

Design, Fabrication and Testing of Active Suspension System Using an Electromagnetic Actuator

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1. INTRODUCTION

Suspension is the term given to the system of springs, shock absorbers and linkages that connects a vehicle to its wheels. Suspension systems serve a dual purpose — contributing to the vehicle's road holding/handling and braking for good active safety and driving pleasure, and keeping vehicle occupants comfortable and reasonably well isolated from road noise, bumps, and vibrations, etc. These goals are generally at odds, so the tuning of suspensions involves finding the right compromise. It is important for the suspension to keep the road wheel in contact with the road surface as much as possible, because all the forces acting on the vehicle do so through the contact patches of the tires. The suspension also protects the vehicle itself and any cargo or luggage from damage and wear.[1]

The suspension system must support the vehicle, provide directional control during handling manoeuvres and provide effective isolation of passengers/payload from road disturbances. Good ride comfort requires a soft suspension, whereas insensitivity to applied loads requires stiff suspension. Good handling requires a suspension setting somewhere between the two.

Due to these conflicting demands, suspension design has had to be something of a compromise, largely determined by the type of use for which the vehicle was designed. Active suspensions are considered to be a way of increasing the freedom one has to specify independently the characteristics of load carrying, handling and ride quality.

A passive suspension system has the ability to store energy via a spring and to dissipate it via a damper. Its parameters are generally fixed, being chosen to achieve a certain level of compromise between road holding, load carrying and comfort.

An active suspension system has the ability to store, dissipate and to introduce energy to the system. It may vary its parameters depending upon operating conditions and can have knowledge other than the strut deflection the passive system is limited to.[2]

1.1. Driving comfort:

If the vibration of the car body has to be controlled for obtaining optimum ride comfort the body has to be isolated from road disturbances and decrease the resonance peak of sprung mass near 1Hz, which is the sensitive frequency range for human body. To improve ride stability, tire should remain in contact with road surface. For a given suspension spring, better isolation of sprung mass road disturbances can be achieved with soft damping by allowing a larger deflection. However, better road contact can be achieved with a hard damping preventing unnecessary suspension deflections.

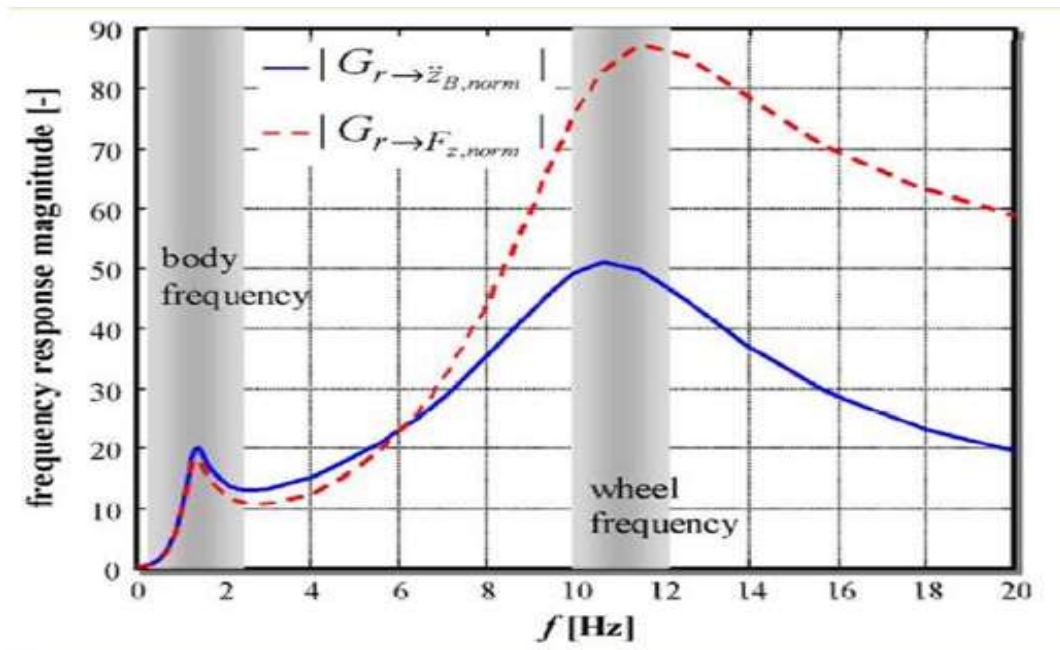


Fig 1.1 Frequency Response magnitude for Normalized Body Acceleration and Tire Load for a Passive Suspension System [1]

Drive comfort and drive safety are thus two conflicting criteria. Active suspensions try to solve this problem by adaption of damping or stiffness of spring of actual demands.

A parameter adaptive suspension system uses a controller that refers to a control system which adapts its behaviour to changing settings of the system to be controlled and its signals.

Main objective of the ground vehicle suspension system in this project is to isolate vehicle body from road irregularities in order to maximize passenger ride comfort and produce continuous road wheel contact, improving vehicle handling quality.[1]

1.2. Passive Suspension Systems:

Passive suspension is composed of the non-controlled spring and the shock absorbing damper. Both components work mechanically in parallel and are fixed between the wheel supporting structure (unsprung mass) and the vehicle body (sprung mass).The damper is a cylinder filled with hydraulic oil or Compressed gas. Inside the cylinder there is a piston driven by a rod. This fluid or gas flow generates a reaction force that is proportional to the relative speed between sprung and unsprung masses. The damping is achieved by converting the energy of the oscillations in heat.

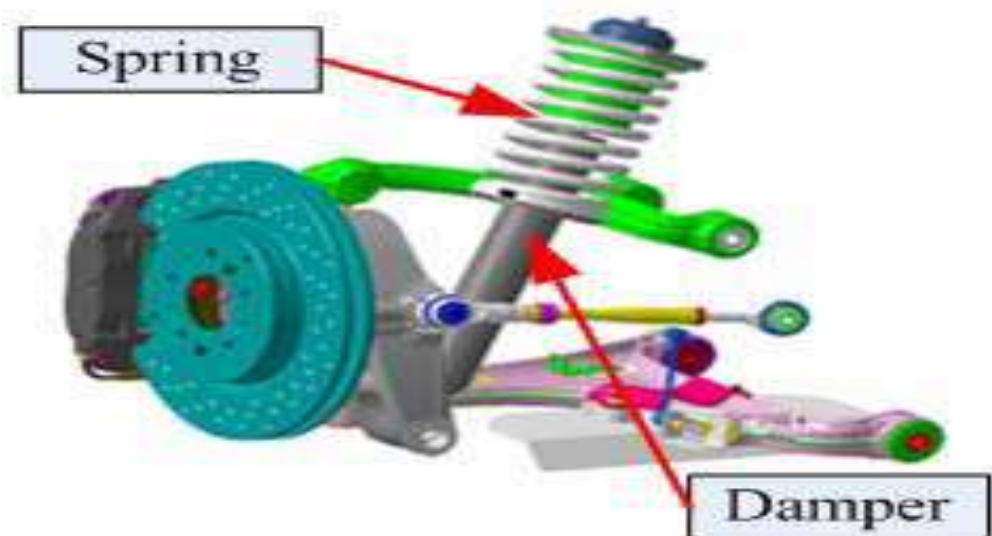


Figure 1.2: Profile of a Commercial Independent Passive System [2]

1.2.2 Advantages of Passive Suspension Systems:

Passive suspension systems are popular in vehicles due to their simplicity, small volume, low cost and high reliability. On the plus side, the system is very predictable. Over time, you will develop a familiarity with your car's suspension. You will understand its capabilities and its limitations.

1.2.3 Disadvantages of Passive Suspension Systems:

Passive suspensions cannot give satisfactory results for suspension problems,

Because passive suspensions have constant elastic characteristics and damping characteristics, and only respond to the external excitation passively, and do not allow any active control.[2] Also once the system has reached these limits, it has no way of compensating for situations beyond its design parameters. Thus shock absorbers bottom out, struts overextend, and springs respond sluggishly, torsion bars get tweaked.

1.3. Active Suspension Systems:

An active suspension system has the ability to store, dissipate and to introduce energy to the system. It may vary its parameters depending upon operating conditions. An active suspension includes an actuator that can supply active force, which is regulated by a control algorithm using data from sensors attached to the vehicle.

An active suspension is composed of an actuator and a mechanical spring, or an actuator, a mechanical spring and a damper. An active suspension controls both the sprung mass and the unsprung mass with the actuator working mechanically in parallel with the spring.

The active suspension is named generally as the hydraulic or pneumatic one if the actuator is selected as the hydraulic or pneumatic actuator, and the active suspension is named generally as the electromagnetic one if the actuator is an electromagnetic actuator. Active suspensions commercially implemented in automobiles today are based on the hydraulic or pneumatic one.[2]

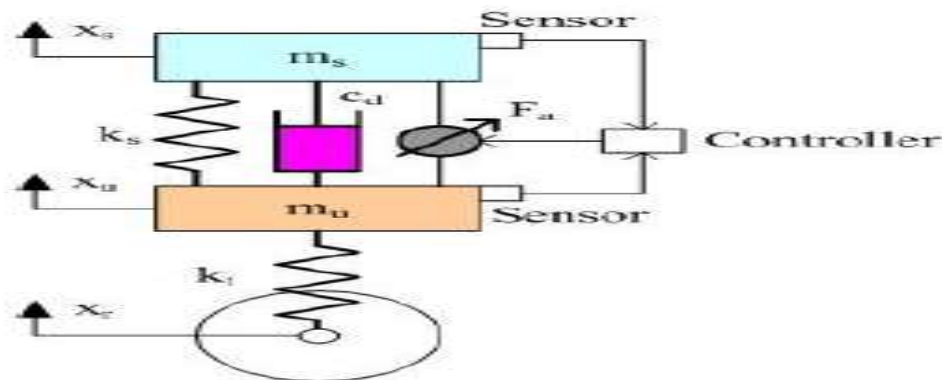


Fig 1.3: Diagrammatic representation of an Active Suspension System [2]

1.4. PID Controller:

A proportional–integral–derivative controller (PID controller) is a generic control loop feedback mechanism (controller) widely used in industrial control systems – a PID is the most commonly used feedback controller. A PID controller calculates an "error" value as the difference between a measured process variable and a desired setpoint. The controller attempts to minimize the error by adjusting the process control inputs.[3]

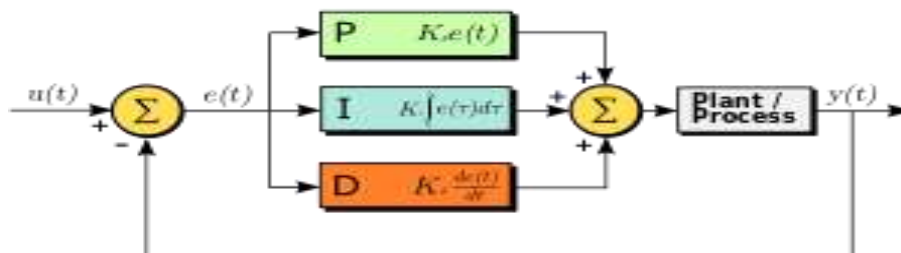


Fig1.4: Block Diagram of PID Controller

1.4.1 PID controller theory:

The PID control scheme is named after its three correcting terms, whose sum constitutes the manipulated variable (MV).

The proportional, integral, and derivative terms are summed to calculate the output of the PID controller. Defining $u(t)$ as the controller output, the final form of the PID algorithm is:

$$u(t) = MV(t) = K_p e(t) + K_i \int_0^t e(\tau) d\tau + K_d \frac{d}{dt} e(t)$$

Where:

K_p : Proportional gain, a tuning parameter

K_i : Integral gain, a tuning parameter

K_d : Derivative gain, a tuning parameter

e : Error = $SP - PV$; SP: Set Point, PV: Process Variable

t : Time or instantaneous time (the present)

Proportional term

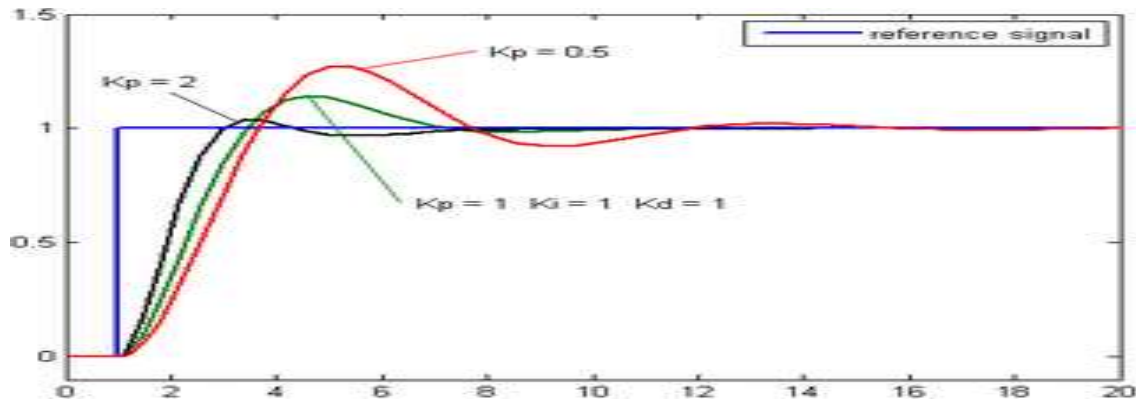


Fig1.5: Plot of PV vs time, for three values of K_p (K_i and K_d held constant)

The proportional term is given by:

$$P_{out} = K_p e(t)$$

A high proportional gain results in a large change in the output for a given change in the error. If the proportional gain is too high, the system can become unstable. In contrast, a small gain results in a small output response to a large input error, and a less responsive or less sensitive controller. If the proportional gain is too low, the control action may be too small when responding to system disturbances.

Integral term:

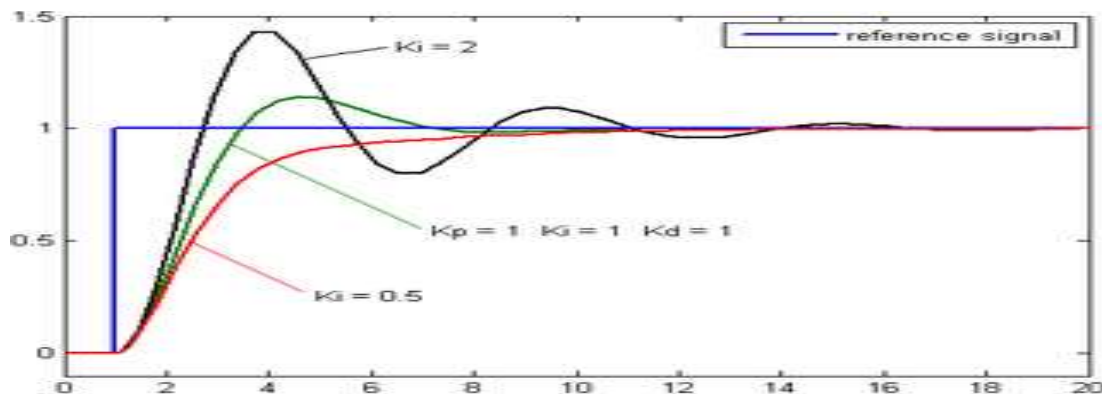


Fig1.6: Plot of PV vs time, for three values of K_i (K_p and K_d held constant)

The integral term is given by:

$$I_{out} = K_i \int_0^t e(\tau) d\tau$$

The integral term accelerates the movement of the process towards setpoint and eliminates the residual steady-state error that occurs with a pure proportional controller. However, since the integral term responds to accumulated errors from the past, it can cause the present value to overshoot the setpoint value.

Derivative term:

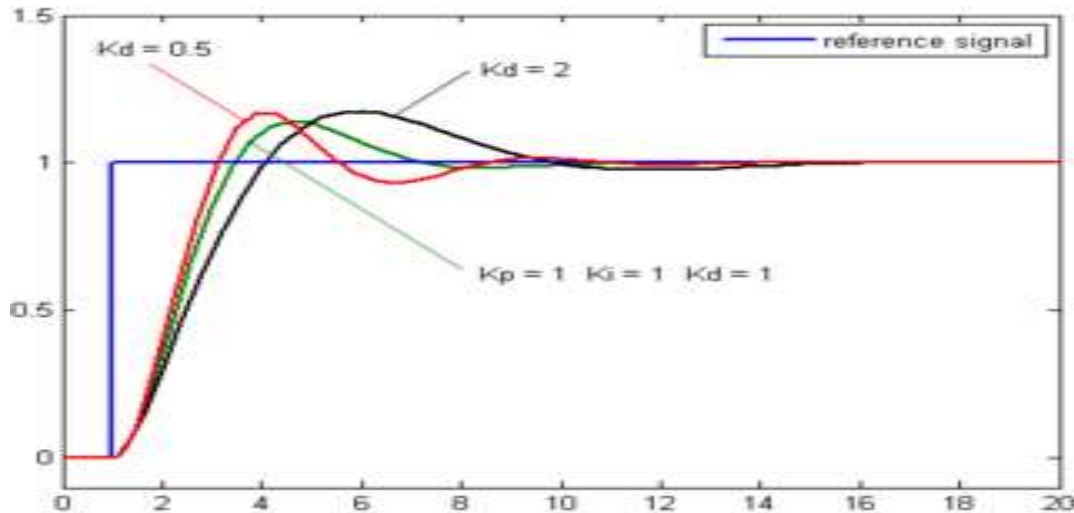


Fig1.7: Plot of PV vs time, for three values of K_d (K_p and K_i held constant)

The derivative term is given by :

$$D_{out} = K_d \frac{d}{dt} e(t)$$

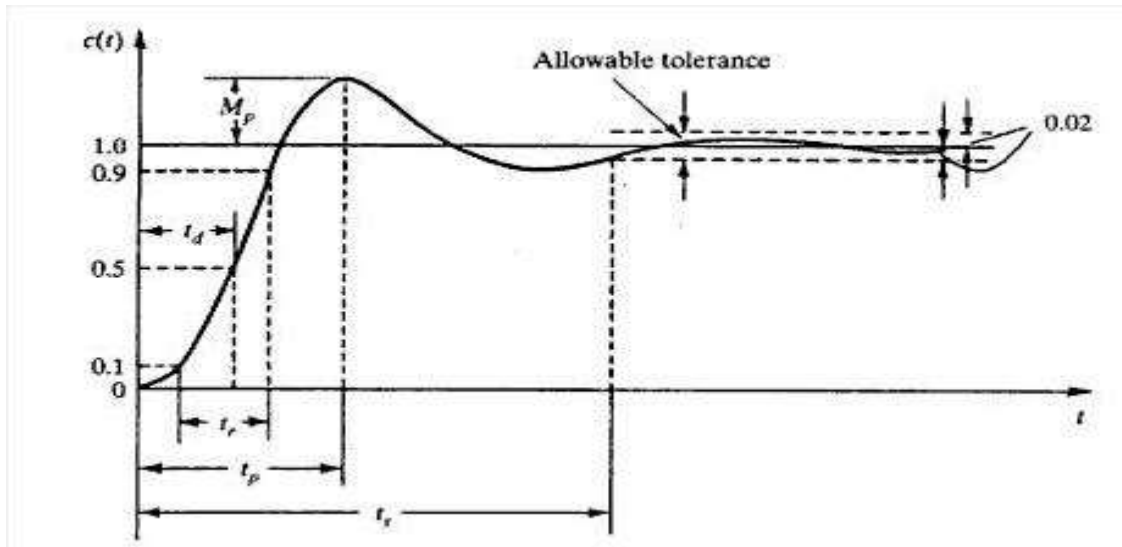
The derivative term slows the rate of change of the controller output. Derivative control is used to reduce the magnitude of the overshoot produced by the integral component and improve the combined controller-process stability. However, the derivative term slows the transient response of the controller. Also, differentiation of a signal amplifies noise and thus this term in the controller is highly sensitive to noise in the error term, and can cause a process to become unstable if the noise and the derivative gain are sufficiently large. Hence an approximation to a differentiator with a limited bandwidth is more commonly used. Such a circuit is known as a phase-lead compensator.[7]

1.4.2 Overview of the Manual Tuning Method:

There are several methods for tuning a PID loop The Manual Method, Ziegler-Nichols, and Cohen-Coon Method etc.[1] The most effective methods generally involve the development of some form of process model, and then choosing P, I, and D based on the dynamic model parameters. Manual tuning methods can be relatively inefficient, particularly if the loops have response times on the order of minutes or longer.

If the system must remain online, one tuning method is to first set K_i and K_d values to zero. Increase the K_p until the output of the loop oscillates, then the K_p should be set to approximately half of that value for a "quarter amplitude decay" type response. Then increase K_i until any offset is corrected in sufficient time for the process. However, too much K_i will cause instability. Finally, increase K_d , if required, until the loop is acceptably quick to reach its reference after a load disturbance. However, too much K_d will cause excessive response and overshoot. A fast PID loop tuning usually overshoots slightly to reach the setpoint more quickly; however, some systems cannot accept overshoot, in which case an over-damped closed-loop system is required, which will require a K_p setting significantly less than half that of the K_p setting causing oscillation.

1.4.3 Transient Response Specifications:



Frequently, the performance characteristics of a control system are specified in terms of the transient response to a unit step input, since it is easy to generate and is sufficiently drastic. (If the response to a step input is known, it is mathematically possible to compute the response to any input.)

The transient response of a practical control system often exhibits damped oscillations before reaching steady state. In specifying the transient-response characteristics of a control system to a unit-step input, it is common to specify the following:

1. Delay time, t_d : The delay time is the time required for the response to reach half the final value the very first time.
2. Rise time, t_r : The rise time is the time required for the response to rise from 10% to 90%, or 0% to 100% of its final value. For underdamped second order systems, the 0% to 100% rise time is normally used. For overdamped systems, the 10% to 90% rise time is commonly used.
3. Peak time, t_p : The peak time is the time required for the response to reach the first peak of the overshoot.
4. Maximum (percent) overshoot, M_p : The maximum overshoot is the maximum peak value of the response curve measured from unity. If the final steady-state value of the response differs from unity it is common to use the maximum percent overshoot. The amount of the maximum overshoot directly indicates the relative stability of the system.
5. Settling time, t_s : The settling time is the time required for the response curve to reach and stay within a range about the final value of size specified by absolute percentage of the final value (usually 2% or 5%). The settling time is related to the largest time constant of the control system.

The time domain specifications just given are quite important, since most control systems are time-domain systems; that is, they must exhibit acceptable time responses.

Also it is interesting to note the maximum overshoot and rise time conflict with each other and both cannot be made smaller simultaneously.[7]

1.4.4 Phase Margin, Gain Margin and Stability using Bode Plot

Phase Margin: The phase margin is that amount of additional phase lag at the crossover frequency required to bring the system to the verge of instability. The gain crossover frequency at which $|G(j\omega)|$, the magnitude of the open loop transfer function is unity. The phase margin is 180° plus the phase angle of the open loop transfer function at the gain cross over frequency [8], or $\gamma = 180^\circ + \Phi$

Gain Margin: The Gain Margin is the reciprocal of the magnitude $|G(j\omega)|$ at the frequency at which the phase angle is -180° [7].

$$K_g = \frac{1}{|G(j\omega)|}$$

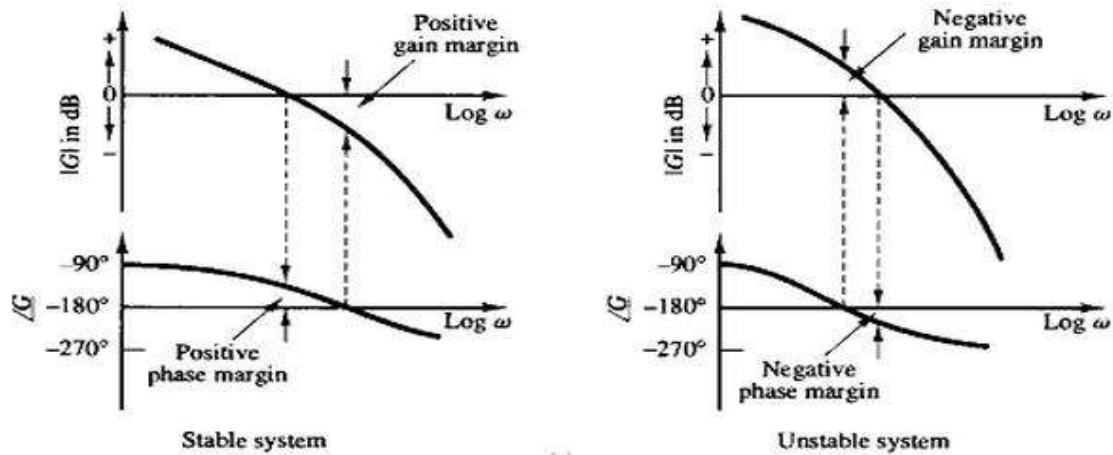


Fig1.9: Phase and Gain Margin of Stable and Unstable systems [7]

1.5. Nyquist Sampling Theorem:

The Nyquist–Shannon sampling theorem, after Harry Nyquist and Claude Shannon, is a fundamental result in the field of information theory, in particular telecommunications and signal processing. Sampling is the process of converting a signal (for example, a function of continuous time or space) into a numeric sequence (a function of discrete time or space). The theorem states [4]:

If a function $x(t)$ contains no frequencies higher than B hertz, it is completely determined by giving its ordinates at a series of points spaced $1/(2B)$ seconds apart.

