

Investigation of Variable Damping Shock Absorber Performance of Automotives in Shock Prone Areas

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Abstract—Shock absorbers allow the use of springs and hydraulic shock absorbers to absorb and dissipate vibration. A shock absorber consists of a helical spring and a damper. A spring has its spring constant and a damper has a damping coefficient. In this project a variable shock absorber model is developed by varying the spring constant which is achieved by using helical spring with variable pitch in coils and varying coil diameter and a variable damper. Research involved improving the shock absorber by using variable spring and a variable damper to come up with variable shock absorber model which adjusts depending on the shock. The governing equations were solved; Analysis of various parameters such as damping coefficient, spring constant and damper fluid temperature are considered and their effect on damping. The results obtained are discussed and presented graphically

Keywords—Crank Nicolson Numerical Scheme; Finite Difference Method and Variable Shock Absorber.

Abbreviations—Crank Nicolson Scheme (CNS); Forward Difference Scheme (FDS); Partial Differential Equation (PDE).

I. INTRODUCTION

THIS section gives the Background to the study and Geometry of the problem.

1.1. Background to the Study

In the design of automotive suspensions, engineers seek for a good compromise among competing objectives of driving safety, driving fun and ride comfort. The spring and damper within suspension system play an important role in achieving the desirable vehicle performance. The suspension spring stores and releases energy temporarily, providing the vehicle body insulation against high acceleration peaks. The suspension damper modulates the energy dissipation, ensuring any induced oscillation to die off quickly. Even though active and semi-active suspensions promise to provide improved ride quality and better handling performance of vehicle, other considerations which include high cost, complex implementation, increased power consumption,

reliability and weight have restricted their use in today's automotive industry. Passive springs and dampers still hold on to their importance. Today, majority of the passenger cars produced still rely on passive springs and damper in the suspension system. In determining automotive suspension's characteristics, the current practice adopted in the automotive industry makes use of prototypes built up using available springs and dampers deemed suitable based on prior experience with similar vehicles.

The motivation of this research is to allow for quick assessment of design variations in suspension spring and damper characteristics for a given vehicle desirable concept. The objectives of this research are to:

a) Determine the spring model with variable spring constant and dampers with varying damping coefficient b) investigate the relationship between the vehicle velocity, damping coefficient and spring constant.

The contribution of this manuscript are a) to developed a model that will provide information to engineers and designers for making decisions associated with damping and shock absorbers so as to increase comfortability and durability of vehicles) to provided information to researchers for making decision associated with damping in sound, tall building, bridges where dampers are required.

1.2. Geometry of the Problem

The Figure 1 shows the design of a variable shock absorber with variable spring, K and variable Damper, γ based on the influence of non-linear characteristics of spring and damper. The unsprung mass consists mass below the shock absorbers which include the chassis and sprung mass include the vehicle body and its load.

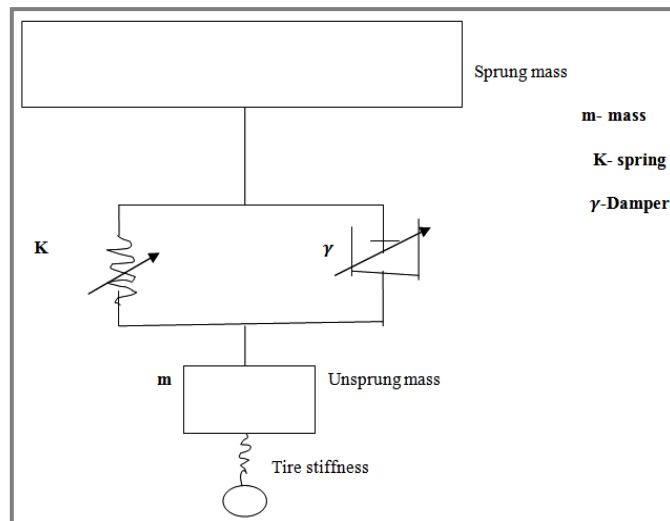


Figure 1: Variable Shock Absorber

In the next section the governing equations of the model are discussed in detail.

II. LITERATURE REVIEW

Damping is an influence within or upon an oscillatory system that has the effect of reducing, restricting or preventing its oscillations. Shock absorbers are an important part of the suspension systems of automobiles, motorcycles and other wheeled or tracked vehicles, as well as aircraft landing gear and the supports for many industrial machines. Large shock absorbers have also been used in structural engineering to reduce the susceptibility of structures to earthquake damage and resonance as studied by Charhan & Burande [1]. Driving a vehicle down a bumpy or potholed road would be extremely bouncy and uncomfortable without shock absorbers. Important properties that affect the shock absorption included,

2.1. Spring Rate/Suspension Rate

It is a component in setting the vehicle's ride height or its location in the suspension shock. The compression or stretch length is proportional to the force.

