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Frequency Response of Semi Independent Automobile Suspension System

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Abstract: Suspension system is one of the important systems in automobile. Its helps to support the car body, engine, passengers, and at the same time it tries to isolate the vehicle body from the road bumps and absorbs the vibration while vehicle is moving on uneven surface of the road. In this paper, attention is focused towards a semi independent suspension vehicle. Describing the vehicle using seven independent coordinates, equations of motion are derived and the dynamic behaviour is studied through frequency response.

Keywords : Semi independent suspension system, Equations of motion, Frequency response, Displacement disturbance

I. INTRODUCTION

The vehicle suspension forms one very important system for an automobile. This is the one which helps to support the engine, vehicle body and passengers and at the same time absorbs the shocks arising due to roughness of the road.

The usual arrangement consists of supporting the chassis through springs and dampers, by the axle. The engine and the body of the vehicle are attached to the chassis rigidly. Hence, the chassis along with the body of the vehicle, engine and the passengers may be considered to be one unit only. The springs and dampers which connect the axle and chassis play an important role in absorbing shocks and keeping chassis affected to a minimum level.

The review of the past literature reveals that this has been a subject of interest to many researchers. Attempts have been made to perform the studies on *quarter car model* and *half car model* of the vehicle. The *quarter car model* involves only one wheel, supporting springs and damper, and mass ***m*** as shown in the Figure 1(a). In *half car model*, one front wheel and one rear wheel are considered to support a mass ***m*** through springs and dampers as shown in Figure 1(b). Such a half car model is called *pitch bounce* model. The *half car model* may involve either only two front wheels or only two rear wheels. Such a model is called *roll bounce* model. The mass ***m*** in Figures 1 includes mass of the chassis, and masses of various things attached to it. Apart from the springs and dampers of suspension system, the pneumatic tyres also have some amount of mass, stiffness and damping. Hence, the *quarter car model* and *half car model* shown in Figures 1 may further be represented as in Figure 2. The mass m_1 in the figures represents the mass of the tyre and a percentage of mass of the axle. These models represent the original system, in an approximate and simplified manner. However, they are not realistic models.

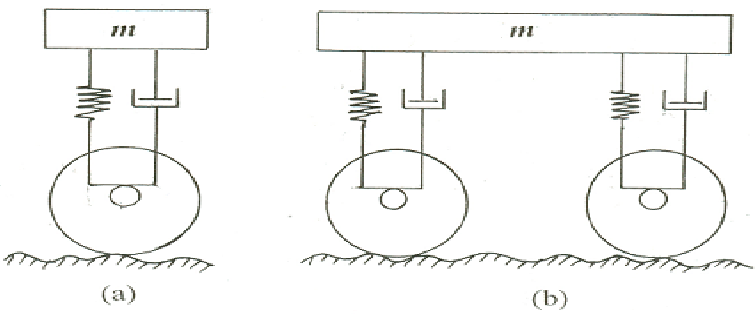

Fig 1 Simplified models of car (a) Quarter car model (b) Half car model

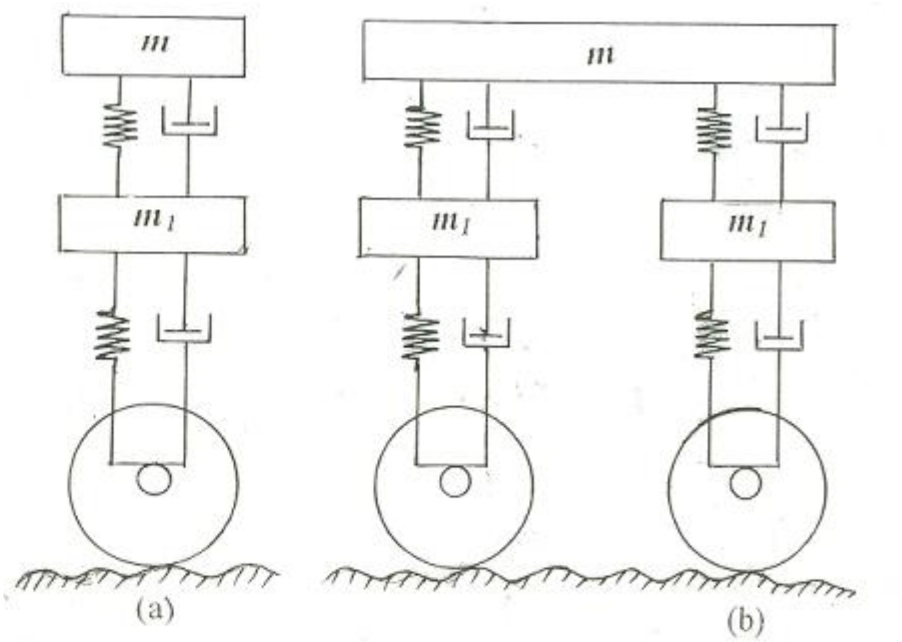

Fig 2 The models including the effect of pneumatic tyres (a) Quarter car model (b) Half car model