1.6. Accelerometer:

An accelerometer is an electromechanical device that will measure acceleration forces. These forces may be static, like the constant force of gravity pulling at your feet, or they could be dynamic - caused by moving or vibrating the accelerometer. The variety of accelerometers used results from different applications with requirements of range, natural frequency, and damping. The basic difference is in the method of mass displacement measurement. In general, the specification sheets for an accelerometer will give the natural frequency, damping coefficient, and a scale factor that relates the output to an acceleration input. A few examples of the commonly available accelerometers are Potentiometric, LVDT, Variable reluctance and Piezoelectric type.[9]

1.5.1 Piezoelectric type accelerometer:

The piezoelectric accelerometer is based on a property exhibited by certain crystals where a voltage is generated across the crystal when stressed. This property is also the basis for such familiar sensors as crystal phonograph cartridges and crystal microphones. For accelerometers, the principle is shown in Figure 1.8. Here, a piezoelectric crystal is spring-loaded with a test mass in contact with the crystal. When exposed to an acceleration, the test mass stresses the crystal by a force ($F = ma$), resulting in a voltage generated across the crystal. A measure of this voltage is then a measure of the acceleration. The crystal per se is a very high-impedance source, and thus requires a high-input impedance, low-noise detector. Output levels are typically in the millivolt range. The natural frequency of these devices may exceed 5 kHz, so that they can be used for vibration and shock measurements.[9]

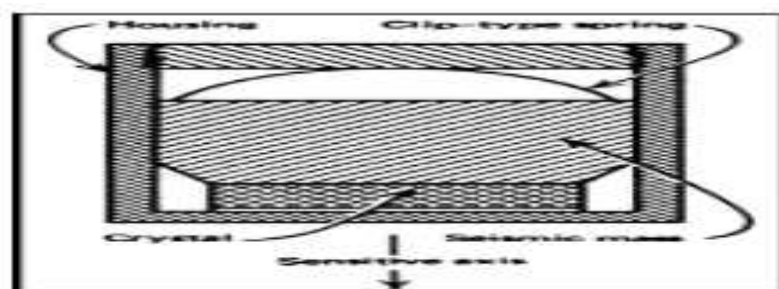


Fig1.10: Piezoelectric type accelerometer [9]

1.7. Solenoid used as an Electromagnetic Actuator:

Electromagnetic solenoid is the common electromechanical actuator for linear (translational) motion. It is consisted of the magnetic circuit (body of solenoid, magnetic subsystem), one or several electrical inputs (electrical subsystem) and mechanical circuit (mechanical subsystem). The construction of the solenoids could be quite sophisticated and typical application are digitally controlled compact pumps, electromagnetic valves, circuit breakers, micro machines, robotics.

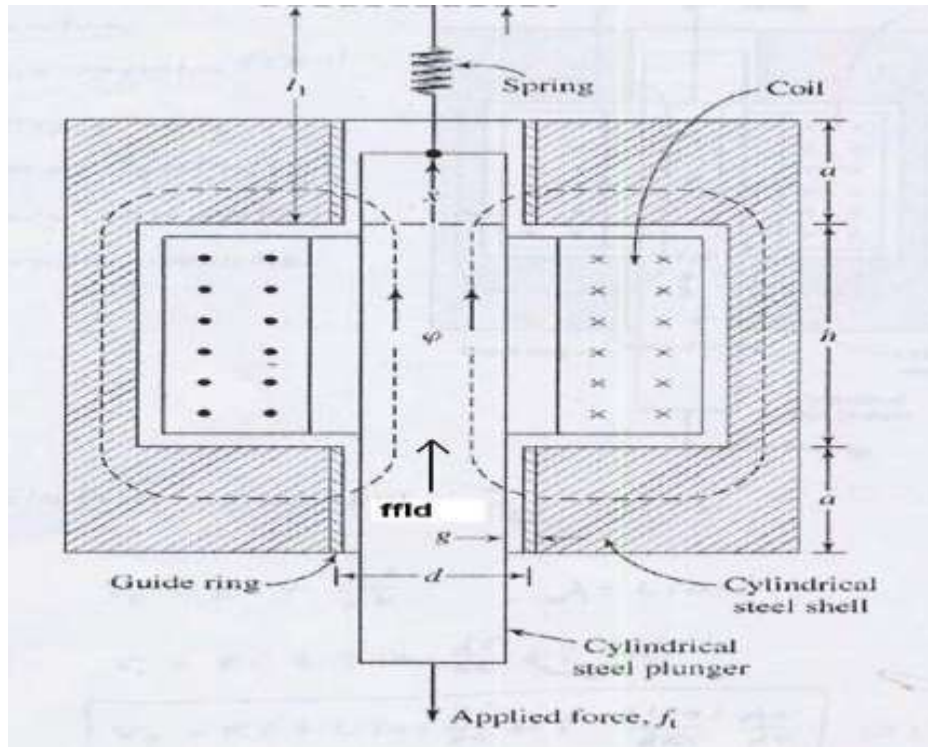


Fig 1.11: Electromagnetic Solenoid [8]

As shown in Figure 1.9; in a cylindrical ferromagnetic steel shell a cylindrical movable steel plunger has been placed. The plunger is connected to spring. Inside the shell, a coil is connected to DC voltage source. As a result of energizing coil, the electric energy will be transferred to magnetic field energy. The electromagnetic force will move plunger in a positive direction of coordinate (x) and decrease the reluctance of the magnetic circuit (increase inductance). At the end of this process, electromagnetic force will be equal to spring force.

If external mechanical force is suddenly applied to the plunger, it will decrease inductance and magnetic energy of the coupling field will be transferred to mechanical energy of the spring. When electromagnetic force equals restraining force, new operation point will be established. When mechanical force f_o drops to zero, system will go back to starting operation point and mechanical energy of the spring will be transferred to coupling field. Part of energy will be dissipated in the electric circuit during transients and due to friction losses.[8]

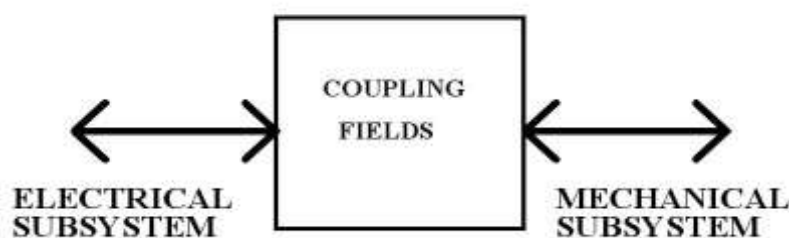


Fig 1.12: Coupled Systems in an Electromagnetic Solenoid [8]

1.8. Typical Test Signals:

The commonly used test input signals are step functions, ramp functions, acceleration functions, impulse functions, sinusoidal functions and white noise. With these test signals mathematical and experimental analysis of control systems can be carried out easily. Since the signals are very simple functions of time. Once a control system is designed on the basis of test signals the performance of the system in response to actual inputs is generally satisfactory. The use of such test signals enables one to compare the performance of many systems on the same basis.[7]

1.9. Data Acquisition System:

Data acquisition is the process of sampling signals that measure real world physical conditions and converting the resulting samples into digital numeric values that can be manipulated by a computer. Data acquisition systems (abbreviated with the acronym DAS or DAQ) typically convert analog waveforms into digital values for processing. The components of data acquisition systems include:

- Sensors that convert physical parameters to electrical signals.
- Signal conditioning circuitry to convert sensor signals into a form that can be converted to digital values.
- Analog-to-digital converters, which convert conditioned sensor signals to digital values.

Data acquisition applications are controlled by software programs developed using various general purpose programming languages such as BASIC, C, Fortran, Java, Lisp, Pascal.

Specialized software tools used for building large-scale data acquisition systems include EPICS. Graphical programming environments include ladder logic, Visual C++, Visual Basic, and LabVIEW.[4]

1.10. Serial Port Communication:

In computing, a serial port is a serial communication physical interface through which information transfers in or out one bit at a time (in contrast to a parallel port). Throughout most of the history of personal computers, data transfer through serial ports connected the computer to devices such as terminals and various peripherals.

While such interfaces as Ethernet, FireWire, and USB all send data as a serial stream, the term "serial port" usually identifies hardware more or less compliant to the RS-232 standard, intended to interface with a modem or with a similar communication device.

Modern computers without serial ports may require serial-to-USB converters to allow compatibility with RS 232 serial devices.[4]

2. SIMULATIONS AND STUDY OF THE RESPONSE AND TRANSMISSIBILITY CHARACTERISTICS

2.1. Introduction:

In the designing of any physical system, the first step is to set-up the governing mathematical relations by understanding the system and then equation of motion or equilibrium equations are written using the Free Body diagram. Now SimMechanics can directly model the physical system into a Simulink Model and using appropriate solver even non-linear problems can be solved easily. However the proposed actuator cannot directly be represented using the tools available in Simulink and hence the need for setting up the governing equations and then converting them into Simulink Model arose and subsequently the study of the existing Passive System and the Proposed System were carried out using Mathematical Models.

2.2. Simulink Physical Modelling of Passive System:

The Physical Modelling of The Passive System was done using Simulink and SimMechanics in Matlab to understand the existing system and accordingly to propose a better system to ensure the transmissibility at the Passenger Seat is greatly reduced and hence a rather comfortable ride is achieved. The Signal Builder block was used to generate the Typical Test Signals such as Step, Impulse, Ramp and Sinusoidal Inputs as shown in Figure 2.1 for studying the response of the systems and their transmissibility for the various cases.

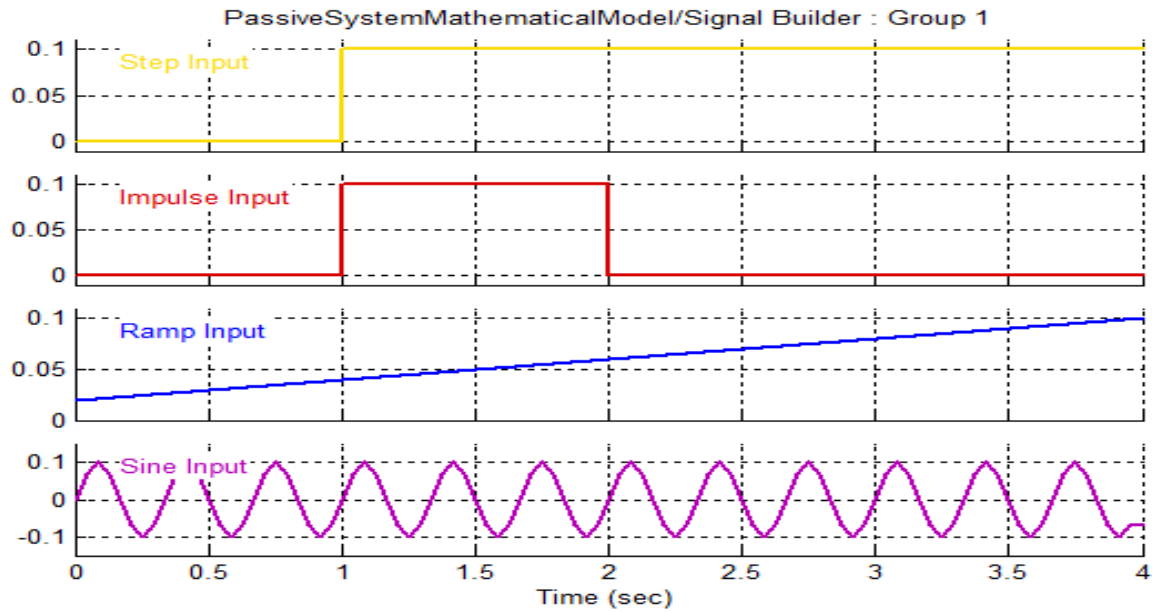


Fig2.1: Signal Builder Block

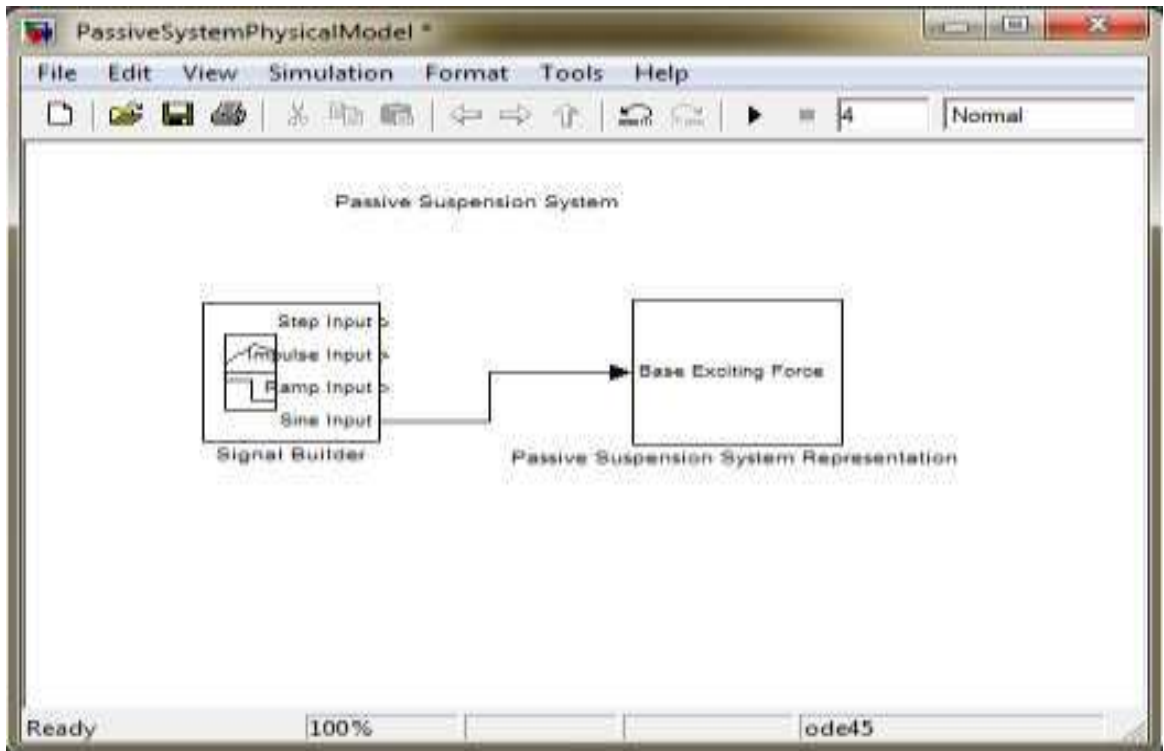


Fig 2.2: Simulink Model: Passive System (Physical)

Figure 2.2 and Figure 2.3 show the basic architecture used for generating the Simulink code. Figure 2.2 is the main model and the signal builder block and the Passive Suspension System Representation blocks are called sub systems. When the Passive Suspension System Representation block is double clicked Figure 2.3, opens up and the Physical Modelling of the system can now be seen. When the signal builder sub system is opened, Figure 2.1 opens up. Wiring the Passive Suspension System Representation to different inputs one at a time the response of the system is studied. Similarly for all the subsequent Simulink Codes similar architecture and hierarchy follows. All the Blocks in the Simulink Codes have configured as per the values of the system. The main basic difference in the Physical Model and the Mathematical Model of the Passive System is the type of nature of input. In the Physical Model a unit force is applied at the base for the study, whereas a displacement is the input parameter for the Mathematical Models. The results can be displayed by opening the various scopes as shown in the model in Figure 2.3.

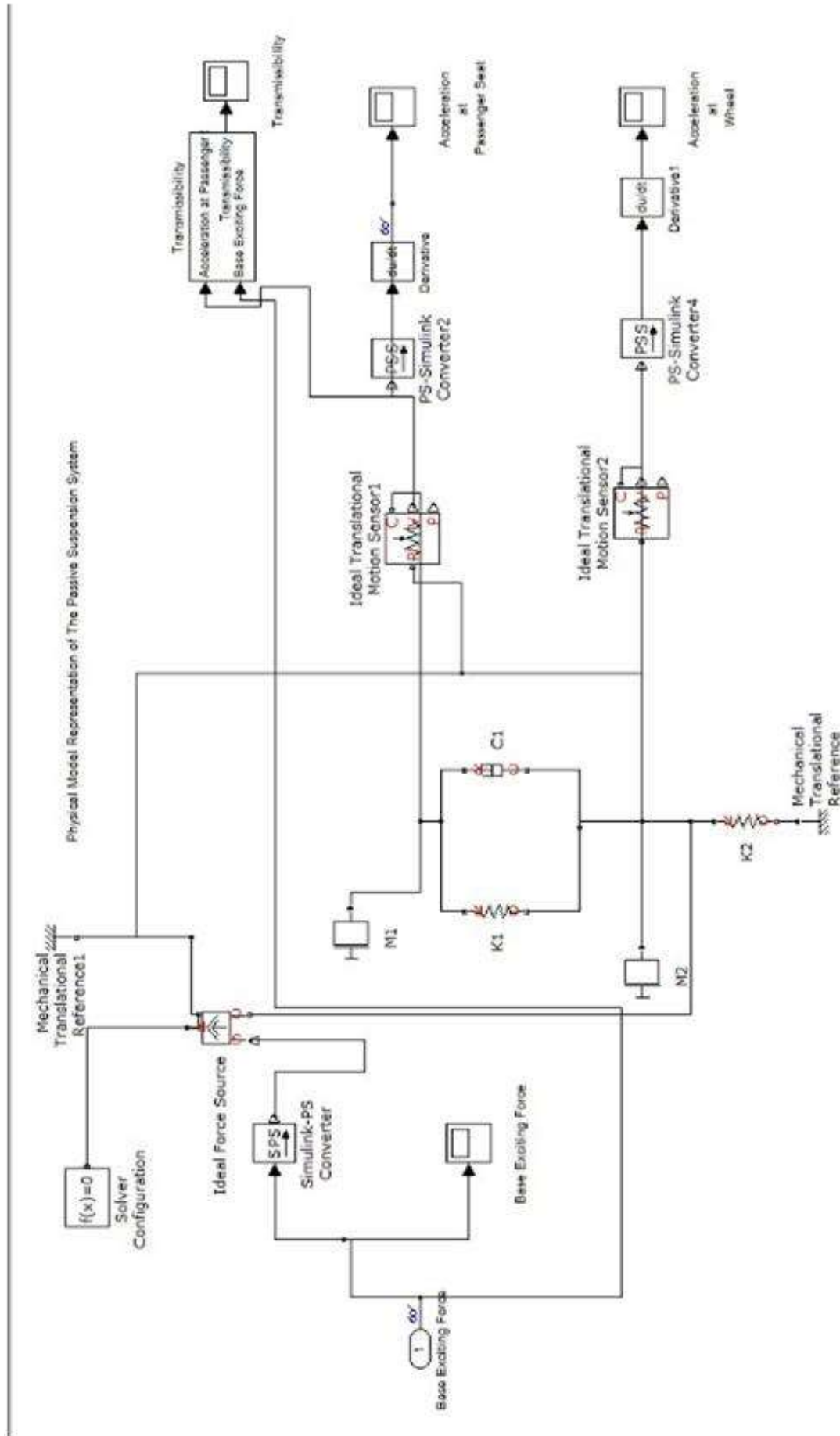


Fig 2.3: Simulink Sub-system: Passive System (Physical)

2.2.1 Results:

i) Step Input

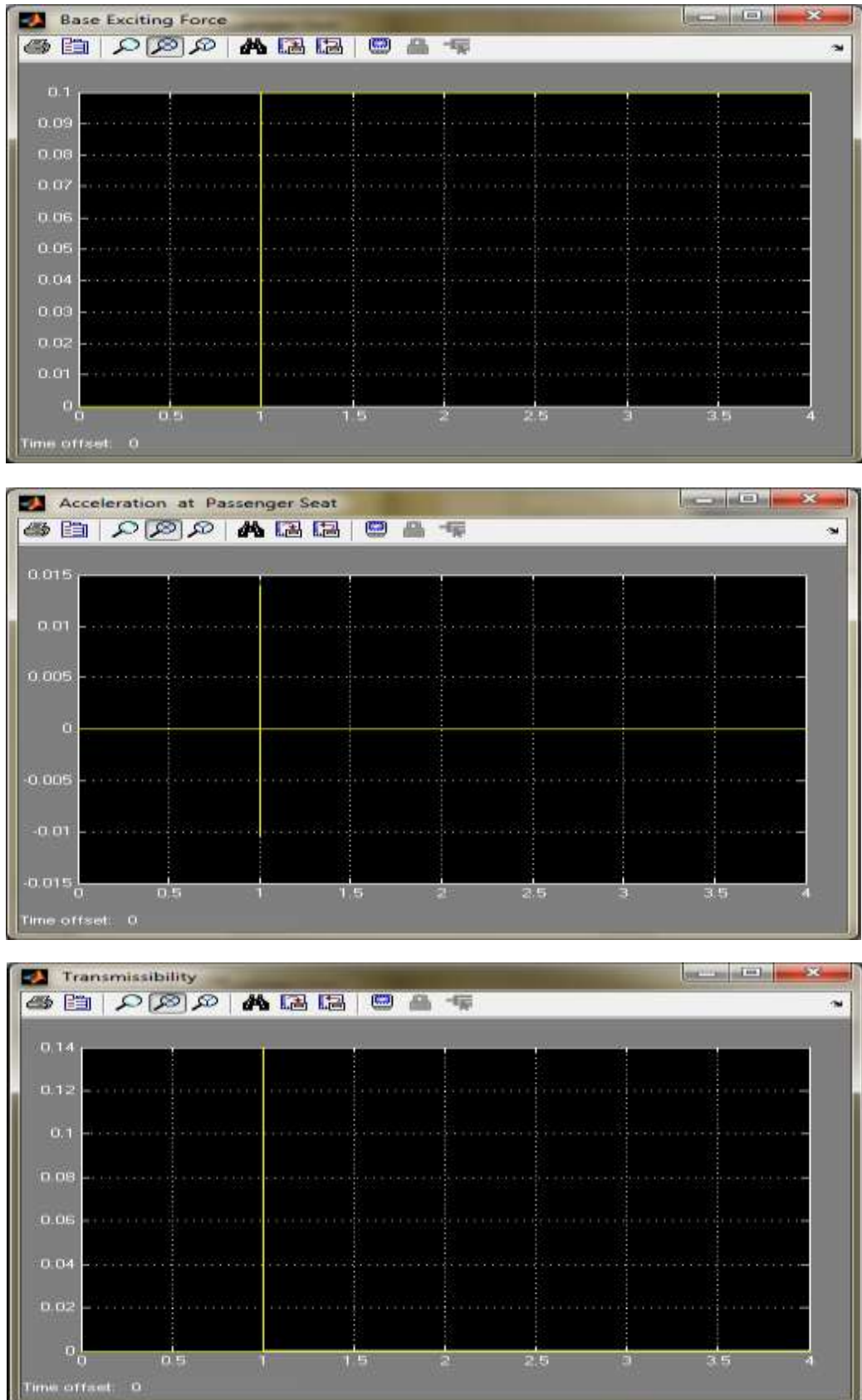


Fig 2.4: Step I/P: a) Base Excitation Force b) Acceleration at Passenger Seat c) Transmissibility

b) Impulse Input

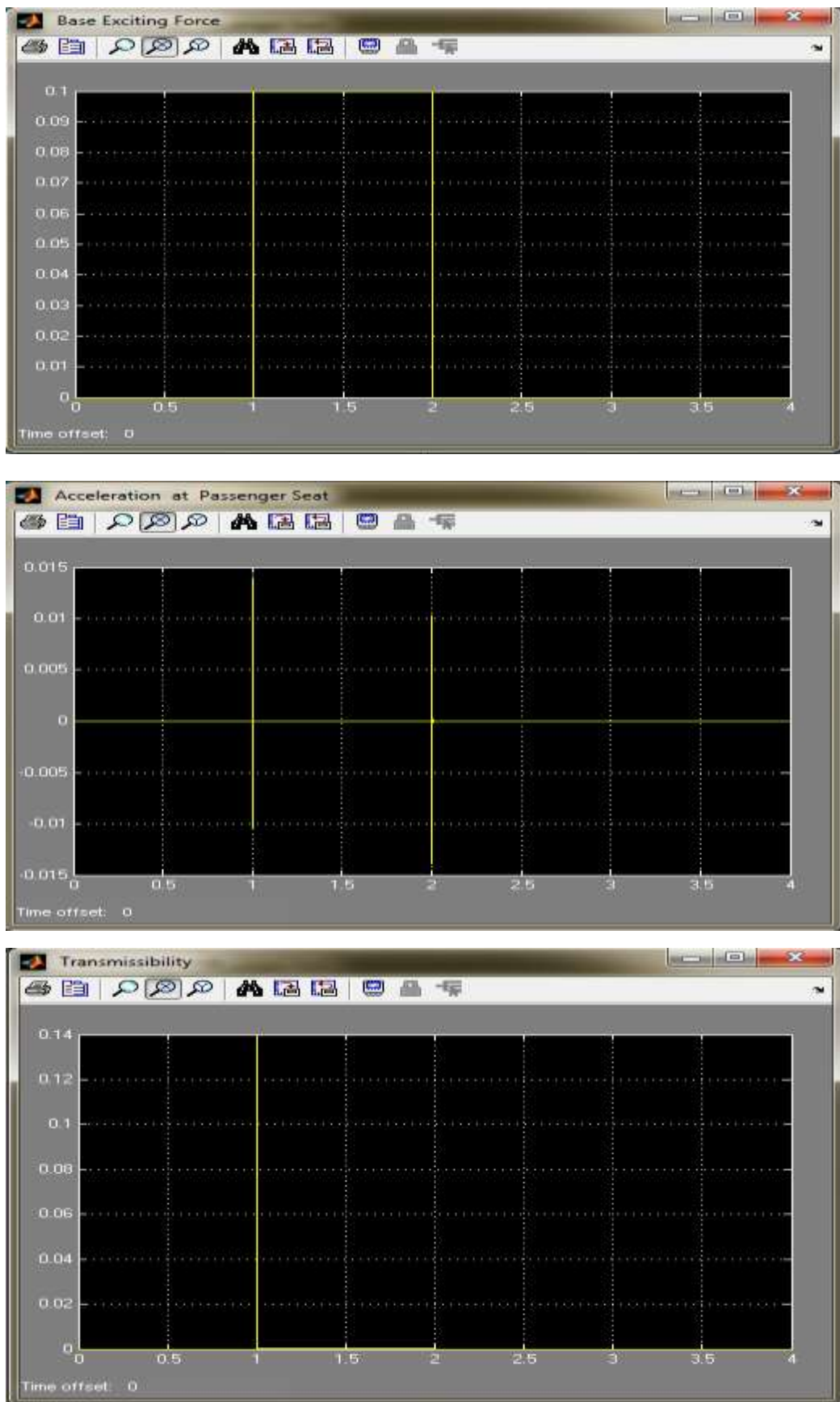


Fig 2.5: Impulse I/P a) Base Excitation Force b) Acceleration at Passenger Seat c) Transmissibility

