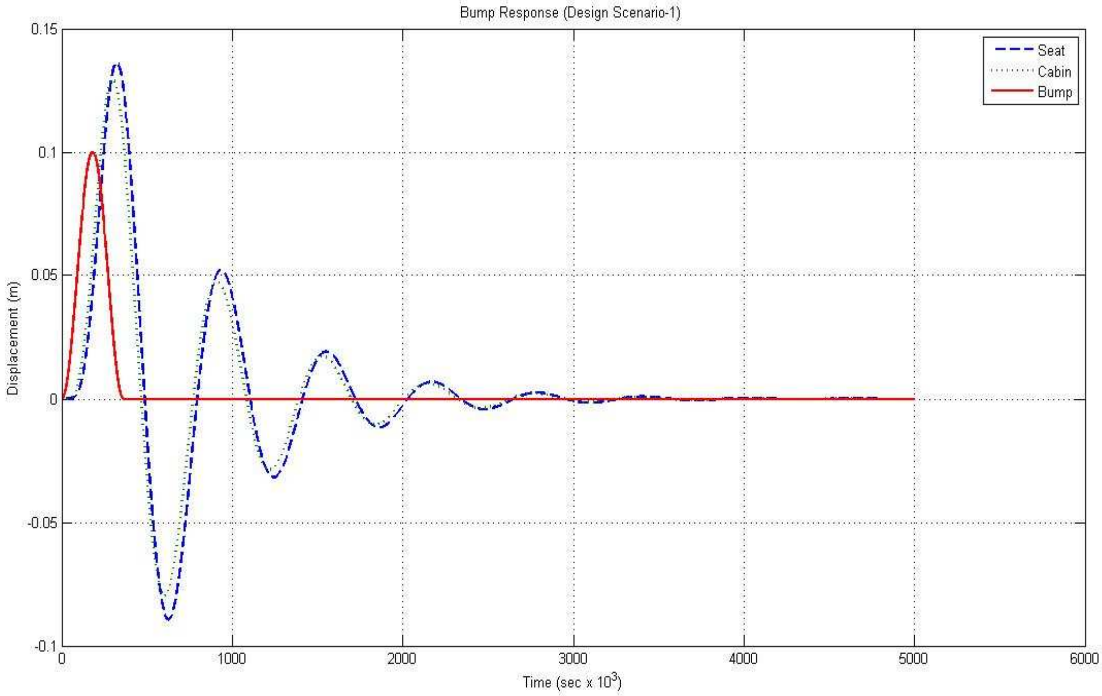

Figure 4.2 Bump response of seat and cabin for design scenario-1.

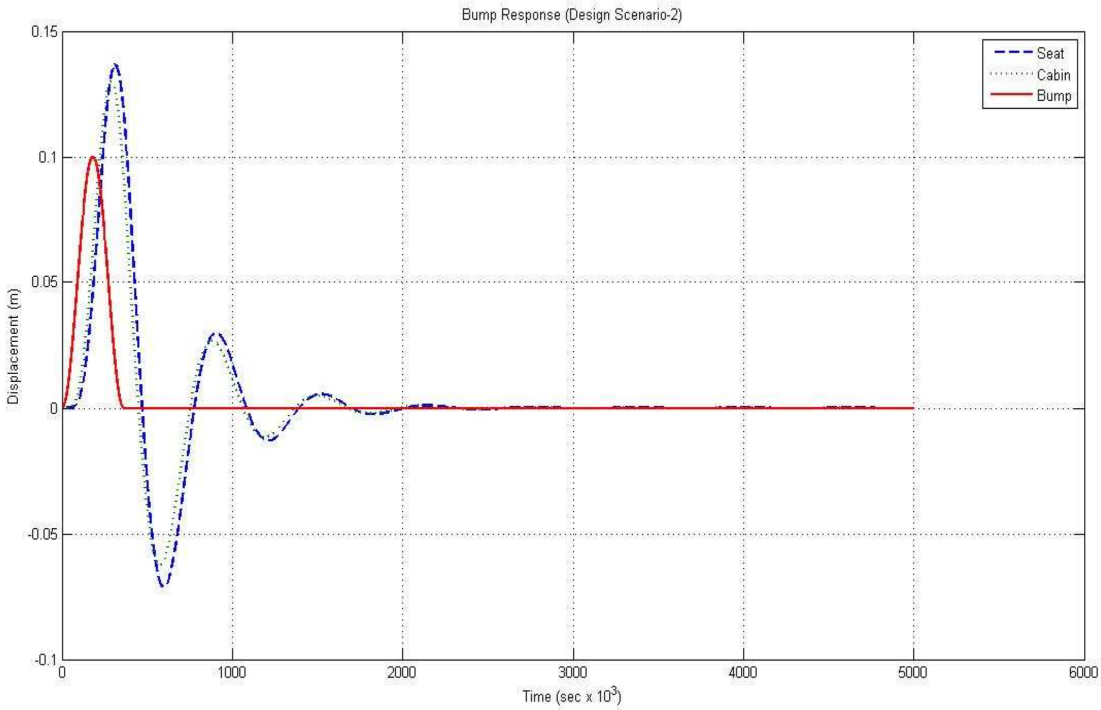

Figure 4.3 Bump response of seat and cabin for design scenario-2.

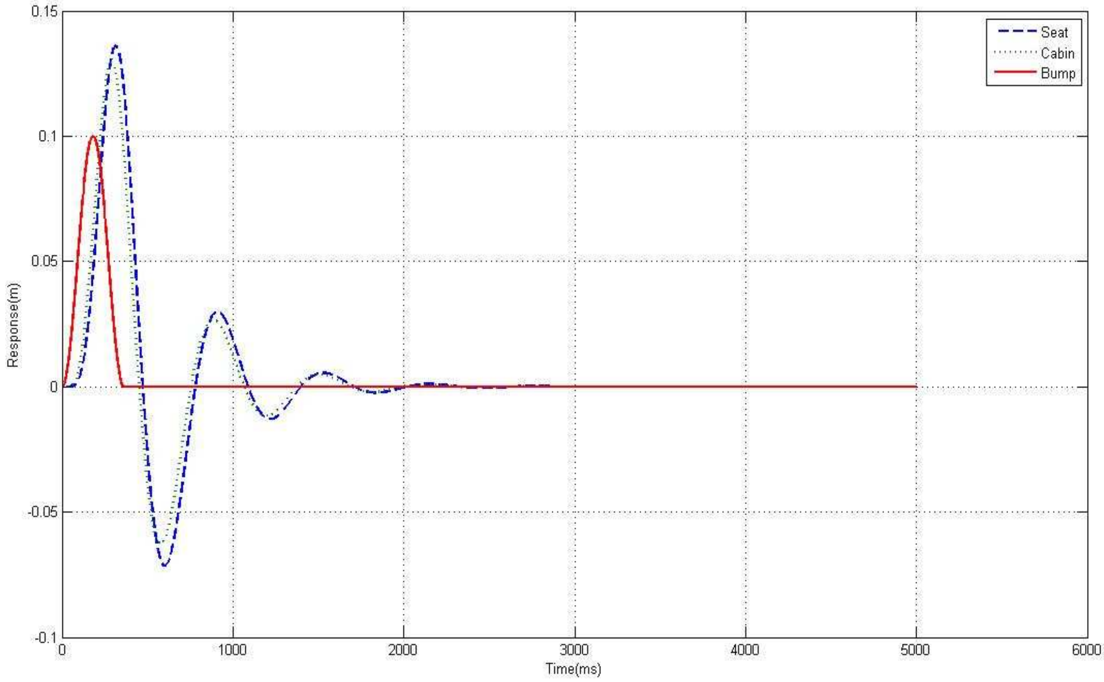

Figure 4.4 Bump response of seat and cabin for design scenario-3.

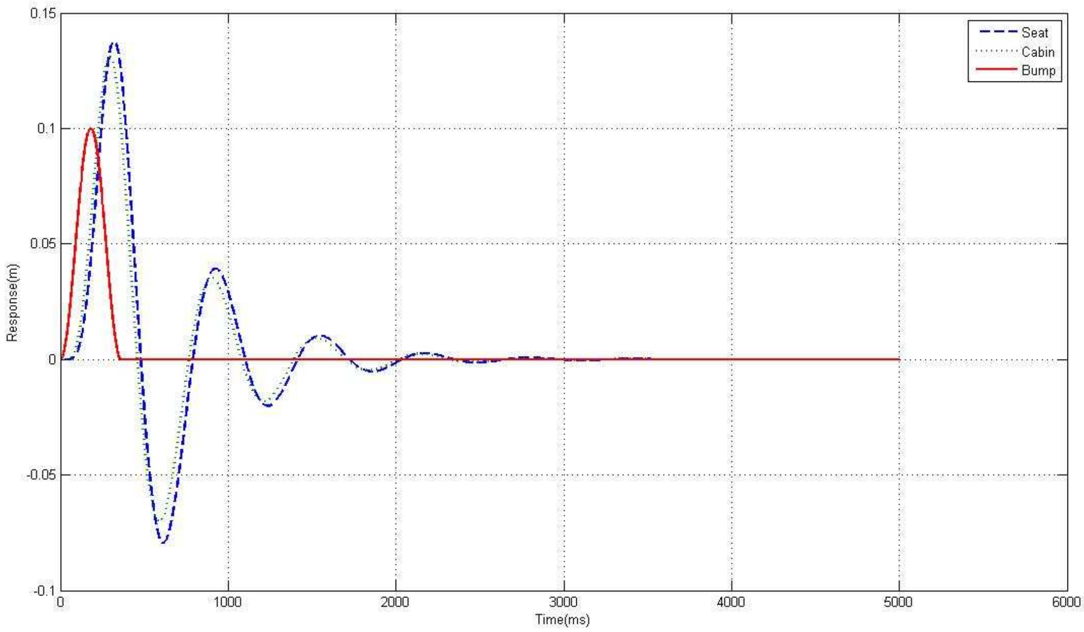

Figure 4.5 Bump response of seat and cabin for design scenario-4.

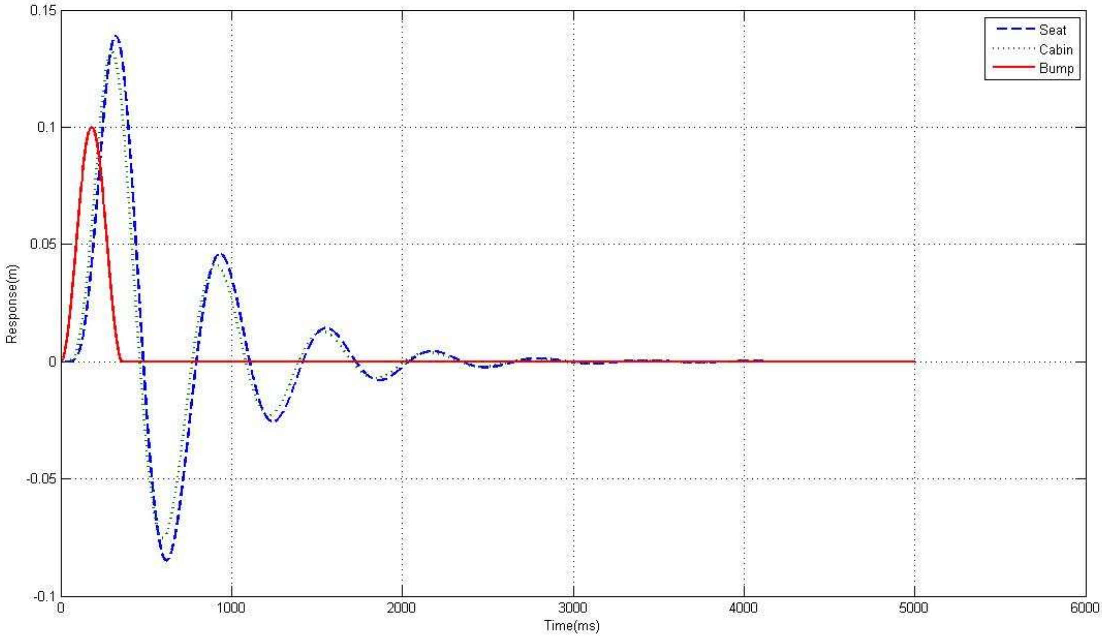

Figure 4.6 Bump response of seat and cabin for design scenario-5.

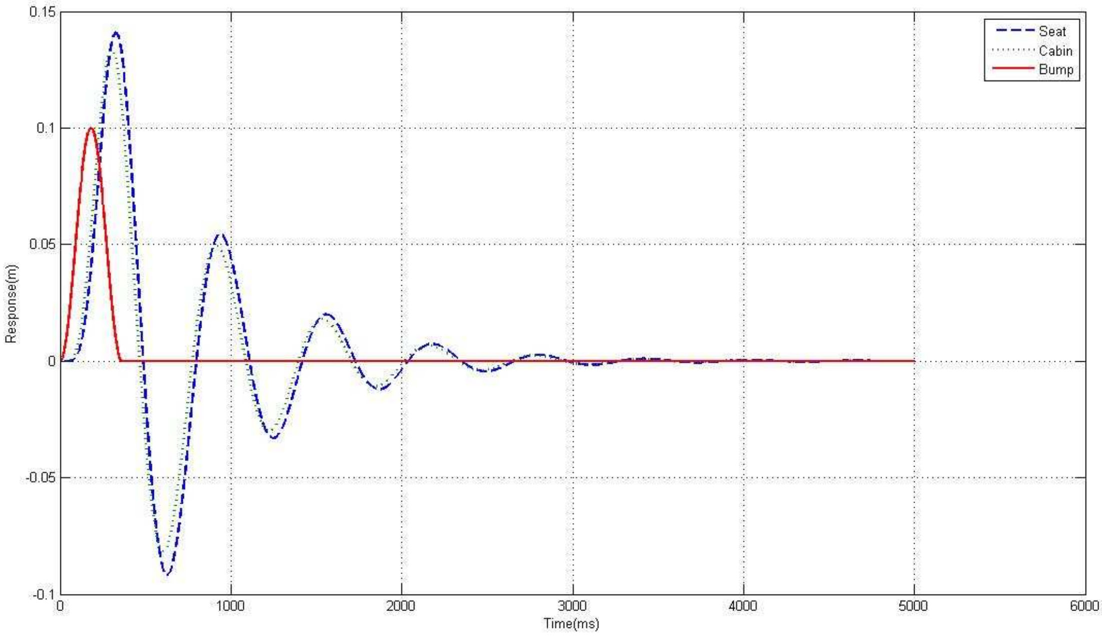

Figure 4.7 Bump response of seat and cabin for design scenario-6.

