# Minimization of Human Body Responses Due to Automobile Vibrations in Quarter Car and Half Car Models Using PID Controller

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***Abstract:****Vibration characteristics due to parametric changes are investigated for heavy vehicle suspension system. Biodynamic response of the lumped parameter passenger seat model coupled with simplified models of ground vehicles is investigated. This paper presents an optimal vehicle and seat suspension design for a quarter car and half car vehicle models to reduce human body vibrations. The model is simulated by MATLAB/SIMULINK to observe the vibration phenomenon. Ride Comfort is investigated by amplitude of vibration and ride comfort level. The application of (Proportional – Integral - Derivative) PID Controller is done to control the vibrations of suspension system. In solving the problem, usage of PID controller have consistently found near optimal solutions for different road excitations of sinusoidal random inputs. The solutions obtained were compared to the ones of passive suspensions. The ride behaviors in terms of vertical accelerations of different segments of the human are compared with and without the use of PID controller and are obtained optimum results.*

**Keywords:***control systems, matlab simulation, active suspension, ride comfort*

## I. Introduction:

In a vehicle with a suspension, sprung mass (or sprung weight) is the portion of the vehicle's total mass that is supported above the suspension. The sprung weight typically includes the body, frame, the internal components, passengers, and cargo. In a ground vehicle with a suspension, the unsprung mass is the mass of the suspension, wheels or tracks and other components directly connected to them, rather than supported by the suspension. Unsprung weight includes the mass of components such as the wheel axles, wheel bearings, wheel hubs, tires, and a portion of the weight of drive shafts, springs, shock absorbers, and suspension links. The larger the ratio of sprung weight to unsprung weight, the less the body and vehicle occupants are affected by bumps, dips, and other surface imperfections such as small bridges. However, a large sprung weight to unsprung weight ratio can also be deleterious to vehicle control. The purpose of a car suspension is to maximize the friction between the tires and the road surface, to provide steering stability with good handling and to ensure the comfort of the passengers. The roads were not perfectly flat but are subjected to subtle imperfections that can interact with the wheels of a car. It's these imperfections that apply forces to the wheels. According to Newton's laws of motion, all forces have both magnitude and direction. A bump in the road causes the wheel to move up and down perpendicular to the road surface. The magnitude, of course, depends on whether the wheel is striking a giant bump or a tiny speck. Either way, the car wheel experiences a **vertical acceleration** as it passes over an imperfection.

Without an intervening structure, all of wheel's vertical energy is transferred to the frame, which moves in the same direction. In such a situation, the wheels can lose contact with the road completely. Then, under the downward force of gravity, the wheels can slam back into the road surface. What we need is a system that will absorb the energy of the vertically accelerated wheel, allowing the frame and body to ride undisturbed while the wheels follow bumps in the road.As a consequence, in addition to performing analytical or numerical studies, a good understanding of the relation between exposure to whole-body human vibration and comfort requires extensive laboratory and real environment experimental studies on the response of the human body to single or multi frequency deterministic excitations or appropriately selected random forcing. Due to its large practical importance, there is a vast literature devoted to the subject. However, most of the previous studies on ride comfort in road vehicles employed either simplified (lumped parameter) or more involved (finite-element) mechanical models of the driver- or the passenger-seat subsystem, without considering the effects from the coupling with the dynamics of the vehicle. Hence, to control the dynamic tire load with acceptable suspension working space to enhance the vehicle stability and safety, the use of "PID Controller" is introduced.

The performance of the passive and active suspension were compared and proved that active suspension has greater damping capability to reduce the pick overshoot of sprung mass [7]. Experiments were done practically with humans and different body postures such that the vibration effect can be analyzed [9, 10]. It showed that seating position is an important factor and the contact between the seat and pelvis plays a dominant role, resulting in nonlinear type vibrations [1]. The mathematical model of the vehicle and human body were developed and the vibration characteristics were investigated [1,2], as to become aware of the critical areas of the vibrations impact at different road inputs of sinusoidal, harmonic and transient type road profiles. This is to analyze the ride comfort in the accepted frequency range of travel [4]. A controller is thereby needed to be designed to control the impact of vibrations from the vehicle body to the human [5]. The use of Genetic algorithm is done to reduce the effect transmission of vibration pulses from vehicle wheels to different parts of human, but was not so accurate in results [3]. In the part of research, PID (Proportional Integral Derivative) Controller is found to be effective in attaining at good results. Hence, it is designed [5] and is integrated to the vehicle (either quarter car or half car) and tested for results [6, 7, 11, 12]. Comparative analysis shown that overshoot of vibration pulses is reduced and also steady state is attained at early range.Hence, in this paper, the main objective is to develop the mathematical model of the 2 DOF Quarter Car and 4 DOF Half Car Models integrated with 4 DOF Human Body and also a PID Controller is attached to reduce the impact of vibrations at different areas of influence.