c) Ramp Input

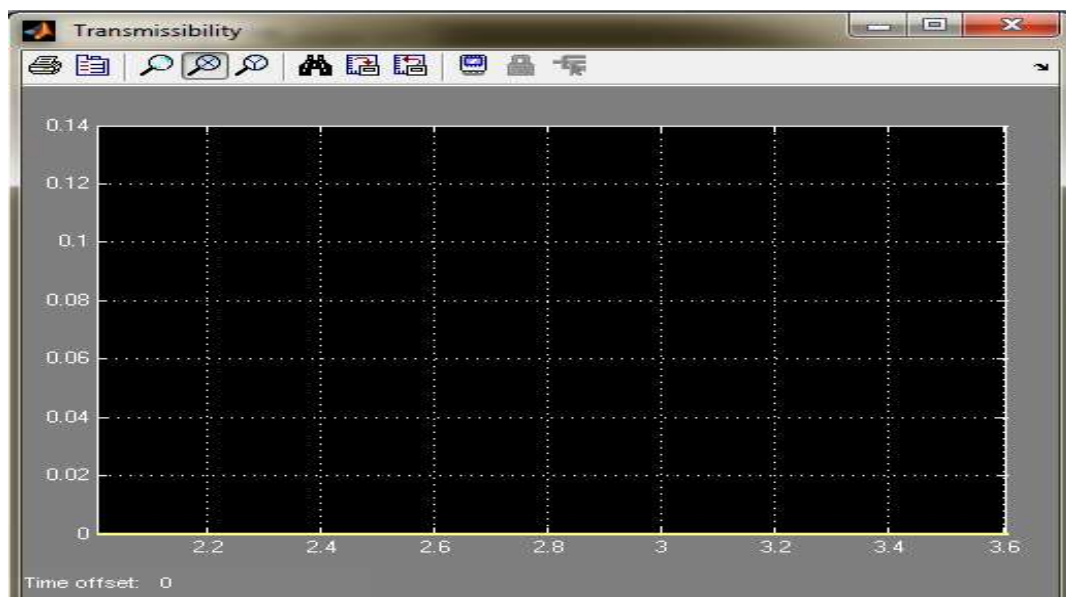
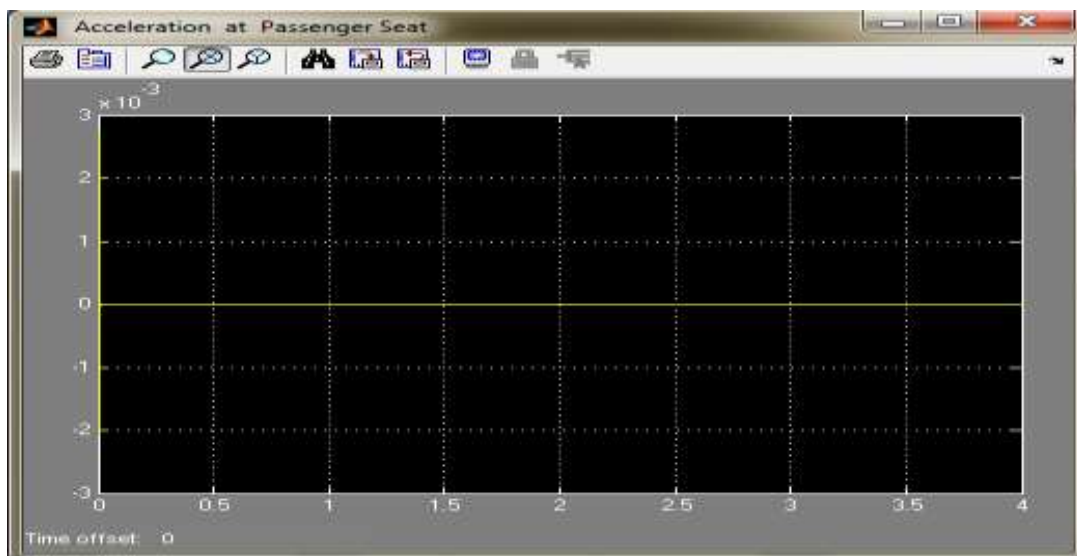
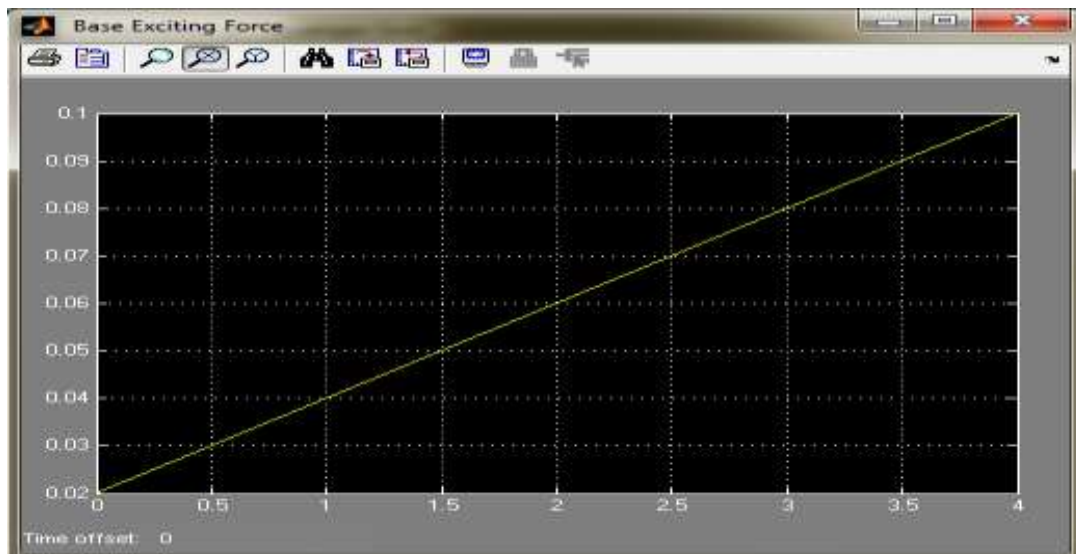


Fig 2.6: Ramp I/P a) Base Excitation Force b) Acceleration at Passenger Seat c) Transmissibility

d) Sine Input:

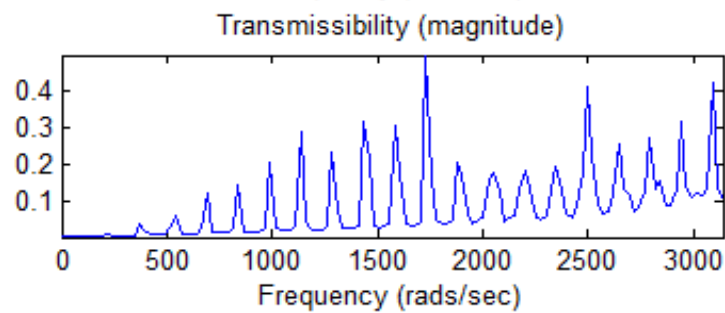
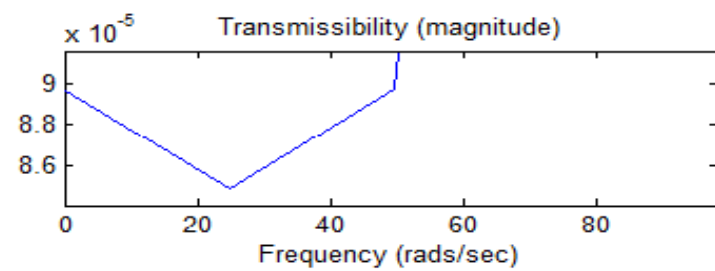
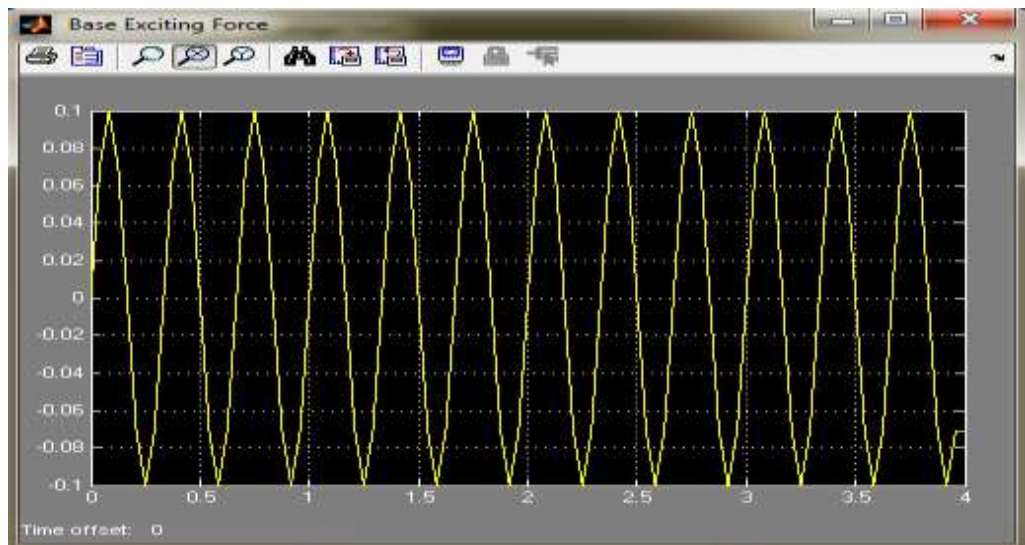


Fig 2.7: Sine I/P a) Base Excitation Force b) Acceleration at Passenger Seat c) Transmissibility

2.2.2 Conclusions about the Passive System from the Simulation:

The above results to the Typical Test Inputs show that the Passive System which is currently existing has been designed for a good driving comfort. However it is desired that the driving comfort be increased beyond the above state or if we need to switch from driving comfort to safety in real time i.e. while the vehicle is running it is impossible and hence for such cases where the demand from the Suspension System changes the need for an Active Suspension System becomes inevitable.

2.3. Simulink Mathematical Modelling of Passive System:

The Mathematical Model of any System simulated on Simulink is basically the representation of the governing equations in the form of blocks representing the constants and the mathematical or algebraic operations. Hence for the Mathematical Modelling the equations have to be set-up. Hence for the Mathematical Modelling of the Passive and Active Systems the first step is to formulate the governing equations using Newton's Second Law of Motion.

2.3.1 Mathematical Model of the Passive System:

Assuming a quarter car model with only two degrees of freedom (DOF), two equations have been formulated for both the DOF's from their respective Free Body Diagrams (FBD). The Quarter car model can be represented as shown in Figure 2.8, where the road disturbance is in the form of displacements; the other terms are as mentioned. The use of a two DOF model is justified mainly because it suffices our preliminary study and if the Suspension System works satisfactorily for this basic model then if necessary the model can be further sub-divided for further research.

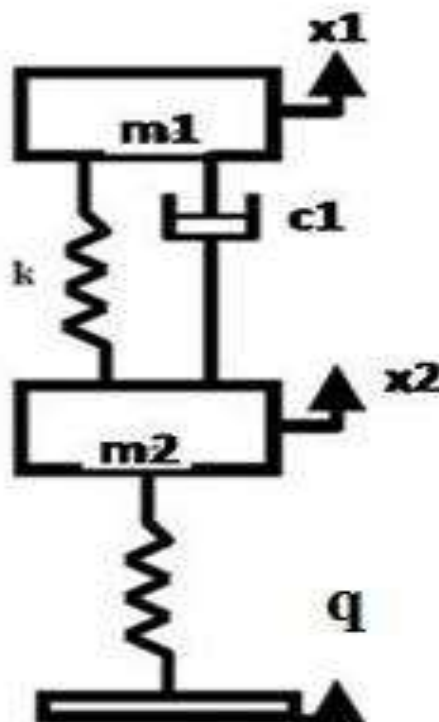


Fig 2.8: Quarter Model of Passive System [3]

c_1 : damping coefficient, N s/m

k_1 : spring stiffness, kN/m

k_2 : tire stiffness, kN/m

m_1 : quarter car sprung mass, kg

m_2 : unsprung mass, kg

x_1 : sprung mass vertical displacement, m

x_2 : unsprung mass vertical displacement, m

$x_1 - x_2$: suspension travel, m

$x_2 - q$: tire deflection, m

\ddot{x}_1 : sprung mass acceleration, m/s²

According to Newton's second law and free body diagram approach, the equations of motion for the system are written as [5],

For spring mass at passenger seat m_1 :

$$m_1 \ddot{x}_1 + c_1(\dot{x}_1 - \dot{x}_2) + k_1(x_1 - x_2) = 0 \quad (1)$$

For unsprung mass at wheel m_2 :

$$m_2 \ddot{x}_2 - c_1(\dot{x}_1 - \dot{x}_2) - k_1(x_1 - x_2) - k_2(q - x_2) = 0 \quad (2)$$

2.3.2 Simulink Block Diagram :

As mentioned in Section 2.1, the Mathematical Model when opened the window will look as below and when the Sub system Mathematical model of The Passive System is opened it will be as shown in Figure 2.9, which is purely a Simulink Block Diagram of the above Mathematical Equations. The Simulation is run and the results generated can be viewed by opening the respective Scope shown in Figure 2.10. The 'q' in Equation (2) is the Road disturbances and is simulated using the Signal Builder Block. The Typical Test Signals as used for the Physical Modelling are used in this Mathematical Model too. Also with the difference that these will be used as inputs which are displacements in meters and in the Physical Model they were used as inputs which were Acceleration or Unit Force in m/s².

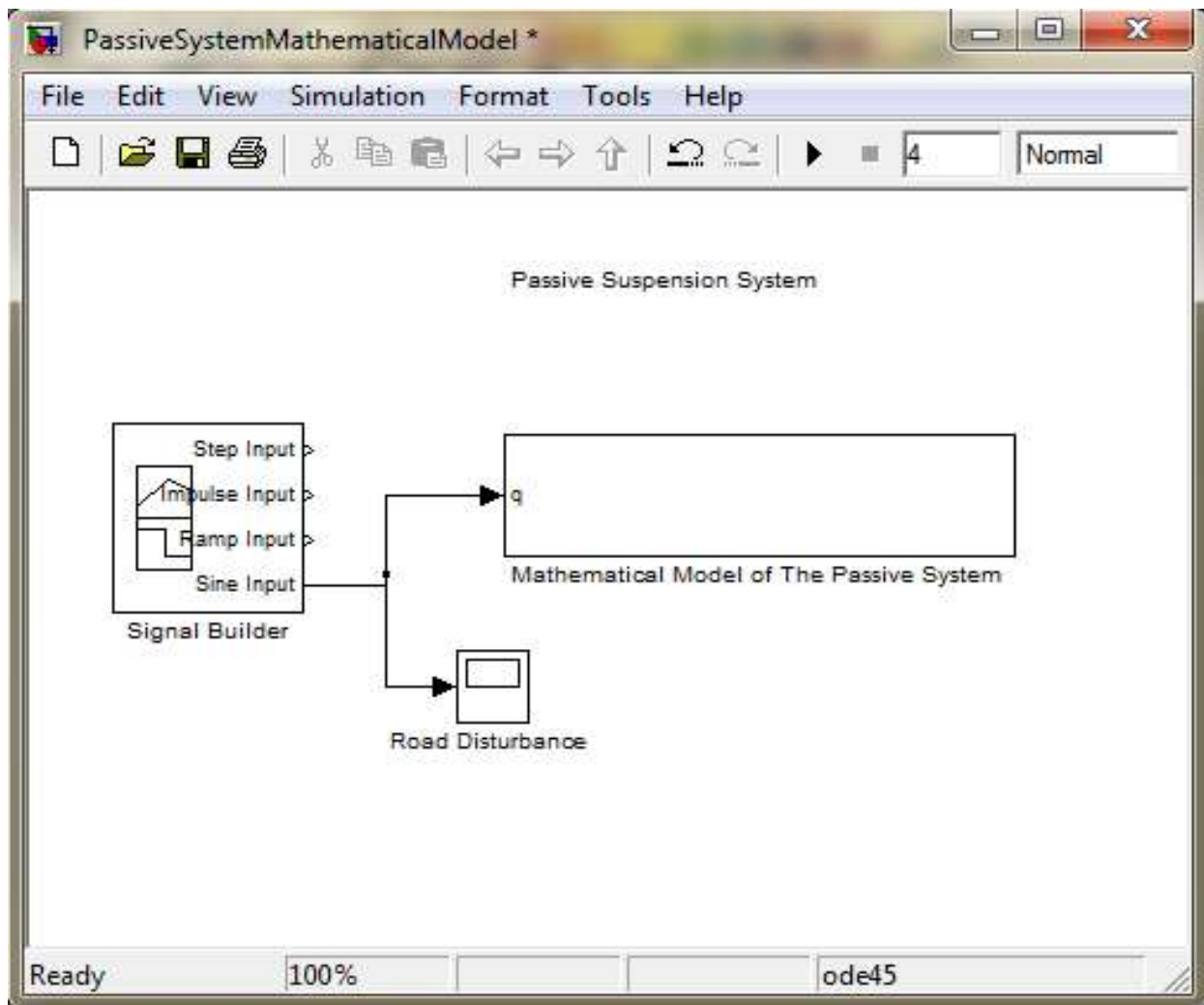


Fig 2.9: Simulink Model: Passive System (Mathematical)

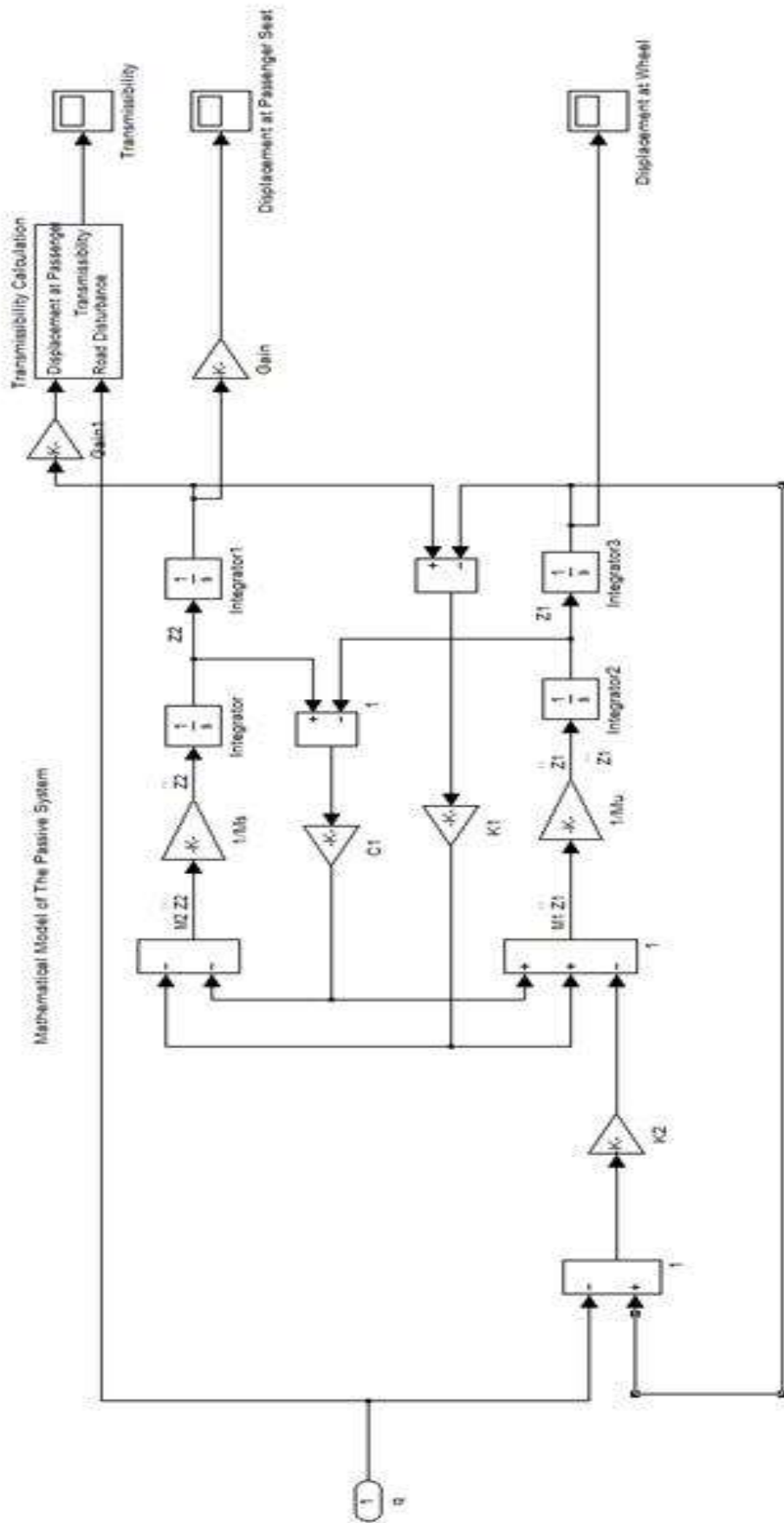


Fig 2.10: Simulink Sub-system : Passive System (Mathematical)