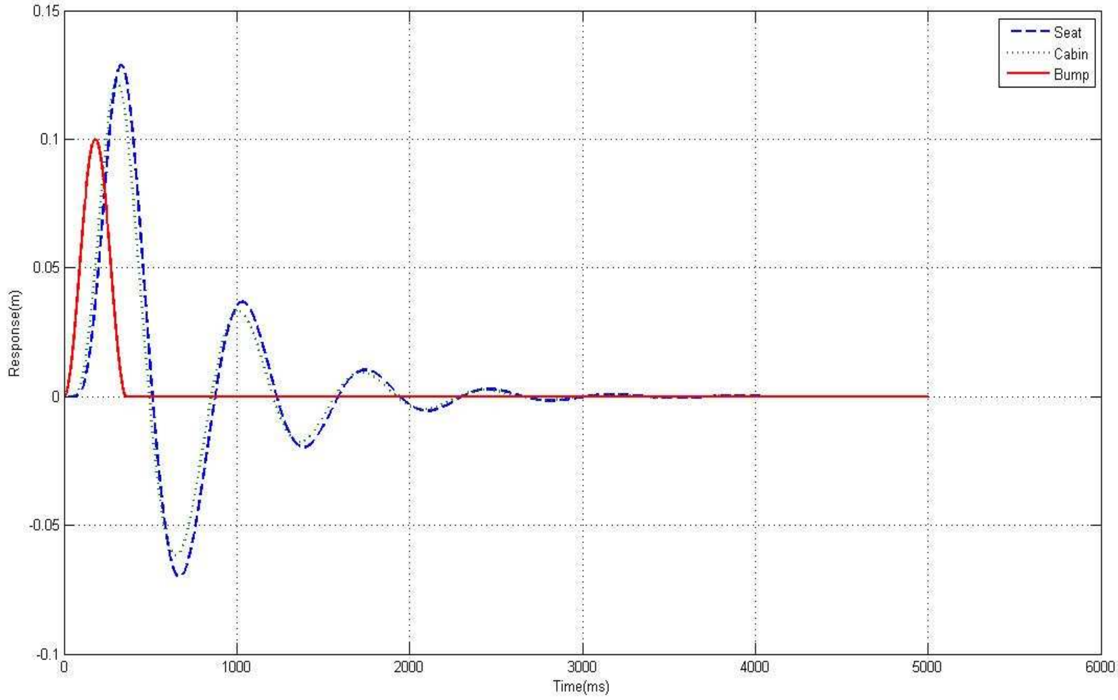

Figure 4.8 Bump response of seat and cabin for design scenario-7.

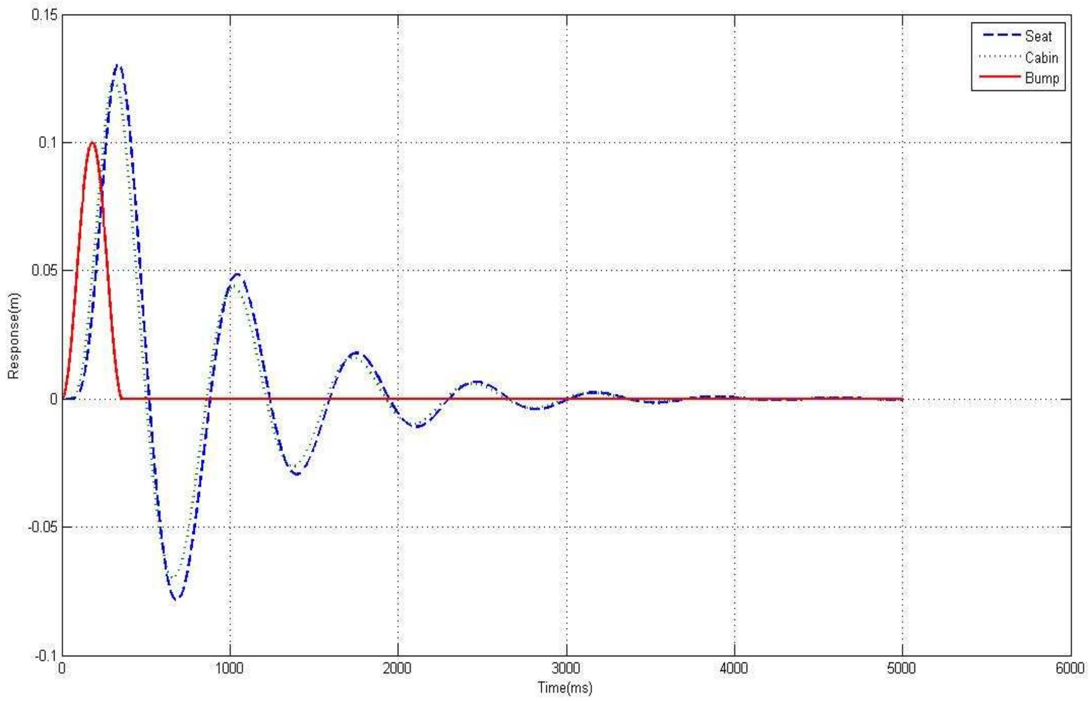

Figure 4.9 Bump response of seat and cabin for design scenario-8.

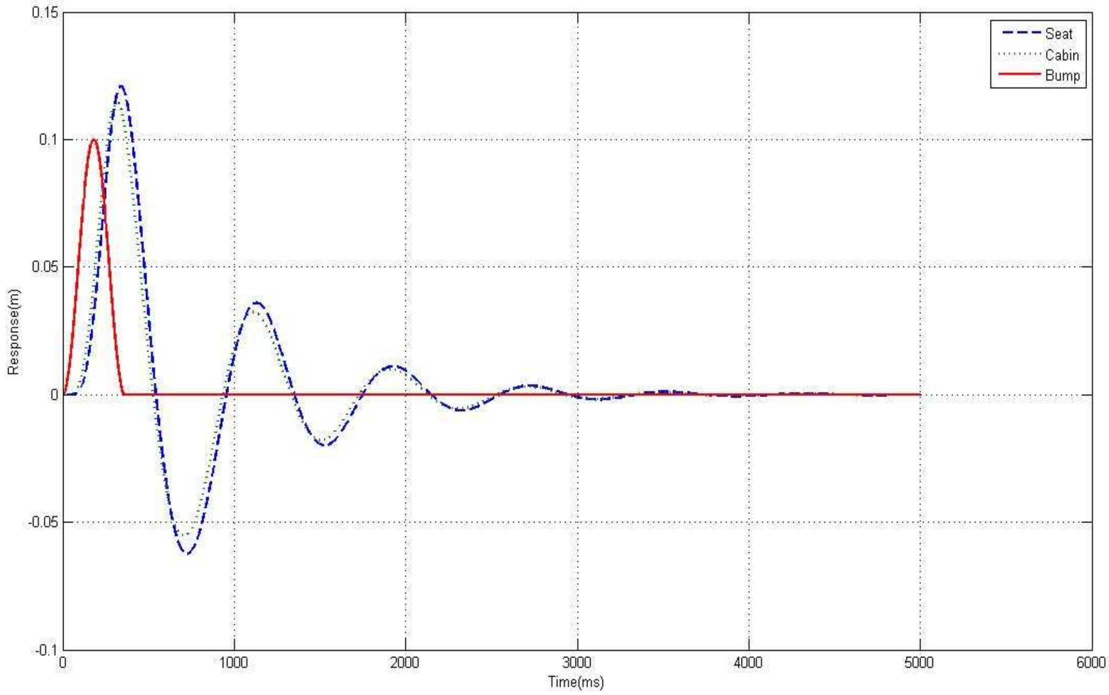

Figure 4.10 Bump response of seat and cabin for design scenario-9.

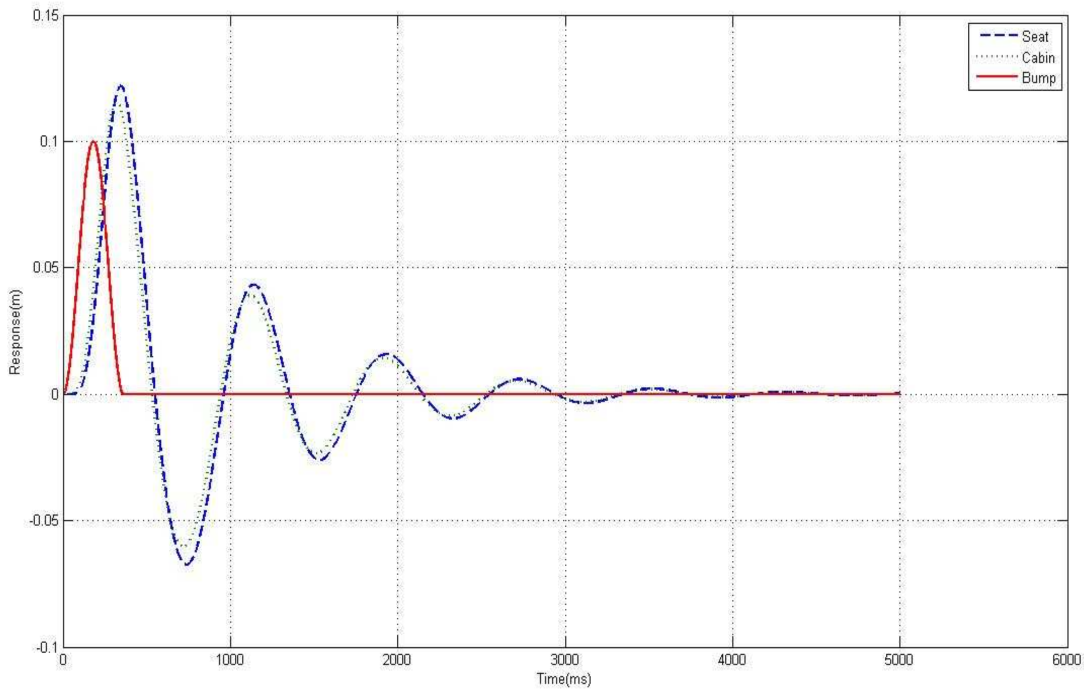

Figure 4.11 Bump response of seat and cabin for design scenario-10.

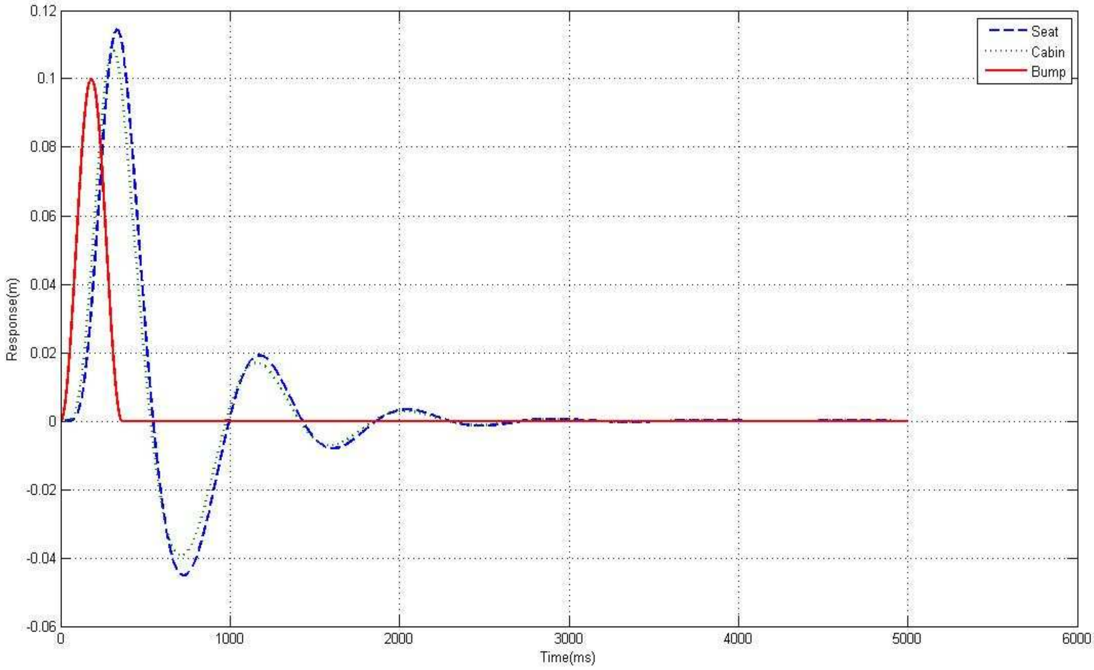

Figure 4.12 Bump response of seat and cabin for design scenario-11.