## II. Mathematical Model Formulation:

***A. Proposed model:***The following assumptions have been considered while modeling the vehicle:

- Center of gravity of all masses lies in central plane so that the vehicle is assumed to possess longitudinal plane of symmetry.
- All springs and dampers are assumed to be linear, viscous dampers with constant coefficients.
- Overall structure of the vehicle deformation is negligible. All mass assemblies are assumed as rigid bodies and Suspension system joint friction is considered to be negligible in comparison to shock absorber damping with suspension deflections to be small.
- The road wheels are rigid discs each making point contact with the road, and they roll without any slip on a flat level road surface.

The above assumptions hold good for both the "quarter car" and "half car" models. For the quarter car motion, there exists "two degree of freedom", which are the "bounce motions of sprung and unsprung masses". It is followed by vertical motion of the vehicle seat and the human body elements. The "pitch motion of sprung mass" is included in the Half Car model.

***B. Equations of motion - Quarter Car – Proposed Model:***A Quarter Car model with 2 DOF is considered, taking into account the bounce motions of the vehicle body and the tire.

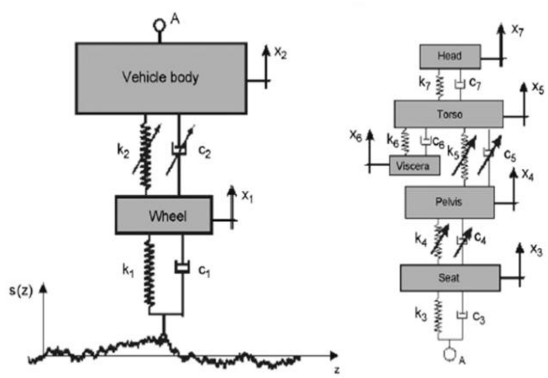



**Fig 1. Quarter Car Model -Biodynamic Model**

The human body has a 4-DOF that proposed by C.Papalukopoulos and S.Natsiavas [1]. In this model, the seated human-body was constructed with four separate mass segments inter- connected by four sets of springs and dampers. The four masses represent the body segments pelvis ($m_4$), torso ($m_5$), viscera ($m_6$), Head ($m_7$) respectively. The Vehicle body mass is $m_2$ and the mass of the tire is $m_1$ respectively. The schematic of the model is shown in the below figure and the biomechanical properties of the model are listed in the table.

**Table 1: Properties of Quarter Car coupled with Human Body model [1]:**

| S.No: | Part of the model | Mass, kg | Symbol | Stiffness, N/m | | Damping Co-efficient, Ns/m | |
|---|---|---|---|---|---|---|---|
| 1. | Wheel (Unsprung Mass) | 60 | $m_1$ | $K_1$ | 200000 | $C_1$ | 7 |
| 2. | Vehicle Body (Sprung mass) | 375 | $m_2$ | $K_2$ | 15000 | $C_2$ | 475 |
| 3. | Seat | 8 | $m_3$ | $K_3$ | 36000 | $C_3$ | 2000 |
| 4. | Pelvis | 29 | $m_4$ | $K_4$ | 33000 | $C_4$ | 488 |
| 5. | Torso | 21.8 | $m_5$ | $K_5$ | 6800000 | $C_5$ | 2674 |
| 6. | Viscera | 6.8 | $m_6$ | $K_6$ | 2900 | $C_6$ | 165 |
| 7. | Head | 5.5 | $m_7$ | $K_7$ | 202200 | $C_7$ | 210 |

When both the vehicle and the human models shown in Fig. 1 are assumed to possess linear characteristics, the equations of motion of the resulting coupled model can be put in the classical matrix form, $M\ddot{x} + C\dot{x} + Kx = 0$

Hence, the equations of motion would be,

$$M_1\ddot{x}_1 + C_1\dot{x}_1 + C_2(\dot{x}_1 - \dot{x}_2) + K_1x_1 + K_2(x_1 - x_2) = 0 \quad \text{---- (1)}$$

$$M_2\ddot{x}_2 + C_2(\dot{x}_2 - \dot{x}_1) + C_3(\dot{x}_2 - \dot{x}_3) + K_2(x_2 - x_1) + K_3(x_2 - x_3) = 0 \quad \text{---- (2)}$$

$$M_3\ddot{x}_3 + C_3(\dot{x}_3 - \dot{x}_2) + C_4(\dot{x}_3 - \dot{x}_4) + K_3(x_3 - x_2) + K_4(x_3 - x_4) = 0 \quad \text{---- (3)}$$

$$M_4\ddot{x}_4 + C_4(\dot{x}_4 - \dot{x}_3) + C_5(\dot{x}_4 - \dot{x}_5) + K_4(x_4 - x_3) + K_5(x_4 - x_5) = 0 \quad \text{---- (4)}$$

$$M_5\ddot{x}_5 + C_5(\dot{x}_5 - \dot{x}_6) + C_4(\dot{x}_5 - \dot{x}_4) + C_7(\dot{x}_5 - \dot{x}_7) + K_5(x_5 - x_6) + K_4(x_5 - x_4) + K_5(x_5 - x_6) + K_7(x_5 - x_7) = 0 \quad \text{---- (5)}$$

$$M_6\ddot{x}_6 + C_6(\dot{x}_6 - \dot{x}_5) + K_6(x_6 - x_5) = 0 \quad \text{---- (6)}$$

$$M_7\ddot{x}_7 + C_7(\dot{x}_7 - \dot{x}_5) + K_7(x_7 - x_5) = 0 \quad \text{---- (7)}$$