2.3.3 Results

a) Step Input

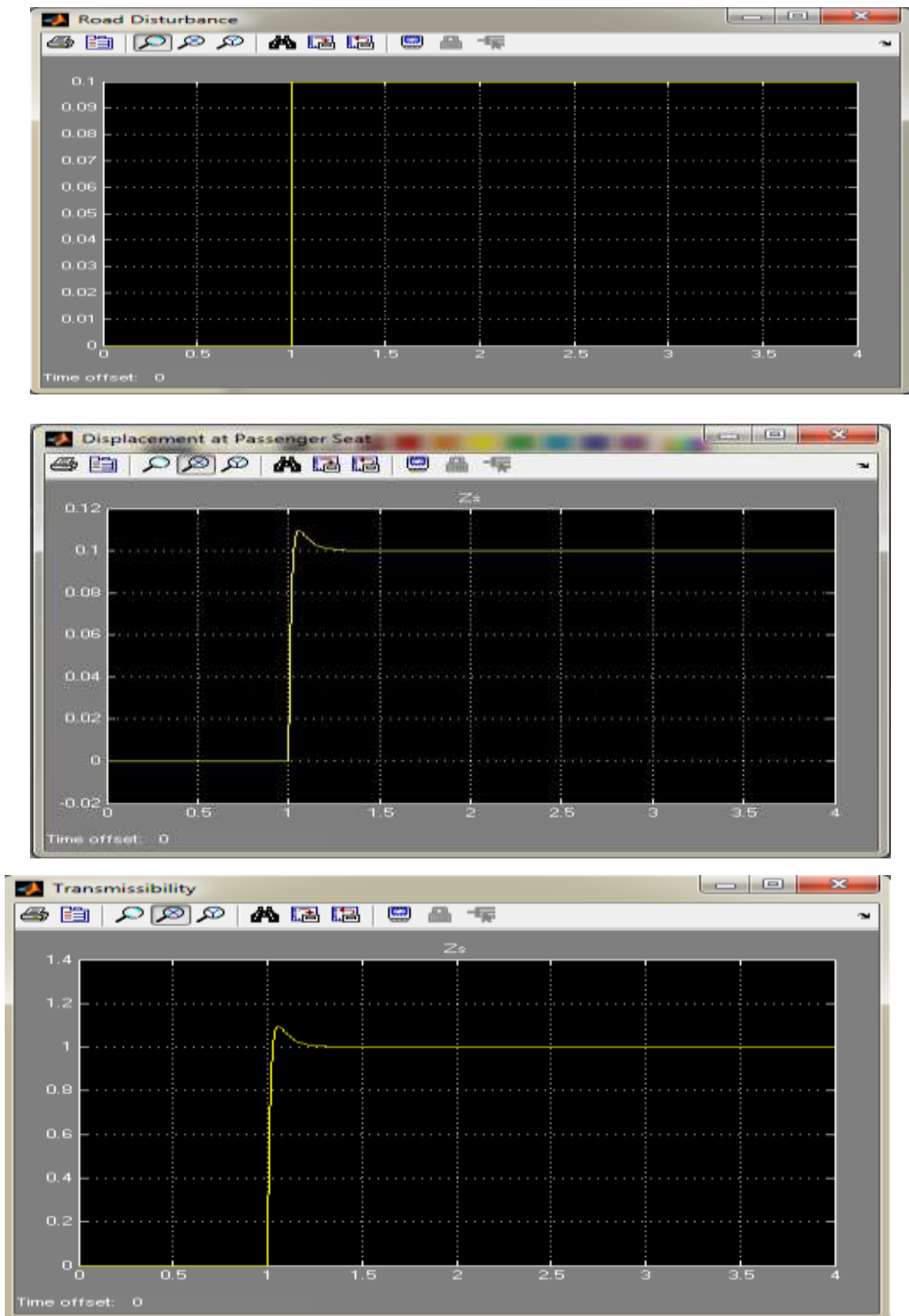


Fig 2.11: Step I/P a)Road Disturbance b)Displacement at Passenger Seat c)Transmissibility

b) Impulse Input

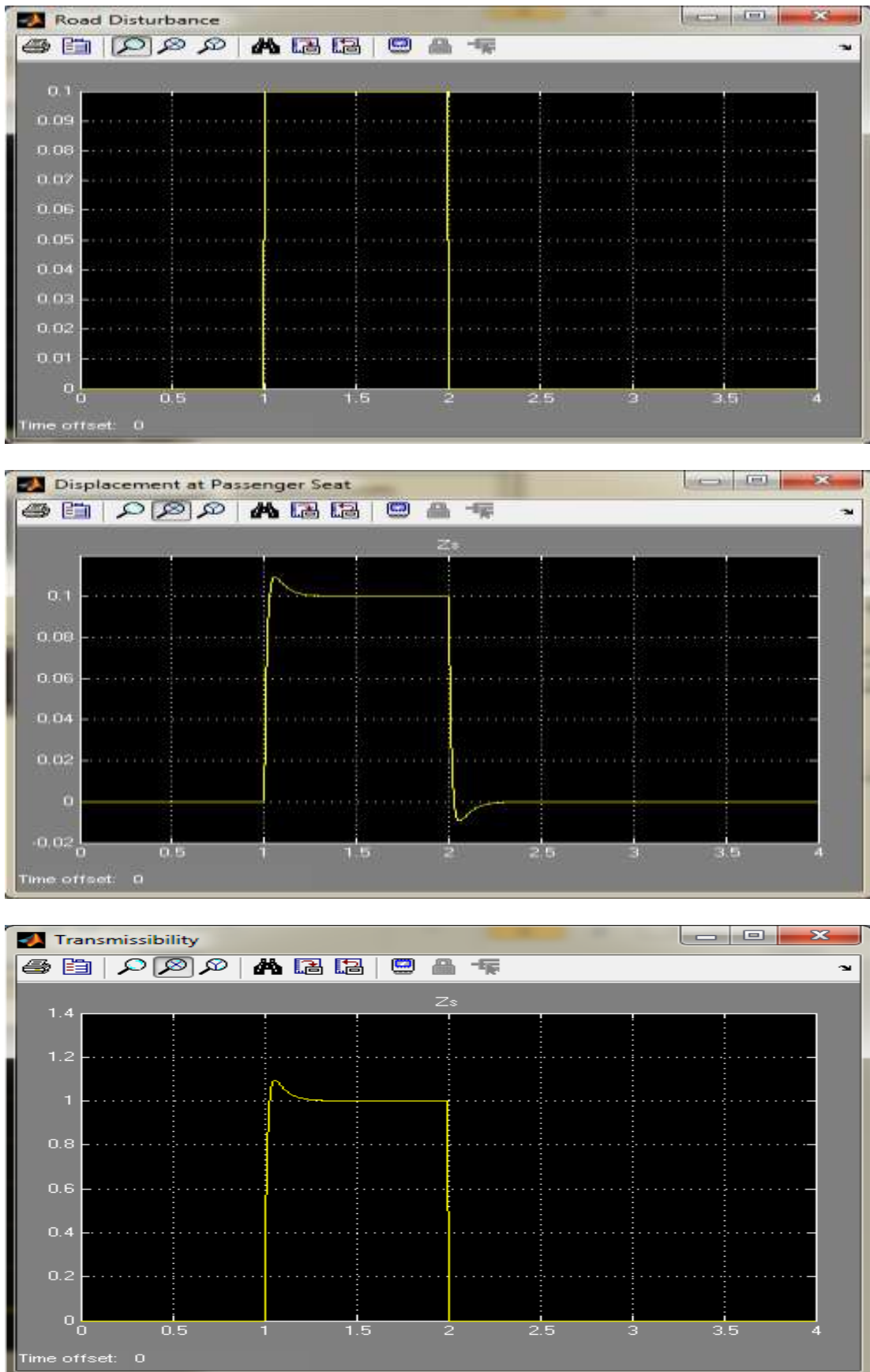


Fig 2.12: Impulse I/P a)Road Disturbance b)Displacement at Passenger Seat c)Transmissibility

c) Ramp Input

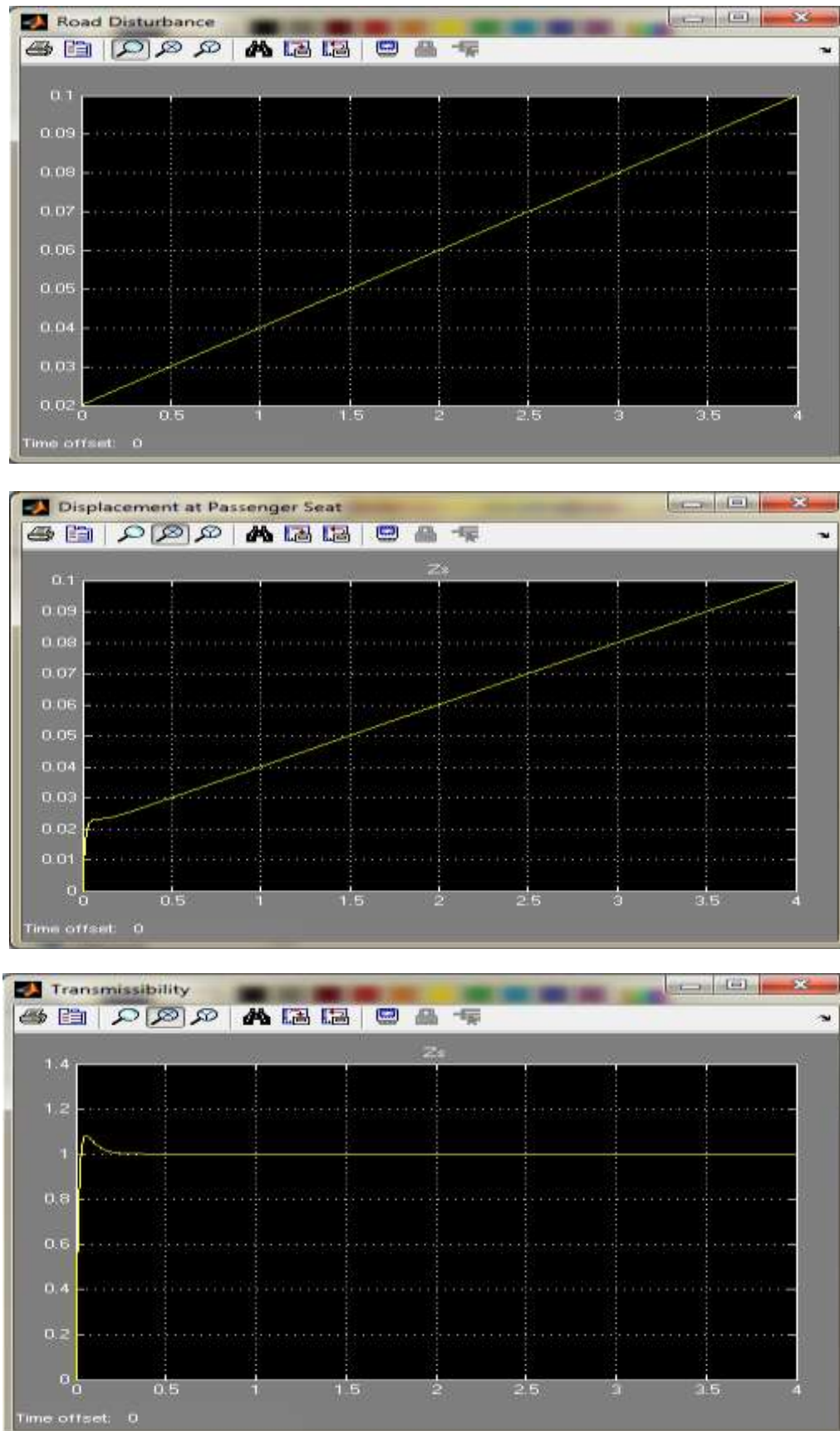


Fig 2.13: Ramp I/P a)Road Disturbance b)Displacement at Passenger Seat c)Transmissibility

d) Sine Input

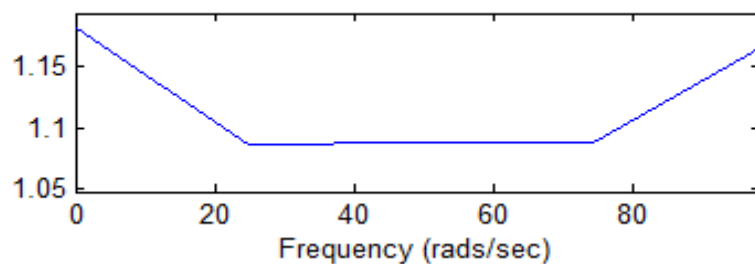
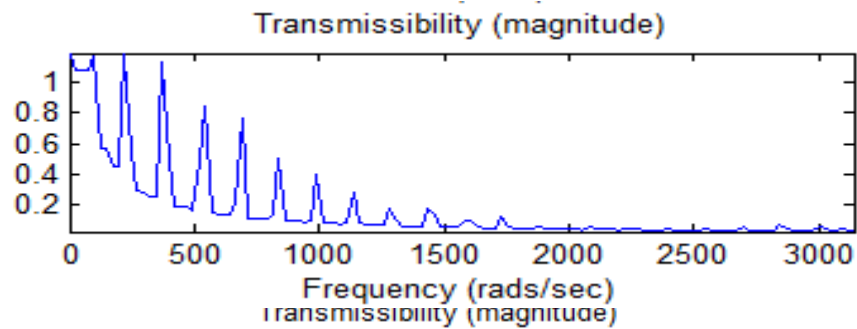
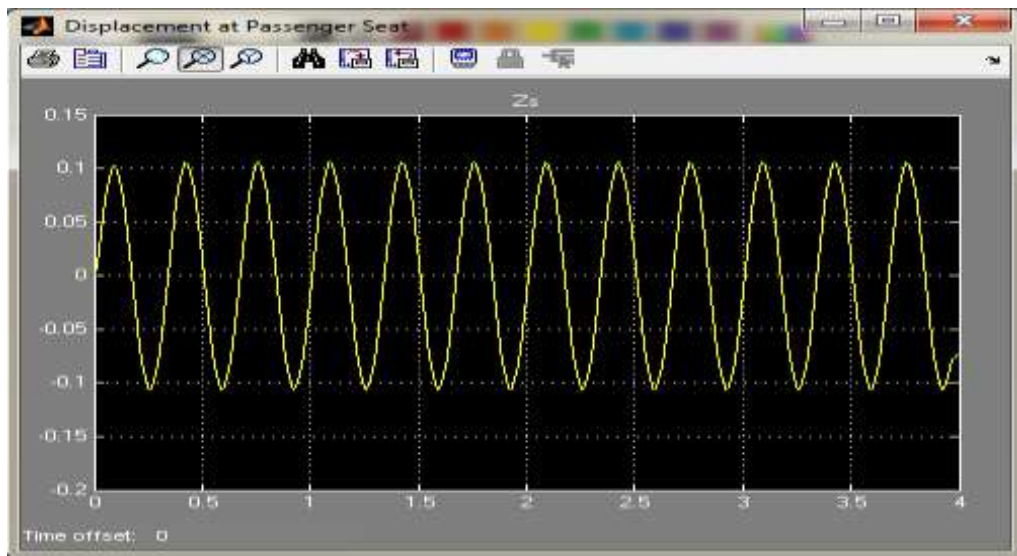
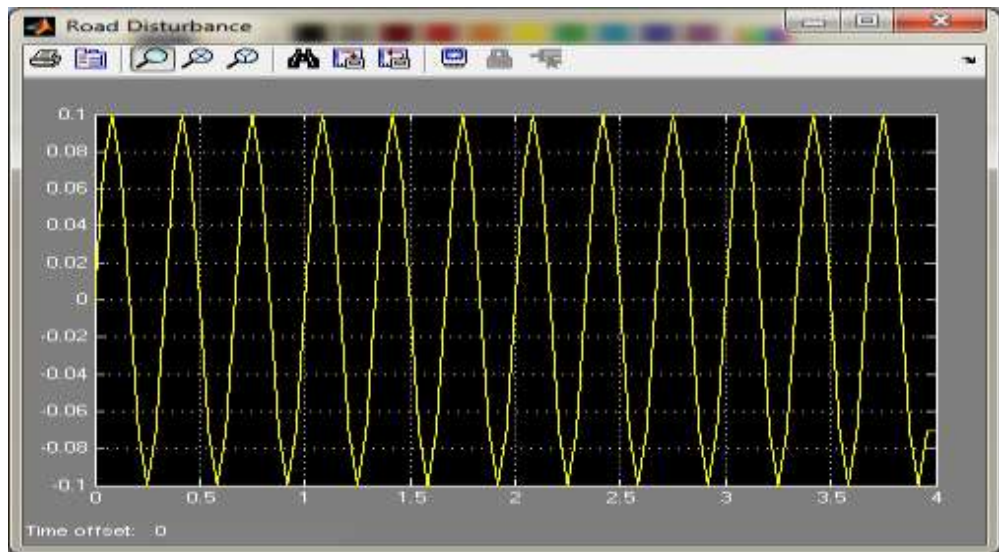


Fig 2.14: Sine I/P a)Road Disturbance b)Displacement at Passenger Seat c)Transmissibility

2.3.4 Conclusions about the Passive System from the above Simulations:

The Response of the Passive System and also the Transmissibility of the System to the Typical Test Signals shows that the Suspension system is not so hard that that the car will be thrown off the road due to Road Disturbances and also it is not so soft that it results in the car swinging dangerously so that the wheels and the road loose contact eventually. From the Transmissibility in the frequency domain it can be seen that the Passive System shows considerable peaks at lower frequencies and at higher frequency excitations the transmissibility is very low. However at no frequencies the Passive System will show any abrupt results and hence the above conclusions are drawn.

2.4. Simulink Mathematical Modelling of Active System:

Similar to the case of Section 2.3 about the Simulink Mathematical Modelling of Passive System, the governing equations for the Active System are formulated for building the Simulink Block Diagram Model of the Proposed Active System.

2.4.1 Mathematical Model of the Active System:

As shown in Figure 2.15 the active suspension includes an actuator attached in addition to the components of the Passive System, the actuator can supply active force, which is regulated by a control algorithm using data from sensors attached to the vehicle. The active suspension is named generally as the hydraulic or pneumatic one if the actuator is selected as the hydraulic or pneumatic actuator, and the active suspension is named generally as the electromagnetic one if the actuator is an electromagnetic actuator. Apart from applying the Newton's Second Law of Motion, equations need to be formulated using the Kirchoff's Voltage Law and other concepts of Electromagnetism for the actuator.

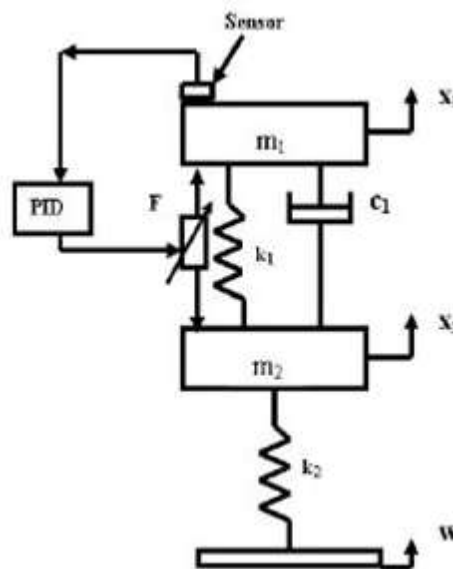


Fig 2.15: Quarter Car Model of Active Suspension [3]

Using the variables as defined in the Passive System in Section 2.3.1, here and also;

F_e : actuator force,

x : linear displacement of the piston = x_1 ,

L : inductance of the electromagnet coil,

R : resistance of the electromagnet coil,

I : electromagnet current, and

v_0 : supply voltage to the electromagnet coil.

For active suspension shown in Figure 2.15, using the Newton's second law of motion and free-body diagram concept, the following equations of motion are derived [5]

For test mass at passenger seat m_1 :

$$m_1 \ddot{x}_1 + c_1(\dot{x}_1 - \dot{x}_2) + k_1(x_1 - x_2) - F_e = 0 \quad (1)$$

For unsprung mass at wheel m2:

$$m_2 \ddot{x}_2 - c_1(\dot{x}_1 - \dot{x}_2) - k_1(x_1 - x_2) - k_2(q - x_2) + F_e = 0 \quad (2)$$

Governing Equations for the Actuator are as follows;

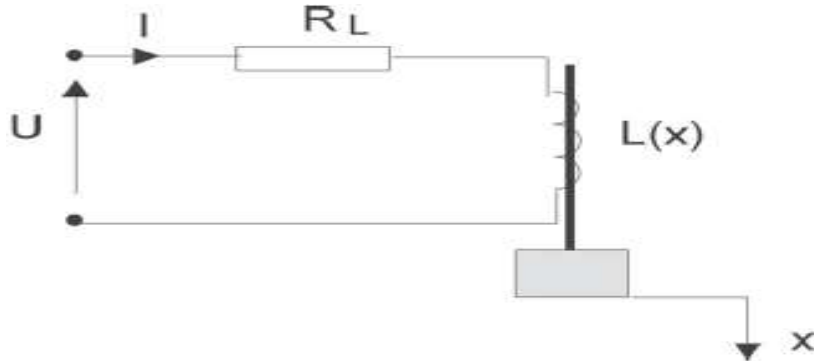


Fig2.16: Equivalent diagram of Electrical Circuit of the Solenoid Actuator [9]

Electrical Sub System is described by the 2nd Kirchoff's Voltage Law, assuming magnetically linear system [8]:

$$\begin{aligned} v_0 &= R \cdot i + \frac{d\lambda}{dt}; \\ \lambda &= L(x) \cdot i; \\ v_0 &= R \cdot i + L(x) \cdot \frac{di}{dt} + i \cdot \frac{dL(x)}{dx} \cdot \frac{dx}{dt}; \\ \frac{di}{dt} &= \frac{1}{L(x)} \cdot \left[v_0 - R \cdot i - i \cdot \frac{dL(x)}{dx} \cdot \frac{dx}{dt} \right]; \end{aligned} \quad (3)$$

Inductance could be derived knowing reluctance of the system[8]:

$$\begin{aligned} R_M &= \frac{g}{\mu_0 \cdot \pi \cdot x \cdot d} + \frac{g}{\mu_0 \cdot \pi \cdot a \cdot d} = \frac{g}{\mu_0 \cdot \pi \cdot a \cdot d} \cdot \left(\frac{a+x}{x} \right); \\ L(x) &= \frac{N^2}{R_M} = \frac{\mu_0 \cdot \pi \cdot a \cdot d \cdot N^2}{g} \cdot \left(\frac{x}{a+x} \right) = L' \cdot \left(\frac{x}{a+x} \right); \\ L' &= \frac{\mu_0 \cdot \pi \cdot a \cdot d \cdot N^2}{g}; \end{aligned} \quad (4)$$

The electromagnetic force could be derived knowing that the magnetic force is linear and that current is constant for a given operating point:

$$f_{fd} = \frac{dW_f}{dx} = \frac{i^2}{2} \cdot \frac{dL(x)}{dx} = \frac{i^2}{2} \cdot \frac{a \cdot L'}{(a+x)^2}; \quad (5)$$

2.4.2 Simulink Block Diagram:

The Simulink Block Diagram for the Active System also can be modified from the Passive System by introducing the Block Diagram of the Electromagnetic Actuator derived from the governing equations and it is coupled directly to the existing Passive System. The left hand side of Figure 2.18 represents the actuator equations or the Electrical and Magnetic sub system and the right hand side represents the Mechanical sub system. Now for controlling the input to this actuator, a PID Controller is introduced into our Simulink Block Diagram as shown in Figure2.17 and this PID Controller is tuned. The tuning has been explained in Section 2.4.3. Once the PID Controller has been tuned the Simulation is run for the Typical Test Signals and the results are generated and studied. A comparative study of the two systems is done in Chapter no 3.

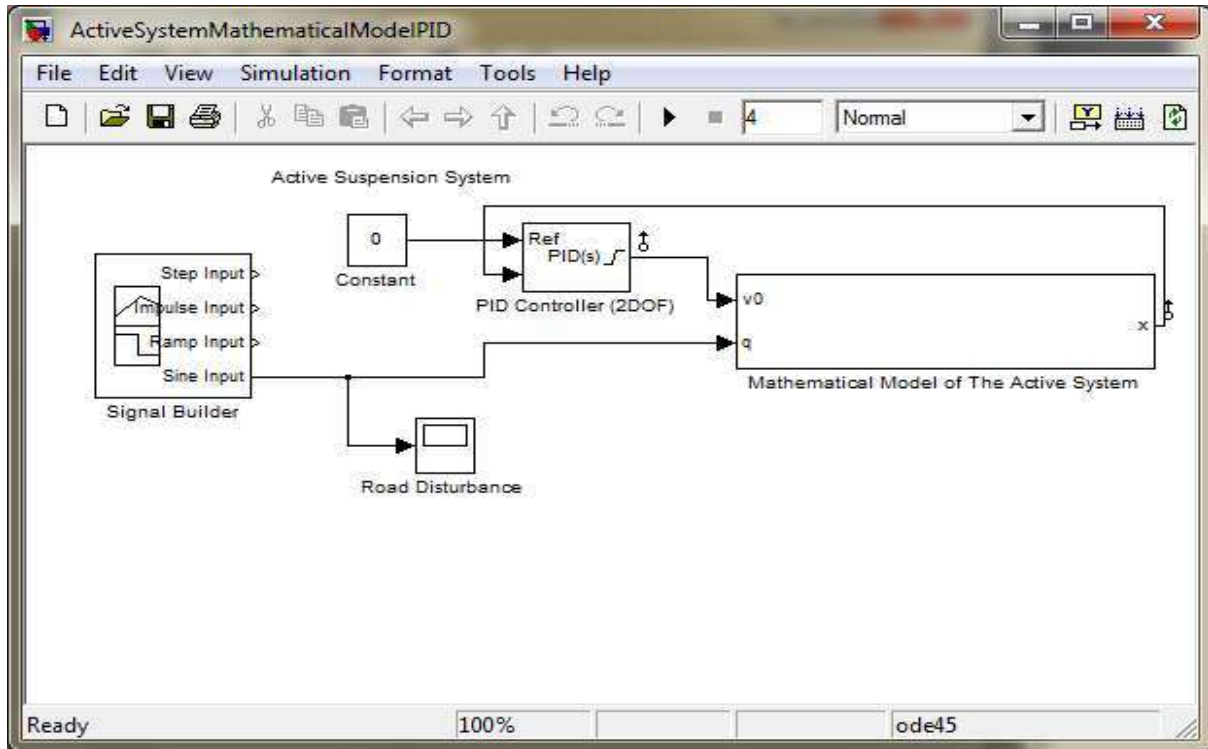


Fig 2.17: Simulink Model: Active System (Mathematical)

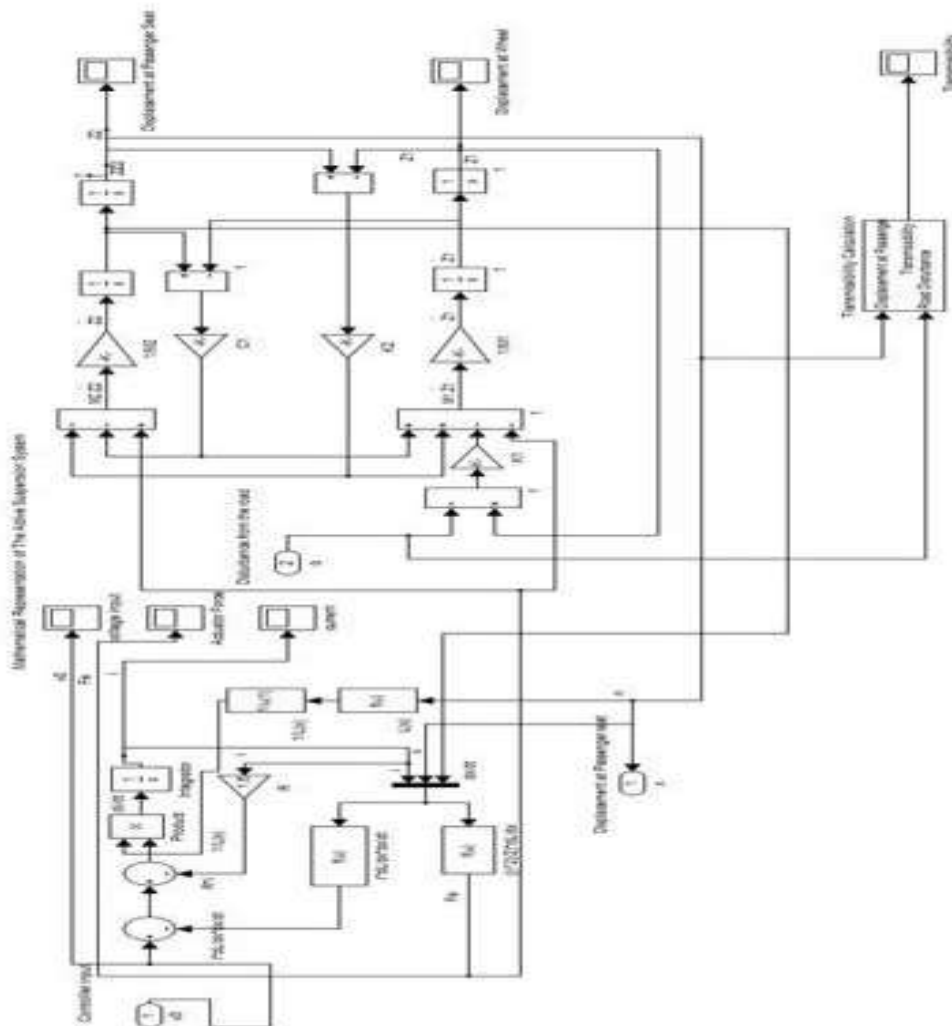


Fig 2.18: Simulink Sub-system :Active System (Mathematical)

2.4.3 PID Controller Tuning:

On opening the PID Controller Sub system from the Simulink Main Block Diagram as shown in Figure 2.17 a dialogue box appears in which if the values of the Proportional Constant (P), Integral Constant (I), Derivative Constant (D), Filter Co-efficient (N) and the Controller Constants (b and c) are already known for a tuned controller they can be directly fed or else the controller can be tuned online.