Hedrick[1] considered a quarter car model with hydraulic actuator acting under the effect of coulomb friction. An absorber based nonlinear controller and adaptive nonlinear controller are proposed. Employing two sensors, one for displacement and other for velocity measurements, Majjad [2] considered a quarter car model and estimated the nonlinear damping parameters. Gobbi and Mastin [3], Wei Gao et al. [8] studied dynamic behaviour of passively suspended vehicles running on rough roads. The road profile is considered to give random inputs to the suspension system. Rajalingam and Rakheja [4] studied the dynamic behaviour of quarter car model under nonlinear suspension damper. Ahmed Faheem [5] studied the dynamic behaviour using quarter car model and half car model for different excitations given by the road. Jacquelien et al. [6] used electrical analogy in conjunction with quarter car model and studied the control scheme of the suspension system. Wei Gao et al. [7] also studied the dynamic characteristics considering the mass, damping and tyre stiffness as random variables. Kamalakannan et al. [9] tried adaptive control by varying damping properties according to the road conditions. Sawant et al. [10] developed an experimental procedure for determining the suspension parameters using a quarter car model. Thite [11] refined the quarter car model to include the effect of series stiffness. State space equations are employed to calculate the natural frequency and model damping ratios. Gadhia et al. [12] analysed quarter car model for rear suspension using ADAMS software.

Wei Gao [13] investigated dynamic response of cars due to road roughness treating it as random excitation. Lin [14] performed a time domain direct identification for vehicle mass, damping and stiffness. Husiyno Akcay[15] studied multi objective control of half car suspension system. It is observed that when the tyre damping coefficients are precisely estimated, the road holding quality of the suspension system can be improved to some extent. Li-Xing Gao [16] considered a half car model in conjunction with pseudoexcitation for the road conditions and studied the dynamic response of the vehicle. Thite et al. [17] used a frequency domain method for estimating suspension system parameters. Roberto Barbosa [18] studied the frequency response of half car model due to pavement roughness. Roberto Barbosa [19] also investigated vibrations of vehicles subjected to a long wave measured pavement irregularity.

The quarter car model and half car model represent the actual car in an approximate way. An automobile, in principle can exhibit three rotations, roll, yaw and pitch, in addition to up and down vertical linear motion and linear motion in the horizontal plane. In total, it can have three displacements and three rotations. However, due to stiffness in the suspension system, the two components of displacements in the horizontal plane and yaw motion may be ignored totally. Any disturbance due to roughness of the road naturally effects up and down linear motion and, roll and pitch rotations. The quarter car model cannot consider roll and pitch motion. A half car model can consider either roll motion or pitch motion, depending on whether it is roll bounce model or pitch bounce model. The coupling of motions cannot be seen in these models. That is way these models cannot be realistic. However, they form a quick and good initial start to the actual problem. Balaraju and Venkatachalam [20, 21] have presented an analysis of a full car model.

The work referred so for is mostly concerned to conventional suspension system in which the chassis of the vehicle is supported on shock absorbers (spring and damper) by the axles which are supported by the wheels of the vehicle, as shown in Figure 3(a). There is yet another type of suspension system called independent suspension system. In this, the axle of the wheel is hinged to the body of the vehicle. The axle is also connected to the body through spring and damper. The body of the vehicle acts as the chassis, as shown in the Figure 3(b).