### *C. Half Car – Proposed Model:*

A Half Car model with 4 DOF is considered, taking into account the pitch motion of the vehicle body. The human body has a 4-DOF that proposed by Wael Abbas et al [3]. In this model, the seated human-body was constructed with four separate mass segments inter- connected by five sets of springs and dampers. The four masses represent the body segments pelvis ($m_4$), lower torso ($m_5$), upper torso ($m_6$), Head ($m_7$) respectively. The Vehicle body mass is $m_3$and the mass of the rear wheel is $m_1$and front wheel is $m_2$respectively. The schematic of the model is shown in the below figure and the biomechanical properties of the model are listed in the table.

**Table 2: Properties of Half Car coupled with Human Body model [3]:**

| S.No: | Part of the model | Mass, kg | Symbol | Stiffness, N/m | | Damping Co-efficient, Ns/m | |
|---|---|---|---|---|---|---|---|
| 1. | Rear Axle | 54.32 | $m_1$ | $K_1$ | 155900 | $C_1$ | 0 |
| 2. | Front Axle | 28.58 | $m_2$ | $K_2$ | 155900 | $C_2$ | 0 |
| 3. | Vehicle Body | 505.1 | $m_s$ | | | | |
| 4. | Seat | 35 | $m_3$ | $K_3$ | 15000 | $C_3$ | 150 |
| 5. | Pelvis | 36 | $m_4$ | $K_4$ | 49340 | $C_4$ | 2475 |

| | | | | | | | |
|---|---|---|---|---|---|---|---|
| 6. | Lower Torso | 5.5 | $m_5$ | $K_5$ | 164000 | $C_5$ | 1239.1 |
| 7. | Upper Torso | 15 | $m_6$ | $K_6$ | 100000 | $C_6$ | 200 |
| 8. | Head | 4.17 | $m_7$ | $K_7$ | 166900 | $C_7$ | 310 |

- Vehicle Body Moment of inertia = 651 kg.$m^2$
- Distance between Centre of gravity and front axle = 1.098m
- Distance between Centre of gravity and rear axle = 1.468m

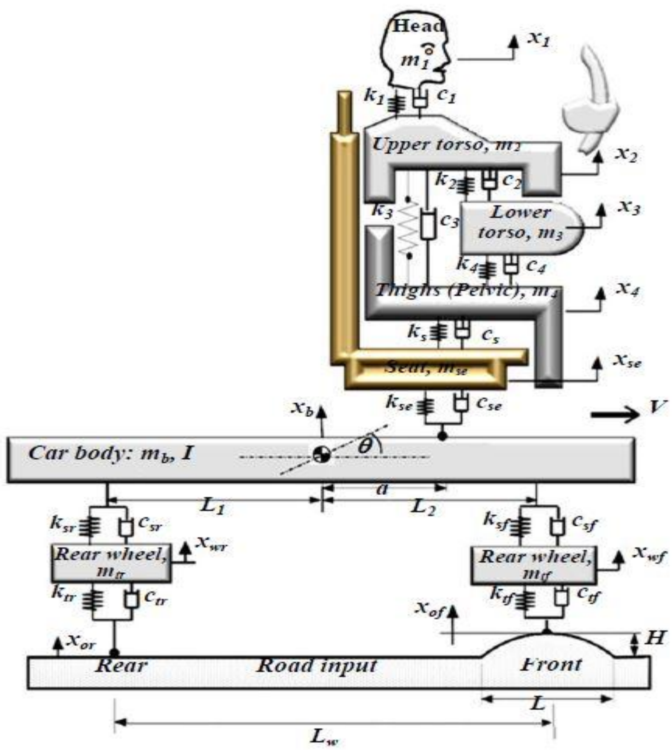



**Fig 2.Half car coupled to Human body**

***D. Equations of motion - Quarter Car – Proposed Model:*** The equations of motion would be,

$$M_1\ddot{y}_1 + (C_1 + C_3)\dot{y}_1 + C_3a\dot{\theta} – C_3\dot{z}_s + (K_1 + K_3)y_1 +K_3a\theta-K_3z_s = 0 \quad \text{---- (1)}$$

$$M_2\ddot{y}_2 + (C_2 + C_4)\dot{y}_2 – C_4b\dot{\theta} – C_4\dot{z}_s + K_4 + Z_s+ K_4 b\theta - (K_4 + K_2)y_2 = 0 \quad \text{---- (2)}$$