To tune the PID Controller the Manual Tuning Method shall be applied which is the most generalized method of tuning PID Controller. The transient-response specifications of the second order like the rise time, settling time, peak and maximum percent over shoot are configured to allow for a very few oscillations since the response of the system without oscillations cannot be tolerated. Figure 2.19 shows the response as well as the configured values of the tuned system and the originally untuned system used as a reference for tuning.

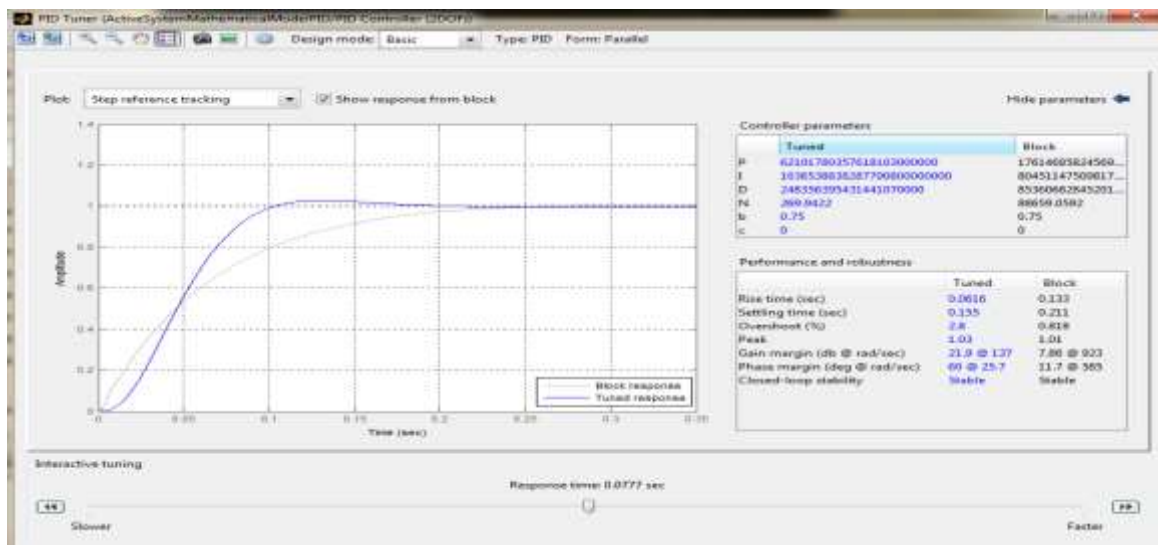


Fig 2.19: Tuning the PID Controller using the Transient Response Specifications

The Bode Plots are also to be used to verify the stability of the Control System in the frequency domain and also to get knowledge about the Gain Margin and the Phase Margin of the control system which helps in determining stability of the system. Figure 2.20 shows the Bode Plot of the Active System and also the Gain and Phase margins.

Figure 2.19 and figure 2.20 were thus used to tune the PID controller and get acceptable values of the Proportional Constant (P), Integral Constant (I), Derivative Constant (D), Filter Co-efficient (N) and the Controller Constants (b and c) and to ensure that the transient response is satisfactory.

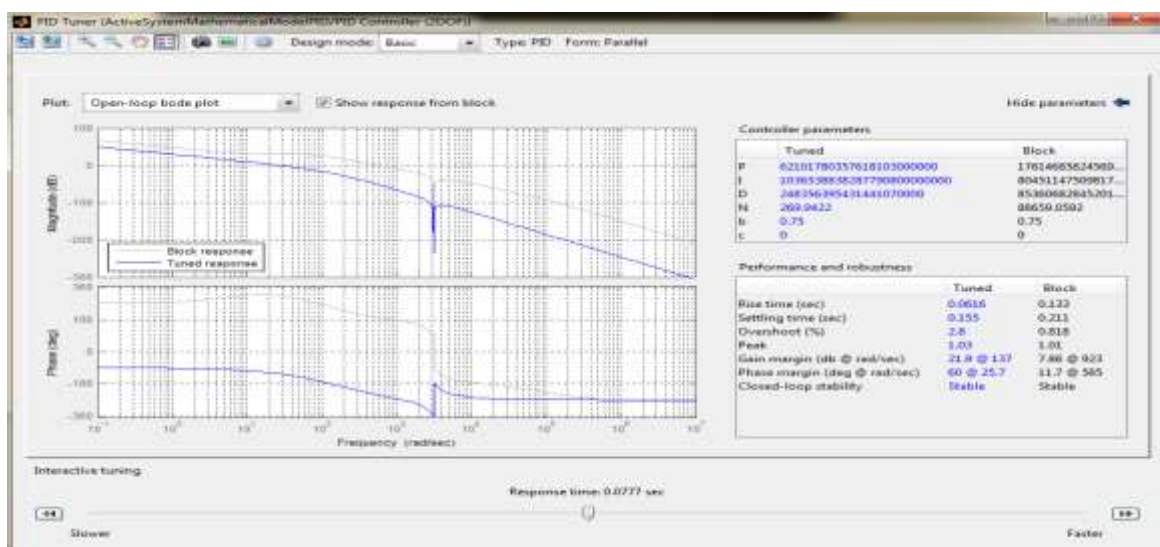


Fig 2.20: Bode Plot

2.4.4 Results

a) Step Input

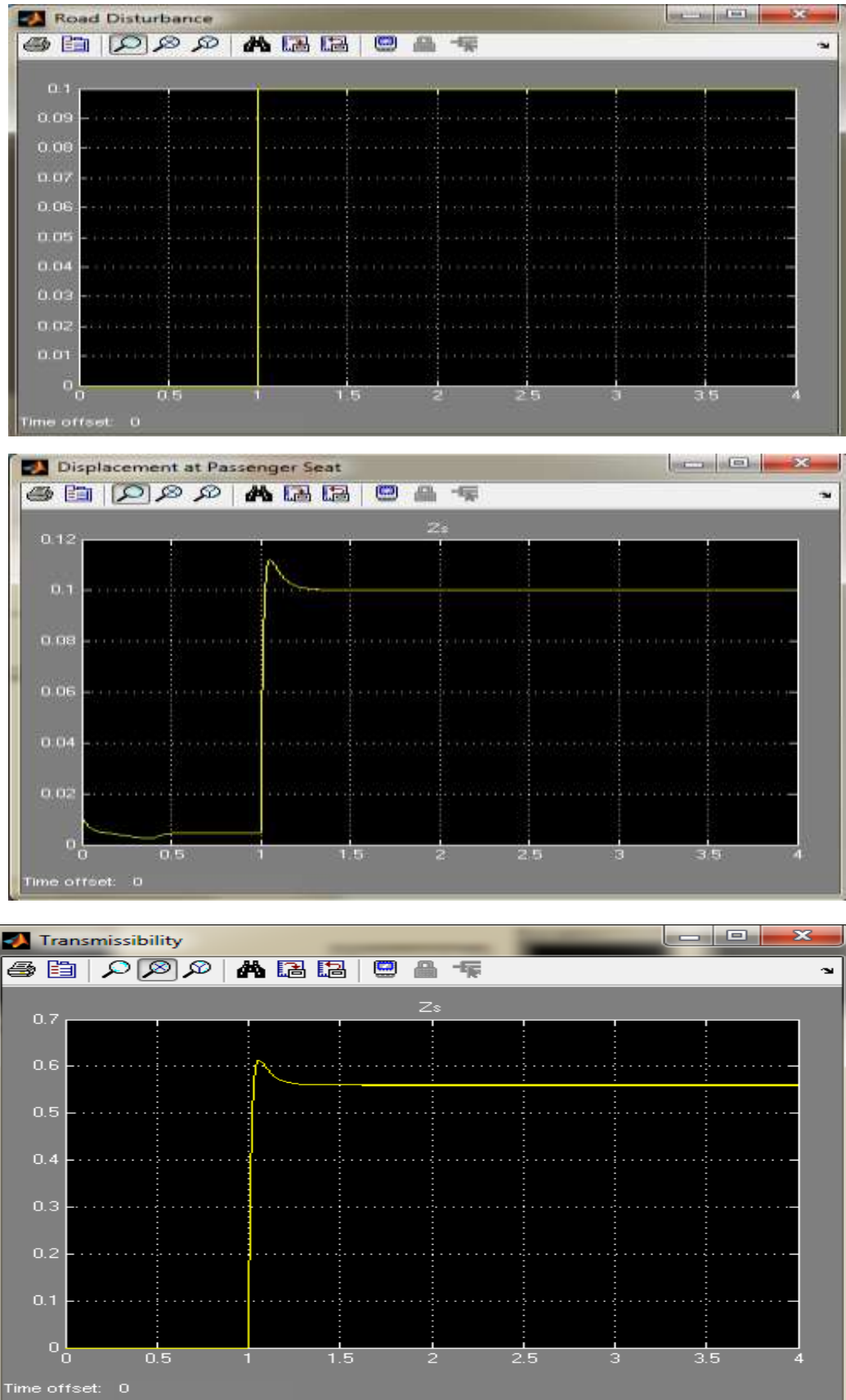


Fig 2.21: Step I/P a)Road Disturbance b)Displacement at Passenger Seat c)Transmissibility

b) Impulse Input

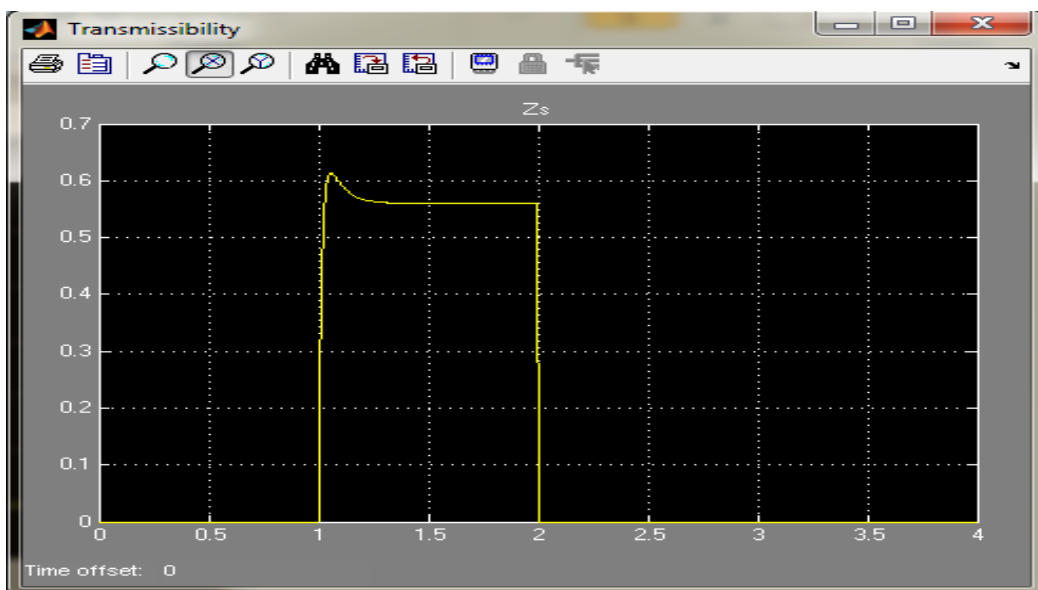
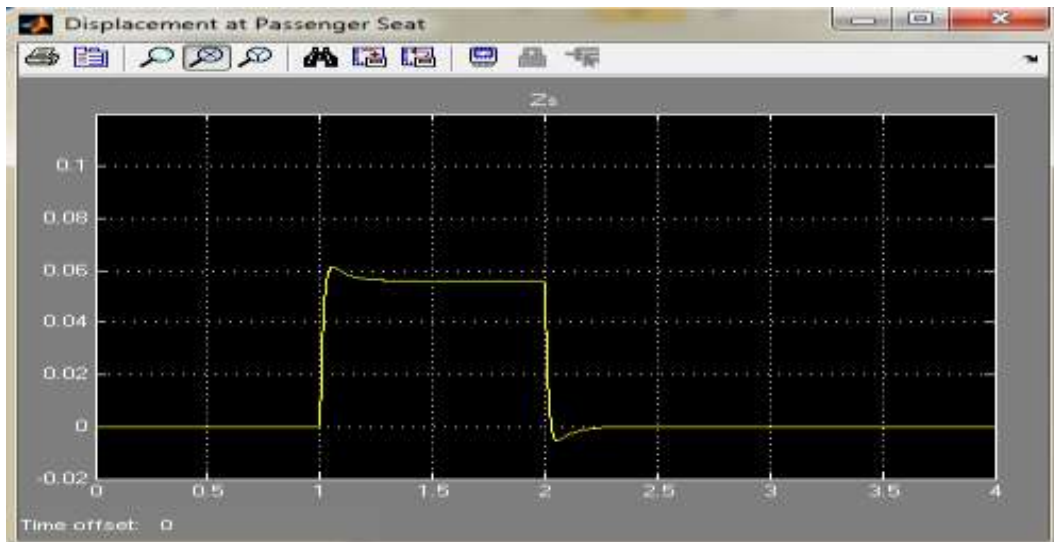
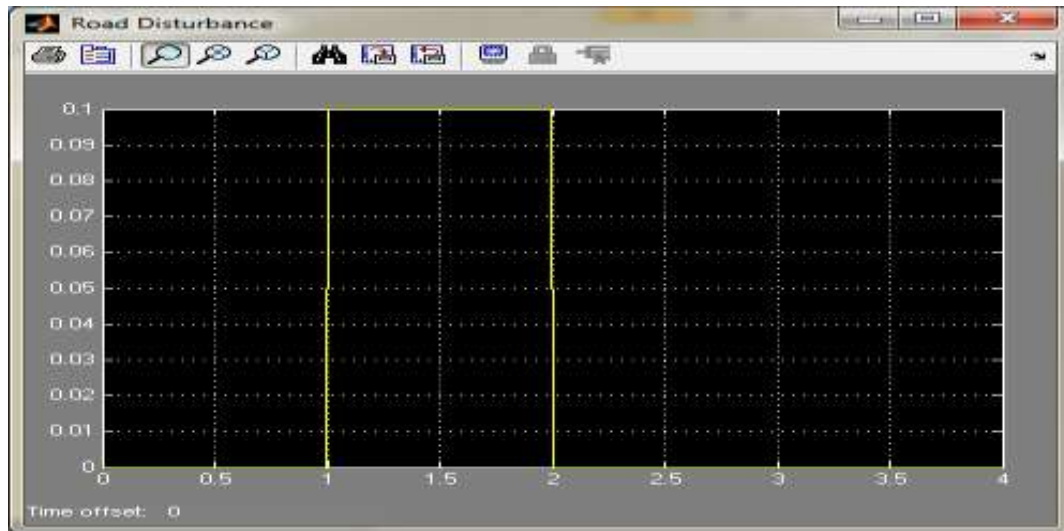


Fig 2.22: Impulse I/P a)Road Disturbance b)Displacement at Passenger Seat c)Transmissibility

c) Ramp Input

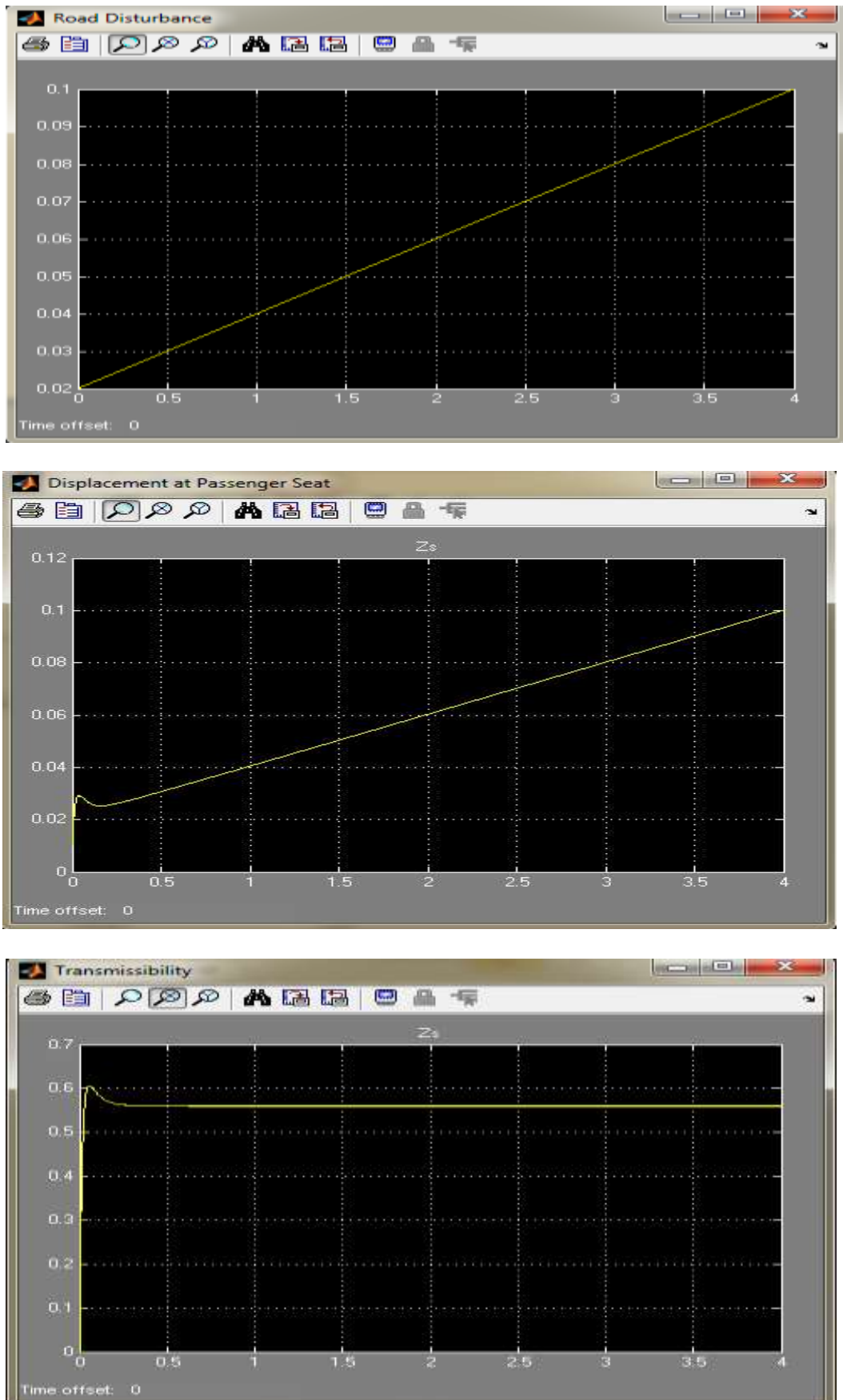


Fig 2.23: Ramp I/P a)Road Disturbance b)Displacement at Passenger Seat c)Transmissibility

d) Sine Input

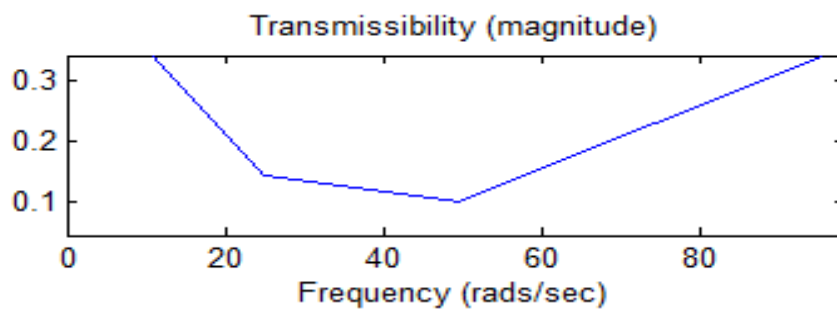
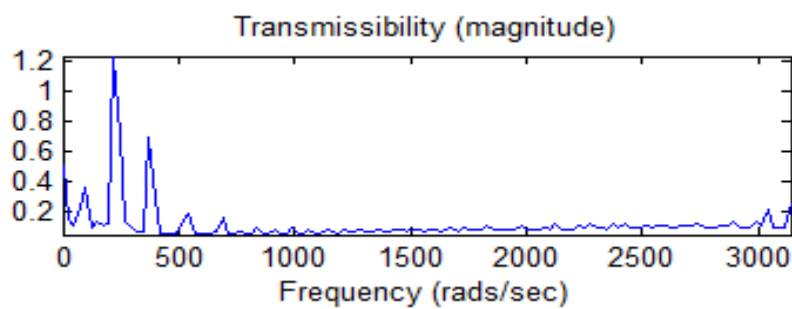
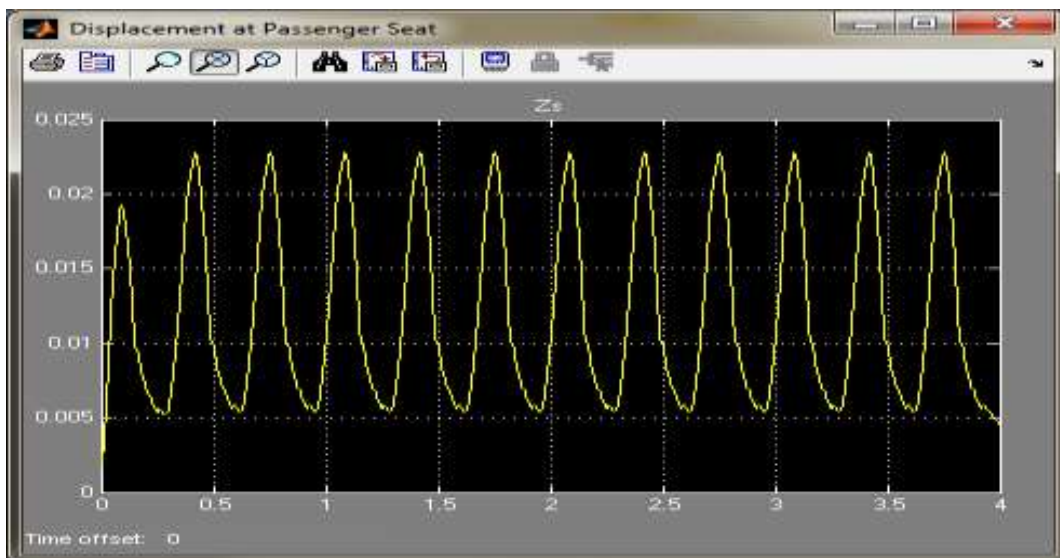
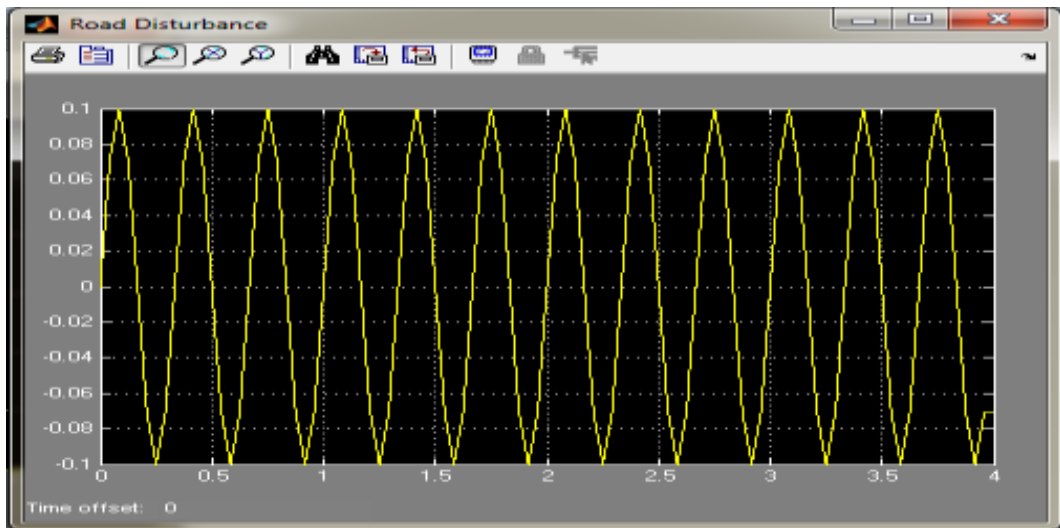


Fig 2.24: Sine I/P a)Road Disturbance b)Displacement at Passenger Seat c)Transmissibility

2.4.5 Conclusions about the Active System from the above Simulation:

The Response of the Active System and the transmissibility from the Simulations show that as compared to the Passive Suspension System; as a result of addition of the Electromagnetic Actuator System and its proper tuning using the PID Controller, considerably stronger suppression of vibrations can be achieved. Also from the transmissibility in the frequency domain it is observed that such a system allows considerable reduction of amplitude of the oscillation to be achieved, especially in high-frequency region. The Active System is also quite effective in the lower frequency ranges of excitation and for the other types of excitation and thus it can be used to achieve a very satisfactory and comfortable ride.