2.2. Weight Transfer

Weight transferring during cornering acceleration or braking is usually calculated per individual wheel and compared with the static weights for the same wheels. Factors affecting weight transfer include: the distance between wheel centers, height of centre of gravity, the mass of the vehicle and the amount of acceleration experienced.

2.3. Jacking Forces

These are sum of the vertical force components by the suspension springs. The resultant force acts to lift the sprung mass.

2.4. Damping

It is the control of motion or oscillation. This is done using hydraulic gates and valves in a vehicle's shock absorbers. Like spring rate, the optimal damping for comfort may be less than control.

In most vehicles, Damping and spring rates controls the travel speed. In this model, damping and spring are variable such that the travel speed is maintained constant.

Variable shock absorbers allow the shock to adjust to type of road on which the vehicle is driven, ensuring a safe and comfortable ride. Today, the current practice adopted in the automotive industry makes use of prototypes built up using available springs and dampers deemed suitable based on prior experience with similar vehicles. Professional test drivers then carried out road testing with these prototypes. The tuning and the final specification of the automotive suspension design hinge mainly on the subjective opinion of the test drivers, with priority placed on handling safety [Tan Chun Wei, 10]. In most parts of the world most Automotives are characterized with reduction of velocity as a way of dealing with shocks, Eshleman & Rao [3] investigated that when driving in roads with potholes and pumps. Potholes is a type of failure in an asphalt pavement caused by the presence of water in the underlying soil structure [Pinjarla & Lakshmana Kishore, 6]. Damping coefficients and spring constants are always considered when designing shock absorbers such that they reduce or eliminate shock for comfort riding, prevention of damage or outright structural failure. The two main function of suspension are firstly to insulate the rider and the bulk of the machine from the road shocks – the first for this compact and the second for mechanized reliability and longevity secondary, to keep the wheels in the vehicles in the closest possible contact with the ground. Many Factors contribute to this including spring damping, suspension geometry and the ratio of sprung to unstrung mass as researched by Reybrouck [7]. Any four wheel vehicle needs suspension for both the front wheel and the rear suspension, but in two wheel drive vehicles that can be very different configuration [Patil & Gawaade, 5]. For front wheel drive cars rear suspension has few constraints and variety of been axles and independent suspensions are used. For rear- wheel drive car, rear suspension has many constraints and the development of the superior move

expensive independent suspension layout has been difficult. Four wheel driver have suspension that are similar for both the front and rear wheels as investigated by Saikrishma Kishore [8]. Rhoades et al., (2010) in his thesis, Development and Experimental Verification of Parametric Model of Automotive Damper, a detailed model of adjustable variable damper is established and explained how it works and results simulated. This contains compression, rebound and gas chambers which are key in its working. Wei et al., (2010) in their paper, they have researched on how to achieve a variable spring constant. Different strategies could be employed to achieve variable spring constant. They include; Superimposing the primary spring with auxiliary smaller springs, using a spring with a variable pitch in the helical spring, using a spring having a series arrangement of two springs of different materials, a spring with varying coil diameter, varying wire diameter or any of the two or more combined. In this project, we adopted variable spring of varying pitch and coil diameter.

To combine a variable damper with a variable spring to come up with variable shock absorber, we use Becedas et al., (2010) model that he has made specific applications in the determination of the parameters mass-spring-damper system. The method is suitable for simultaneously identifying both the spring constant and the damping coefficient. Using Mohan D. Rao & Scott Gruenberg [4] analysis and performance of a shock absorber formula, we now get effective spring constant and damping coefficient of the shock absorber. In decades efforts have been made to produce more efficient machines with an aim of reducing shocks. Applications of this knowledge have emerged in modern machines in motor vehicle Assembly companies and its relevance in industrial engineering has motivated research interest.

III. METHOD OF SOLUTIONS

In this section, the partial differential equations governing the problem will be solved using the Finite Difference. This will assist in the formulation for appropriate mathematical models and produce data on various performance characteristics. The knowledge gained is applied to scaled prototype geometries in vehicle.

3.1. General Governing Equations

A variable shock absorber is achieved by coming up with variable spring constant which is achieved by using a helical springs with variable pitch in coils or superimposing more than one spring with different spring constants or using springs with varying coil diameter.