$$M_s\ddot{z}_s + (C_4 + C_3 + C_5)\dot{z}_s - K_3y_1 - K_5y_3 + ( bC_4 – aC_3 –dC_5)\dot{\theta} - C_4\dot{y}_2 - C_3\dot{y} + (K_4+K_4+K_5)Z_s+ (bK_4-aK_3-dK_5) Z_s + (bK_4-aK_3-dK_5)\theta - C5\dot{y}_3 - K_4y_2 = 0 \quad \text{---- (3)}$$

$$I\ddot{\theta} + (C_4b-aC_3+dC_5)\dot{z}_s - K_5dy_3 - C_5d\dot{y}_3 + (b^2C_4+a^2C_3+d^2C_5)\dot{\theta} - bC_4\dot{y}_2 + aC_3\dot{y}_1 - bK_4y_2 + (bK_4-aK_3+dK_5)Z_s+aK_3y_1 + (b^2K_4 +a^2K_3+d^2K_5)\theta = 0 \quad \text{---- (4)}$$

$$M_3\ddot{y}_3 + (C_3+C_6) \dot{y}_3 - C_6\dot{y}_4 - C_3\dot{z}_s + dC_3\dot{\theta} + (K_3+K_6) y_3 + dK_3\theta - K_3Z_s - K_6y_4 = 0 \quad \text{---- (5)}$$

$$M_4\ddot{y}_4 + (C_6+C_7+C_8)\dot{y}_4 – C_7\dot{y}_5 - C_8\dot{y}_6 - C_6\dot{y}_3 - (K_6+K_7+K_8)y_4 - K_7y_5- K_8y_6 - K_6y_3 = 0 \quad \text{---- (6)}$$

$M_5\ddot{y}_5 - C_9\dot{y}_6 + (C_9 + C_7)\,\dot{y}_5 - K_9\dot{y}_6 - C_7\dot{y}_4 - C_7\dot{y}_4 - K_7\dot{y}_4 + (K_9+K_7)y_5 = 0$ ---- (7)

$M_6\ddot{y}_6 - C_{10}\dot{y}_7 + (C_8+C_9+C_{10})\dot{y}_6 - C_9\dot{y}_5 - K_{10}y_7 + (K_8+K_9+K_{10})\,y_6 - K_9y_5 - C_8\dot{y}_4 - K_8y_4 = 0$ ---- (8)

$M_7\ddot{y}_7 + C_{10}\dot{y}_7 - C_{10}\dot{y}_6 + K_{10}y_7 - K_{10}y_6 = 0$ ---- (9)

## III. Input Profile Excitations:

### A. Random Road Input:

In this work, sinusoidal and random road profile excitations are adopted to evaluate the proposed system. For the random profile, the relationship of the road Power Spectral Density(PSD) to the spatial frequency is approximated by the equation, $S_g(\Omega) = C_{sp}\Omega^{-N}$

Where $S_g(\Omega)$ is the power spectral density function, $C_{sp}$and N are constants, Ω is the spatial frequency

Random road profile has been generated in time domain in MATLAB workspace using sinusoidal approximation method in which a single track can be approximated by a superposition of k sine waves.The sine waves are then summed to create a surface which is used as the road profile input.The longitudinal position of each contact point with the road surface, front and rear axles and wheel contact, will be accessed at each moment in time.This longitudinal position can then be related to a vertical displacement of the random road profile at that particular point. Thus, the road inputs to the vehicle model can be established.

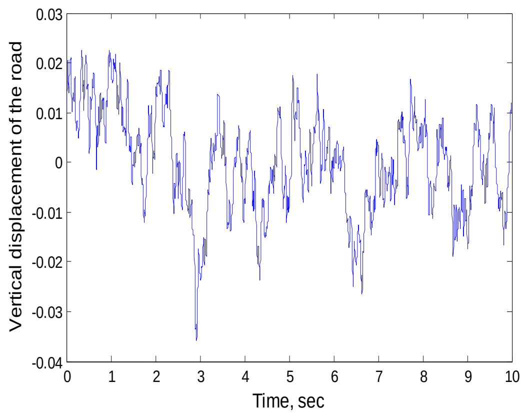



**Fig 3. Random road profile in time domain**

### B. Sinusoidal Road Input:

The vehicle model under examination is made to pass through constant horizontal speed, $V_o$, over an isolated road irregularity. Among all the possible choices that are available, one of the most characteristic types of irregularities correspond to a road with a harmonic profile of the form:

$$s(z) = \hat{x}_g \sin(\Pi z/l) \text{ for } 0 \le z \le l$$

Where, 'l'is the "length of the road irregularity". This induces a base excitation on the wheel of the car with the form:

$$s(z) = \hat{x}_g \sin(\omega t+\Phi) \text{ for } 0 \le t \le T$$

where, $\Phi$ is an arbitrary phase angle, while the parameters

$$\omega = \Pi V_o / l \text{ and } T = l/V_o$$

correspond to the forcing frequency and the time it takes for the vehicle to run over the irregularity respectively. In a half car, the rear wheel follows the same front wheel track with a time delay of 'T', which is explained above as the ratio of "wheel base to the vehicle speed".The sine input profile of frequency 10Hz i.e. 62.83 rad/s and the frequency of 78.95m is given as road input to the wheels of quarter car and half car models in Simulink model using the "Sine function" block.