3. COMPARISON BETWEEN THE EXISTING PASSIVE SYSTEM AND THE PROPOSED ACTIVE SYSTEM BASED ON SIMULATIONS

The nature of the response of the Passive and Active Systems for the four Typical Test signals are pretty much similar, however differences in the Road Disturbances transmitted for both the cases is noteworthy and a comparison has been drawn between existing Passive System and the Proposed Active System based on the Transmissibility calculated by the Simulations. Main objective of the suspension system in this project is to isolate vehicle body from road irregularities in order to maximize passenger ride comfort and produce continuous road wheel contact, improving vehicle handling quality.

The response of the Passive and Active Systems show that both the systems are designed well enough to prevent being thrown off the road due to sudden disturbances and also that they do not oscillate extensively to make the wheels loose contact with the road. Thus continuous road wheel contact will be achieved considerably well by both the systems.

However when it comes to passenger ride comfort, for all the four Typical Test Signals, namely Step, Impulse, Ramp and Sinusoidal Road Disturbances; there is a marked difference in the isolation property of the Suspension Systems. The Passive Suspension system often transmits the entire Road disturbance to the passenger seat but very gently. Whereas the active suspension transmits only around 60% of the Road Disturbances and sometimes even much lesser Road Disturbances are transmitted to the Passenger seat and hence a great level of comfort of the ride is obtained in the case of the Active System.

It does become imperative to now actually fabricate the proposed Active System using the Electromagnetic Actuator and to validate the Simulation results to ensure that the proposed system in fact performs as proposed.

a) Step Input

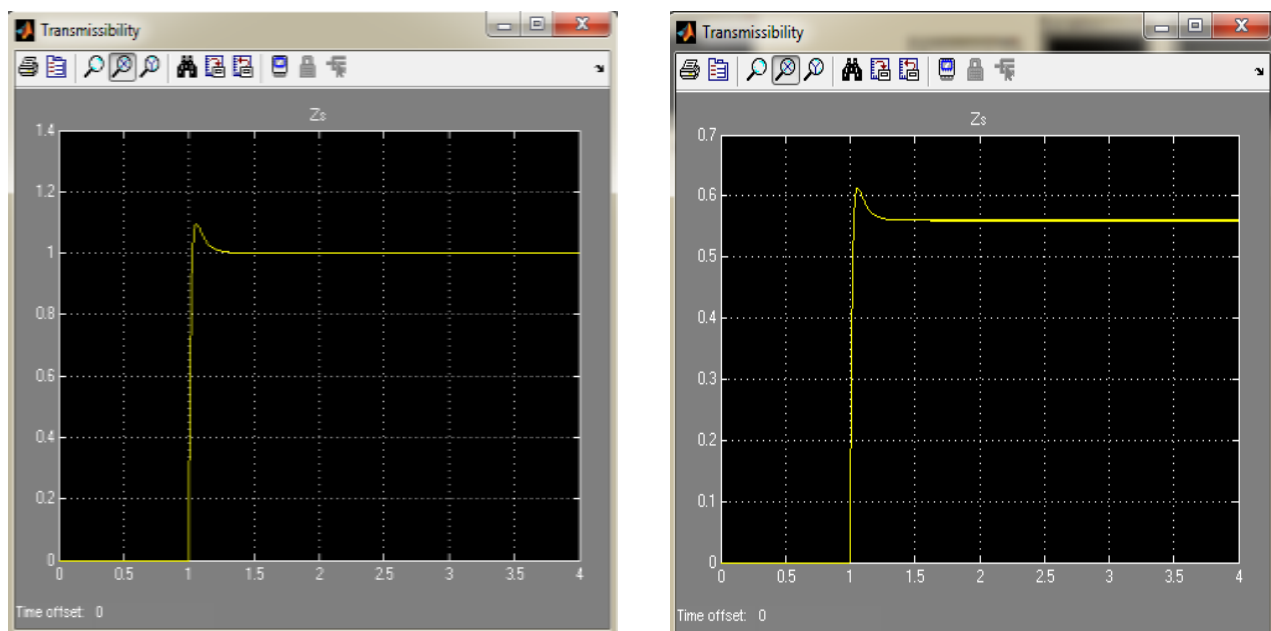


Fig 3.1 Comparison of Transmissibility for Passive and Active Systems respectively (Step Input)

b) Impulse Input

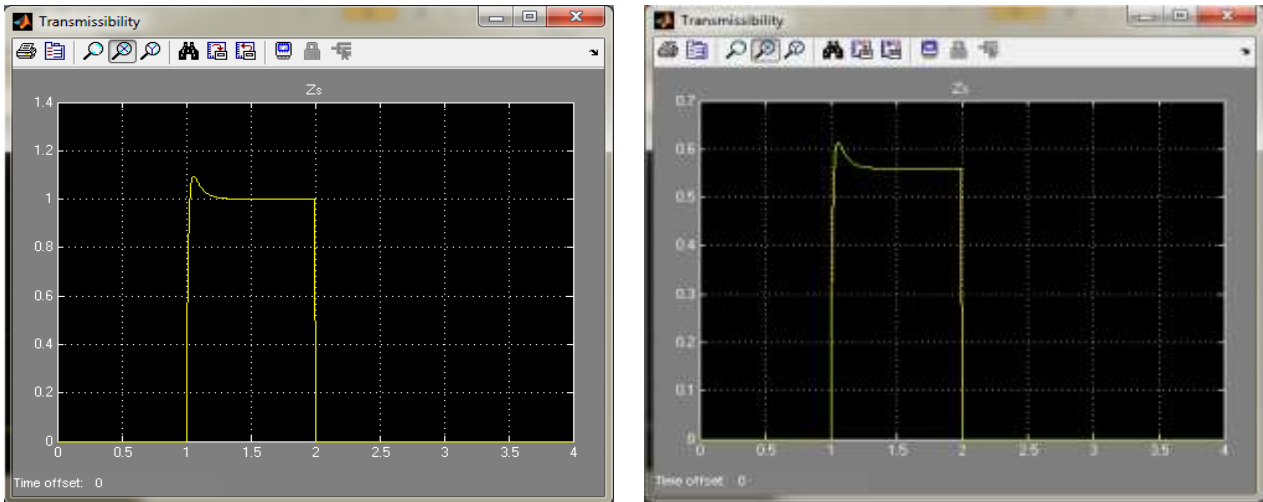


Fig 3.2 Comparison of Transmissibility for Passive and Active Systems respectively (Impulse Input)

c) Ramp Input

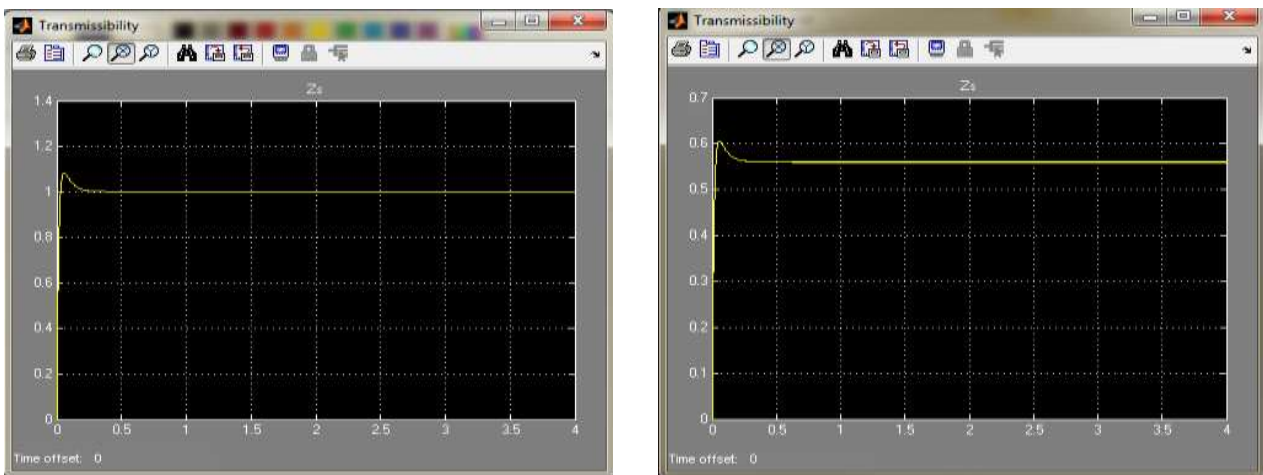


Fig 3.3 Comparison of Transmissibility for Passive and Active Systems respectively (Ramp Input)

d) Sine Input

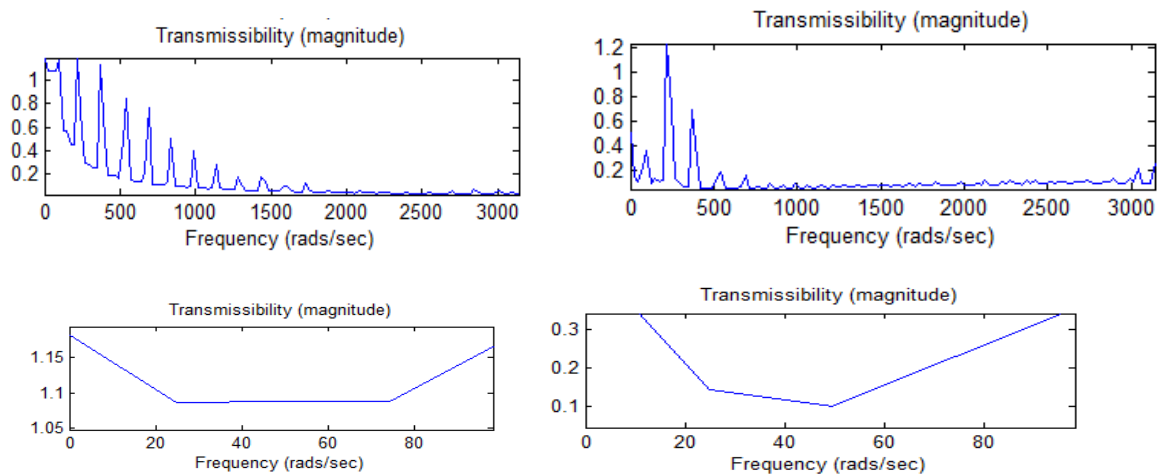


Fig 3.4 Comparison of Transmissibility for Passive and Active Systems respectively (Sine Input)

4. FABRICATION AND ASSEMBLY

4.1 Fabrication:

Now that the Simulations have shown that the coupling of the Electromagnetic Actuator with the Passive System does in fact help us to improve the Ride Comfort, validation of the Simulation results can only be done by actually fabricating the proposed Actuator and using the tuning of the PID Controller to drive it. Hence to be able to couple the electromagnetic actuator familiarity with the basic components present in the Passive System and their attributes like dimensions, etc. is required. Therefore the existing Passive Suspension System was dismantled and reverse engineered. Once a basic idea of this system is understood, the Electrical and Magnetic Sub System and the Mechanical Sub System are coupled together.

a) Disassembly of the Passive Suspension System: Shock Absorber:

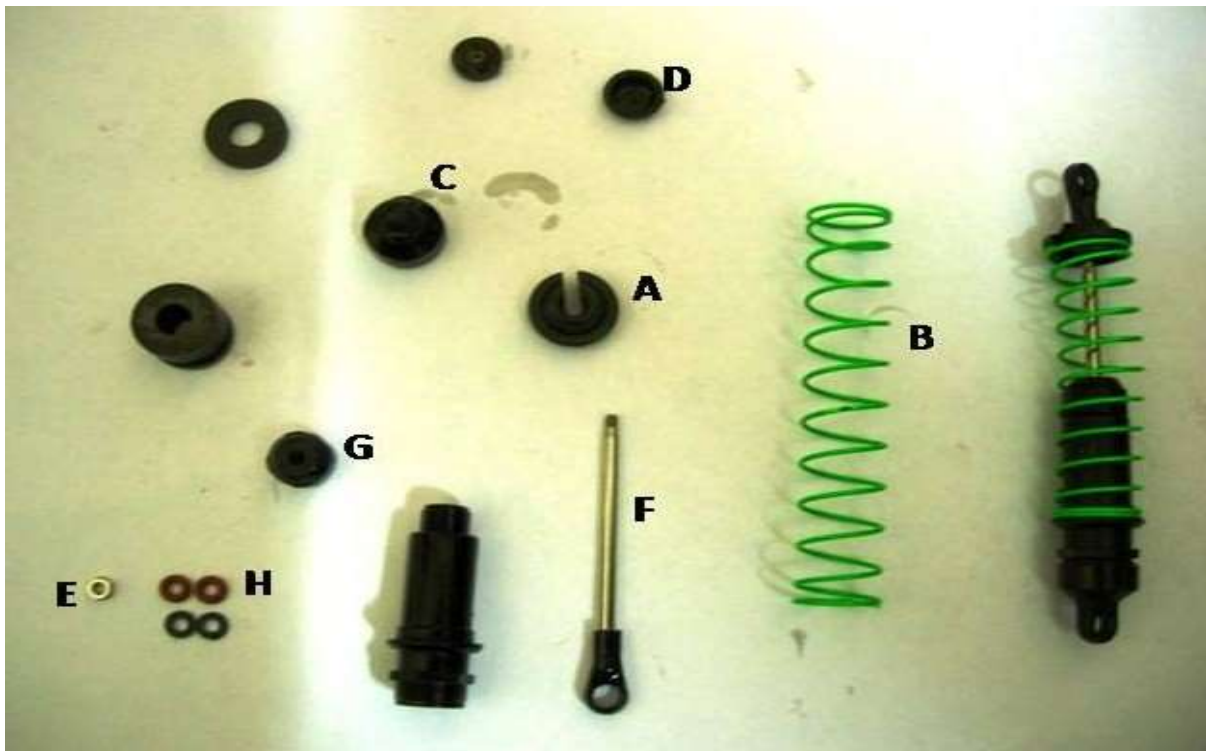


Fig 4.1: Components of Passive Suspension System: Shock Absorber

Figure 4.2 shows an assembled Shock Absorber of the existing Passive Suspension System. To disassemble the Shock Absorber the bottom head (A) is first pushed against the spring and pulled out, due to this the spring (B) becomes uncompressed and can be removed easily. Now the upper head (C) is unscrewed to reveal the upper head of the cylinder (D). The upper head of the cylinder is removed and the damping oil is carefully drained out. The nut (E) at the head of the piston (F) is removed and the piston is removed. The lower head of the cylinder (G) is then unscrewed and the washers (H) are removed and cleaned.

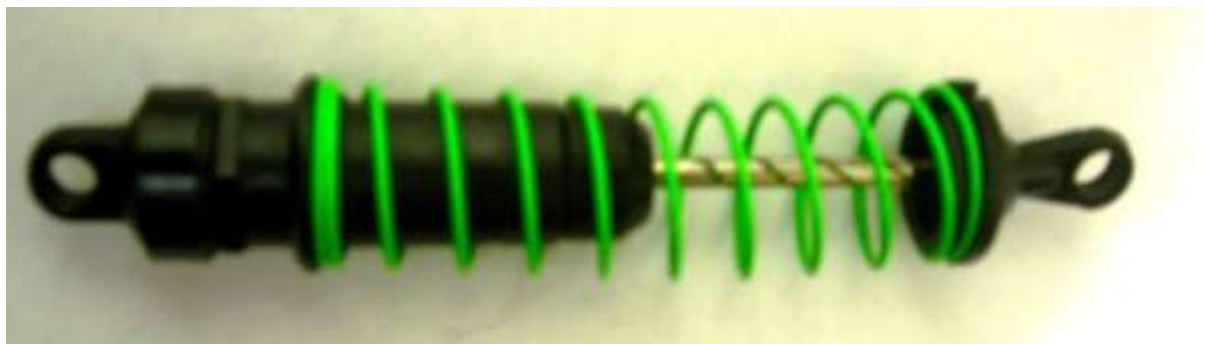


Fig 4.2: Assembled Passive Suspension System

b) Fabrication and Assembly of the Active Suspension System



Fig 4.3: Components of Active Suspension System

A solenoid of 25 turns is inserted into the cylinder and its end pulled out from two holes made at the side-top and side-bottom of the cylinder to get the modified cylinder assembly (A). Now the damping oil is carefully filled in the cylinder after sealing the lower head using the washers and all the other parts are assembled in the reverse order of their disassembly. Thus if all these parts have been assembled properly the shock absorber for the Active System will look like the one in Figure 4.4. The ends of the wire coming out from the cylinder are connected to a RS 232 Serial Port using the connecting wires whose resistance is measured for calculations.

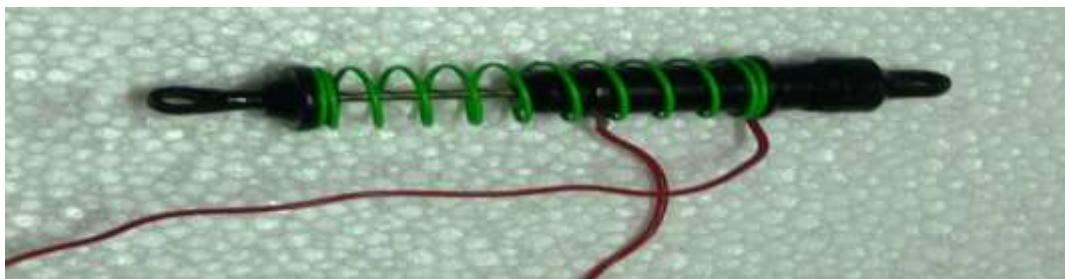


Fig 4.4: Assembled Active Suspension System



Fig 4.5: Passive and Active System

5. EXPERIMENTAL SETUP AND VALIDATION RESULTS

5.1: Experimental Setup:

A brief explanation of the instrumentation and the other laboratory apparatus has been presented to gain familiarity with the experimental set-up, before the experimental validation of the Simulation.

5.1.1 Excitation Source:



Fig 5.1: Source of Excitation

While validation can be done using all the four typical Test Signals that have been used for simulations, verifying for a sinusoidal input is fairly sufficient and that it is very easy and economically done. An unbalanced fan was used to generate the sinusoidal displacements at the base of the Systems. Figure 5.1 shows the source of excitation used for simulating the sinusoidal road disturbances. Although an unbalanced also produces harmonic excitations at frequencies other than the primary component, however the contribution of the primary component of vibration is so large that effect of the other contributions can be neglected.

5.1.2 Accelerometer:

The accelerometer that was available in the laboratory and compatible with the NI-DAQ is the one shown in Figure 5.2. It is a piezoelectric type accelerometer with voltage as its measuring medium. Ideally two accelerometers are best suited for the application however only one could be procured and hence once the base excitation reading was taken using the accelerometer and then the acceleration at the passenger seat was taken for both the cases of Active and Passive Systems



Fig5.2: Accelerometer

5.1.3 Data Acquisition System:

The National Instruments product (compact) cDAQ -9174 with the module NI-9234 for measuring accelerations was used for the Data Acquisition as shown in Figure 5.3.a and Figure 5.3.b The cable from the Accelerometer was directly connected to the port a1 as shown in Figure 5.3.a



Fig5.3.b: NI-DAQ Accelerometer Connection Port

5.1.4 USB to RS-232 Serial Port Communication Connector



Fig5.4: USB to RS-232 Converter

Since Laptops do not have Serial Ports for communication and Serial Ports are one of the easiest ways of communicating with other devices the cable shown in Figure 5.4 was used to give an output signal from the Laptop to the Actuator.

5.1. Setting up the Data Acquisition System:

The accelerometer was connected to the NI-DAQ and the NI-DAQ was then connected to the Laptop using the USB port. To ensure that the module and its ports are connected properly and that the NI-DAQ is reading the values from the accelerometer the National Instruments Measurements and Automation Explorer (MAX) is opened and the NI-DAQ was set-up as shown in Figure 5.5. Under My System/ Devices and Interfaces the cDAQ can be observed and using an analog accelerometer test panel the cDAQ module NI-9234 was configured and tested to ensure that it is responding to the signals from the accelerometer. Parameters such as the Channel name, the Max and Min values of the input, and input configuration are generally configured by MAX automatically, however if the accelerometer reading shown on the graph of the test panel is incorrect-this is verified using a known test signal from a function generator. However for our experiments the NI-DAQ was configured well as seen in the Figure5.5. The sampling rate is configured as per the Nyquist Sampling Theorem.

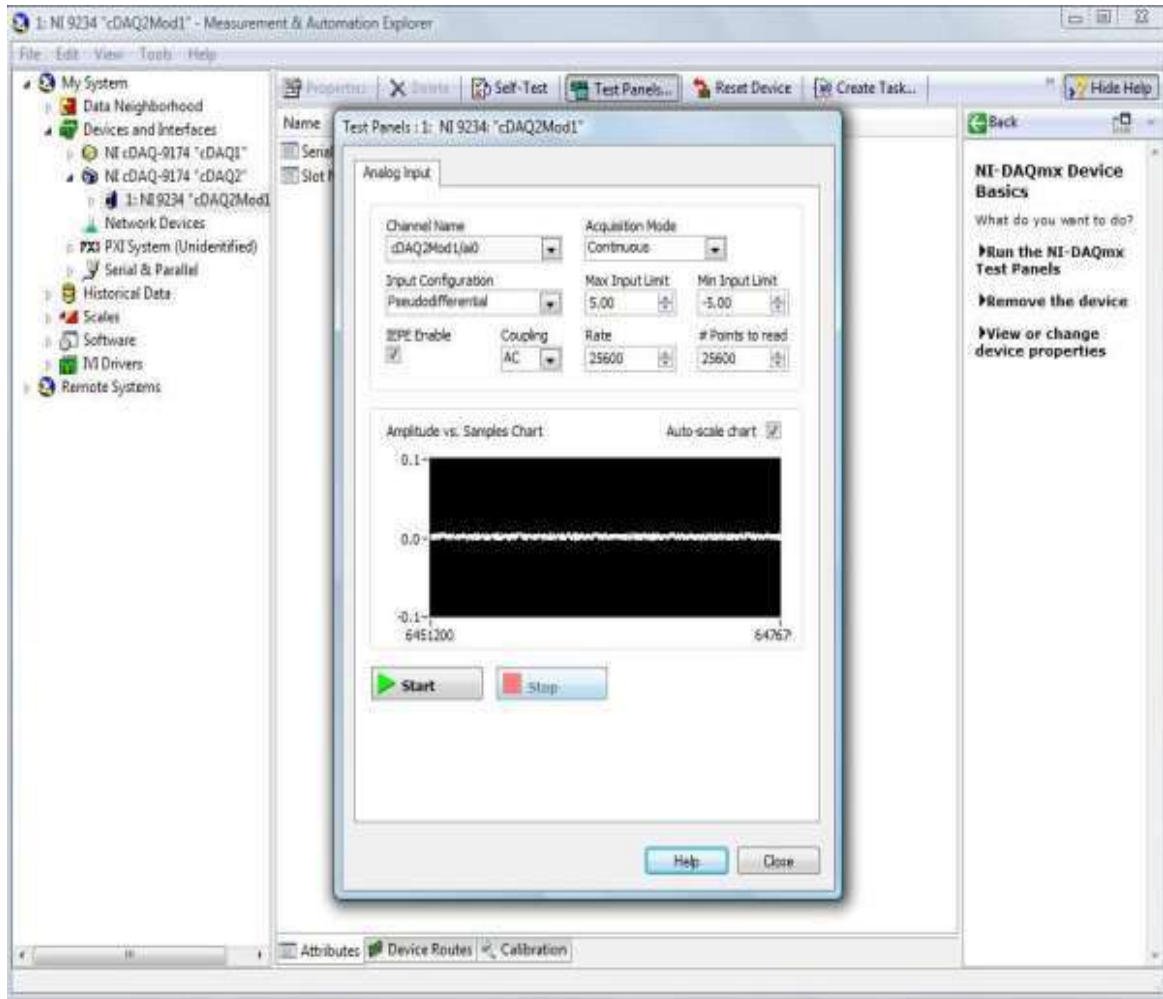


Fig 5.5: Setting up the cDAQ

5.2. Passive Suspension System Setup and Testing:



Fig 5.6.a: Road Disturbance Measurement



Fig 5.6.b: Passenger Seat Measurement

Now that all the instrumentation for the Testing was set-up, LabVIEW was used for acquiring the data, processing the data and storing the data.

However before beginning with the testing the entire Passive Suspension System was set-up as shown in Figure 5.6.a and Figure 5.6.b. and the Set-up was help firmly on the exciting base created for simulating the sinusoidal input. Now due to availability of only a single accelerometer, the input acceleration at the excitation source is taken first and follow the process below. Similarly the accelerometer is then kept at the mass representing the passenger seat mass and the steps below are repeated.

