International Journal of Engineering Applied Sciences and Technology, 2017
Vol. 2, Issue 6, ISSN No. 2455-2143, Pages 44-50
Published Online in IJEAST (http://www.ijeast.com)

VIBRATION ANALYSIS OF SEMI ACTIVE SUSPENSION SYSTEM FOR AN AUTO RICKSHAW BY USING MATLAB/SIMULINK

P. Srinivasa Rao
Department of Mechanical Engineering
Anits, Visakhapatnam

M. Amareswari Reddy
Department of Mechanical Engineering
Anits, Visakhapatnam

M. Ramkrishna
Department of Mechanical Engineering
Anits, Visakhapatnam

V.S.C Varma
Department of Mechanical Engineering
Anits, Visakhapatnam

***Abstract*— An auto rickshaw is a three wheeled motor vehicle with one front steering wheel. Auto rickshaws are most commonly found in developing countries as they are a very cheap form of transportation due to low price, low maintenance cost, and low operation costs. Auto rickshaws needs to be developed to take the step into the 21-century. To develop a new product in the modern world one of the most important challenges is safety of the users. As for the automotive industries this challenge has a great importance since the outcome can be devastating. One important category from the safety point of view is the vehicle suspensions, as the suspensions control the movement of the wheels and thus keeping the vehicle on the road.**

Suspension system design is a challenging task for the automobile designers in view of multiple control parameters, complex objectives and stochastic disturbances. The objective of this project is to develop a MATLAB/SIMULINK model of one third auto rickshaw suspension to analyze the ride comfort and vehicle handling. A theoretical model of the human seated model is developed in order to simulate the vertical motion of the Passenger in an auto rickshaw when the vehicle passing over various road disturbances. This project used a new approach in designing the suspension system which is semi-active suspension. The semi active suspension system uses a varying damping force as a control force. Ride comfort of off-road vehicles can be estimated by replacing the normal passive dampers in the vehicle suspension system with controllable, two-state, semi-active dampers. Skyhook controller is developed for controlling the damping force of the suspension system. Comprehensive analysis of passive and semi-active suspension system in terms of human body vibrational displacements and accelerations has been done. The semi-active suspension with skyhook controller reduces the sprung mass acceleration and displacement hence improving the passengers comfort.

Keywords—damper, semi active, skyhook,comfort

Abbreviations and Acronyms

Z_{se} Vertical displacement of the seat,
Z_S Vertical displacement of the sprung mass,
Z_u Vertical displacement of the unsprung mass
Z_r Road displacement
M_{se} Masse of the seat,
M_S Sprung mass
M_u Unsprung mass.
K_{se} Spring stiffness of seat
Cp_{se} Damping coefficient of seat
Cp_S and K_S Suspension damping and stiffness
K_t Tire stiffness.

I. INTRODUCTION

Suspension is a term that given for a system that contained spring, shock absorber and few linkages that connected the body to vehicles to the tire. Suspension system can be divided into three categories which is passive, semi-active and fully active suspension system. This suspension system categorizing depends on the external power input and/or the control bandwidth into the system. A passive suspension system is conventional suspension system consist of non-controlled spring and shock-absorbing damper which means the damping criteria is fixed. Semi-active suspension system has equally same configuration as the passive suspension but with a controllable damping rate for the shock-absorbing damper. An active suspension is one in which the passive components are augmented by actuators that supply additional force.

II. PROBLEM STATEMENT

Passive suspension system is very common in the passenger's auto rickshaw vehicles. The main problem for passive suspension system is it cannot give comfort to the Passengers without sacrificing the traction force between the tire and the road. Figure shows the relation of ride comfort and vehicle stability in a vehicle passive suspension system design. The passive suspension system performance also is variable subject to road profile and added passengers weight. It is because passive suspension system has fixed spring constant and damping coefficient thus its damping force is not adjustable. This project developed a vehicle suspension system as semi active suspension system that can adjust its damping force by replacing standard hydraulic damper with a continuously adjustable damper or on and off adjustable damper to overcome the problem. The main focus is to make the vehicle passenger feel more comfortable without sacrificing the vehicle handling abilities.