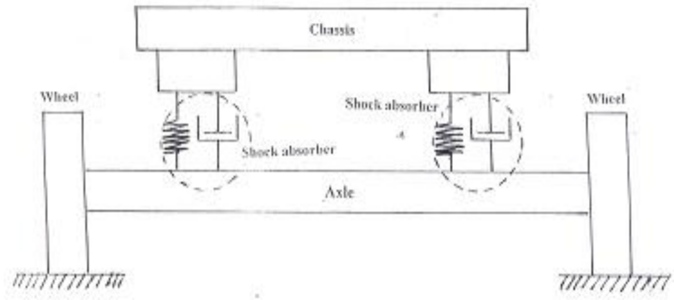

Fig 3(a) Conventional suspension system

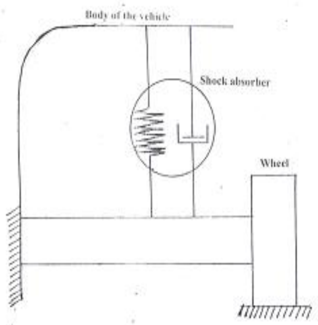

Fig 3(b) Independent suspension system

When an independent suspension system is used, the effect of the disturbance caused by the roughness of the road is confined mostly to that part (corner) of the vehicle. The present trend an automobile industry is to go for independent suspension. Among the Indian vehicles, heavy trucks and lorries, Ambassador car, Bolero van, Scarpio, Safari, Innova car etc. use conventional suspension system. The cars such as Swift. Santro -xing, Maurti-Alto, i20, Tata-indica, maruti800, Firo, use semi independent suspension system. That is, the front wheels are independently suspended and rear wheels are conventionally suspended. The cars such as Accent use independent suspension for all the four wheels. Attempts are being made to analyze the four wheeler with fully independent suspension system. Libin Li [22] performed computer simulation studies through multi body model, identifying twenty degrees of freedom. Pater Gaspar [23] considered full car model and proposed a method for identifying suspension parameters taking in to account nonlinear nature of the components. Anil Shirahatt et al.[24] attempted to maximize the comfort level considering a full car model. Genetic algorithms have been employed to perform optimization to arrive at optimum values of suspension parameters. Hajkurami et al. [25] studied the frequency response of a full car model as a system of seven degrees of freedom. Ikbal Eski [26] obtained neural network base control system for full car model. Guidaa et al. [27] proposed a method of identifying parameter of a full car model. The analysis has been developed for designing an active suspension system. All these papers are concerned with fully independent suspension system. Regarding semi independent suspension system, very little work has been reported in the literature. In this paper an attempt is made to derive equations of motion for a semi independent suspension system, and study the dynamic behaviour of the vehicle.

II. NOMENCLATURE

B, L_1, L_2	Dimensions of the main body see Figure 4
c_2 and c_3	Damping coefficients of dampers in suspension system, *N.s/m*
G and G_2	Centres of mass of main body and rear axle
I_2	Mass moment of inertia of the rear axle about G_2, $kg.m^2$
I_r and I_p	Mass moments of inertia of the main body about roll and pitch axes , respectively, $kg.m^2$
k_1 , c_1	Stiffness and damping of the tyre, *N/m* and *N.s/m*
k_2 and k_3	Stiffness's of springs in the suspension system, *N/m*
m	Mass of chassis and various things attached to the chassis, such as engine, body of the vehicle, *kg*
M , C , K	Mass, damping and stiffness matrices
m_1	Mass of the tyre, *kg*
m_2	Mass of the rear axle, *kg*
x	Displacement of the centre of mass G of the main body
x_1 and x_2	Displacements of masses m_1
x_3	Displacement of G_2
y_i	The vertical displacements caused by roughness of the road at different wheels, $i = 1 \text{ to } 4$, *m*
γ and λ	Roll and Pitch motions of the main body

III. DEVELOPMENT OF EQUATIONS OF MOTION

Figure 4 shows the schematic arrangement of the semi independent suspension system. The mass m of the main body, is supported at its four corners. The front side suspension system characteristics are specified by the spring constant k_2 and the damping constant c_2. The rear side suspension system characteristics are denoted by the spring constant k_3 and the damping constant c_3. The mass m_1 indicates mass of the tyres. The tyre characteristics are indicated by the spring constant k_1 and the damping constant c_1. The mass m_2 indicates the mass of the rear wheel axle which includes the masses of the tyres. The displacement inputs given by the road roughness are described by y_i , i = 1 , 2 , 3 and 4.