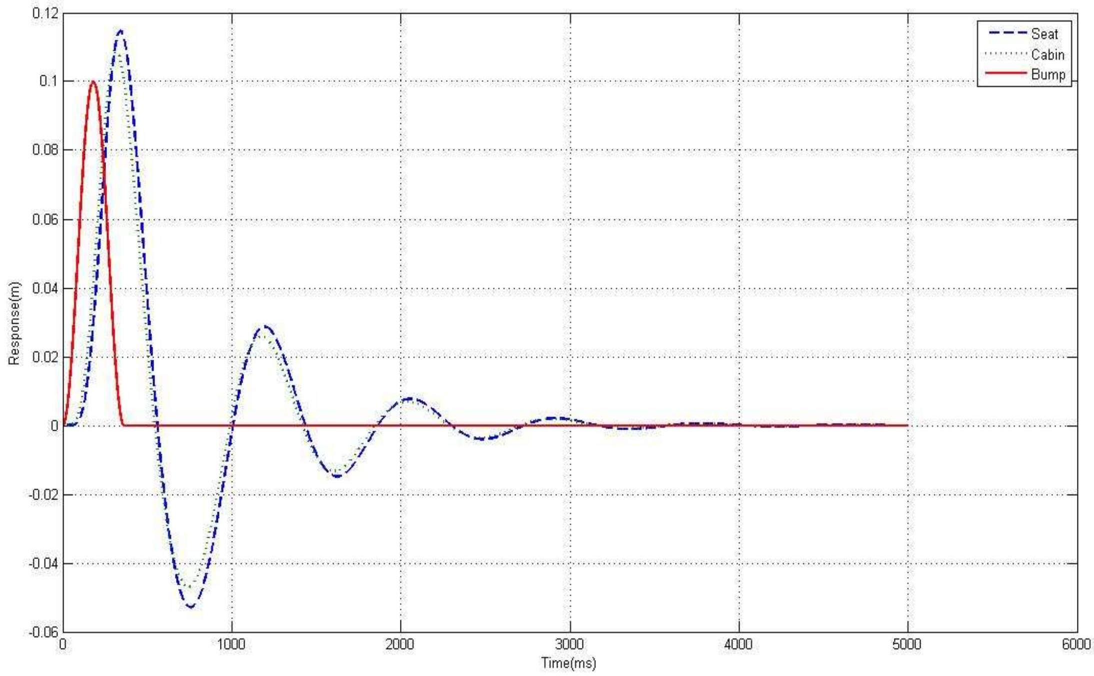

Figure 4.13 Bump response of seat and cabin for design scenario-12.

According to simulation made with 12 design scenarios, using softer springs in cabin suspension helps decresing the displacements. But the settling time increases. Thus keeping the damping in a reasonable value (2000 N.s/m seems to perform good) decreases the settling time. Increasing damping so much can cause problems in high frequencies such as hardening in suspension and noise. So design scenario-11 seems to be a good solution for time response of the vehicle for a bump. In next section the time response for harmonic inputs will be investigated.

4.3.2 Time Response for Potholes

For the simulation of potholes, the negative values of H is used in equation 18. The response diagram of design secenario-1, for a pothole with depth of 0.1 m and length of 0.5 m while the vehicle speed is 10 km/h, is shown below in Figure 4.14.

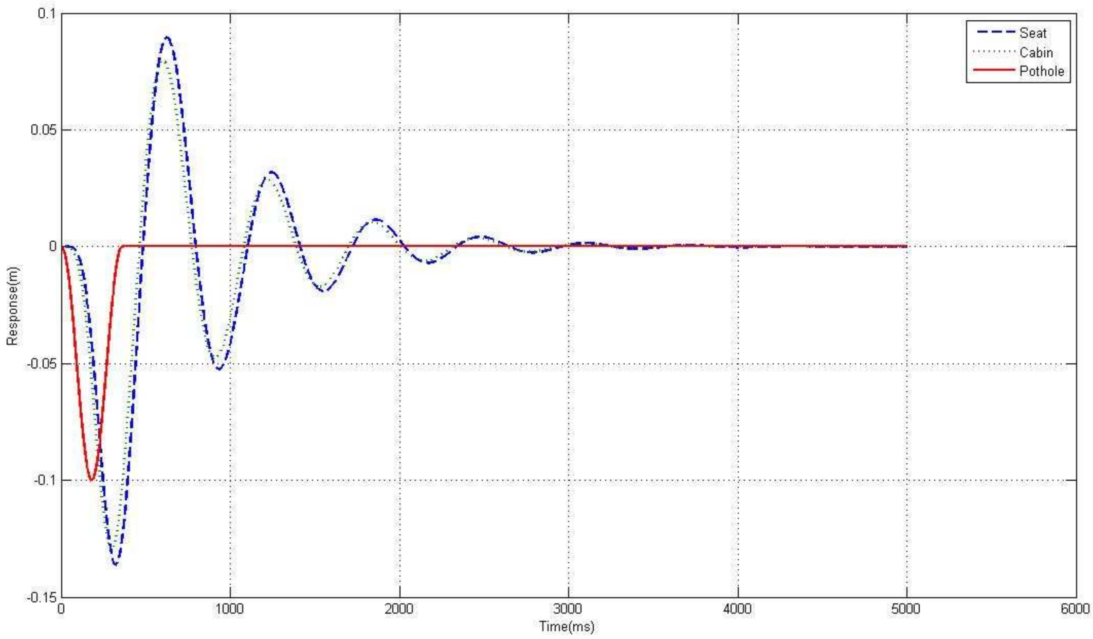

Figure 4.14 The response of the model for pothole depth of 0.1 m and for a length of 0.5 m.

As seen in Figure 4.14 the response is very similar to the bump response shown in Figure 4.2. The only difference is the direction of the excitaion. In the next section the frequency response of the system will be investigated.

4.3 Frequency Response and Transfer Function of Model

The frequency response analyses gives us ideas about the behaviour of the vehicle under vibrations. In different frequencies the system gives different responses. The idea of transfer function is to visulize the behavior of the system for a frequency range. The gain of the system which is the ratio between the output and input magnitudes, is ploted for the desired frequency range. The transfer function representations of the system for the given design scenarios are below in the Figures.

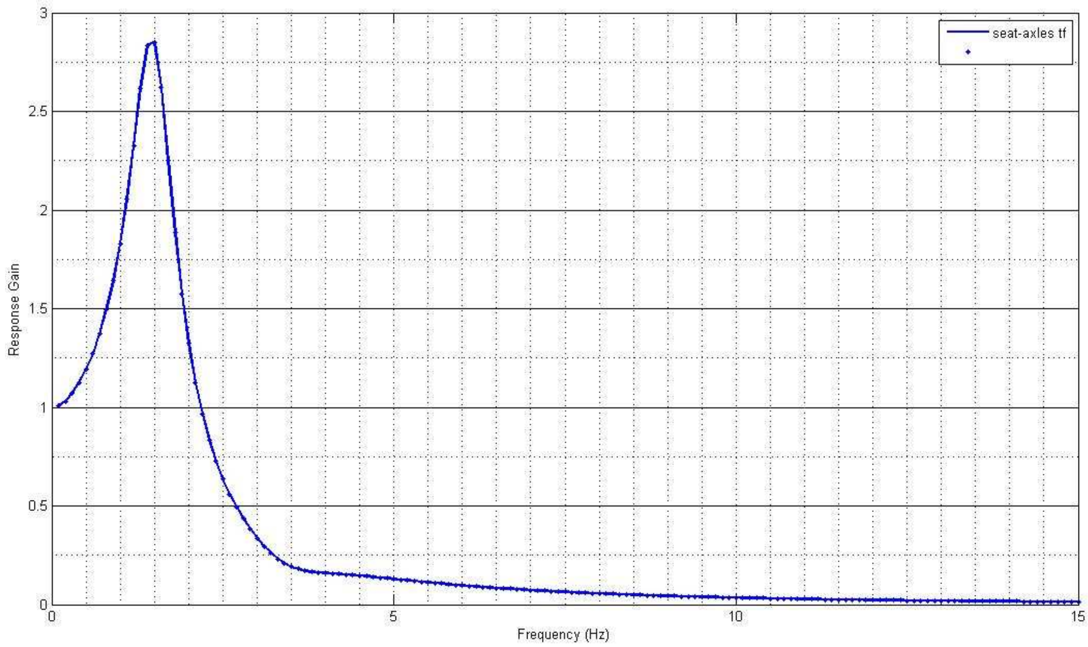

Figure 4.15 Transfer function between seat and axles for design scenario-1.

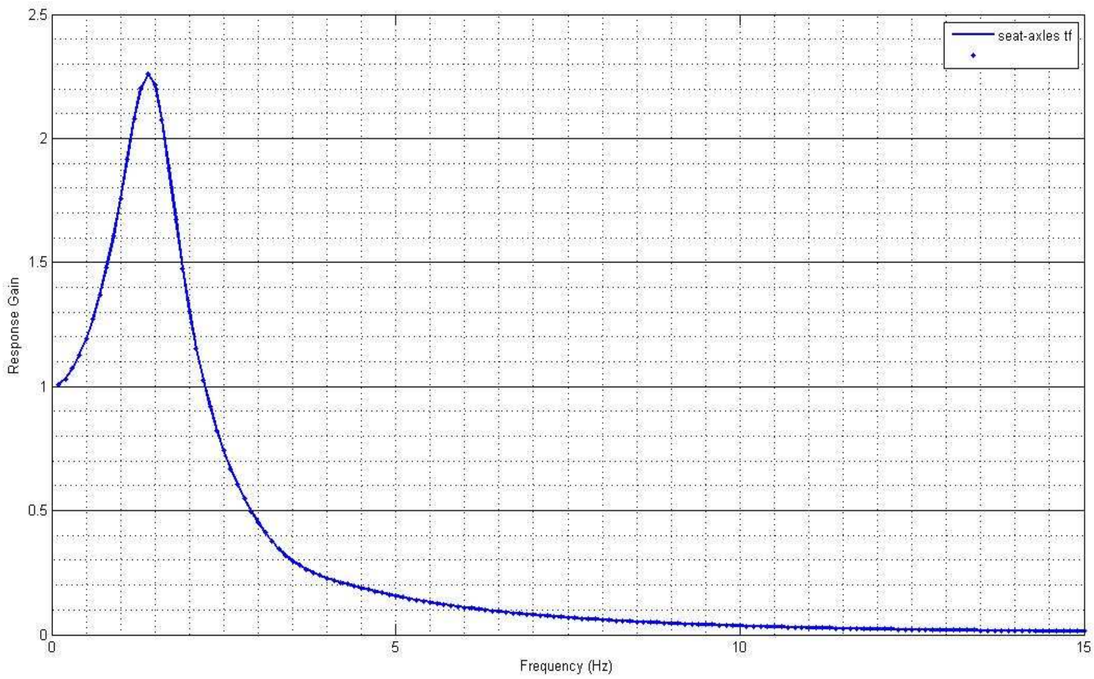

Figure 4.16 Transfer function between seat and axles for design scenario-2.

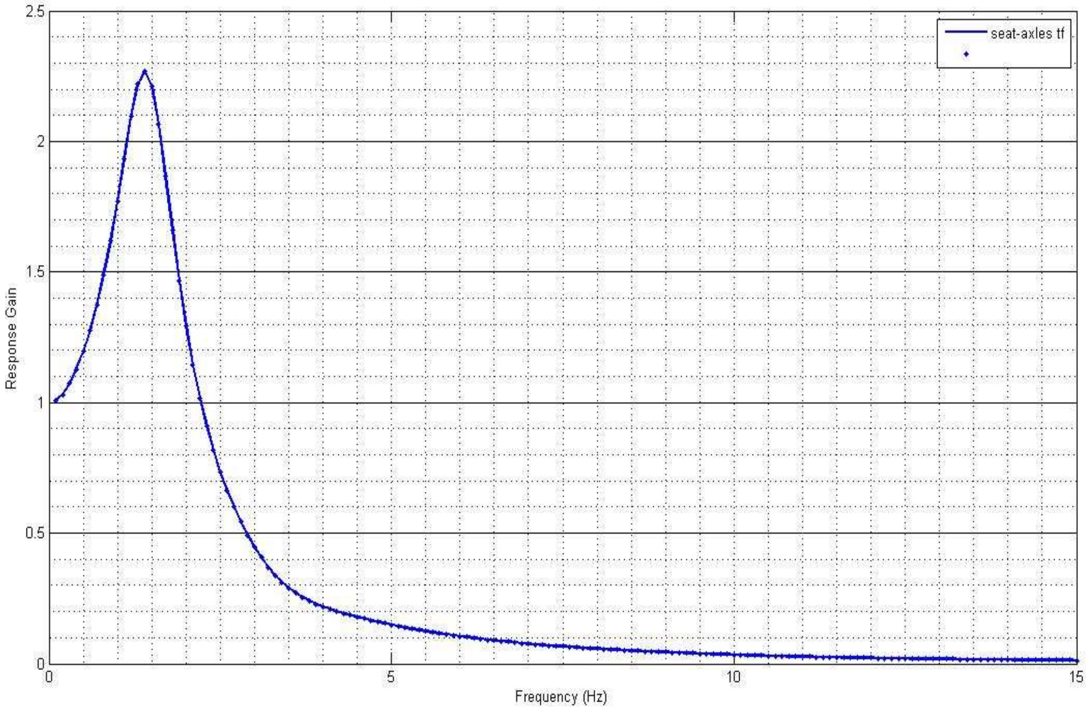

Figure 4.17 Transfer function between seat and axles for design scenario-3.