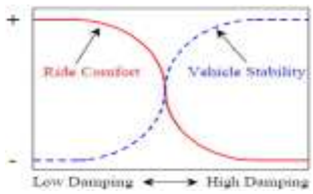

Fig 1: Passive Suspension Design compromise

III. DYNAMIC MODELS OF A QUARTER CAR

Physical models for the investigation of vertical dynamics of suspension systems are most commonly built on the quarter-car model.

3.1 Passive system model along with seat

The piecewise linear model of the passive viscous damper used in the simulation is shown in fig 2.
Equation of motion for combined occupant and seat mass is given as:

$$M_{se}\ddot{Z}_{se} + K_{se}(Z_{se} - Z_s) + Cp_{se}(\dot{Z}_{se} - \dot{Z}_s) = 0$$

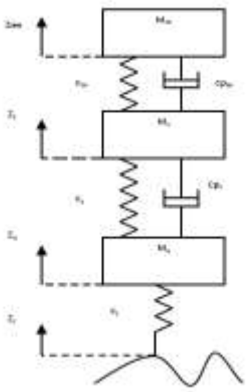

Fig 2: One third Auto Rickshaw passive model.

Equation of motion for sprung mass is

$$M_s\ddot{Z}_s + K_s(Z_s - Z_u) + Cp_s(\dot{Z}_s - \dot{Z}_u) - K_{se}(Z_{se} - Z_s) - Cp_{se}(\dot{Z}_{se} - \dot{Z}_s) = 0$$

Similarly, the equation for unsprung mass is

$$M_u\ddot{Z}_u + K_t(Z_u - Z_r) - K_s(Z_s - Z_u) - Cp_s(\dot{Z}_s - \dot{Z}_u) = 0$$

3.2 Semi-Active Model

Semi-active suspension systems are the adaptation of the damping and/or the stiffness of the spring to the actual demands. Figure 6 shows a schematic diagram of a one third of auto rich shaw semi-active suspension control system. The common concept of semi-active springs is based on a system containing an air spring or hydro pneumatic system.

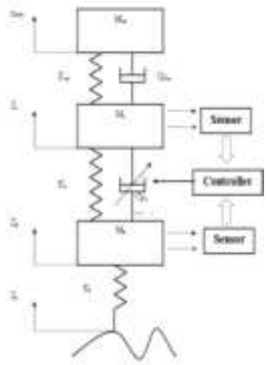

Fig 3: One third of Auto Rickshaw Semi active suspension control system.

3.2.1. Semi active control strategies

There are different controls strategies adopted under the semi active suspension system, each one having its own characteristics

1 .Limited Relative Displacement Control Method

The ideal goal of an optimal suspension is to minimize the sprung mass relative displacement and acceleration. However, these two criteria are in conflict. In general, a suspension system with a small relative displacement corresponds to a high sprung mass acceleration, and a large relative displacement corresponds to a low sprung mass acceleration. For this reason, the control strategy is set in a way that the damper is switched to a high damping ratio when the relative displacement is higher than a specific value and a low damping value otherwise. This on–off control law can be expressed as

$$\zeta_s = \begin{cases} \zeta_{max} & |Z_s - Z_u| \geq 0 \\ \zeta_{min} = 0 & |Z_s - Z_u| < 0 \end{cases}$$

Where
$\zeta\ s$ is the equivalent damping ratio of the suspension system.

$$\zeta_S = \frac{C_{PS}}{\sqrt[2]{K_S M_S}}$$

This method can limit the relative displacement of the suspension by adjusting two parameters, ζ max and ζ min. This is a simple approach and the results are matching with skyhook control .Therefore they are not presented here. In this paper simulations are run for skyhook (SH) and modified skyhook (MSH) methods only.