## IV. Optimal Linear Seat Suspension Design:

### A. Numerical Solutions:

The displacement, velocity, and acceleration for the model in terms of time domain are obtained by solving equations of motion represented by Equations of Quarter Car and Half Car models, using Mat lab/Simulink software (R2011a). The initial conditions are assumed at equilibrium position. In this assumption, the driver is seated, where the input excitation has not been provided to the seat. Therefore, the initial velocity and displacement for each mass are equal to zero.

### *B. Optimization through PID Controller - PID Controller Design*

The model is represented in the form of transfer function and is serially coupled with PID Controller. The output is fed back to the input to form a closed loop system as shown in the figure 4.

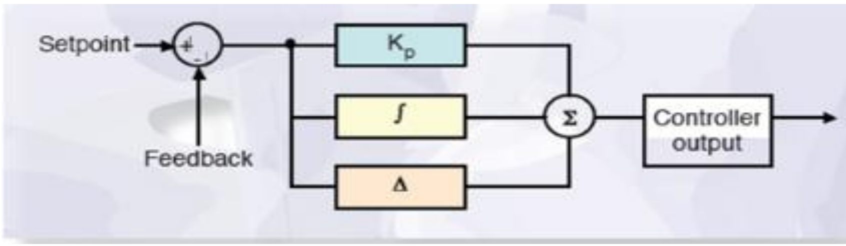



**Fig 4. PID Controller**

The PID Controller output in relation to the input and control gains is expressed as, Error, $e(t) = Y(t) - R(t)$

$$U(t) = Kp\, e(t) + K_i \int e(t)\, dt + K_d \frac{d}{dt} e(t)$$

For the Quarter car model considered, the PID controller would be as shown in the figure 5.

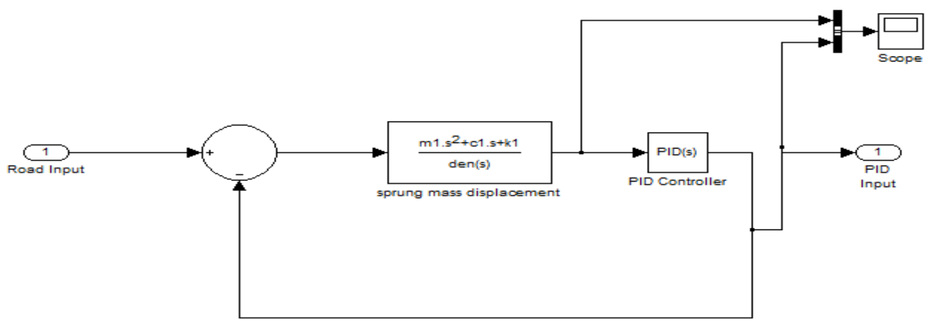
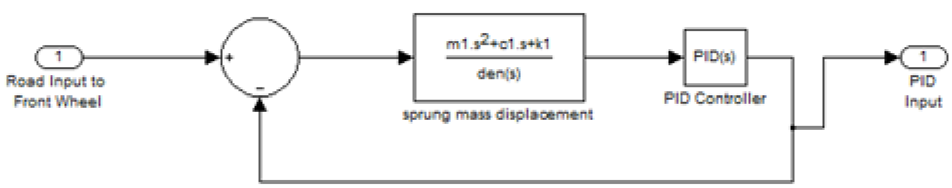



**Fig 5. Simulink PID Controller of a Quarter Car Model**

The PID parameters were set by applying "Ziegler – Nicholas" method and also Direct tuning. The parameters were set as P = 94.205, I = 15.741, D = 51.358. The parameters above were used in decreasing the acceleration responses and obtaining the steady state with less number of pulses.Similarly, the Half Car model is derived in terms of transfer function, two of the outputs are feedback to the input to form the double closed loop system. It would be as shown in the figure. The PID parameters set for tuning are found to be P = 152.32, I = 68.921, D = 19.826.



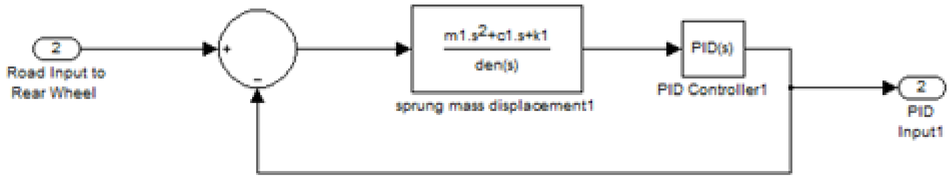



**Fig 6. Simulink PID Controller for Half Car Model**

The effect of increasing the different parameters on the four main system characteristics is summarized in the table 3.