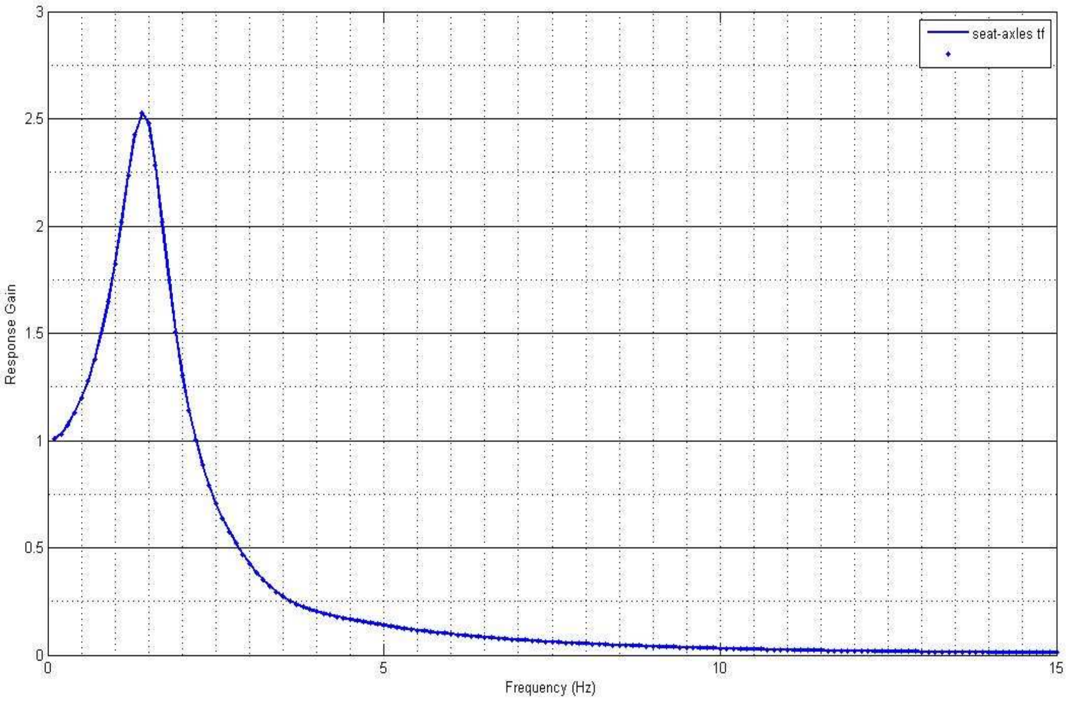

Figure 4.18 Transfer function between seat and axles for design scenario-4.

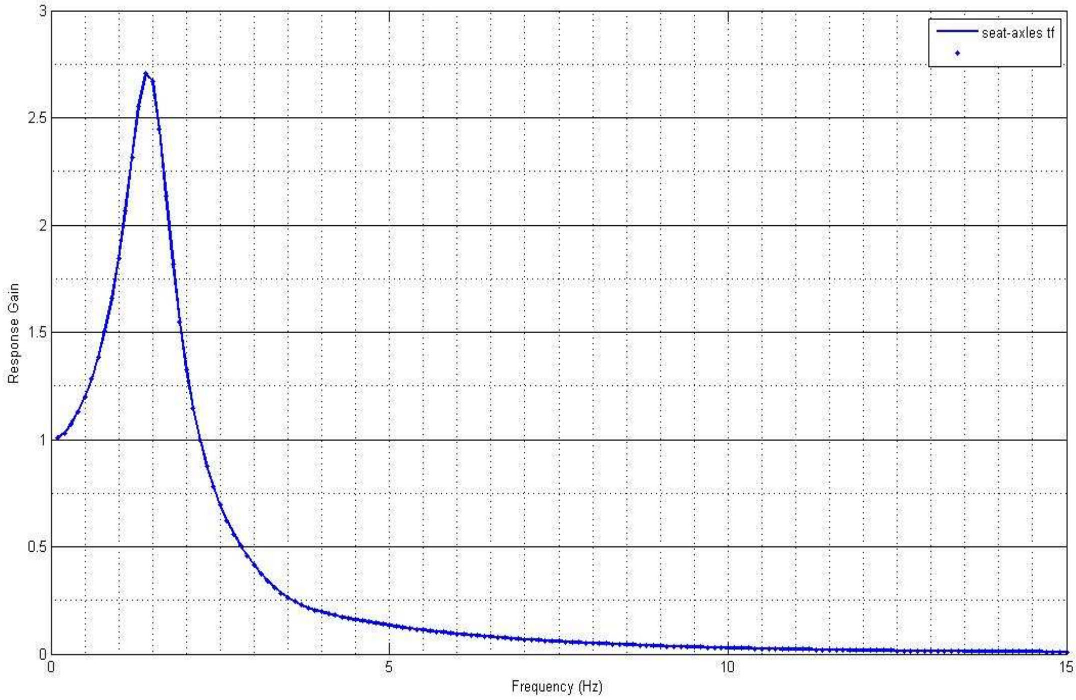

Figure 4.19 Transfer function between seat and axles for design scenario-5.

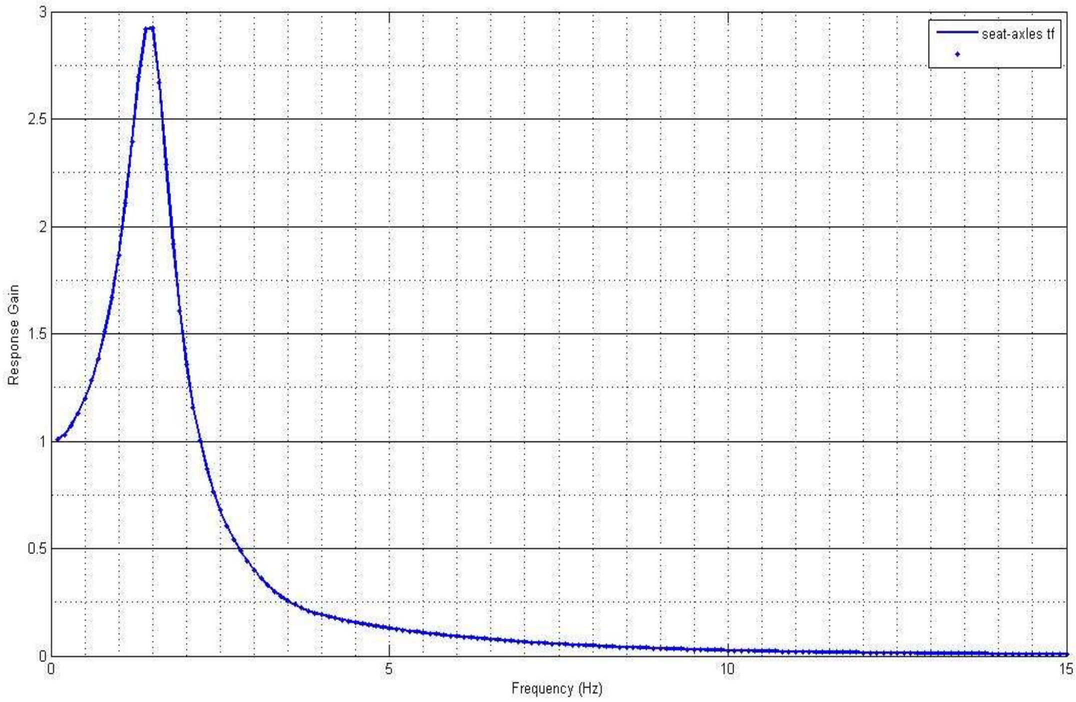

Figure 4.20 Transfer function between seat and axles for design scenario-6.

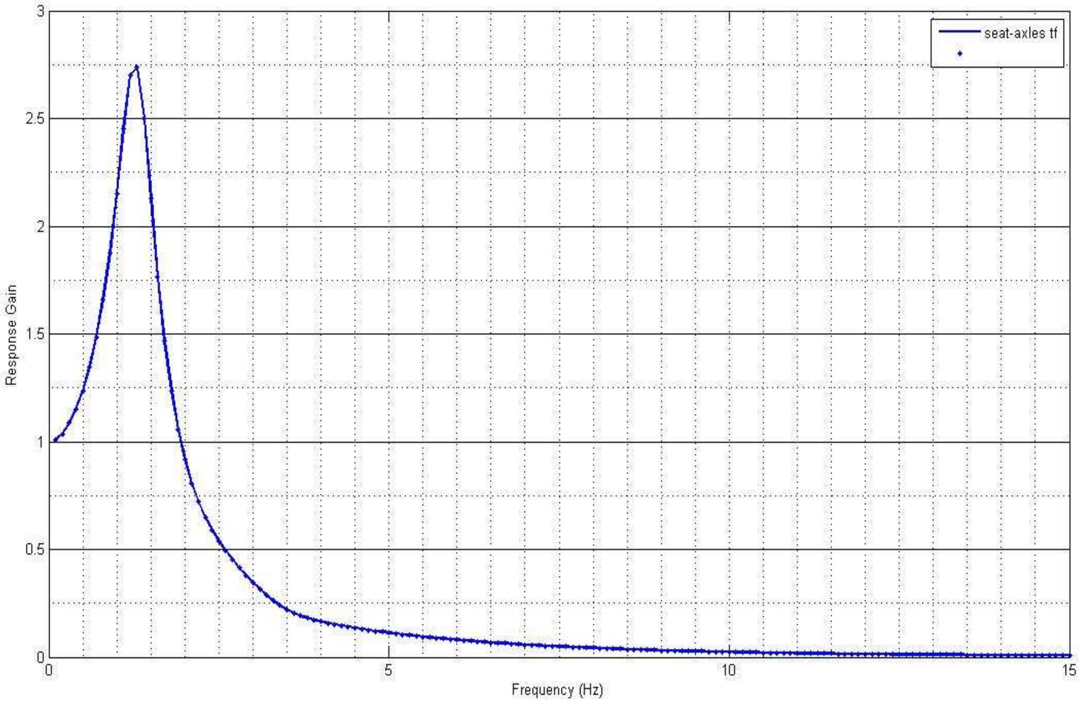

Figure 4.21 Transfer function between seat and axles for design scenario-7.

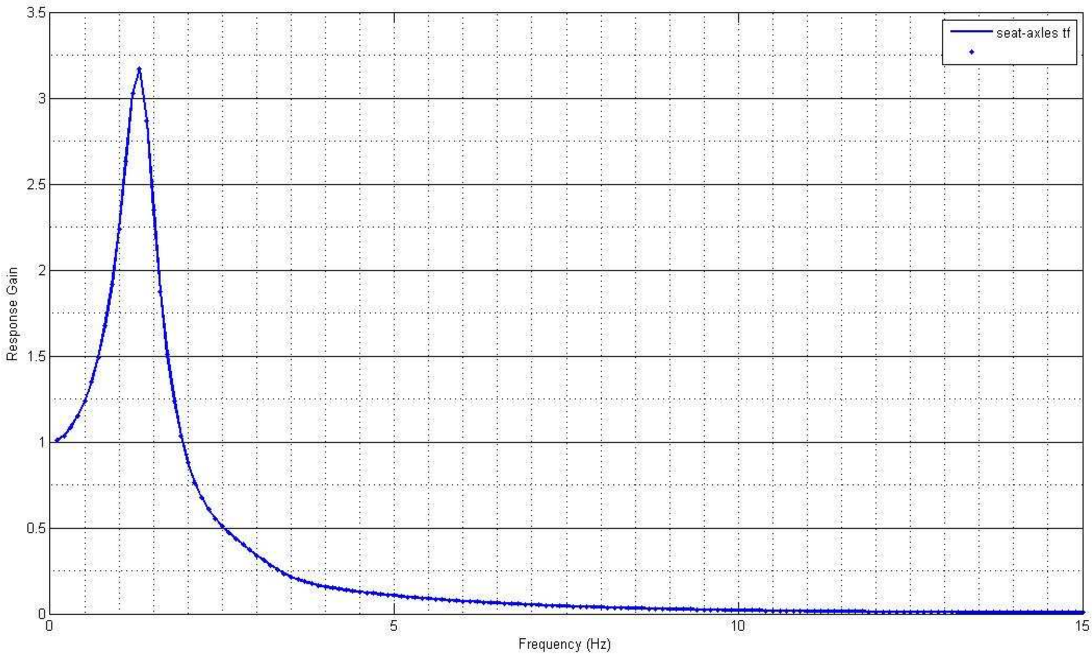

Figure 4.22 Transfer function between seat and axles for design scenario-8.

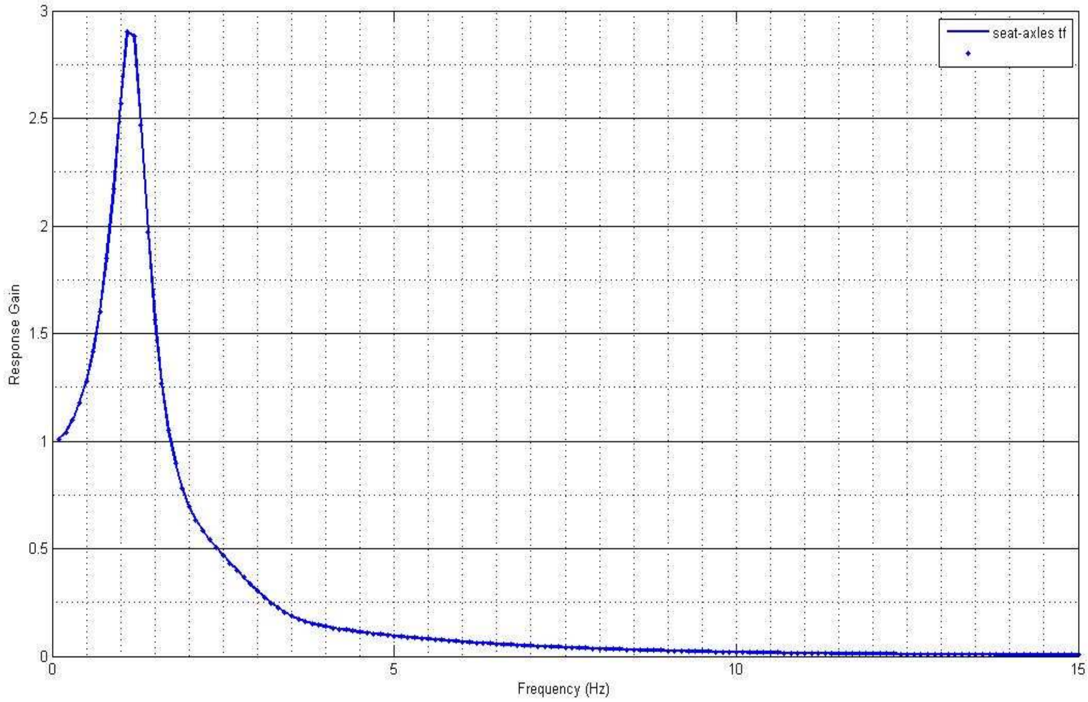

Figure 4.23 Transfer function between seat and axles for design scenario-9.