The 'Write measured signal to file.vi' is opened and its Front Panel is as shown in Figure 5.7.a and the Block Diagram is as shown in Figure 5.7.b. For all the Virtual Instrumentation (VI) files developed on LabVIEW the Front Panel is always used only for Control and Indication of variables and parameters so that any input from the user can be directly taken from the Front Panel and the method of taking the input is well organised and easy to understand for the end-user. The Block Diagram is always the window where the graphical coding takes place and thus it contains the entire code algorithm and the blocks used to implement it.

As is seen in the Figure5.7.b the 'DAQ Assistant' Express VI is used to directly read the acceleration data from the NI-DAQ into our present VI and then it is written to the file specified by the user on the Front Panel. The measurement data is also displayed on the front panel while it is being acquired and the write to spread sheet block on the block diagram then writes it to the specified file in .csv format.

Now this written data is filtered using a low pass 3rd order Chebyshev filter to remove the noise content from the measured signal using 'Filter Signal.vi'. From Figure5.8.b the block diagram shows the Filter Express VI which was used to Digitally filter the signal. The filtered and the unfiltered measurement data can also be seen in Figure 5.8.a; that is, the Front Panel of the VI.

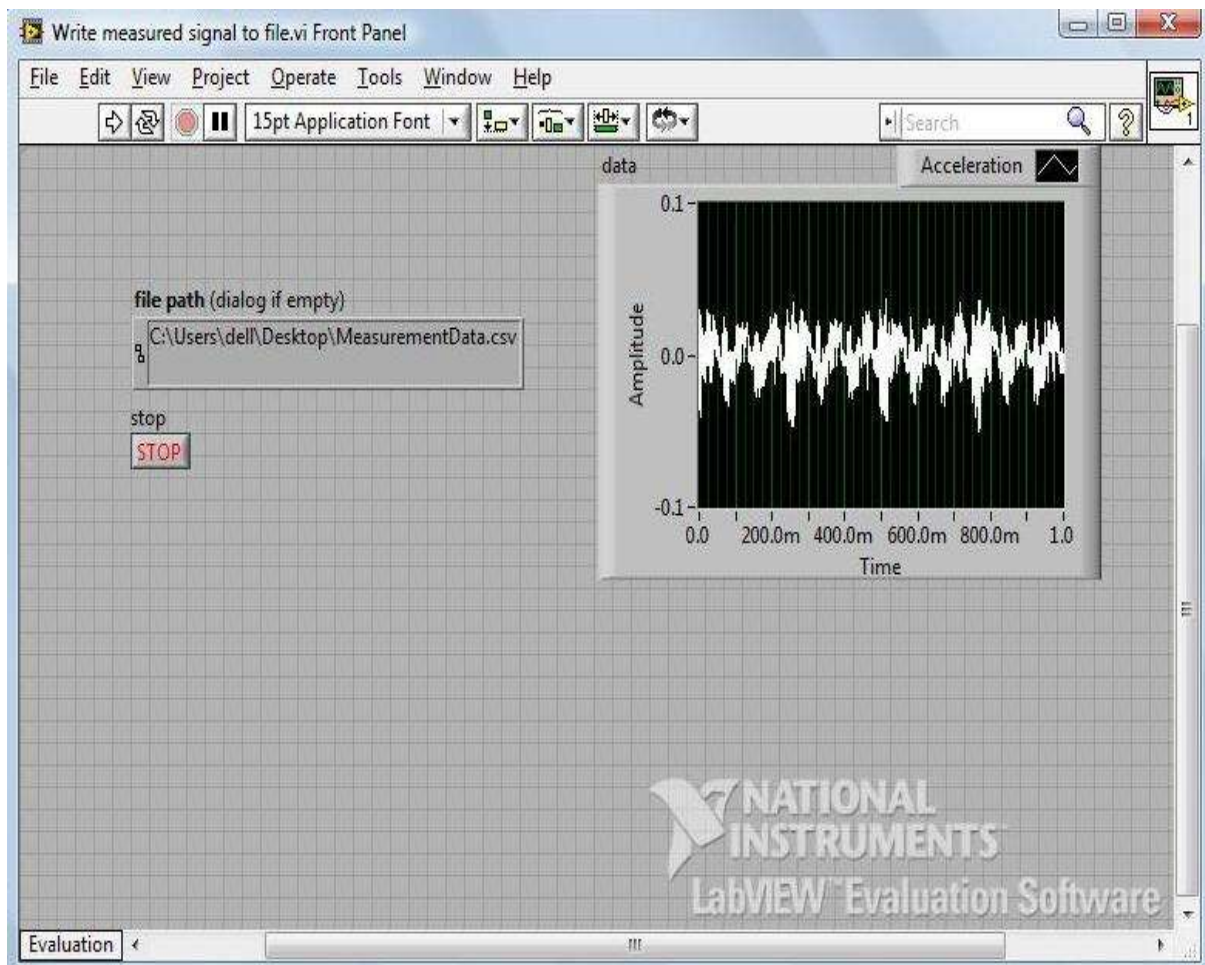


Fig 5.7.a: Write measured signal to file.vi (Front Panel)

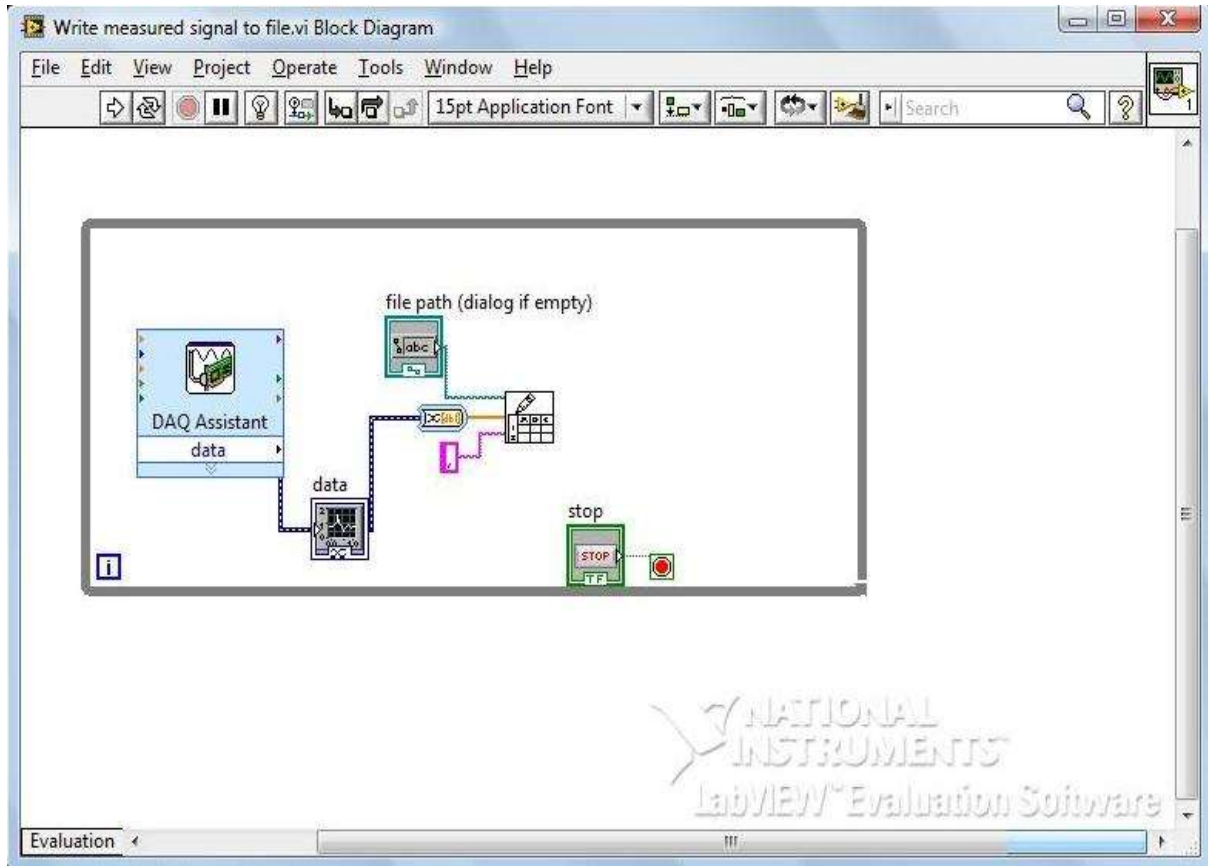


Fig 5.7.b: Write measured signal to file.vi (Block Diagram)

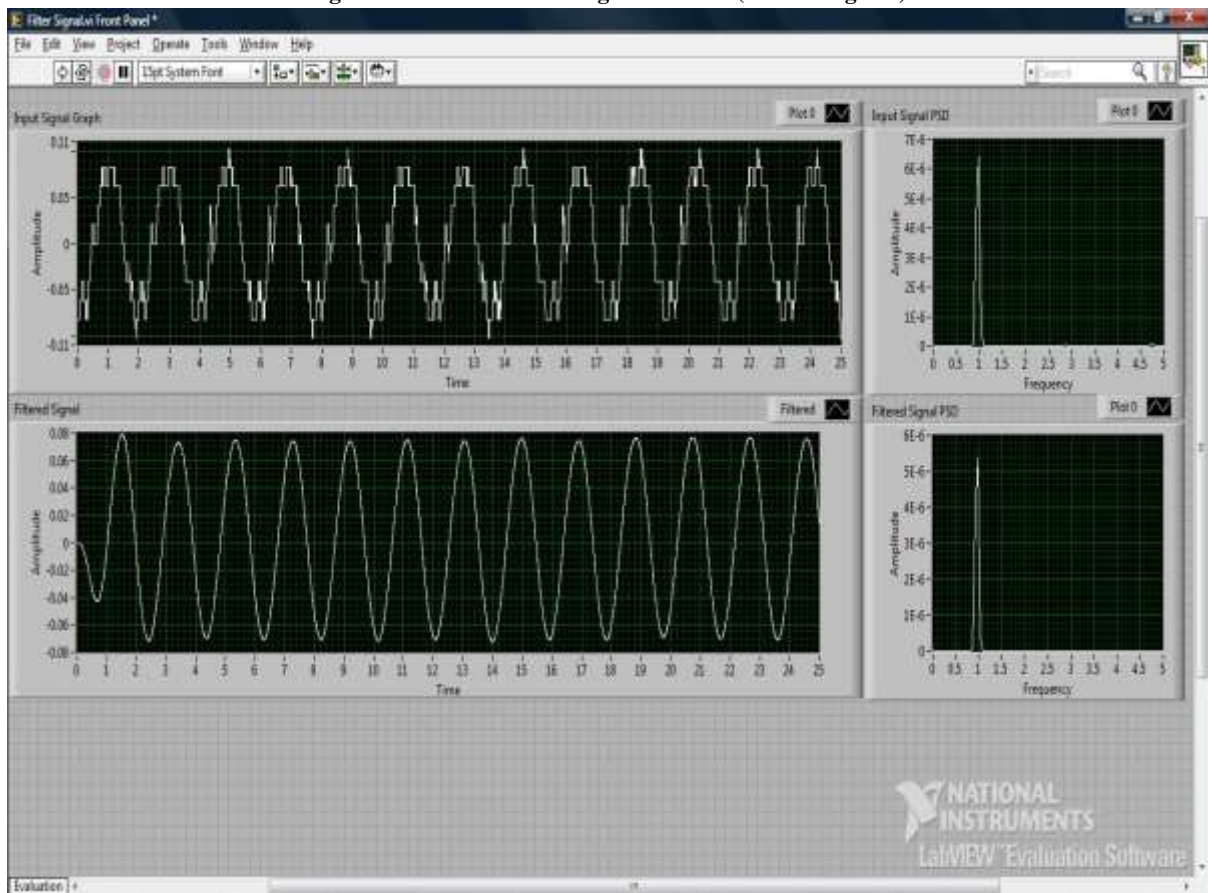


Fig5.8.a: Filter Signal.VI (Front Panel)

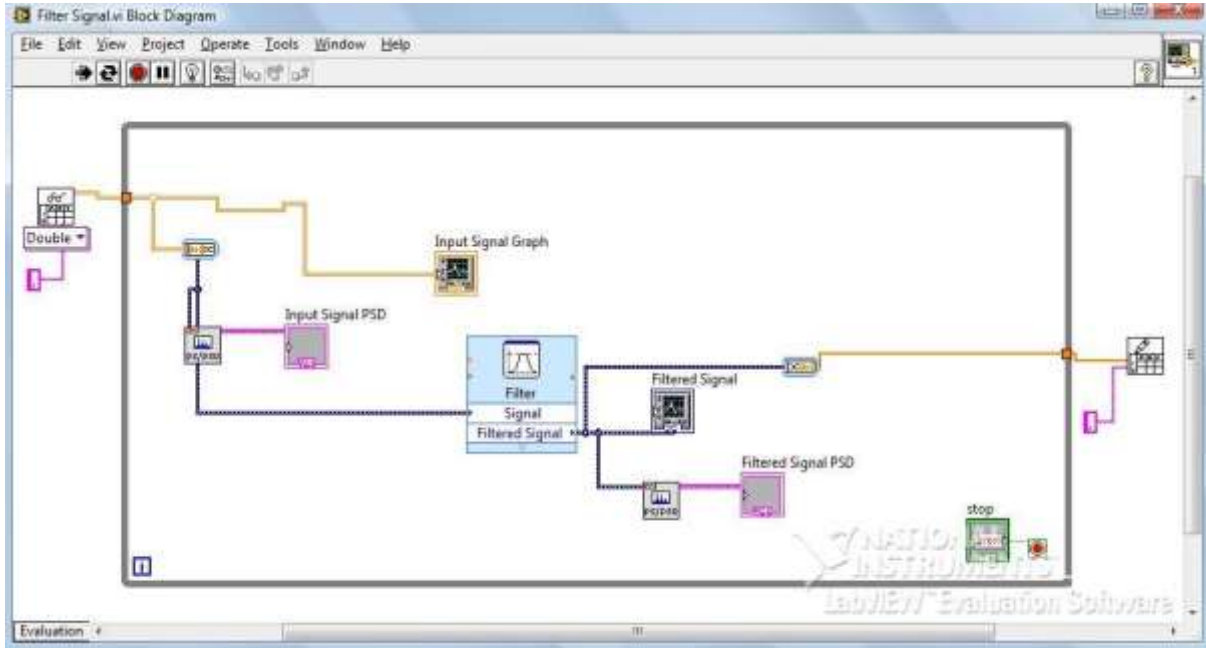


Fig5.8.b: Filter Signal.VI (Block Diagram)

Thus by following the above methodology the graphs for the Input Signal and the Output Signal are generated. Input Signal

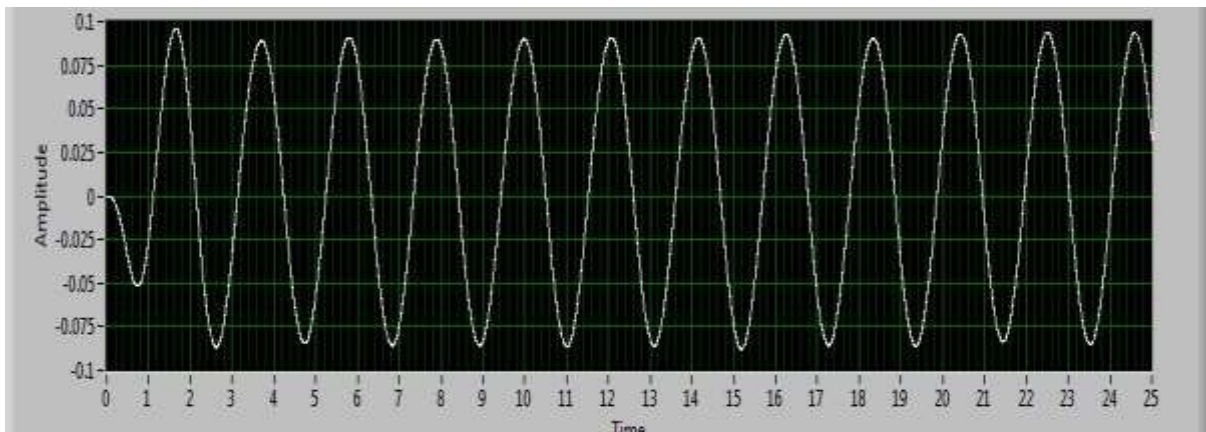


Fig5.9: Filtered signal (Acceleration) of the exciting source (Passive)

Output Signal

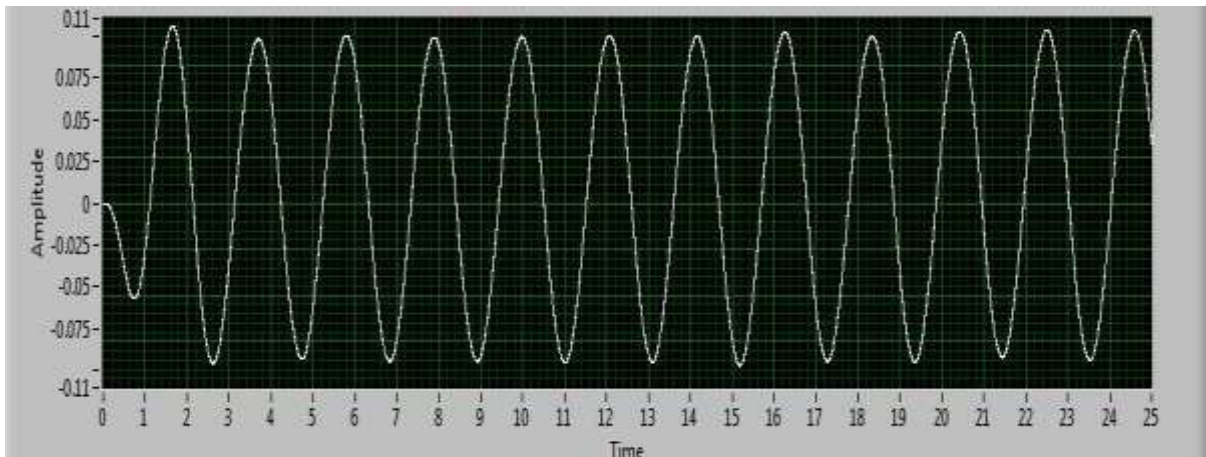


Fig5.10: Filtered signal (Acceleration) of the passenger seat mass (Passive)

Transmissibility

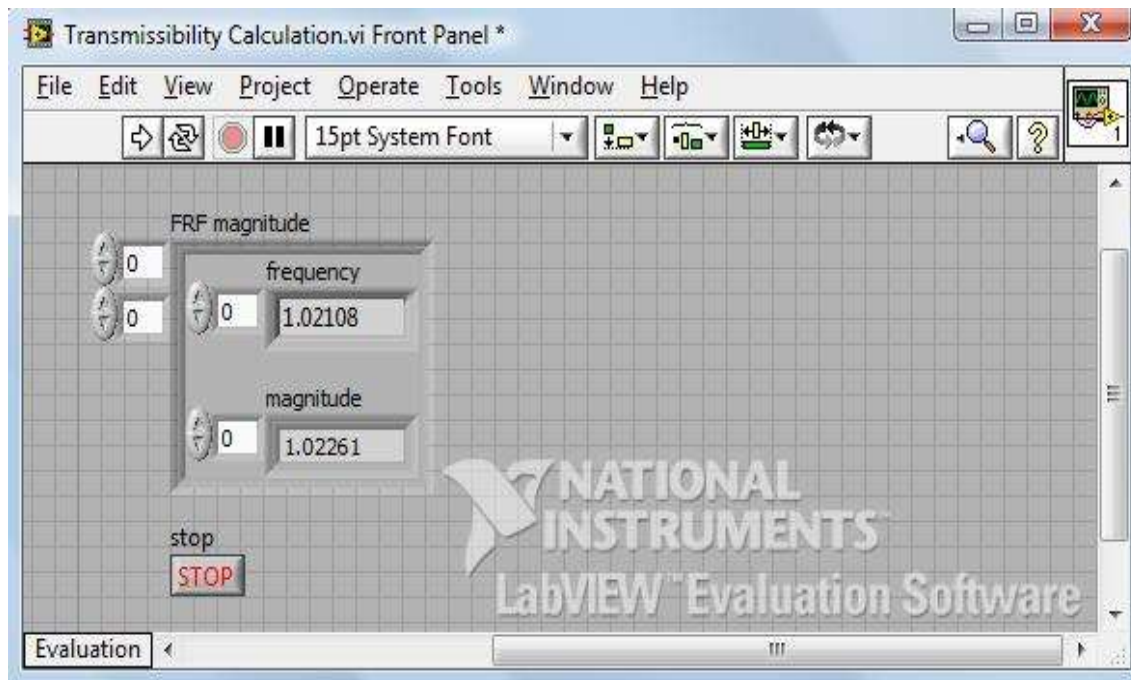


Fig5.11.a: Transmissibility.vi (Front Panel)

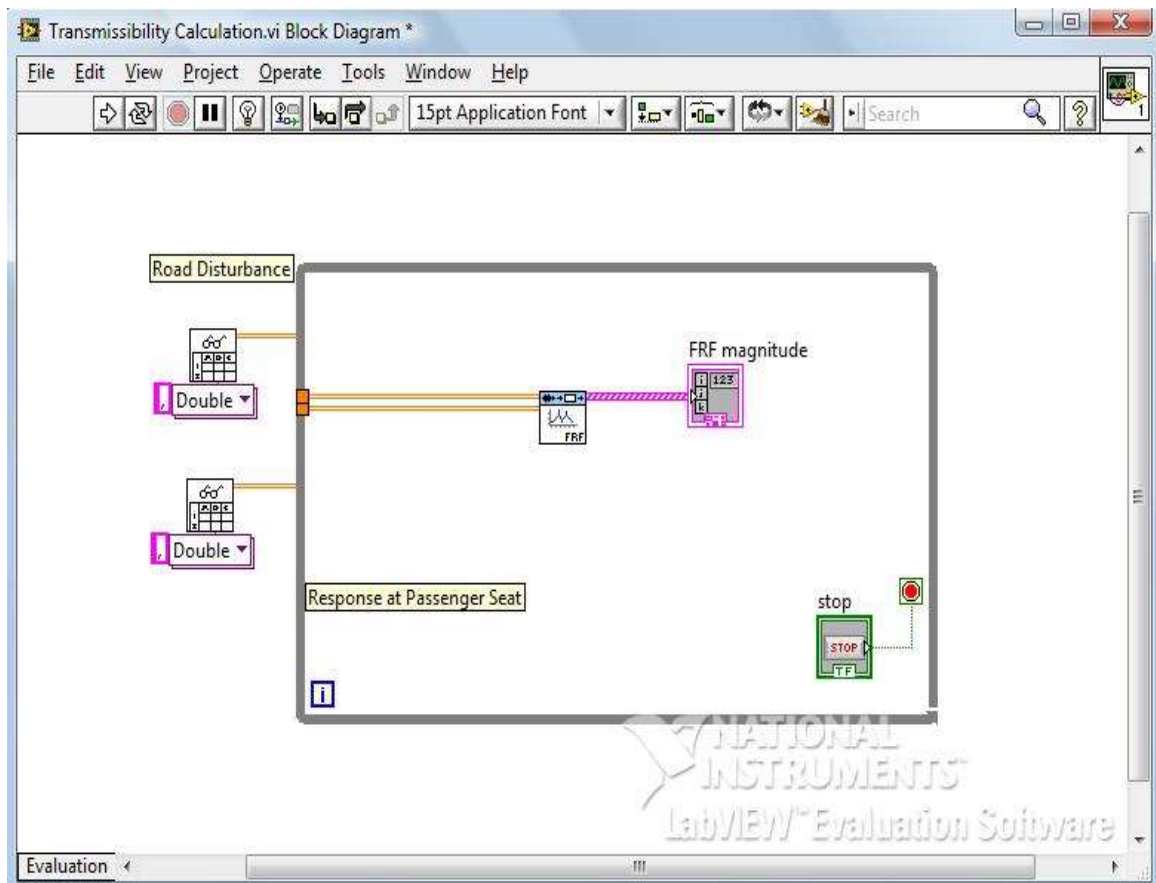


Fig5.11.b: Transmissibility.vi (Block Diagram)

For calculating the transmissibility, the 'Transmissibility.vi' was used which can be seen in Figure5.11.a and Figure5.11.b. The Read from Spread Sheet VI prompts the user to first select the Input File and then the Output File and then the transmissibility is calculated in the frequency domain as the ratio of the output signal to the input signal.