2. Skyhook Control Method

Skyhook control is a popular and effective vibration control method because it can dissipate system energy at a high rate.
It is typically classified as continuous skyhook control and on-off skyhook control. The on-off Skyhook controller is usually simpler and better suited for the

International Journal of Engineering Applied Sciences and Technology, 2017
Vol. 2, Issue 6, ISSN No. 2455-2143, Pages 44-50
Published Online in IJEAST (http://www.ijeast.com)

industrial applications. In this study, on-off skyhook control is implemented. The control law can be described as follows:
This strategy indicates that if the relative velocity of the body with respect to the wheel is in the same direction as that of the body velocity, then a maximum damping force should be applied to reduce the body acceleration. On the other hand, if the two velocities are in the opposite directions, the damping force should be at a minimum to minimize body acceleration. This control strategy requires the measurement of the absolute Velocity of body.

Skyhook Controller

Semi-active dampers allow for the damping coefficient, and therefore the damping force, to be varied between high and low levels of damping. Early semi- active dampers were mechanically adjustable by opening or closing a bypass valve. The only power required for the damper is the relatively small power to actuate the valve. For this research, a magneto-rheological damper which varies the damping by electrically changing the magnetic field applied to the magneto-rheological fluid is used. With a semi-active damper, the 2DOF model modifies to where the damping coefficient, $C_{ontrollable}$, can be varied in time. This configuration is referred to as a semi-active suspension.

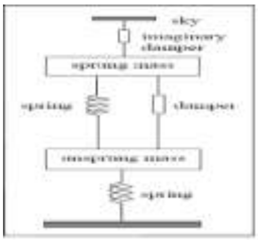

Fig 4: 2 Degree of freedom skyhook Damper Configuration

Once it is decided that a semi-active damper is used, the means of modulating the damper such that it emulates a skyhook damper must be determined. We first define the velocity of the sprung mass relative to the unsprung mass, V_{12}, to be positive when the sprung mass and unsprung mass are separating (i.e., when V_1 is greater than V_2) for the systems. Now assume that for both systems, the sprung mass is moving upwards with a positive velocity V_1. If we consider the force that is applied by the skyhook damper to the sprung mass, we notice that it is in the negative

$$F_{sky} = -C_{sky}V_1$$

Where, F_{sky} is the skyhook force and C_{sky} is the skyhook damping coefficient. Next, is to determine if the semi-active damper is able to provide the same force. If the sprung and unsprung masses in Fig. 3.1 are separating, then the semi-active damper is in tension. Thus, the force applied to the sprung mass is in the negative X_1 direction, or

$$F_{controllable} = -C_{controllable}V_{12}$$

Where $F_{controllable}$ is the force applied to the sprung mass. Since we are able to Generate a force in the proper direction, the only requirement to match the skyhook suspension is

$$C_{controllable} = C_{sky}$$

To summarize, if V_1 and V_{12} are positive, $C_{controllable}$ should be defined as in equation above. Now consider the case in which the sprung and unsprung masses are still separating, but the sprung mass is moving downwards with a negative velocity V_1. In the skyhook configuration, the damping force will now be applied in the upwards, or positive, X_1 direction. In the semi-active configuration, however, the semi-active damper is still in tension, and the damping force will still be applied in the downwards, or negative, direction. Since the semi-active damping force cannot possibly be applied in the same direction as the skyhook damping force, the best that can be achieved is to minimize the damping force. Ideally, the semi- active damper is desired to be set so that there is no damping force, but in reality there is some small damping force present and it is not in the same direction as the skyhook damping force. Thus, if V_{12} is positive and V_1 is negative, we need to minimize the semi-active damping force.
We can apply the same simple analysis to the other two combinations of V_1 and V_{12}, resulting in the well-known semi-active skyhook control policy:

$$\begin{cases} V_1V_{12} > 0 & \mathrm{F_{SA}} = C_{SKY}V_1 \\ V_1V_{12} < 0 & \mathrm{F_{SA}} = 0 \end{cases}$$