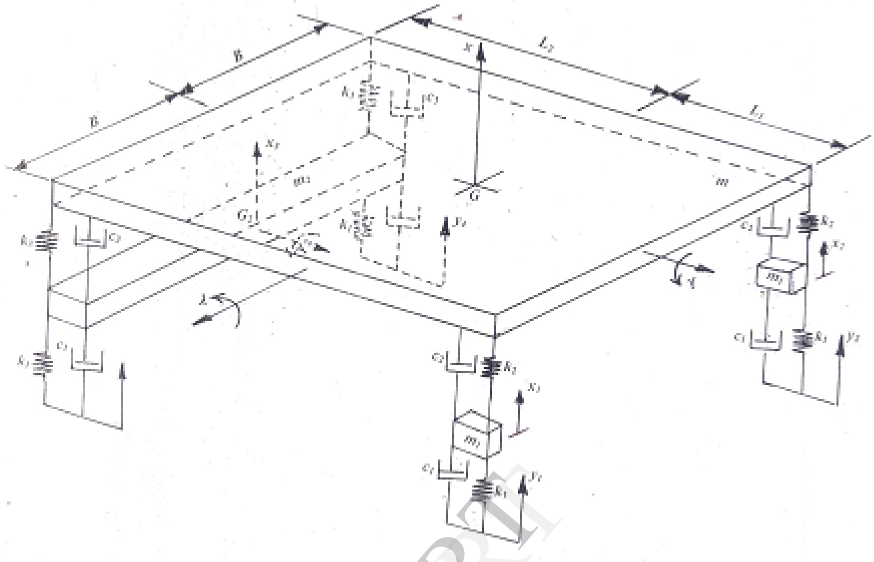

Fig 4 Semi Independent suspension of full car model

The displacements of the masses m_1 may be described using the coordinates, x_1 , x_2 . The mass m_2 is the mass rear wheel axle which includes masses of the two rear wheels. The motion of the mass m_2 may be described using the coordinates, x_3 to describe the up and down motion of its centre of mass G_2 and γ_2 to describe its roll motion. The displacement of the centre of mass G of the main body of mass m is described by the coordinate x. The roll and pitch rotations of the main body are described by the coordinates by γ and λ. The total suspension system is described by seven coordinates, namely, x_1 , x_2 , x_3 , x , γ , γ_2 and λ.

Giving positive displacements to all the seven coordinates, the free body diagrams may be drawn as in Figures 5 and the equations of motion may be written as,

$$m_1\ddot{x}_1 + (k_1+k_2)x_1 - k_2x + k_2B\gamma - k_2L_1\lambda + (c_1+c_2)\dot{x}_1 - c_2\dot{x} + c_2B\dot{\gamma} - c_2L_1\dot{\lambda} = k_1y_1 + c_1\dot{y}_1 \quad (1)$$

$$m_1\ddot{x}_2 + (k_1+k_2)x_2 - k_2x - k_2B\gamma - k_2L_1\lambda + (c_1+c_2)\dot{x}_2 - c_2\dot{x} - c_2B\dot{\gamma} - c_2L_1\dot{\lambda} = k_1y_2 + c_1\dot{y}_2 \quad (2)$$

$$m_2\ddot{x}_3 + 2(k_1+k_3)x_3 - 2k_3x + 2k_3L_2\lambda + 2(c_1+c_3)\dot{x}_3 - 2c_3\dot{x} + 2c_3L_2\dot{\lambda} = 0 \quad (3)$$

$$m\ddot{x} - k_2x_1 - k_2x_2 - 2k_3x_3 + 2(k_2+k_3)x + 2(k_2L_1 - k_3L_2)\lambda - c_2\dot{x}_1 - c_2\dot{x}_2 - 2c_3\dot{x}_3 + 2(c_2+c_3)\dot{x} + 2(c_2L_1 - c_3L_2)\dot{\lambda} = 0 \quad (4)$$

$$I_r\ddot{\gamma} + k_2Bx_1 - k_2Bx_2 + 2B^2(k_2+k_3)x - 2k_3B^2\gamma_2 + c_2B\dot{x}_1 - c_2B\dot{x}_2 + 2B^2(c_2+c_3)\dot{x} - 2c_3B^2\dot{\gamma}_2 = 0 \quad (5)$$

$$I_2\ddot{\gamma}_2 - 2k_3B^2\gamma + 2(k_1+k_3)B^2\gamma_2 - 2c_3B^2\dot{\gamma} + 2(c_1+c_3)B^2\dot{\gamma}_2 = k_1(y_3 - y_4) + c_1(\dot{y}_3 - \dot{y}_4) \quad (6)$$

$$I_p\ddot{\lambda} - k_2L_1x_1 - k_2L_1x_2 + 2k_3L_2x_3 + 2(k_2L_1 - k_3L_2)x + 2(k_2L_1^2 + k_3L_2^2)\lambda - c_2L_1\dot{x}_1 - c_2L_1\dot{x}_2 + 2c_3L_2\dot{x}_3 + 2(c_2L_1 - c_3L_2)\dot{x} + 2(c_2L_1^2 + c_3L_2^2)\dot{\lambda} = 0 \quad (7)$$