5.3. Comparison between the Simulated and Experimental result:

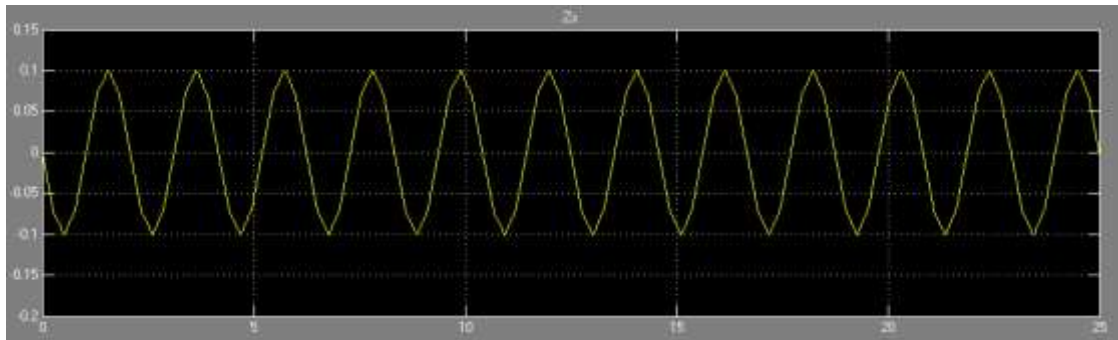


Fig 5.12 a) Result obtained using Simulation

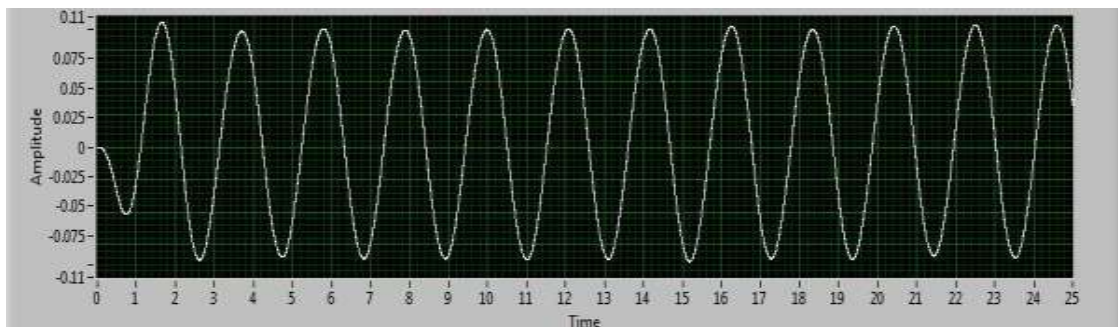


Fig 5.12 b) Result obtained from Experiment

From Figures 5.12.a and 5.12.b it can be seen that the Results obtained from Simulation and Results obtained from the Experiment are in good accordance with each other and hence results for the Passive Suspension System have been validated. Also the transmissibility from Simulation is found out to be 1.12 from Figure 2.14.c, whereas the transmissibility from the testing is found out to be 1.02261 from Figure 5.11.a and hence the Conclusions drawn in Section 2.3.4 hold good.

5.4. Setting up the Serial Port Communication:

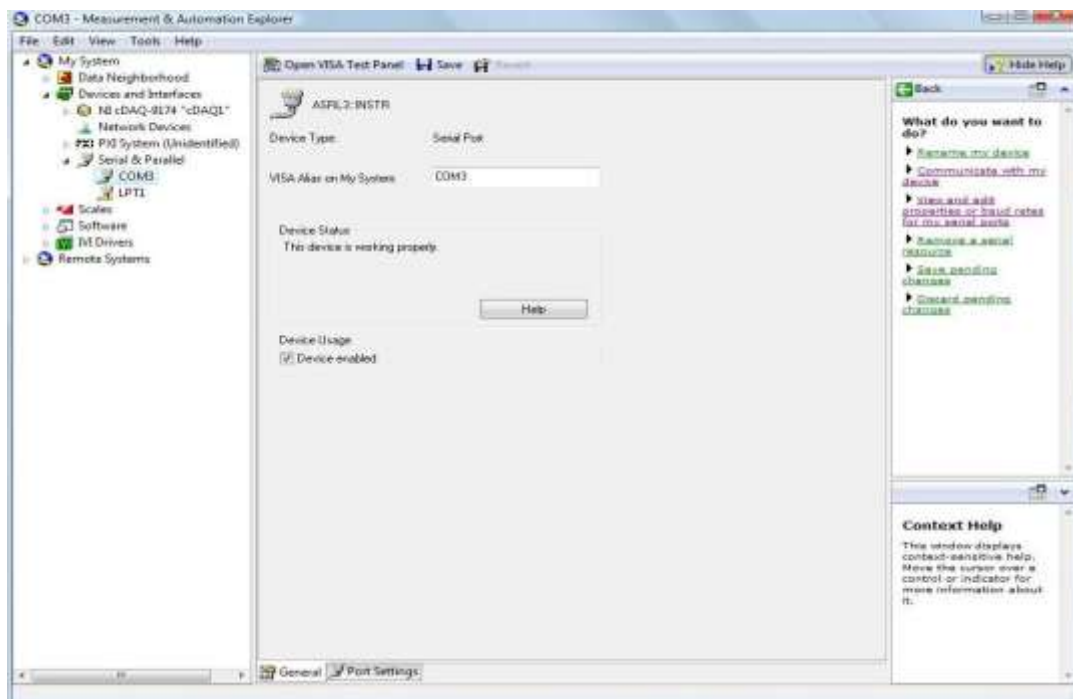


Fig5.13.a: Setting up the Serial Port Communication COM Port

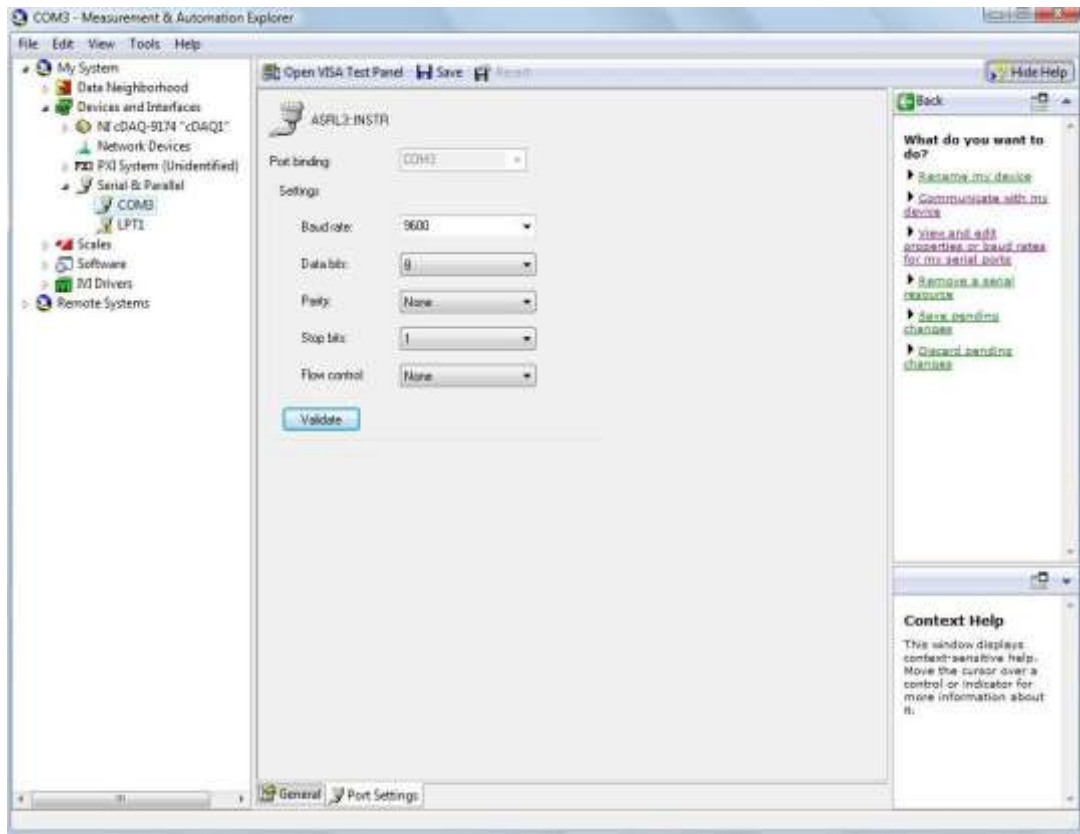


Fig5.13.b: Setting up the Serial Port Communication Parameters

The USB to RS-232 Convertor was connected to the USB port of the Laptop and after the drivers for the Convertor were installed; using MAX and an analog multimeter the Port was tested whether it is working. As shown in Figure 5.12.a the port appears under My System/ Serial and Parallel. The Port Parameters such as the Baud rate, Data Bits and Control Flow were configured to ensure compatibility with the VI which will use it to send signals to the Actuator.

5.5. Active Suspension System Setup and Testing:



Fig5.14: Active System Setup

The set-up of the Active System was similar to the Passive System as shown in Figure 5.14. The only difference being the connection of the Actuator inputs to the Laptop which was done in Section 5.5. Also the same procedure was followed for the testing, only that the VI used for the acquiring the data, processing the data and writing measurement files was different since it had to involve the PID Controller algorithm and Serial Port Communication Blocks. The Values of the Gains (Kp, Ki and Kd) found out from simulations had to be input in the Front Panel of the 'Active System(PID, write and serial communication).vi' The PID controller block in the block diagram contains the Control Algorithm used for controlling the PID Controller output as shown in Figure 5.15.b. Also the SERIAL Write Block is used to send the voltage values in the form of a Pulse Width to the actuator to control the magnetic field strength.

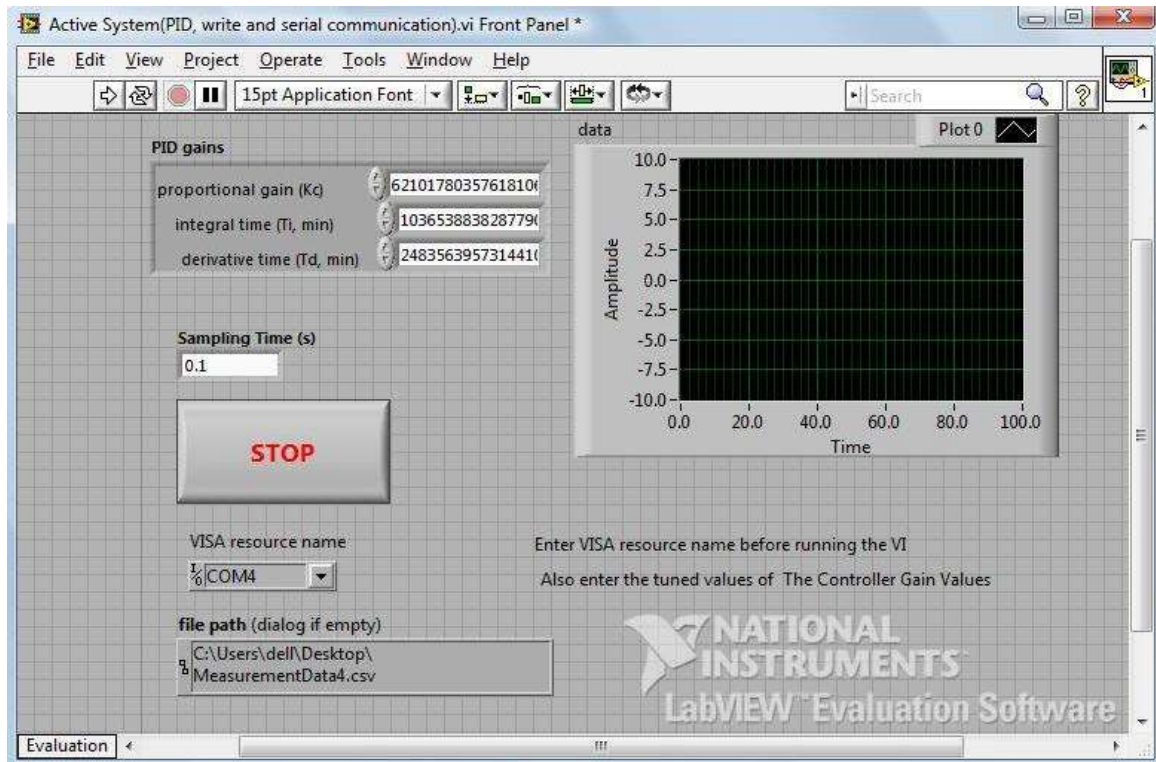


Fig5.15.a: Active System(PID, write and serial communication).vi (Front Panel)

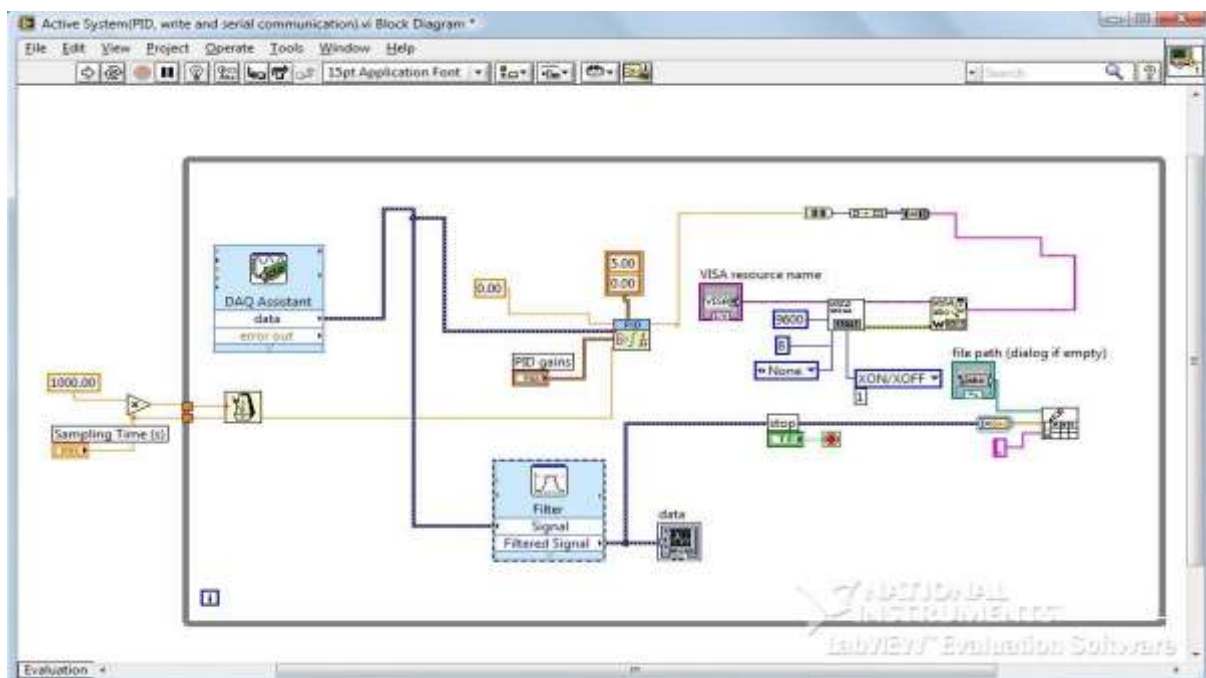


Fig5.15.b: Active System(PID, write and serial communication).vi (Block Diagram)

Input Signal

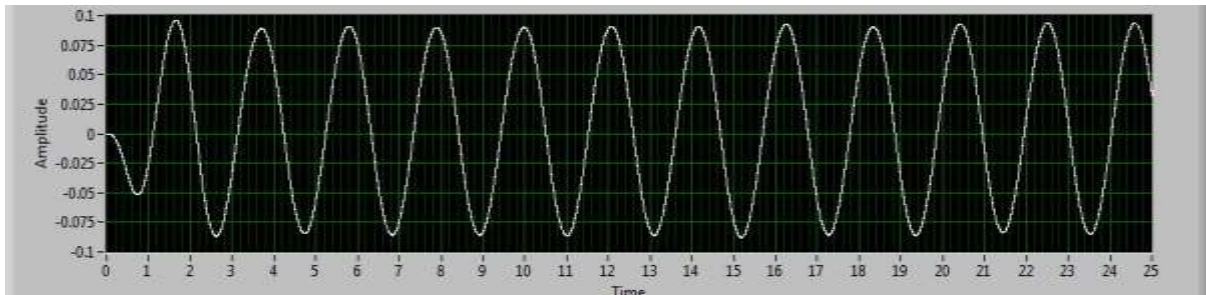


Fig5.16: Filtered signal (Acceleration) of the exciting source

Output Signal

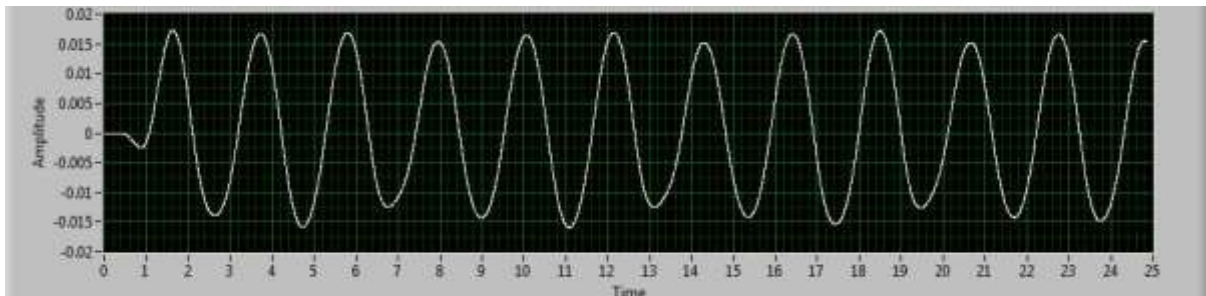


Fig5.17: Filtered signal (Acceleration) of the passenger seat mass

Transmissibility

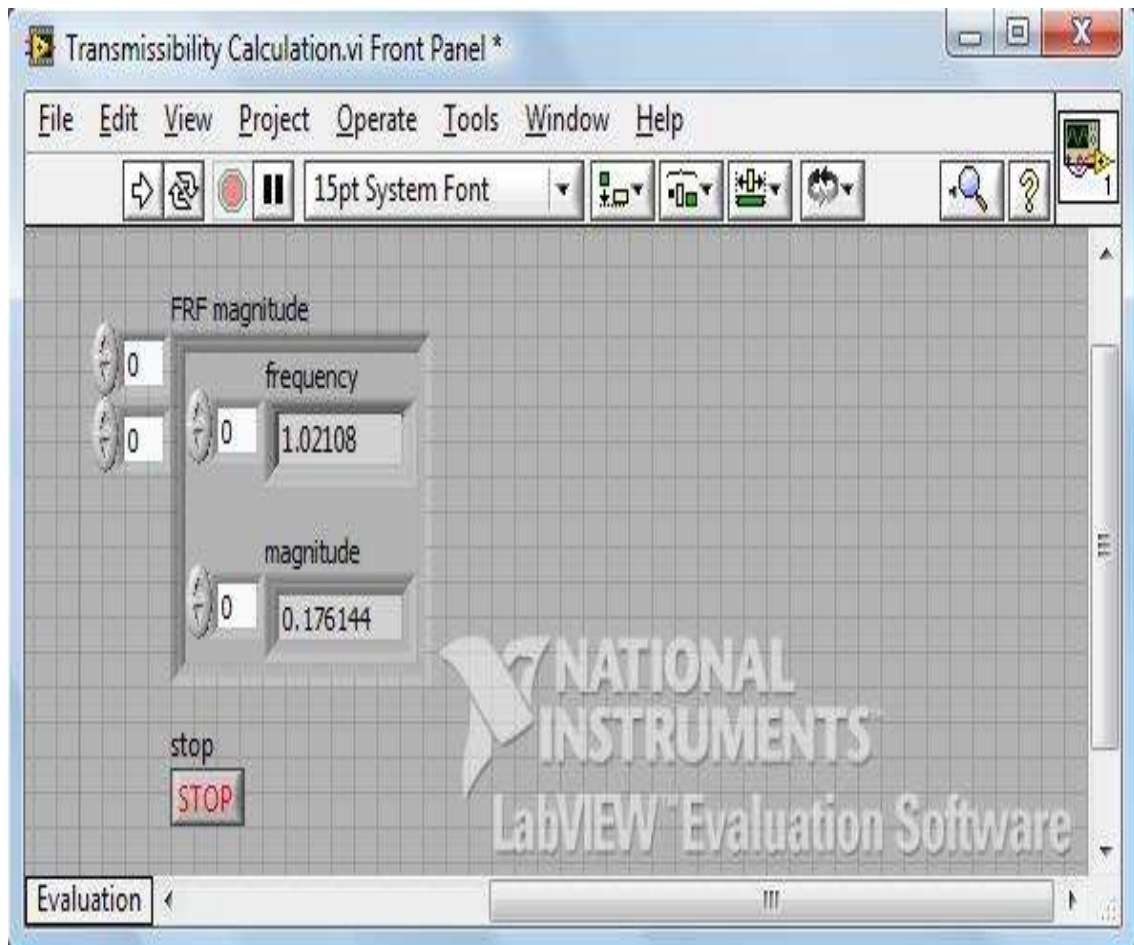


Fig5.18: Transmissibility.vi (Front Panel)

The Transmissibility was calculated using the 'Transmissibility.vi' in the same manner as for the Passive Suspension case.

5.6. Comparison between the Simulated and Experimental result:

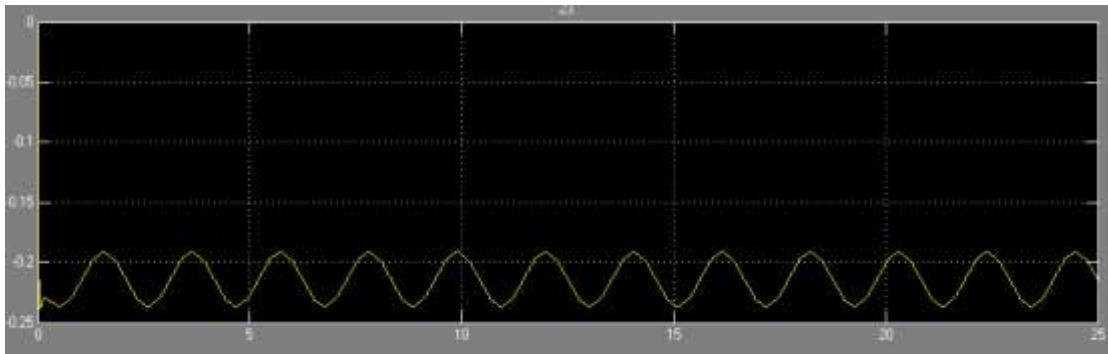


Fig 5.19.a) Result obtained using Simulation

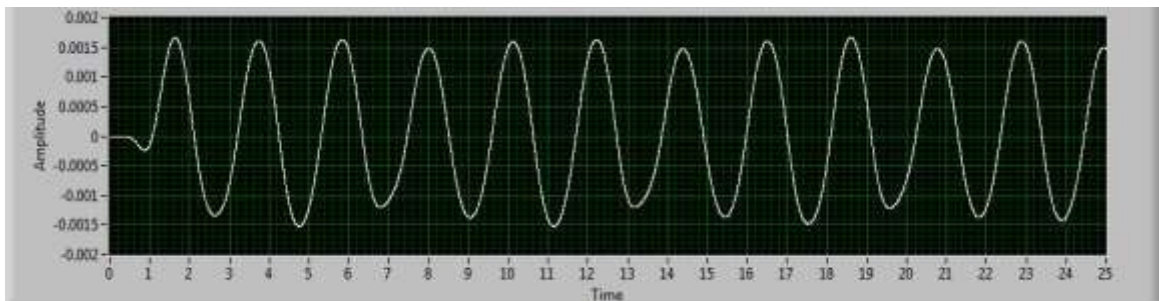


Fig 5.19.b) Result obtained from Experiment

Figures 5.19.a and 5.19.b show that although the results obtained from the Simulation and Experiment appear to be different from the Screen Shots, however when closely observed without the offset in the Simulation result the response characteristics of both the graphs are in fact similar. The offset in the Simulation was provided to prevent Solver Issues. The Solver Used was ode45 which is a fourth order Runge-Kutta method for solving non-linear equations and it returned Inf (Infinite) results because the value of the current from the actuator sometimes dropped to values as low as of the order of 10^{-9} . Hence the response characteristics are similar for both the cases. Now the transmissibility obtained from Simulation is 0.157 from Figure 2.24.c and the transmissibility found out using Experimental results is 0.176144 From Figure 5.18. Thus even the Active System performed as proposed within the tolerance limits and hence the conclusions drawn in Section 2.4.5 hold good for the Active Suspension System Case.

6. SUMMARY AND CONCLUSIONS ABOUT THE PERFORMANCE OF THE PROPOSED ACTIVE SUSPENSION SYSTEM

The results of simulation performed by means of the mathematical model developed on Simulink comply with the results of experimental test carried out using the solenoid actuator that was designed, which can be used to state that the actuator model on Simulink reflects the dynamic properties of the actuator. Tests by means of the simulation model of the solenoid actuator can be used to fully discover the dynamic phenomena which occur during the actuator's action.

In this project, an electromagnetic actuator and correspondingly a PID controller has been designed for a quarter car model of a passenger car to improve the ride comfort and road holding ability. Manual tuning technique is used for series of tunings to improve the ride comfort and road holding ability. For step input the of 0.01 m, the sprung mass displacement has been reduced by 85% which shows the improvement in ride comfort and sprung mass acceleration reduced by 85%.The suspension travel has been reduced by 74% and tire deflection has reduced by 36% which will improve the life of the suspension system and shows the better road holding ability of the active suspension system. Therefore, it is concluded that the active suspension system has better performance capabilities over the passive suspension system.

Due to the change in vehicle concepts to the more electric car, the suspension system becomes ever more important due to changes in the sprung and unsprung masses. Active electromagnetic suspension systems can maintain the required stability and comfort due to the ability of adaptation in correspondence with the state of the vehicle.

It is suggested that specifications must also be drawn from on-road measurements on a passive suspension system to be able to better design the Active Suspension Model and also that it directly validates and helps us to optimize the Control Algorithms used for controlling the Controller.

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