Where, F_{SA} is the semi-active skyhook damper force. Equation (4) implies that when the relative velocity across the suspension (V_{12}) and the sprung mass absolute velocity (V_1) have the same sign, a damping force proportional to V_1 is desired. The skyhook damper configuration attempts to eliminate the trade-off between resonance control and high frequency isolation common to passive suspensions . Consider the arrangement . The damper is connected to an inertial reference in the sky. Clearly, this arrangement is fictitious, since for this configuration to be implemented, the damper would have to be connected to a reference point which is fixed with respect to the ground but can translate with the vehicle. Such a suspension mounting point does not exist. The end goal of skyhook control is not to physically implement this system, but to command a controllable damper to cause the system to respond in a similar manner to this fictitious system.
In essence, this skyhook configuration is adding more damping to the sprung mass and taking away damping from the unsprung mass. The skyhook configuration is ideal if the primary goal is isolating the sprung mass from base excitations, even at the expense of excessive unsprung mass motion. An additional benefit is apparent in the frequency range between the two natural frequencies. With the skyhook configuration, isolation in this region actually increases with increasing C_{sky}.

IV. IMPLEMENTATION USING MATLAB/SIMULINK

The simulink block diagram for passive suspension system by means of state space approach. The various matrices are entered in to the simulink block and the response for the given input is obtained. Skyhook and modified skyhook controllers are implemented to estimate the passenger comfort when the vehicle passing over the bump with some speed.

In this simulink block diagram an on-off switch is used to actuate control policy. This switch has three input ports which are numbered from top to bottom and one output port. The first and third input ports are data ports and second input port is control port. As per the control algorithm policy, signal passes through input one when input two satisfies the selected criteria; otherwise it passes through input three. In this way the damper switches back and forth between two possible damping states, high damping and low damping. In this analysis, equivalent damping ratio ζ_S value is varied between 0.11 to 0.45 for case and 0.11 to 0.6 for second case. This is to check whether the same results are obtained through adjustment of parameters within the range of maximum and minimum limits.

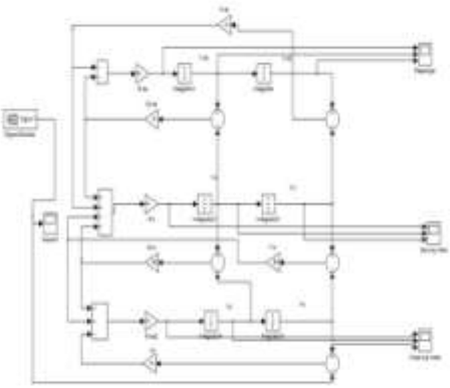

Fig 5: Passive Simulink block diagram

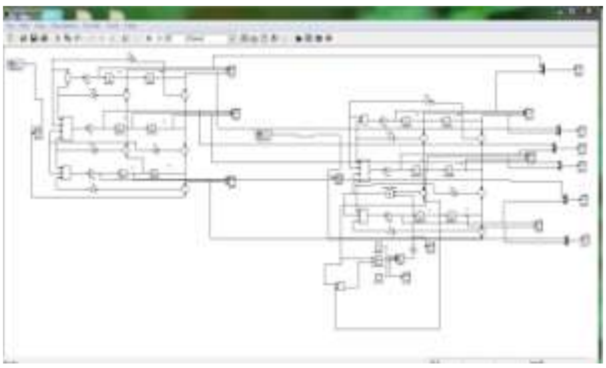

Fig 6: Passive and semi active Simulink block diagram.

V. RESULTS AND DISCUSSION

Here, Matlab/Simulink is used as a computer aided-control system tool for modeling the non-physical one third auto rich Shaw with its modeling as, all included in one analysis loop passive system, and semi active system. The vehicle parameters considered for the analysis are given in Table. For the given in put parameters the response of the system is observed on 10 seconds scale. The simulation results for semi active suspension system with sky hook control policy show, apparent trade off in between displacement, velocity and acceleration. The semi active suspension system response of the skyhook control peak to peak displacement is less compared to passive system. The seat peak to peak accelerations of the vehicle are increased at the cost of peak to peak reduction of displacements between the passive and semi active. The important finding for semi active suspension system with sky hook control and passive is the response of the both seat and sprung mass dies out faster in semi active system.