**Table 3: PID Influencing parameters**

| Response | Rise Time | Overshoot | Settling Time | Steady State Error |
|---|---|---|---|---|
| $K_p$ | Decrease | Increase | Minor | Decrease |
| $K_i$ | Decrease | Increase | Increase | Removes |
| $K_d$ | Minor | Decrease | Decrease | Minor |

## V.RESULTS AND DISCUSSIONS:

The drivers and passengers of vehicles are subjected to vibrations that are directly related to the characteristics of the vehicle and of the road surface. These car body vibrations are transmitted to the occupants along the seat, pelvis, torso, viscera part and to the head. The system response is analyzed considering both active (with the PID) and the passive system simultaneously in order to ascertain the performance of one over the other. Settling time and maximum overshoot are used as parameters in establishing the effectiveness and robustness of the models.

***A. For Quarter Car Model:***From figure at sinusoidal and random inputs, it can be deduced that the maximum suspension travel occurred at the maximum heights of the road disturbance for both the models.

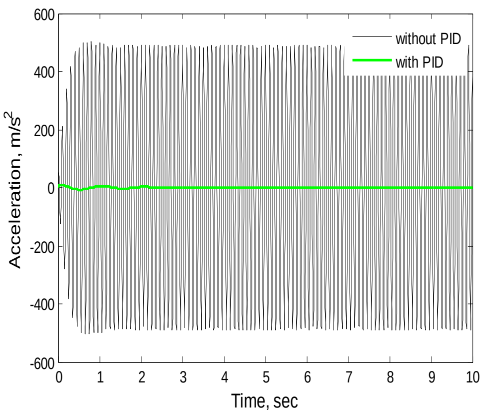



**Fig 7.Sinusoidal Input - PID Wheel acceleration**

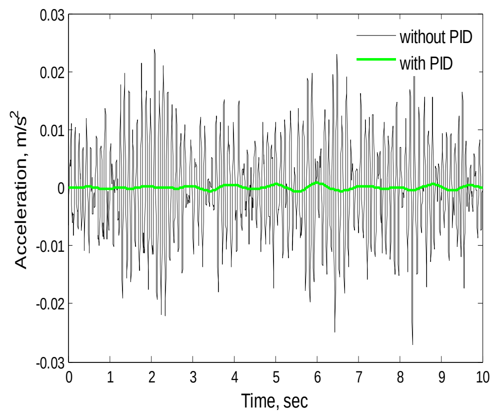



**Fig 8. Random Input – PID Wheel acceleration**

It is worth noting that the active suspension not only lessens the maximum overshoot but also hastens the settling time. For the vehicle body acceleration as shown in the figure, it can be seen that the use of active suspension decreased the overshoot from 30.6 $m/s^2$ to 9.4 $m/s^2$. This decrease aids in speeding up the system stability as it reaches the steady state much earlier than the passive suspension.

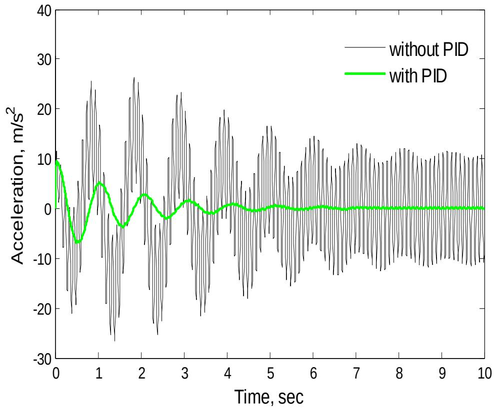



**Fig 9.Sinusoidal Input - PID Vehicle Body acceleration**

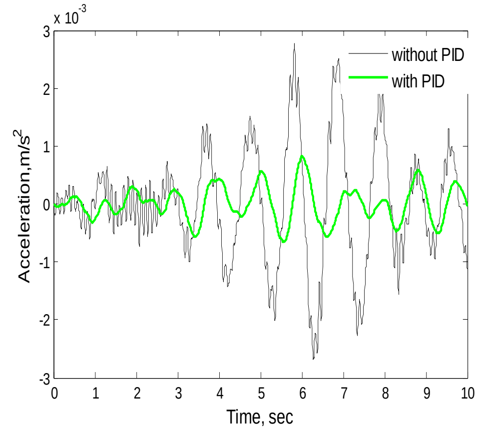



**Fig 10. Random Input – PID Vehicle Body acceleration**

In random type road input, the peak is reduced by 11%. This helps in lesser influence of vibrations at other parts of the body considered. The non- linearity effect is observed at the contact at the seat and the pelvis parts. Its influence can be seen from the figures.