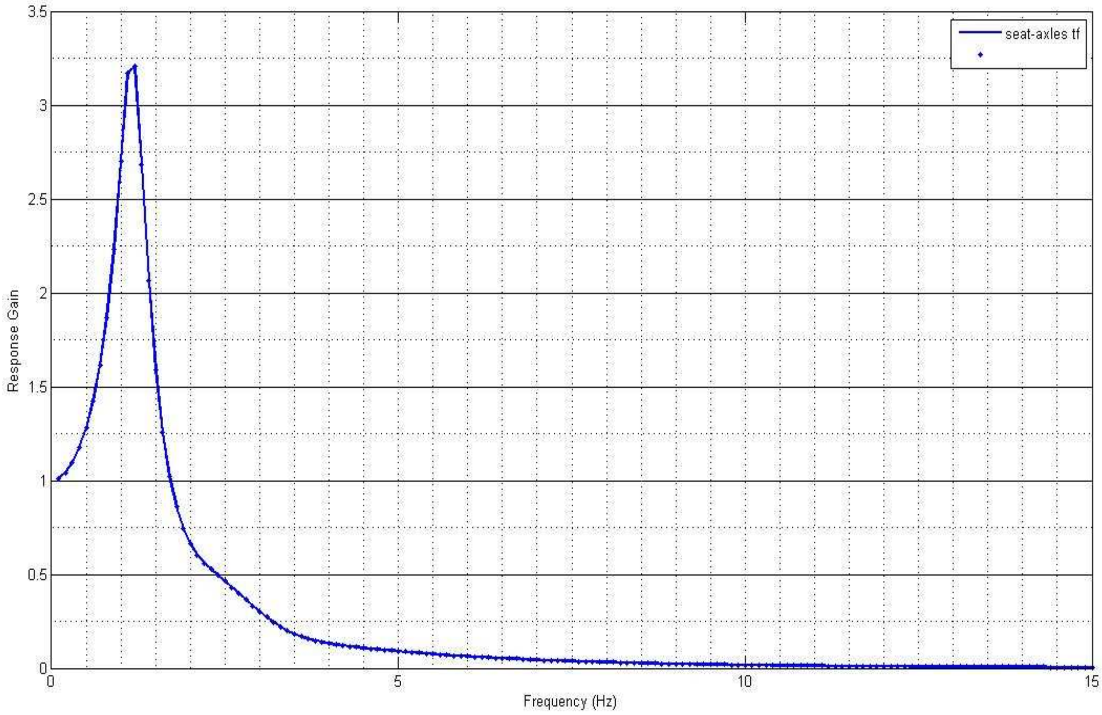

Figure 4.24 Transfer function between seat and axles for design scenario-10.

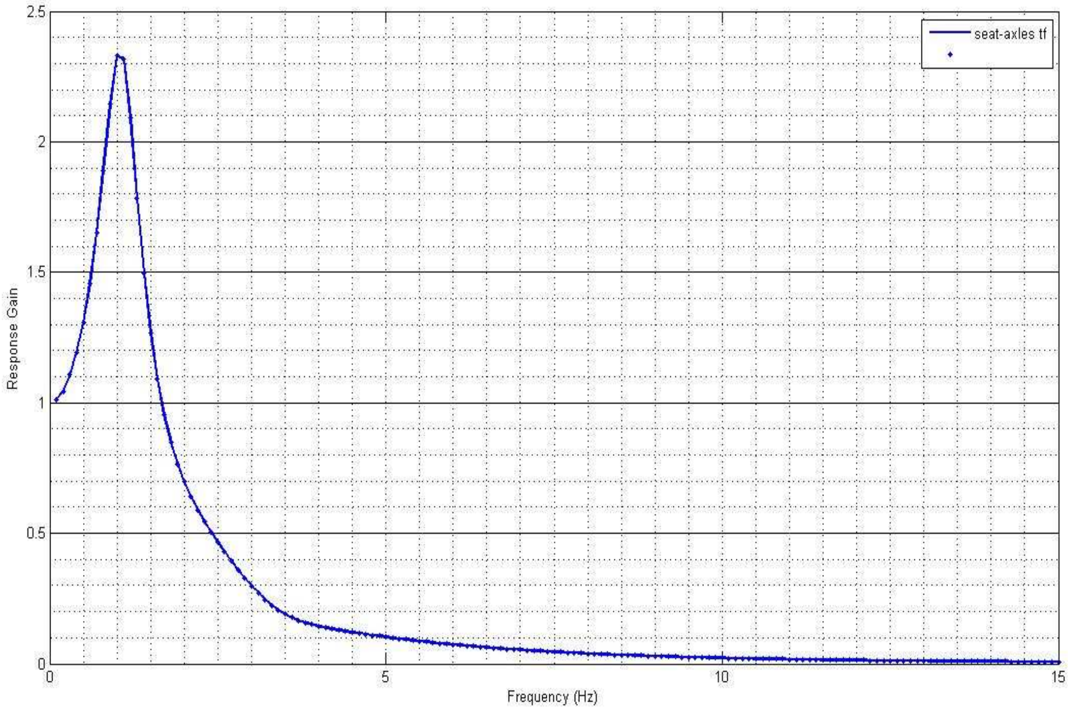

Figure 4.25 Transfer function between seat and axles for design scenario-11.

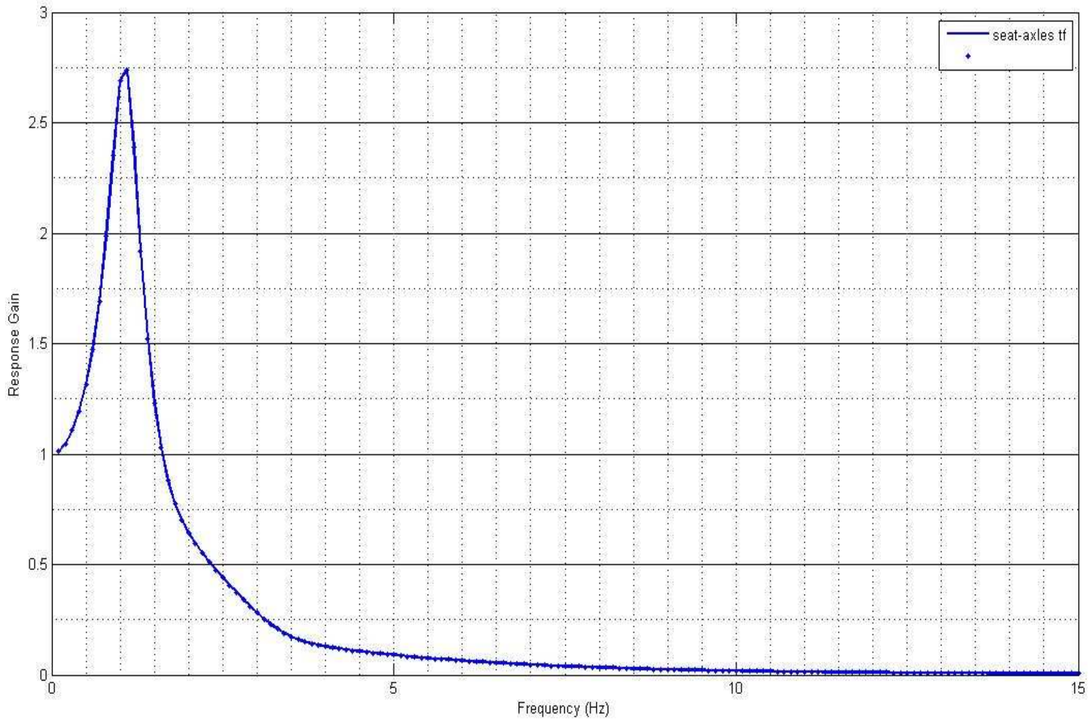

Figure 4.26 Transfer function between seat and axles for design scenario-12.

If the results are compared to each other, design scenario-3 gives the best results when the transfer functions are considered. The crest of the graph (the maximum response gain) is reduced 2.3 where this value is 2.8 in the existing situation. The high damping rate helps the crest of the graph decrease. But high damping causes noise discomfort in high frequencies. Thus seeing the transfer function overlapped woluld give a good idea about high frequecy performance damping rate. But the transfer function approach is not a proof of comfort. The human body response to vibrations has it's own unique characteristics. According to the international standards, the human body is sensitive to vertical vibrations in the range of 4-8 Hz. Thus the damped natural frequency of the vehicle is important and it must be lower than 4 Hz or higher than 8 Hz. In vehicles it is usually between 1 Hz and 2 Hz. But it is an important design fact that has to be taken care of. The detailed investiagation of weighted accelerations and human body exposure to vibration will be made in chapter 5.

4.5 Effects of Natural Frequency to Transfer Function

Because the cabin suspension springs are connected in series with relatively stiff suspension springs and tire, the cabins suspension springs pre-dominates natural frequency of the cabin and the seat in the vertical mode. So changing the cabins suspension effects the damped natural frequency of the cabin and seat greatly. A set of scenarios were prepared before in previous chapter. The damped natural frequencies of the seat for the given design scenarios are below in Table 4.4.

Table 4.4 Existing parameters of a heavy commercial vehicle.

Design Scenario	**Seat Damped Natural Frequency (Hz)**
Design scenario-1	1.45
Design scenario-2	1.42
Design scenario-3	1.41
Design scenario-4	1.41
Design scenario-5	1.42
Design scenario-6	1.43
Design scenario-7	1.26

Design scenario-8	1.27
Design scenario-9	1.15
Design scenario-10	1.15
Design scenario-11	1.05
Design scenario-12	1.07

As it can be understood from the Table 4.4, using stiffer springs in cabin suspensions increases the damped natural frequency greatly, on the other hand increasing damping in cabin suspension decreases the damped natural frequency slightly. The ideal natural frequency for vertical direction for encumburent people on the seat is 1 Hz. But using such soft springs are not practical because of the static weight of the cabin causing so much defleciton and the space needed between the cabin and the chasis for the deflection occurres when a road input exists.

To observe the effect of natural frequency on transfer function, keeping all the parameters constant except cabin suspension springs and plotting transfer functions ovelapped will be helpful. The parameters that are changed given below in Table 4.5 with the corresponding natural frequency. The 7 DOF half truck Simulink model was run in a Matlab code in order to obtain the transfer functions and plotting them overlapped..

Table 4.5 The changed parameters with corresponding damped natural frequency.

Damped Natural Frequency (Hz)	**k5 (N/m)**	**k6 (N/m)**
1.58	90000	90000
1.54	80000	80000
1.49	70000	70000
1.43	60000	60000
1.36	50000	50000
1.26	40000	40000
1.13	30000	30000

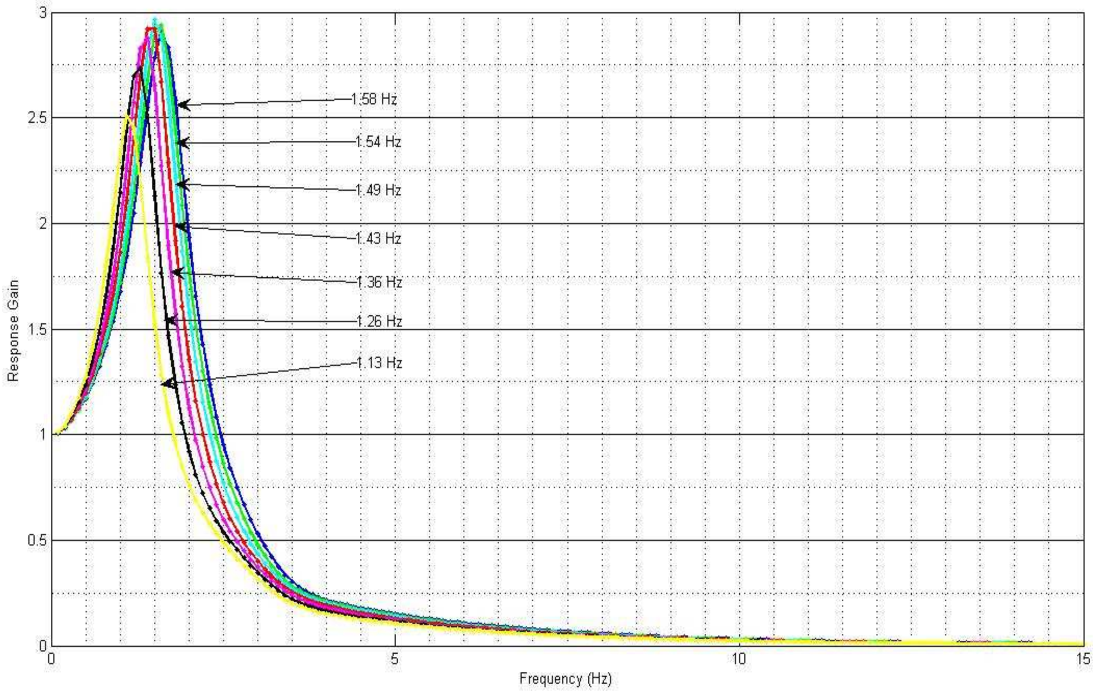

Figure 4.27 The effect of damped natural frequency on transfer function.

As seen in Figure 4.27 higher damped natural frequency effects the transfer function and shifts it up. This causes the response gain increase in all frequency range. Lower natural frequency provides a more comfortable ride according to the transfer function approach. Lowest displacement occurs at the natural frequency of 1.13 Hz. At higher values of natural frequency (stiffer cabin suspension springs) the displacement peak in the 1-5 Hz range increases reflecting a greater transmission of road input.