3.2. Damper Shock Absorber Vibration Equation

The transverse vibration of damper shock absorber $u(x, t)$ can be described by the following equation which is non dimensional in the spatial variable [Dino Sciulli, 2].

$$(1)$$

The system has inhomogeneous boundary conditions represented by $u(x, 0) = 0$.

, k and F in equation (1) are parameters which can represent mass of damper, damping coefficient, damper spring constant and the external disturbance force (in our case, the mass of the vehicle) [Dino Sciulli, 2]. $u(x, t)$ is the displacement of two vibrating springs measured from their equilibrium positions.

3.3. Energy Equation

This equation results from the first law of thermodynamics which states that the amount of heat added to a system ∂Q equals to the change in internal energy ∂E plus the work done ∂w .

$$\partial Q = \partial E + \partial w \tag{2}$$

The energy equation governing the temperature of the fluid in the damper is taken as;

$$\rho C_p \left[\frac{\partial T}{\partial t} + u \frac{\partial \theta}{\partial x} + v \frac{\partial \theta}{\partial y} \right] = \kappa \left[\frac{\partial^2 \theta}{\partial x^2} + \frac{\partial^2 \theta}{\partial y^2} \right] + \mu \phi \tag{3}$$

Equation (4) is known as the dissipation function. For an incompressible fluid flow in two dimensions, we have

$$\Phi = 2 \left\{ \dots \dots \dots \right\} \tag{4}$$

From the Fourier law the conduction can be obtained as

$$= -k \dots \tag{5}$$

Where k is thermal conductivity. Equation (6) can be simplified using the definition of h which is given as

$$h = e + \dots \tag{6}$$

In differential form equation (7) can be written as

$$\partial h = T \partial s + \dots \tag{7}$$

Applying the first and second laws of thermodynamics to equation (8) yields

$$\partial e = T \partial s - p \partial \dots \tag{8}$$

Substituting equation (9) into (7) yields

$$\partial h = T \partial s + \dots \tag{9}$$

Since enthalpy is a property it can be expressed as $s=s(p,T)$ so that on differentiating both sides of this equation yields

$$\partial s = \left(\frac{\partial s}{\partial p} \right) \partial p + \left(\frac{\partial s}{\partial T} \right) \partial T \tag{10}$$

In the next section, the method of solving the equation and parameters used are discussed in detail.

IV. METHOD AND NUMERICAL SCHEMES

In this part, the parameter values used to simulate the results and the method of solution are discussed.

4.1. Damper Shock Absorber Vibration Equation

The transverse vibration of damper shock absorber $u(x, t)$ can be described by the following equation which is non dimensional in the spatial variable.

$$\dots \dots \dots (11)$$

The system has inhomogeneous boundary conditions represented by $u(x, 0) = 0$.

, k and F in equation (12) are parameters which can represent mass of damper, damping coefficient, damper spring constant and the external disturbance force (in our case, the mass of the vehicle) [Dino Sciulli, 2]. $u(x, t)$ is the displacement of two vibrating springs measured from their equilibrium positions.

Using the central finite difference approximations for u and u_t , we discretized Equation (11) to get the explicit scheme shown below;

$$10 \left[\frac{U_{i,j+1} - 2U_{i,j} + U_{i,j-1}}{(\Delta t)^2} \right] + 4 \left[\frac{U_{i,j+1} - 2U_{i,j} + U_{i,j-1}}{2(\Delta t)} \right] + 20 \left[\frac{U_{i,j+1} - U_{i,j}}{2} \right]$$

If we take $\Delta t = \Delta x = 1$, $F = 1000\text{kg}$, $M = 10\text{kg}$, $\gamma = 4$ and multiply both sides by $2\Delta t$ we get the explicit scheme below

$$22U_{i,j+1} - 30\theta_{i,j} + 8U_{i,j-1} = 1000 \quad (12)$$

Taking initial condition as $u(x, 0) = 0$.

And letting $i=1, 2, 3, \dots, 5$ and $j=1$ into the scheme in equation (13) we form the following systems of linear algebraic equations which reduce to matrix form as

$$\begin{bmatrix} -30 & 22 & 0 & 0 & 0 & 0 & 0 & 0 \\ 8 & -30 & 22 & 0 & 0 & 0 & 0 & 0 \\ 0 & 8 & -30 & 22 & 0 & 0 & 0 & 0 \\ 0 & 0 & 8 & -30 & 22 & 0 & 0 & 0 \\ 0 & 0 & 0 & 8 & -30 & 22 & 0 & 0 \\ 0 & 0 & 0 & 0 & 8 & -30 & 22 & 0 \\ 0 & 0 & 0 & 0 & 0 & 8 & -30 & 22 \\ 0 & 0 & 0 & 0 & 0 & 0 & 8 & -30 \end{bmatrix} \begin{bmatrix} U_{11} \\ U_{21} \\ U_{31} \\ U_{41} \\ U_{51} \\ U_{61} \\ U_{71} \\ U_{81} \end{bmatrix} = \begin{bmatrix} 1000 \\ 1000 \\ 1000 \\ 1000 \\ 1000 \\ 1000 \\ 1000 \\ 1000 \end{bmatrix} \quad (13)$$