Parameters of the Passenger Vehicle

Body mass (sprung mass)	120kg
Mass of the wheel/axle assembly(unsprang mass)	25kg
Passenger and seat mass	60kg
Suspension damping	20Ns/mm
Suspension stiffness	60N/mm
Passenger seat Damping	6Ns/mm
Passenger seat Stiffness	100N/mm
Tire stiffness	240N/mm

Different Inputs

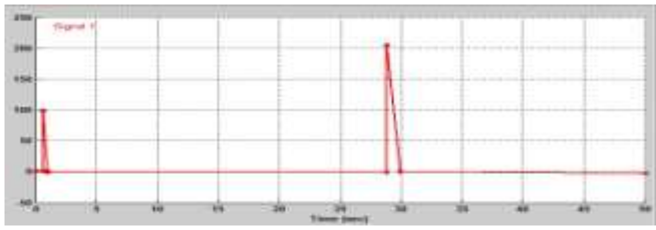

Fig 7: Input signals for Double bump

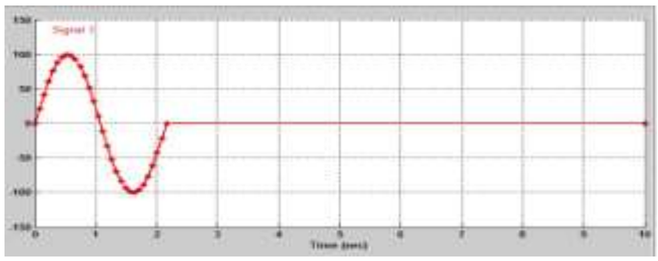

Fig 8: Input signals for sinusoidal bump

5.1 Equivalent Damping Ratio In Between 0.11 To 0.45

For Double bump input:

The maximum passenger displacement under the passive system is 55.3% more as compared to maximum passenger displacement of the semi active system. The sprung mass maximum displacement under the passive is 42.10% more as compared to the semi active system. The unsprung mass maximum displacement under the passive is 5% more as the semi active controllers.

Simulation Responses:

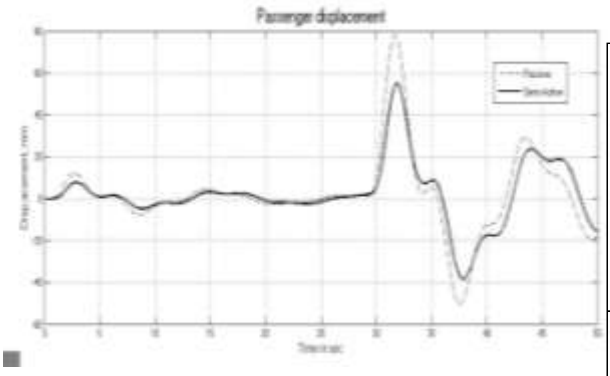

Fig 9: Passenger displacement for double bump.

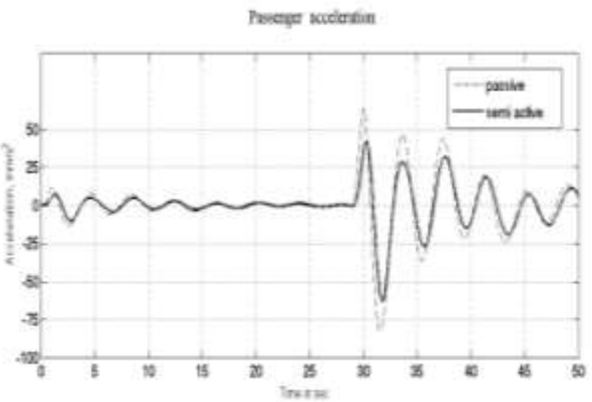

Fig 10: Passenger Acceleration for double bump.