On the other hand the effect of damping on transfer function is different from the springs' effect. Damping in suspensions comes primarily from the action of hydraulic shock absorbers. Contrary to their name, they do not absorb shock from the road bumps. Rather the suspension absorbs the shock and the shock absorber's function then is to dissipate energy put into the system by the bump (Gillespie, 1992). In Figure 4.29, the effect of cabin suspension damping rate to transfer function can be seen.

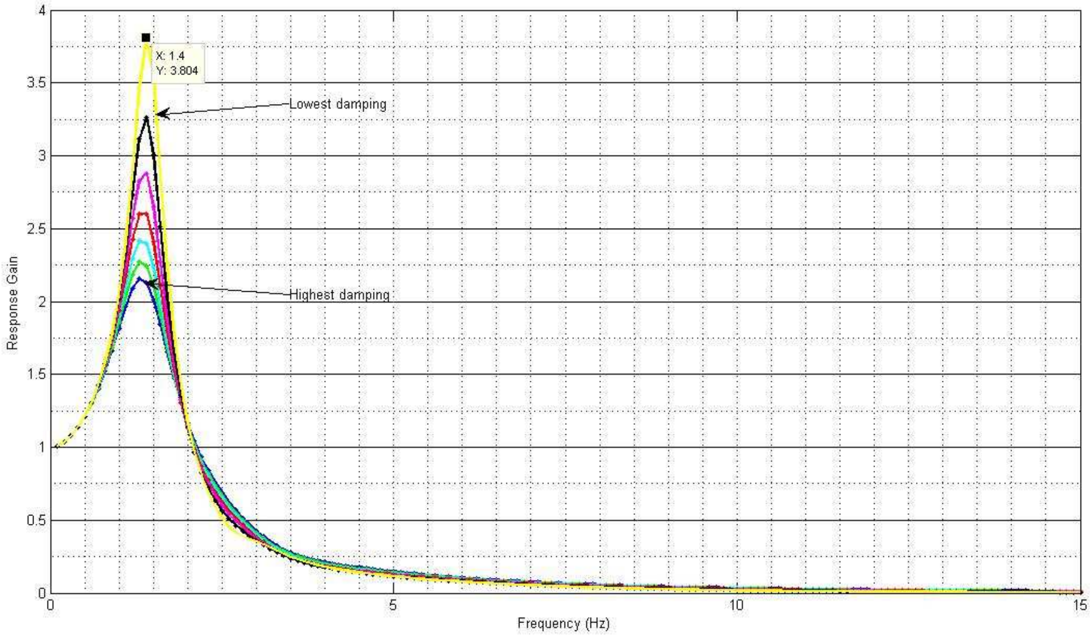

Figure 4.28 The effect of cabin suspension damping rate on transfer function.

The peak of the transfer function increases while the damping in the cabin suspension decreases. But for the higher frequencies lower damping gives a better performance by reflecting lower displacement and acceleration to the cabin and seat. So choosing very hard dampers seems to be good solution at first sight but the isolation at high frequencies might be problem. So choosing a moderate damping rate is good and widespread solution for heavy trucks' cabin suspension systems.

4.6 GUI Design

Designing and engineering of a motor vehicles is time consuming activity. So all the work that had been done is standardized in order to keep information for the next project in most automotive firms. To shorten the suspension design process in heavy duty trucks, a user friendly, Matlab based Guide User Interface has been designed.

The frequency response and the transfer function can be observed by entering the parameters only. The requested transfer function or frequency response can be selected from the selection panel. And the results of the simulation can be saved to a desired location. The GUI designed for simulating the frequency response and transfer functions can be seen in Figure 4.30.

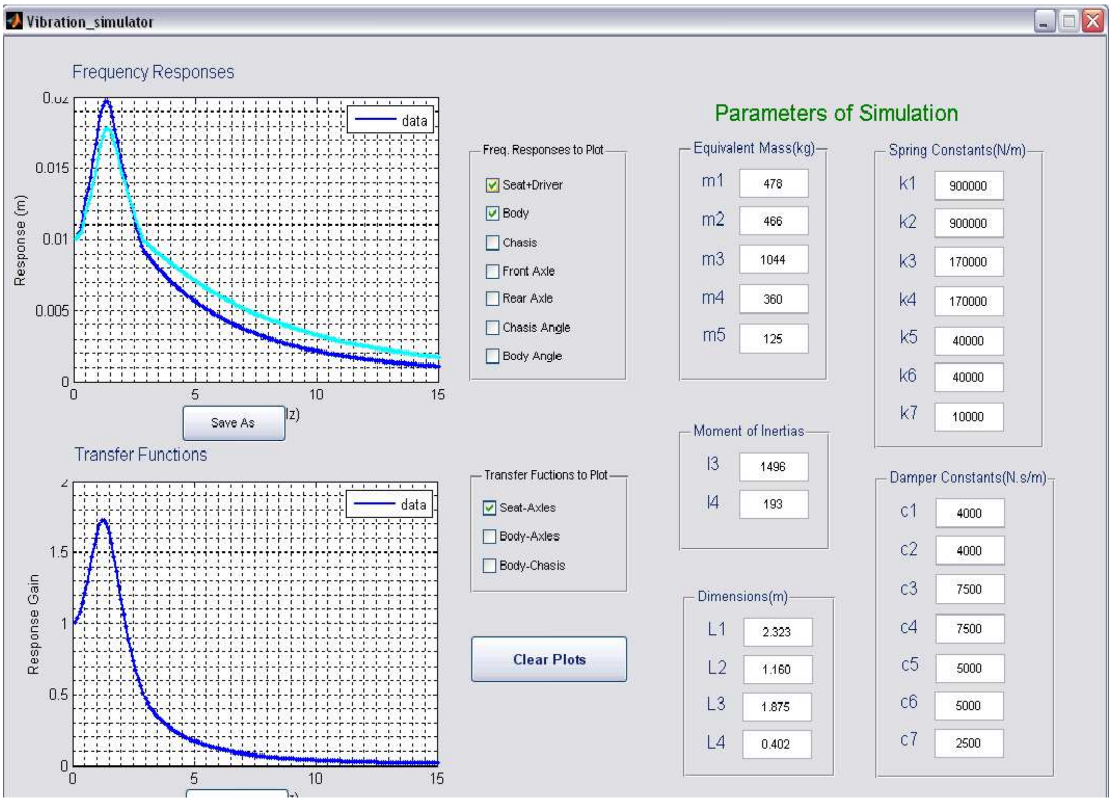

Figure 4.29 The GUI designed for simulations.

For another simulation the plots must be cleared first by pushing the “Clear Plots” button. The GUI uses the 7 DOF half truck Simulink model in a Matlab code. Since the vibration and vehicle dynamics are complicated areas of study, with the aid of a user interface, an expert is not needed to make simulations in order to get the results. The evaluation of the results on the other hand still needs an expert’s observation.

CHAPTER FIVE

EVALUATION OF COMFORT ACCORDING TO INTERNATIONAL STANDARTS

5.1 The wieghted R.M.S Approach and Frequency Weighting

The performance characteristics which are of most interest when designing the vehicle suspension are passenger ride comfort, road holding and suspension travel. The passenger ride comfort is related to passenger acceleration, suspension travel is related to relative distance between the unsprung mass and sprung mass and road handling is related to the tyre displacement (Shirahatt, Prasad & others, 2008).

The vibration evaluation according to ISO 2631 shall always include measurements of weighted root-mean-square (r.m.s) acceleration. The weighted RMS acceleration is expressed in meters per second squared (m/s^2) for translational vibration and radians per second squared (rad/s^2) for rotational vibration. The weighted RMS acceleration shall be caculated in accordance with the following equation or its equivalents in the frequency domain.

$$a_w = \left[\frac{1}{T} \int_0^T a_w^2(t) dt \right]^{\frac{1}{2}} \tag{13}$$

Where $a_w(t)$ is the weighted acceleration (translational or rotational) as a function of time, in meters per second squared (m/s^2) or radians per second squared (rad/s^2), respectively. T is the duration of measurement, in seconds (Intenational standart 2631-1, 1997(E)). Frequency weighting curves reccomended and/or used for various directions and their applications are listed in Table-1 and Table-2 in Intenational standart 2631-1 . Numerical values of the values of the weighting curves are given in tables 3 and 4 in Intenational standart 2631-1.

To design a comfortable ride in simulations for heavy trucks, the RMS values of acceleration for all directions have to used. In the calculation of RMS values of acceleration, r.m.s block of Matlab/Simulink has been used.

5.2 Effects of Vibration to Comfort and Perception

There is no conclusive evidence to support a universal time dependence of vibration effects on comfort. The weighted RMS acceleration shall be determined for each axis of translational vibration (x-,y- and z- axes) at the surface which supports person. Frequency weightings used for the prediction of the effects of vibration on comfort are w_c, w_d, w_e, w_j, and w_k. These weightings should be applied as follows with the multiplying factors W_i as indicated below in equation (24).

$$a_w = \left[\sum_i \left(W_i . a_i \right)^2 \right]^{\frac{1}{2}} \tag{14}$$

Where a_w is the frequency-weighted acceleration, W_i is the weighting factor for the i th one-third octave band, a_i is te RMS acceleration for the i th one-third octave band.

For seated persons:
x-axis (supporting seat surface vibration): $w_{d,k}$=1
y-axis (supporting seat surface vibration): $w_{d,k}$=1
z-axis (supporting seat surface vibration): $w_{k,k}$=1

Since the simulations in this thesis takes into account only the vertical vibrations for seated persons the weighting factor is 1.
Acceptable values of vibration magnitude for comfort depend on many factors which vary with each application. There for a limit is not defined. The following values in Table 5.1 give approximate indications of likely reactions at various magnitudes of overall vibration total values in public transport.

However, the reactions at various magnitudes depend on passenger expectations with regard to trip duration and the type of activities passengers expect to accomplish (e.g. reading eating writing, etc.) and many other factors (acoustic noise, temprature, etc.)

Table 5.1 Approximate indications of likely reactions at various magnitudes of overall vibration total values.

R.M.S acceleration	**Comfort Rate**
Less than 0.315 m/s^2	Not uncomfortable
0.315 to 0.63 m/s^2	A little uncomfortable
0.5 to 1 m/s^2	Fairly uncomfortable
0.8 to 1.6 m/s^2	Uncomfortable
1.25 to 2.5 m/s^2	Very uncomfortable
Greater than 2 m/s^2	Extremely uncomfortable

5.3 Evaluation of Design Scenarios According to ISO 2631-1

In order to reach a conclusion with comfort concept, the results of the simulation must be evaluated according to international standarts which has been mentioned in the previous section. The time and frequency response characteristics were investigated in the previous chapter. But in order to make tangible comparison between design scenarios, the RMS acceleration anaylsis must be completed. For the same inputs, the design scenarios given in previous chapter will be compared. A typical RMS signal is shown with the original acceleration signal in Figure 5.1.

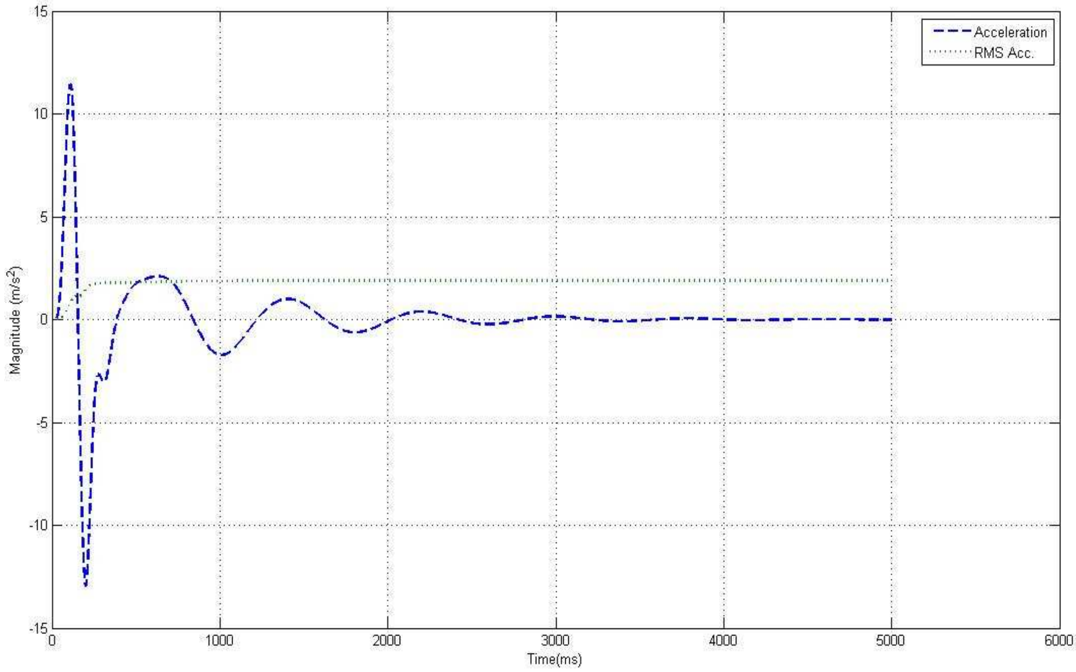

Figure 5.1 A typical RMS acceleration with original acceleration signal.

RMS acceleration analysis is meaningful for time frequency response anaylsis. First, time response of the vehicle model for a bump which has a height of 0.1 m and length of 0.5, will be investigated. Then the weighted RMS method will be applied. For the given design scenarios in the previous chapter the the r.m.s accelerations of the seat are given in the Table 5.2.

Table 5.2 RMS acceleration values for design scenarios.