The motion of the entire suspension system may be described by the Equations (1) to (7). The road roughness is giving inputs to the system through y_i and $\dot{y}_i$, $i = 1, 2, 3, 4$, which influence x_1 , x_2 and γ_2 only. These in turn influence x_3 of the rear axle and , x, γ and λ motions of the main body.

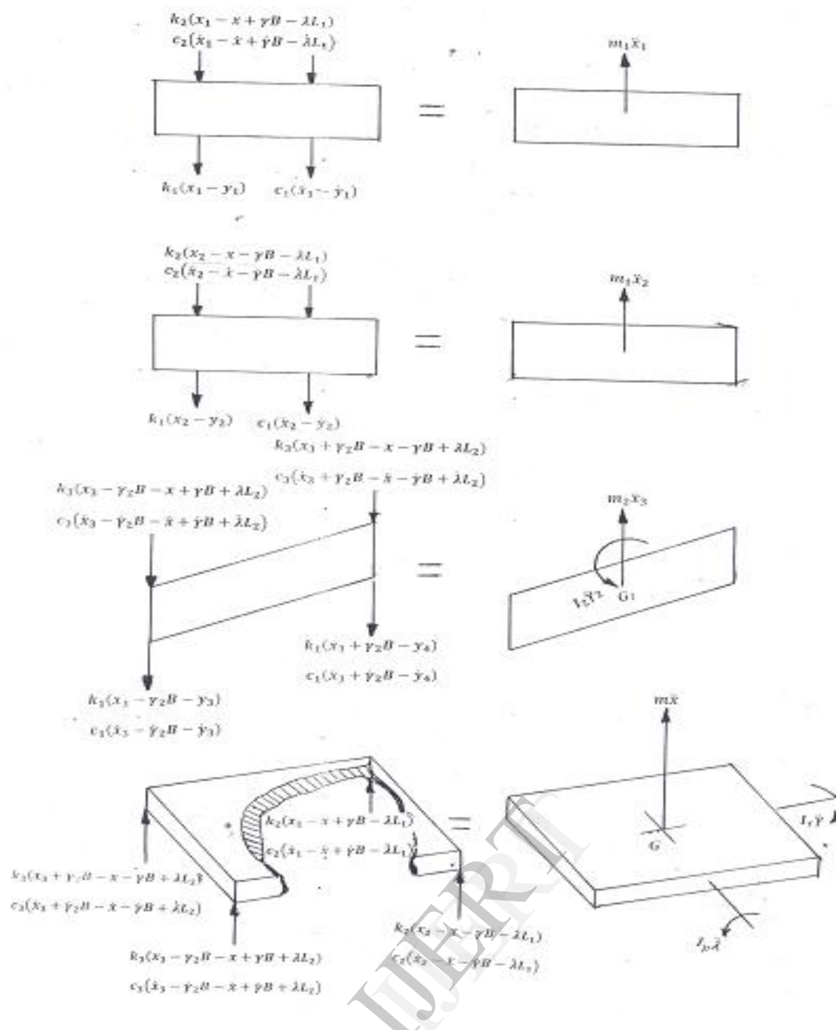

Fig 5 Free body diagrams

Renaming the coordinates as $X_1 = x_1$, $X_2 = x_2$, $X_3 = x_3$, $X_4 = x$, $X_5 = \gamma$, $X_6 = \gamma_2$, and $X_7 = \lambda$, the equations of motion (1) to (7) may be represented in matrix form as,

$$M\ddot{\overline{X}} + C\dot{\overline{X}} + K\overline{X} = \overline{F} \qquad (8)$$

where, M , C and K are mass, damping and stiffness matrices, each of size (7×7) and may be expressed as,

$$M = \begin{bmatrix} m_1 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & m_1 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & m_2 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & m & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & I_r & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & I_2 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & I_p \end{bmatrix} \qquad (9)$$