Table 1: Maximum values of time responses of the Auto rickshaw for 1st bump in double bump

Input	Controller	Max. passenger displacement (mm)	Max. sprung mass displacement (mm)	Max. Unsprung mass displacement (mm)	Max. passenger Acceleration (mm/s^2)	Max. sprung mass Acceleration (mm/s^2)	Max. Unsprung mass Acceleration (mm/s^2)
Double bump	Passive	12.88	9.5	34.8	-13.6	39.55	937
	Semi active	5.8	5.5	32.8	-10.4	22	702

The maximum passenger acceleration under the passive system is 23.52% more as compared to maximum passenger acceleration of the semi active system. The sprung mass maximum acceleration under the passive is 44% more as compared to the semi active system. The unsprung mass maximum acceleration under the passive is 25.08% more as the semi active controllers. Settling time also low in semi active system compared to passive system.

Table 2: Maximum values of time responses of the Auto rickshaw for 2nd bump in double bump

Input	Controller	Max. passenger displacement (mm)	Max. sprung mass displacement (mm)	Max. Unsprung mass displacement (mm)	Max. passenger Acceleration (mm/s^2)	Max. sprung mass Acceleration (mm/s^2)	Max. Unsprung mass Acceleration (mm/s^2)
Double bump	Passive	81	55.8	161.87	-83.2	-157.01	1977.5
	Semi active	43.3	34.2	161.795	-62.45	-150	1484.5

The maximum passenger displacement under the passive system is 46.54% more as compared to maximum passenger displacement of the semi active system. The sprung mass maximum displacement under the passive is 38.66 more as compared to the semi active system. The unsprung mass maximum displacement under the passive is 5% more as the semi active controllers.

The maximum passenger acceleration under the passive system is 24.93% more as compared to maximum passenger acceleration of the semi active system. The sprung mass maximum acceleration under the passive is 4 % more as compared to the semi active system. The unsprung mass maximum acceleration under the passive is 24.93% more as the semi active controllers. Settling time also low in semi active system compared to passive system.

Sinusoidal bump:

Simulation Responses:

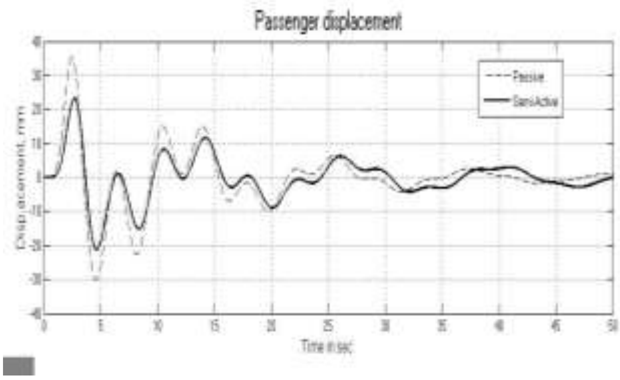

Fig 11: Passenger displacement for sinusoidal bump

International Journal of Engineering Applied Sciences and Technology, 2017
Vol. 2, Issue 6, ISSN No. 2455-2143, Pages 44-50
Published Online in IJEAST (http://www.ijeast.com)

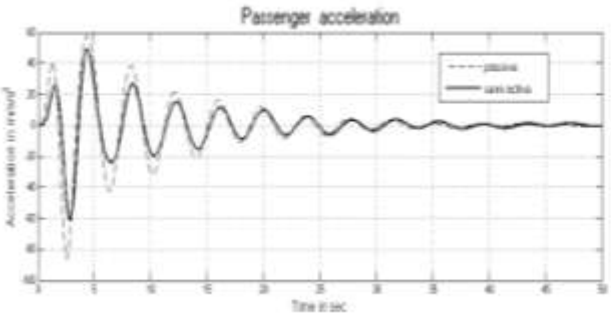

Fig: 12 Passenger Acceleration for sinusoidal bump.

Table 3: Maximum values of time responses of the Auto rickshaw for sinusoidal bump

Input	**Controller**	**Max. passenger displacement (mm)**	**Max. sprung mass displacement (mm)**	**Max. Unsprung mass displacement (mm)**	**Max. passenger Acceleration (mm/s^2)**	**Max. sprung mass Acceleration (mm/s^2)**	**Max. Unsprung mass Acceleration (mm/s^2)**
Sinusoidal (1 kmph)	Passive	34.9	33	-139.65	-85.35	-169	1247
	Semi active	21.12	-18.5	-127.85	-62.85	-131	1090

The maximum passenger displacement under the passive system is 39.48% more as compared to maximum passenger displacement of the semi active system. The sprung mass maximum displacement under the passive is 43.93% more as compared to the semi active system. The unsprung mass maximum displacement under the passive is 8% more as the semi active controllers.