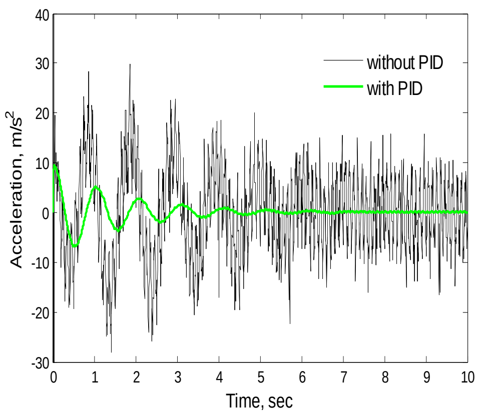



**Fig 11.Sinusoidal Input – PID Seat acceleration**

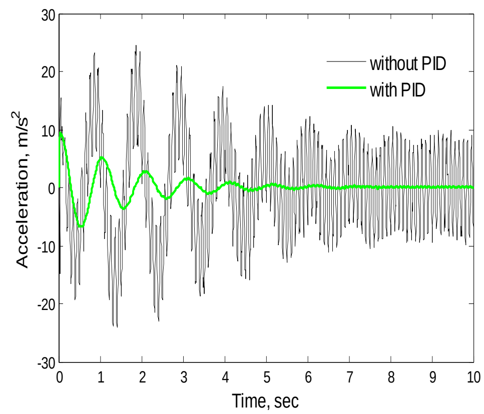



**Fig 12. Sinusoidal Input – PID Pelvis acceleration**

The vehicle body vibrations are transmitted as off to the occupant through the seat. The contact between the both plays a major role when crossing a bump or on a random road. Hence, the role of the controller observed here is that, it decreased the overshoot by 91.5% from the maximum peak of 112.5m/s$^2$ to 9.5 m/s$^2$. Also the steady state is attained in a minimum of 5sec wherein further vibrations are nullified or reduced to minimum such that its influence at the other parts of the human is less observed.

The next part to which the vibrations are transmitted is the "Pelvis" part. Some of the vibration pulses were observed by the seat and then are transmitted to the pelvis. The use of PID Controller reduced the peak acceleration by 81.24% and steady state is attained in a minimum of 4.5 sec. Fig 13. shows the influence of vibrations at "Torso" part of the human. It is the backrest part and the supporting part for neck and head. Hence, it should be less prone to fatigue. This is accomplished with the use of PID Controller. 92.16% of reduction in peaks is observed and the steady state is attained in a minimum of 4.3 sec as shown in the figure below.

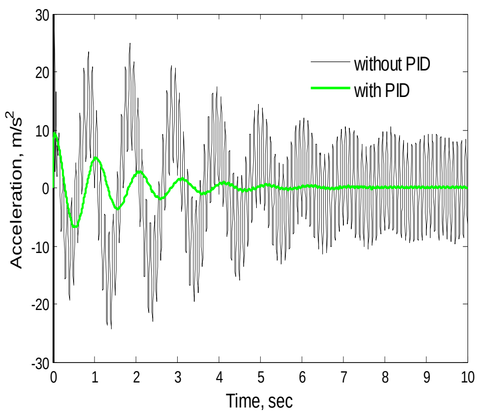



**Fig 13.Sinusoidal Input - PID Torso acceleration**

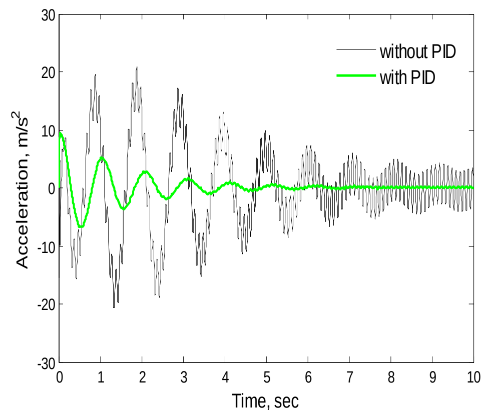



**Fig 14. Sinusoidal Input - PID Viscera acceleration**

"Viscera" is the part that is next prone to vibrations. It is seen from the figure that the intensity of vibrations decreased comparatively. The controller used helps in obtaining the transient output at an earlier steady state. It is shown in the figure. "Head" is another critical area prone to shock impulses. This is to be reduced and is done with PID. It can be shown from the figure. The intensity of vibrations and also the maximum overshoot is reduced from 111.5 $m/s^2$ to 9.45 $m/s^2$. This decrease leads the human to reduce fatigue, drive safer, thereby ride comfort is increased.