$$C = \begin{vmatrix} c_1+c_2 & 0 & 0 & -c_2 & c_2B & 0 & -c_2L_1 \\ 0 & c_1+c_2 & 0 & -c_2 & -c_2B & 0 & -c_2L_1 \\ 0 & 0 & 2(c_1+c_3) & -2c_3 & 0 & 0 & 2c_3L_2 \\ -c_2 & -c_2 & -2c_3 & 2(c_2+c_3) & 0 & 0 & 2(c_2L_1-c_3L_2) \\ c_2B & -c_2B & 0 & 0 & 2(c_2+c_3)B^2 & -2c_3B^2 & 0 \\ 0 & 0 & 0 & 0 & -2c_3B^2 & 2(c_1+c_3)B^2 & 0 \\ -c_2L_1 & -c_2L_1 & 2c_3L_2 & 2(c_2L_1-c_3L_2) & 0 & 0 & 2(c_2L_1^2+c_3L_2^2) \end{vmatrix}$$

(10)

$$K = \begin{vmatrix} k_1+k_2 & 0 & 0 & -k_2 & k_2B & 0 & -k_2L_1 \\ 0 & k_1+k_2 & 0 & -k_2 & -k_2B & 0 & -k_2L_1 \\ 0 & 0 & 2(k_1+k_3) & -2k_3 & 0 & 0 & 2k_3L_2 \\ -k_2 & -k_2 & -2k_3 & 2(k_2+k_3) & 0 & 0 & 2(k_2L_1-k_3L_2) \\ k_2B & -k_2B & 0 & 0 & 2(k_2+k_3)B^2 & -2k_3B^2 & 0 \\ 0 & 0 & 0 & 0 & -2k_3B^2 & 2(k_1+k_3)B^2 & 0 \\ -k_2L_1 & -k_2L_1 & 2k_3L_2 & 2(k_2L_1-k_3L_2) & 0 & 0 & 2(k_2L_1^2+k_3L_2^2) \end{vmatrix} \quad (11)$$

$$\bar{X} = [\, x_1 \quad x_2 \quad x_3 \quad x \quad \gamma \quad \gamma_2 \quad \lambda \,]^T \quad (12)$$

$$\bar{F} = [\, k_1y_1 + c_1\dot{y}_1 \,\vdots\, k_1y_2 + c_1\dot{y}_2 \,\vdots\, 0 \,\vdots\, 0 \,\vdots\, 0 \,\vdots\, k_1(y_3 - y_4) + c_1(\dot{y}_3 - \dot{y}_4) \,\vdots\, 0 \,]^T \quad (13)$$

It is to be observed that $\boldsymbol{M}$, $\boldsymbol{C}$ and $\boldsymbol{K}$ are symmetric matrices. Further, the matrix $\boldsymbol{M}$ is a diagonal matrix. The equations of motion are linear and non homogenous differential equations. A considerable amount of coupling is existing between all the equations.

IV. ANALYSIS OF THE SUSPENSION SYSTEM

In order to determine the natural frequencies of the system, the damping and the excitation are ignored and the remaining part of the equations of motion is considered as,

$$M\ddot{\bar{X}} + K\,\bar{X} = 0 \quad (14)$$

For the sake of numerical computations, a specific set of various parameters, relating to a practical automobile are taken as fallows.

m_1 = 40 kg	m_2 = 100 kg	m = 1000 kg
I_2 = 20 kg.m^2	I_r = 500 kg.m^2	I_p = 1000 kg.m^2
k_1 = 2×10^5 N/m	k_2 = 0.5×10^5 N/m	k_3 = 0.5×10^5 N/m
c_1 = 1000 N.s/m	c_2 = 1000 N.s/m	c_3 = 1000 N.s/m
L_1 = 1.0 m	L_2 = 2.5 m	B = 0.75 m

The natural frequencies, f_n (in Hz) are obtained using MATLAB software as follows

1.793	2.129	3.872	11.434	12.602	12.621	18.883

All these seven frequencies are to be taken as natural frequencies of the total system. The resonance may occur whenever the exciting frequency matches with any one of these natural frequencies. It is observed that the first three frequencies are district.