The maximum passenger acceleration under the passive system is 27% more as compared to maximum passenger acceleration of the semi active system. The sprung mass maximum acceleration under the passive is 27% more as compared to the semi active system. The unsprung mass maximum acceleration under the passive is 12.59% more as the semi active controllers. Settling time also low in semi active system compared to passive system.

VI. CONCLUSIONS

One third model of auto rickshaw with passive and semi active suspension system has been simulated for various road disturbances by using MATLAB/SIMULINK. Comparison between passive and semi active suspension system with different valves of on-off skyhook control has been done. The simulation results show considerable differences between the results of passive and different schemes of semi active suspension system.

1. Semi active suspension with skyhook controller gives lower values of maximum sprung mass acceleration than passive suspension for given road inputs.
2. Also settling time for the passenger under semi active system less than passive suspension system.
3. It can be observed that the skyhook control can achieve substantial reduction of peak displacement for passenger than that of passive suspension.
4. Hence suspension model with semi active suspension provides good passenger comfort and vehicle stability than passive suspension system.

VII. REFERENCES

1. Thomas Gyllendahl and David Tran "Development of an auto rickshaw vehicle suspension" Luleå University of Technology Department of Engineering Sciences and Mathematics
2. T. Ram Mohan Rao , G. Venkata Rao, k.Sreenivasa Rao and A. Purushottam" Analysis Of Passive And Semi Active Controlled Suspension Systems For Ride Comfort In An Omnibus Passing Over A Speed Bump" IJRRAS 5 (1) October 2010 Rao & al. Analysis of Passive & Semi Active Controlled Suspension Systems
3. K.Rajeswari, P.Lakshmi "Simulation of Suspension System with Intelligent Active Force Control" 2010 International Conference on Advances in Recent Technologies in Communication and Computing.
4. Dawei Liu, Huanming Chen , Rongchao Jiang and Wei Liu "Study of Driving Performance of Semi-active Suspension Based on Hybrid Control Strategy" 2010 Second WRI Global Congress on Intelligent Systems.
5. Shu-wei Liu, Gang-Li and Liang-Hao "Optimization Research of Vehicle Damper Matching Parameters" 2013 International Conference on Mechatronic Sciences, Electric Engineering and Computer (MEC) Dec 20-22, 2013, Shenyang, China.
6. Galal Ali Hassaan "Car Dynamics using Quarter Model and Passive Suspension,
Part I: Effect of Suspension Damping and Car Speed" International Journal of Computer Techniques -– Volume 1 Issue 2, 2014.
7. Ramë Likaj, Ahmet Shala, Mirlind Bruqi & Mehmet Qelaj "Optimal Design Of Quarter Car Vehicle Suspension System" 14th International Research/Expert Conference "Trends in the Development of Machinery and Associated Technology" TMT 2010, Mediterranean Cruise, 11-18 September 2010.
8. Thanh-Tung Tran, Chiaki Hori1 and Hiroshi Hasegawa "Integrated Inerter Design And Application To Optimal Vehicle Suspension System" International Journal of Computer-Aided technologies (IJCAx) Vol.1,No.2/3,October 2014.

9. Andronic Florin , Manolache-Rusu Ioan-Cozmin and Pătuleanu Liliana "Pasive Suspension Modeling Using Matlab, Quarter Car Model, Imput Signal Step Type" TEHNOMUS - New Technologies and Products in Machine Manufacturing Technologies.
10. Ahmad Zaim Solehin Bin Che Hasan "Modelling And Simulation Of Modified Skyhook Control For Semi-Active Suspension" Universiti Malaysia Pahang.
11. S. Segla, S. Reich "Optimization and comparison of passive, active, and semi-active vehicle suspension systems" 12th IFToMM World Congress, Besançon (France), June18-21, 2007.