Design Scenario	RMS Acceleration on Seat
Design scenario-1	1.8356
Design scenario-2	1.9373
Design scenario-3	1.9103
Design scenario-4	1.8948
Design scenario-5	1.9020
Design scenario-6	1.9263
Design scenario-7	1.5840
Design scenario-8	1.5870
Design scenario-9	1.3641

Design scenario-10	1.3578
Design scenario-11	1.3138
Design scenario-12	1.2526

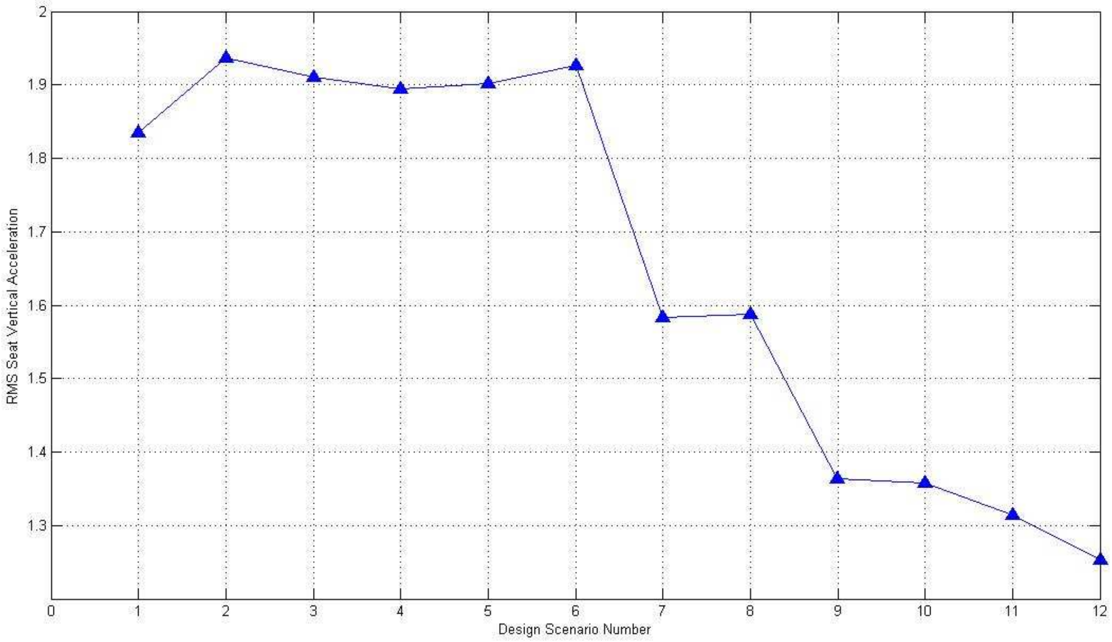

Figure 5.2 RMS seat vertical accelerations for design scenarios.

As it can cleary be seen in Table 5.2, the least RMS acceleration occurs in design scenario-12. Seeing the design scenarios as a chart which is shown in Figure 5.1, would give a better idea about the effects of parameters to rms acceleration. There is a slight increase in transition from design scenario 4 to 5. The reason is the the decrease in the damping rate. But on the other hand, decrasing damping reduces the RMS acceleration. To find the clear effect of cabin suspension damping rate on RMS acceleration and the optimum damping rate in cabin suspension Figure 5.3 can be observed.

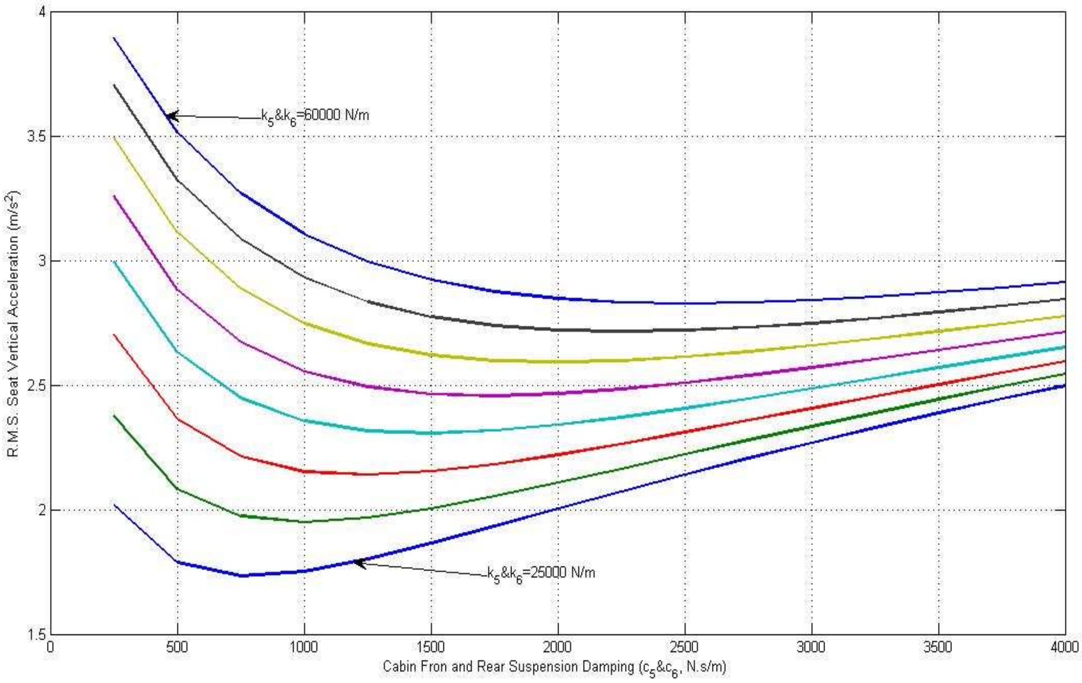

Figure 5.3 The change in RMS seat vertical accelerations versus damping rates of front and rear cabin suspension damping rates for several front and rear suspension stifnesses.

As it can be observed form Figure 5.3, the damping rate has an optimal value which makes the RMS acceleration minimum. For different cabin suspension stiffnesses the minima changes it's location. For stiffer springs minumum RMS acceleration occurs in harder damping rates. The RMS acceleration increses as the damping rate increases after a certain point.

In conclusion, the optimal choice cabin suspension parameters is very important for ride characteristics. The chart in Figure 5.3 gives reasonable idea about the cabin suspension parameter selection since the change of RMS acceleration can clearly be seen according to cabin suspension parameters. According to design scenarios, design scenario-12 gives the best performance. It provides %31.76 better performance in comparison with existing parameters.

CHAPTER SIX

A MULTI DOF MODEL DESIGN USING SIMMECHANICS

6.1 Basic Approach

Designing and solving multi degree of freedom mechanical systems are hard tasks to accomplish since the nonlinearities may occur and the number differential equations to solve increase. So making assumptions and simplifications on the equations provides ways to solve equations. But these assumptions and simplifications may cover the true behaviour of the system. Simmechanics in this aspect is a powerful tool that provides the combination of CAD drawings and assemblies with Simulink environment. Computer-aided design (CAD) tools allow you to model machines geometrically as collections of parts, or assemblies. Simulink® and SimMechanics™ software use a block diagram approach to model control systems around mechanical devices and simulate their dynamics. The block diagram approach does not include full geometric information, nor do CAD assemblies typically incorporate controllers or allow you to perform dynamic simulations (Mathworks, Simmechanics CAD Translator Guide, 2003). Translating the CAD models with the assembly constraints into Simulink environment allows you either define force elements, or actuators. So a complete control system can be simulated.

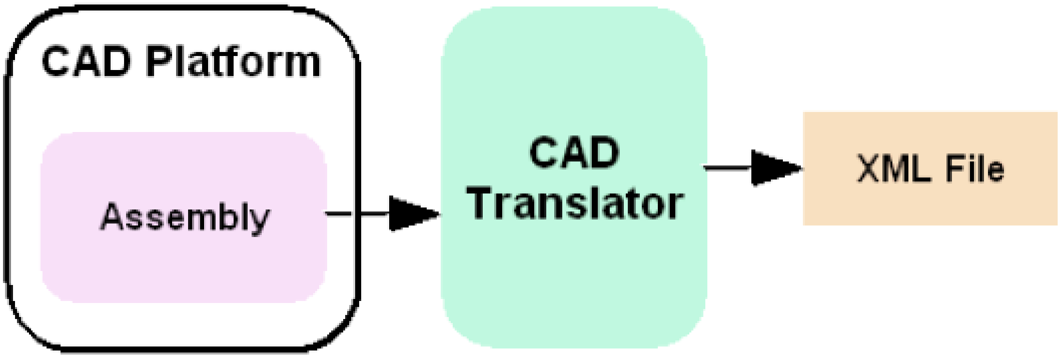

Figure 6.1 Translating CAD data into an Simulink convertable XML file.

The translation operation is made with a software called CAD translator and an XML file is created. Then the XML file is imported in Matlab via ‘mech_import’ command. The imported file contains the mass and inertia tensors, and assembly constraints and it becomes a simulink model.

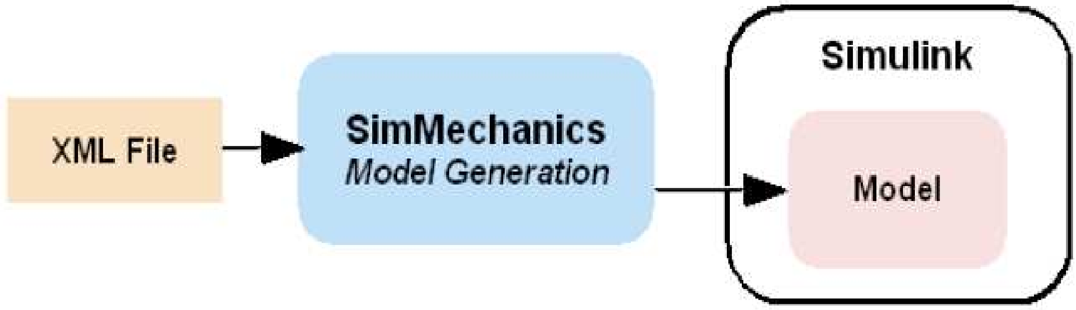

Figure 6.2 Converting the XML file to Simulink model.

The imported model file is crude, so some operations must be done before using. Adding the force elements and the actuators makes the dynamic system complete. The Simmechanics models are compatible with other Simulink blocks so other blocks in Simulink library can be used.

6.2 Joint, Body, Force Elements

The joint, body and force elements used in the Simmechanics model are different from other simulink blocks. They have a compact structure which can contain all the geometric information inside. Thus the explanation of the blocks will be made in order to provide better understanding.

- **Machine Environment:** Defines mechanical simulation environment for the machine to which the block is connected: gravity, dimensionality, analaysis mode, constraint solver type, tolerances, linearization and visulization.
- **Ground:** Grounds one side of a joint to a fixed location in the World coordinate system.
- **Body:** Represents a user-defined rigid body. Body defined by mass m, inertia tensor I, coordinate origins, axes for center of gravity (CG) and other user-specified Body coordinate system. The body dialog box sets body initial

position and orientation unless, body and/or connected joints are actuated seperately. The body dialog box also provides optional settings for customized body geometry and color.

- **Revolute Joint:** Represents one rotational degree of freedom. The follower (F) Body rotates relative to the base (B) Body about a single rotational axis going through collocated Body coordinate system origins. Sensor and actuator ports can be added. Base-follower sequence and axis direction determine sign of forward motion by the right-hand rule.
- **Prismatic Joint:** Represents one translational degree of freedom. The follower (F) body translates relative to the base (B) Body along single translational axis connecting Body coordinate origins. Sensor and actuator ports can be added. Base-follower sequence and axis direction determine sign of forward motion.
- **Body Spring&Damper:** Models a damped linear oscillator between two Bodies, equivalent to a translational spring and damper. The force F between the bodies is projected along the axis connecting the Body coordinate systems and is a function of the relative displacement r and velocity v of these Body coordinate systems, given by F = -k*(r-r0) - b*v. The parameters r0, k, and b represent the spring's natural length, the spring constant, and the damper constant, respectively.

6.3 Simmechanics Model and Simulation Results

The Simmechanics model builded in Matlab consists of many degrees of freedom. The force elements are assumed linear in the blocks, so in the model. But in this model the angle linearizations do not exist since it’s not needed. Simmechanics model contain the body blocks, joint blocks, force element (spring&damper) blocks,actuator blocks and sensor blocks. The actuation signal is the source of excitation from the tyres through the vehicle body. A typical Simmechanics block integration is shown in Figure 6.3.

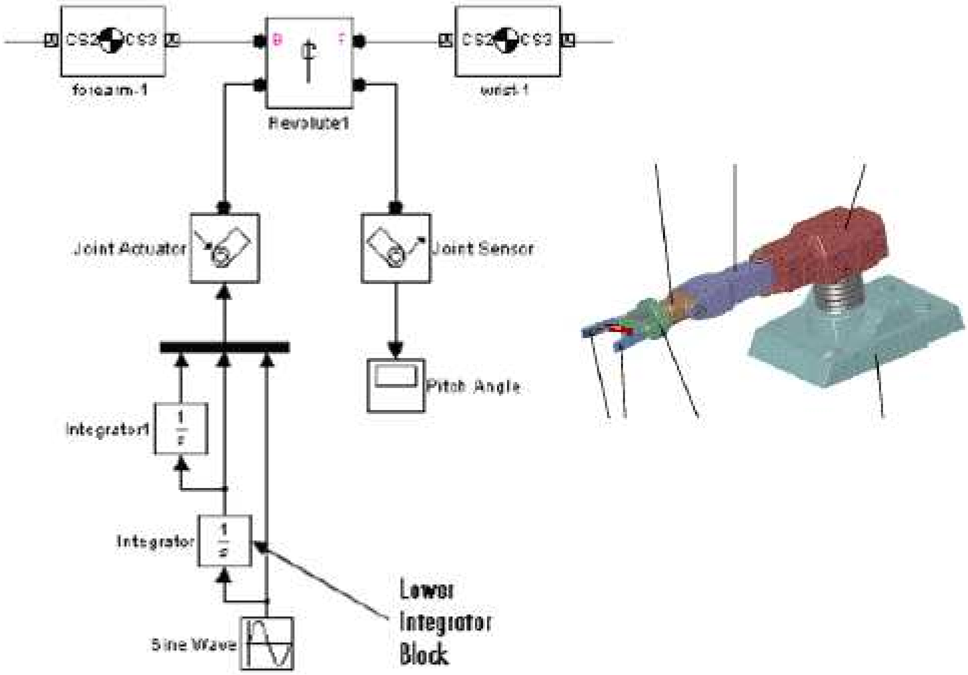

Figure 6.3 Typical Simmechanics block integration o a robot arm.