V. FREQUENCY RESPONSE OF THE SYSTEM

The system is excited with a harmonic force at right side front wheel as,

$$y_1(t) = y_{m_1} \sin(2\pi f)t \quad (15)$$

$$\dot{y}_1(t) = y_{m_1} (2\pi f)\cos(2\pi f)t \quad (16)$$

The set of seven second order differential equations representing the equations of motion as given in the Equation (8) is integrated using fourth order Runge-Kutta method. The response of the system is observed by noting the solutions for x_1 , x_2 , x_3 , x , γ , γ_2 and λ. As expected all these variables are found to vary harmonically with a frequency same as the exciting frequency f. The frequency of excitation is varied and the maximum values of these variables are noted. For the sake of convenience in discussion, the variables are expressed in non dimensional forms as

$$\tilde{x}_1 = \frac{x_1}{y_{m_1}} \;,\; \tilde{x}_2 = \frac{x_2}{y_{m_1}} \;,\; \tilde{x}_3 = \frac{x_3}{y_{m_1}} \;,\; \tilde{x} = \frac{x}{y_{m_1}} \;,\; \tilde{\gamma} = \frac{\gamma}{y_{m_1}} \;,\; \tilde{\gamma}_2 = \frac{B\gamma_2}{y_{m_1}} \;,\; \tilde{\lambda} = \frac{L_2\lambda}{y_{m_1}} \quad (17)$$

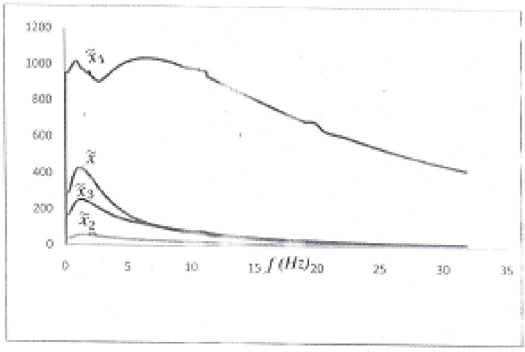

Fig 6 Frequency response

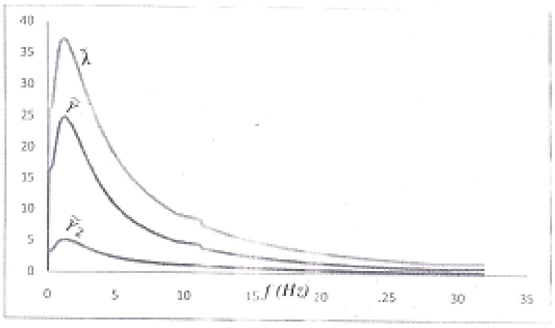

Fig 7 Frequency response

Figures 6 and 7 show the frequency responses using the non dimensional variables. It may be observed from Figure 6 that x_1, x_2, x_3 and x, are showing their peak responses in the region of the lowest frequency 1.796Hz. Figure 7 shows that variables x, γ and λ are also having the peak responses at 1.793Hz. In the remaining part, all the variables are showing low responses. This shows that the first frequency is the most important one, in the practical point of view. It may be observed from Figure 6 that the magnitude of the response of $\tilde{x}_1$ is much higher than the responses of $\tilde{x}_2$, $\tilde{x}_3$ and $\tilde{x}$. The reason could be that the excitation given is at the right side front wheel. The response of $\tilde{x}$ is nearly more than half of that of $\tilde{x}_1$ at the first resonance zone. The $\tilde{x}$ is in between $\tilde{x}_1$ and the other two responses $\tilde{x}_2$ and $\tilde{x}_3$. It may be observed from Figure 7 that $\tilde{\gamma}$ and $\tilde{\lambda}$ are having almost a same kind of response with $\tilde{\lambda}$ being greater than $\tilde{\gamma}$. It is also noted that $\tilde{\gamma}_2$ is smallest of all.

VII. CONCLUDING REMARKS

The work presented in this paper may be summarized as follows.

i. A semi independent suspension system is considered and the motion is described using seven independent coordinates.

ii. The equations of motion are derived. It is found that the seven equations of motion are having considerable amount of coupling and all equations are to be handled together.

iii. Considering only the undamped part of the system of equations, the natural frequencies are obtained.

iv. The first three frequencies are found to be distinct and important. The higher frequencies are found to be very close to each other. The natural frequencies are comparable in magnitude with the frequencies obtained for conventional suspension system [21].

v. Frequency responses are obtained by giving harmonic inputs at one wheel and study the dynamic behaviour of the entire system. The system of equations of motion is solved numerically for this purpose.

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