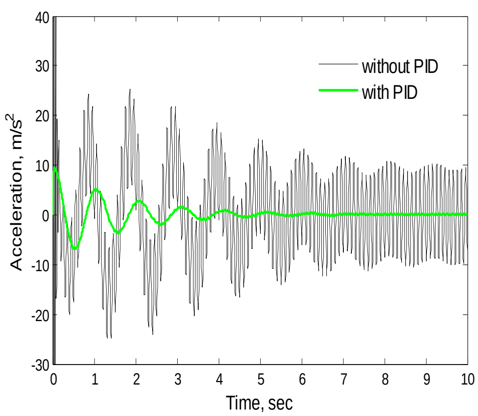



**Fig 15.Sinusoidal Input - PID Head acceleration**

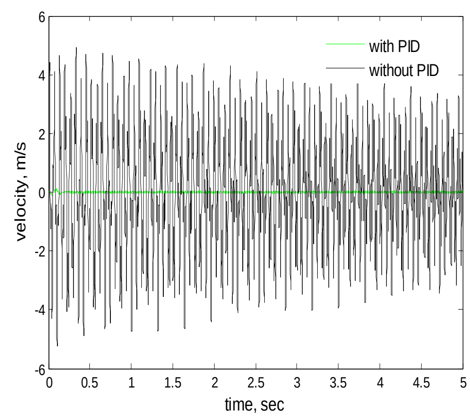



**Fig 16. Sinusoidal Input - Half Car - PID Pitch Velocity**

***B. For Half Car Model:*** Similarly, for the random input of road excitation, the maximum overshoot or the peak values of excitations obtained were reduced to minimum with the help of the PID Controller adopted. This can be observed in the same way in a Half Car that is coupled human body. The most annoying or the deterministic motion in the half car is the "Pitch motion". It is mainly affected in terms of velocity. The role of the controller in reducing the vibratory motion as in the case of pitch velocity can be seen by the above figure16.The high induced vibration pulses create impacts on the ride comfort of the occupants. This is reduced to minimum as shown in the figure. It also describes the vehicle's road handling capability. The controller tends to restore the system to steady state faster and with less overshoot.

## VI. CONCLUSIONS:

Systematic methodologies were developed for determining response of biodynamic driver-seat lumped parameter models riding on quarter car, half car models. The equations of motion of the coupled dynamical system were set up. Modelling and simulation of quarter car and half car models integrated with human body were discussed. PID controller has been successfully implemented to make the suspension system active. Ziegler-Nichols tuning criterion is used in obtaining the PID controller gains used in the model. Road surface conditions are considered while modeling the input to the system, in which the vehicle is excited by sinusoidal and random road inputs. The performance characteristics and the robustness of both passive and active suspension system are evaluated by considering the maximum overshoot and settling time of the responses. From the simulation result, it has been established that the proposed active suspension systemproved to be more effective in controlling the vehicle oscillation and more robust in restoring the system to its steady state as compared to the passive suspension system. The desired objective is proposed as minimization of multi degree of freedom (DOF) system in achieving the best comfort of the driver. In all simulation runs, it can be observed that,

**Table 4: Percentage reduction in acceleration responses using PID controller for Quarter Car for Sinusoidal input**

| S.No: | Component | Without PID Controller | With PID Controller | % Decrease |
|---|---|---|---|---|
| 1. | Wheel | 500 | 9.6 | 98.08% |
| 2. | Vehicle Body | 30.6 | 9.464 | 69.07% |
| 3. | Seat | 112.3 | 9.5 | 91.5% |
| 4. | Pelvis | 50.38 | 9.45 | 81.24% |
| 5. | Torso | 120.6 | 9.45 | 92.16% |
| 6. | Viscera | 25 | 9.45 | 62.2% |
| 7. | Head | 111.50 | 9.44 | 91.8% |

**Table 5: Percentage reduction in acceleration responses using PID controller for Quarter Car for Random input**

| S.No: | Component | Without PID Controller | With PID Controller | % Decrease |
|---|---|---|---|---|
| 1. | Wheel | 0.0223 | $6.85 \times 10^{-4}$ | 3.06% |
| 2. | Vehicle Body | $6.85 \times 10^{-3}$ | $6.85 \times 10^{-4}$ | 11% |
| 3. | Seat | $4 \times 10^{-3}$ | $5 \times 10^{-4}$ | 14.28% |
| 4. | Pelvis | $2.8 \times 10^{-3}$ | $6.85 \times 10^{-4}$ | 32.3% |
| 5. | Torso | $2.5 \times 10^{-3}$ | $5 \times 10^{-4}$ | 25% |
| 6. | Viscera | $2.68 \times 10^{-3}$ | $6.85 \times 10^{-4}$ | 34.3% |
| 7. | Head | $2.5 \times 10^{-3}$ | $6.8 \times 10^{-4}$ | 37.3% |

**Table 6: Percentage reduction in acceleration responses with using PID controller for Half Car for Sinusoidal input**

| S.No: | Component | Without PID Controller | With PID Controller | % Decrease |
|---|---|---|---|---|
| 1. | Rear Wheel | 70.88 | 0.5 | 99.29% |
| 2. | Front wheel | 72.5 | 0.5 | 99.31% |
| 3. | Vehicle Body | 12.5 | 0.73 | 94.16% |
| 4. | Pitch | 2.715 | 0.7 | 74.21% |
| 5. | Seat | 1.724 | 0.773 | 55.16% |
| 6. | Pelvis | 1.65 | 0.771 | 53.27% |
| 7. | Lower Torso | 1.62 | 0.77 | 52.46% |
| 8. | Upper Torso | 1.125 | 0.73 | 35.1% |
| 9. | Head | 1.014 | 0.77 | 24% |

The results of optimal seat and vehicle suspensions system by adopting PID Controller has successfully managed improving for all the dynamic performance parameters. The numerical results and the plots indicate that optimal system is less oscillatory, and have lower values of maximum over shoots than passive suspension system. This is directly related to driver fatigue, discomfort, and safety. These results are encouraging and suggest that PID Controller can be easily used in other complex and realistic designs often encountered in the engineering.

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