The truck model built int Simmechancis can be seen below at Figure 6.4. The model consists of too many bodies and joints connecting the bodies. The joint spring dampers are used as force elements. With the aid of force plates modelled in solid works, the actuation is given to the system. The actuated force plates are in connection with tyres with a spring&damper couple. And the force plates are connected to the ground with prismatic joints. The system resembles a shaker machine which is widely used in most automobile firms. The excitation to the system can be given as 4 independent inputs from 4 tyres. This helps to visualize the effects of inputs independently.

The outputs of the system are as RMS acceleration. But if desired the outputs can be observed as position and velocity. Also all the rotations of rotating bodies can be observed.

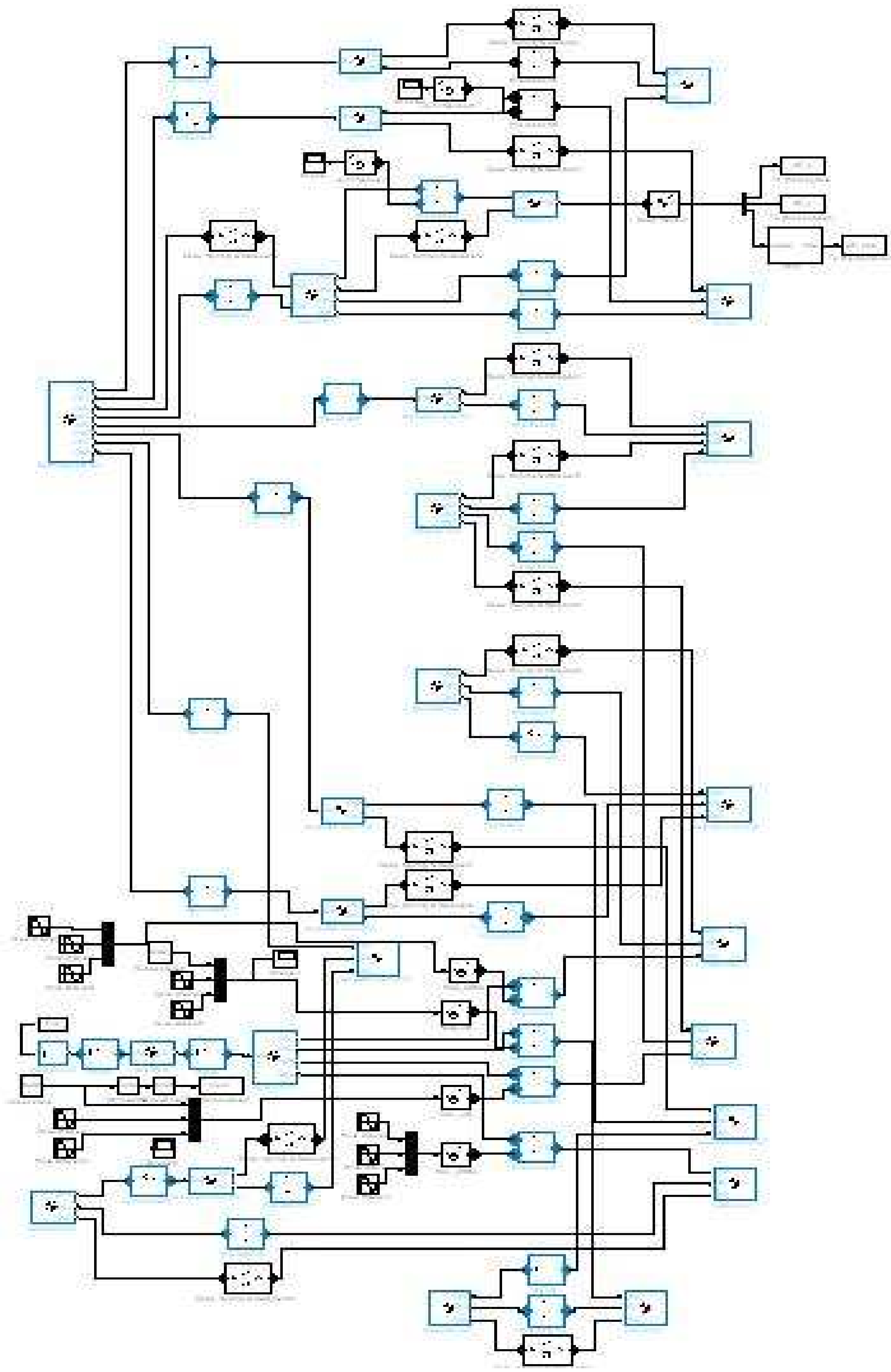

Figure 6.4 Truck model built in Simmechanics.

The simulation can be animated if desired. But the CAD models and the model file must be in the same directory. The animation of the truck model built in Simmechanics can be observed in Figure 6.5.

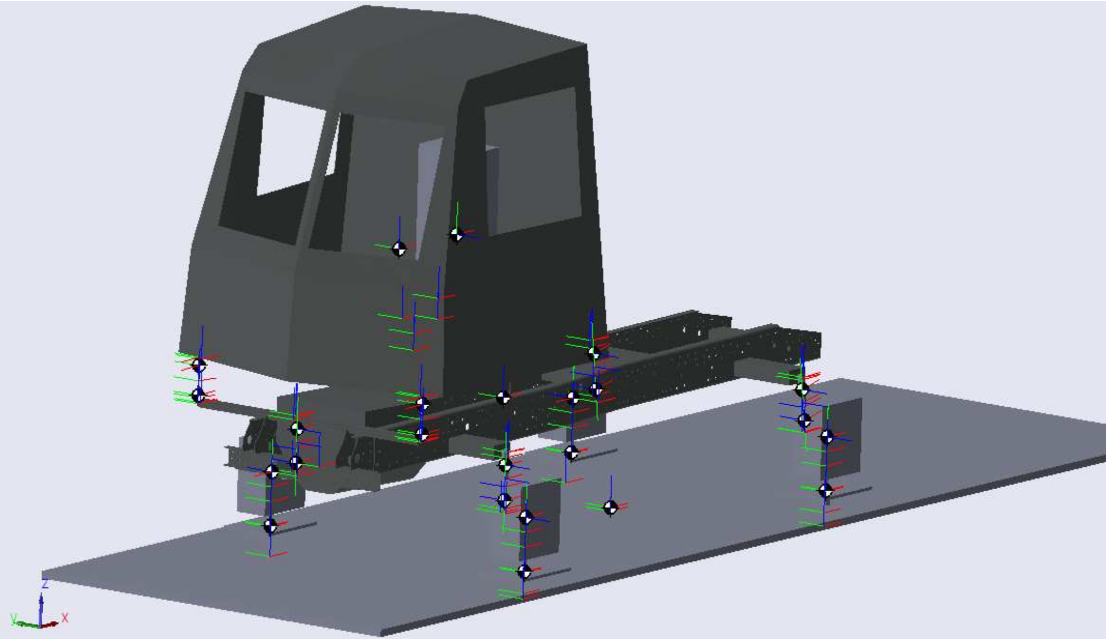

Figure 6.5 Simmechanics animation.

The motion of rigid bodies can be seen in Simmechanics animation which provides a powerful visualization tool. The Simmechanics model built can excited from 4 independent tyres. The three dimensional rotation and translation of chasis, cabin and the seat can be obsereved from simulation. Thus it provides immense observation on the dynamics of the heavy trucks.

In order to make a comparison between the linear 7DOF half truck model and the full model built in Simmechanics some results are printed below in Figure 6.6. The RMS vertical acceleration values are given for the corresponding design scenarios mentioned in chapter 4. The results are not parallel exactly with the results found in 7 DOF half truck model. But the RMS values are almost at the same range. The reasons of the deviation might be the linearization operation and the assumptions made to built the 7 DOF half truck model. The Lowest RMS vertical seat acceleration occurs in design scenario-11. The ideal design scenario was found to be number 12 in linearize half truck model. It is good to be mention the difference between design scenario 11 and 12 is very slight which provides us a better understanding about the optimum cabin suspension parameters. Another result is presented in Figure 6.7 which gives an idea about the effect of damping for different cabin suspension stiffnesses.

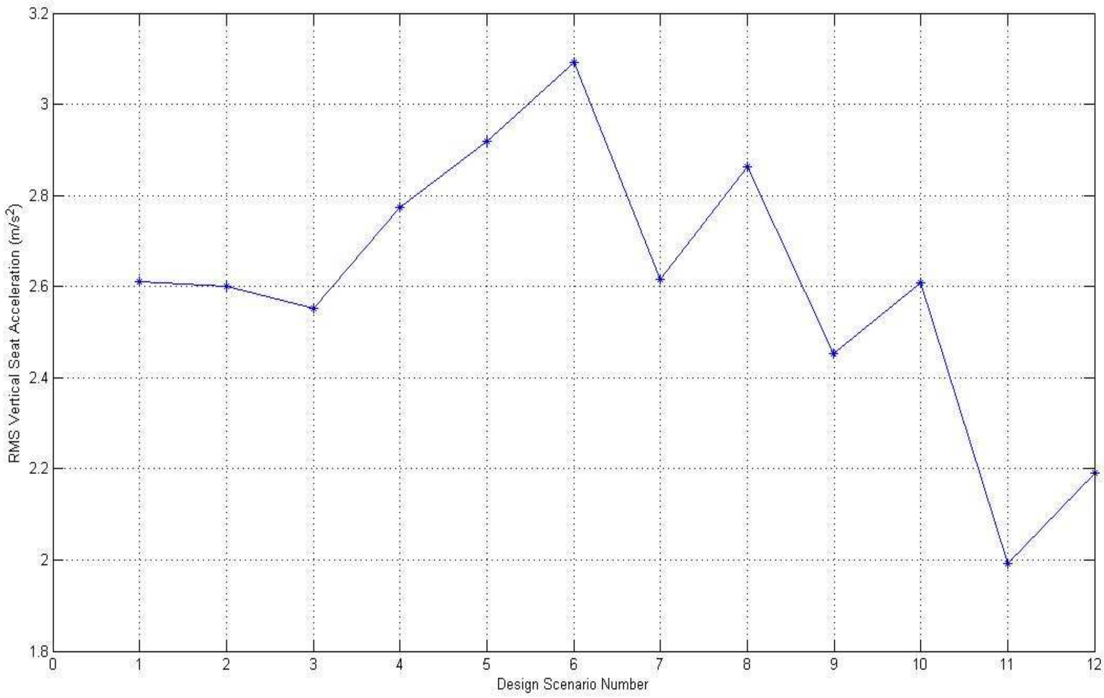

Figure 6.6 RMS seat vertical accelerations for design scenarios (Simmechanics simulation).

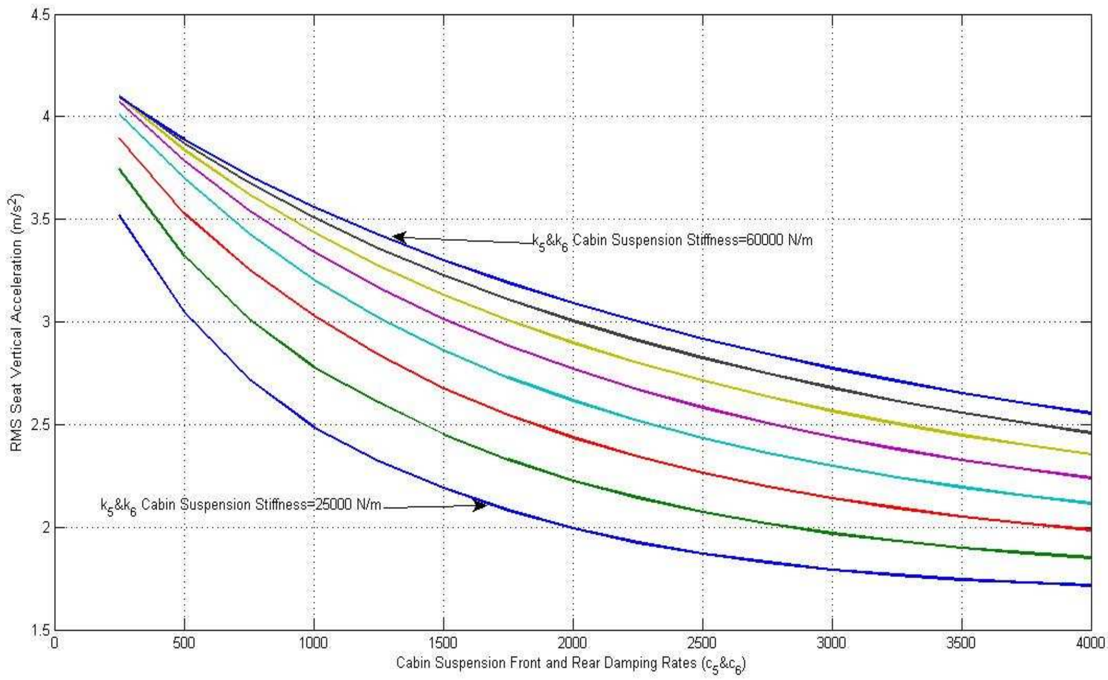

Figure 6.7 The effect of damping for different cabin suspension stiffnesses.

The lowest RMS vertical acceleration occurs in 25000 N/m front and rear cabin suspension stiffnesses. It proves the idea of obtaining more comfortable ride with soft springs. In this simulation the effect of damping is different as shown in Figure 6.7. The increase in damping causes decrease in RMS acceleration. But as it can be understood from the Figure, increasing the damping has no more effect on vertical RMS acceleration after a certain damping rate.

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