4.2. Discretization of Energy Equation

The energy equation governing the temperature of the fluid in the damper is taken as;

$$\rho C_p \left[\frac{\partial T}{\partial t} + u \frac{\partial \theta}{\partial x} + v \frac{\partial \theta}{\partial y} \right] = \kappa \left[\frac{\partial^2 \theta}{\partial x^2} + \frac{\partial^2 \theta}{\partial y^2} \right] + \mu \phi \quad (14)$$

Where ρ is the density of the fluid, C_p is the specific heat capacity, κ is the thermal conductivity, μ is dynamic viscosity. If we let $u = 1$, $v = 0$, $C_p = 1$, $\kappa = 0.5$

$\mu/k, \mu\phi = 0.2$ and $\kappa = 0.5$ we discretize equation (3.2) using Crank Nicholson scheme as follows

$$0.96 = 0.5 \left[\frac{1}{2} \left(\frac{\theta_{i+1,j} - 2\theta_{i,j} + \theta_{i-1,j}}{(\Delta x)^2} + \frac{\theta_{i+1,j+1} - 2\theta_{i,j+1} + \theta_{i-1,j+1}}{(\Delta x)^2} \right) + \frac{\theta_{i,j+1} - 2\theta_{i,j} + \theta_{i,j-1}}{(\Delta t)^2} \right] + 0.2$$

If we take $\Delta t = \Delta x = 1$ and multiply both sides by $2\Delta t$ we get the Crank Nicolson scheme below

$$0.75\theta_{i+1,j+1} - 0.5\theta_{i,j+1} - 0.75\theta_{i-1,j+1} = 1.4 - 0.75\theta_{i+1,j} + 3.5\theta_{i,j} + 0.75\theta_{i-1,j} - \theta_{i,j-1} \quad (15)$$

Taking $\theta(x, 0) = e^{-x}$, $\theta(x, t) = \theta(0, t) = 0$ and letting $i=1, 2, 3, \dots, 5$ and $j=1$ into the scheme in equation (15) we form systems of linear algebraic equations which reduce to matrix form as

$$\begin{bmatrix} 0.5 & 0.75 & 0 & 0 & 0 \\ -0.75 & 0.5 & 0.75 & 0 & 0 \\ 0 & -0.75 & 0.5 & 0.75 & 0 \\ 0 & 0 & -0.75 & 0.5 & 0.75 \\ 0 & 0 & 0 & -0.75 & 0.5 \end{bmatrix} \begin{bmatrix} \theta_{1,2} \\ \theta_{2,2} \\ \theta_{3,2} \\ \theta_{4,2} \\ \theta_{5,2} \end{bmatrix} = \begin{bmatrix} 1.03212 \\ 1.264665 \\ 1.350213 \\ 1.393262 \\ 1.397521 \end{bmatrix} \quad (16)$$

V. RESULTS AND DISCUSSIONS

5.1. Effects of Damper Mass on Damper Shock Absorber Vibrations

We solve matrix equation (13) and get the results as in table one below

Table 1: Damper Vibration Values for Varying Damper Mass at Constant Mass of Vehicle $m=1000\text{kg}$, Damping Coefficient and Spring Constant

Time	Damper Mass M=10 kg	Damper Mass M=8 kg	Damper Mass M=6kg
t = 0	-337.7078	-214.7267	-147.8964
t = 1	-395.0561	-272.09	-190.9384
t = 2	-397.7282	-272.2444	-196.9434
t = 3	-345.9726	-249.5305	-187.8219
t = 4	-281.6979	-215.398	-169.9506
t = 7	-142.3881	-123.7681	-109.2759
t = 8	-71.30349	-66.33815	-62.47357

If we take the mass of the vehicle to be 1000kg , Damping coefficient $\gamma = 4$ and a spring constant $k = 20\text{N/m}$ and simulate, we get results as shown in table 1 above. As the damper mass reduces, damping increases. At $t = 0$, damper vibration is low and as time increases damping also increases. Between $t = 0$ and $t = 8$, the vehicle has gained maximum speed.

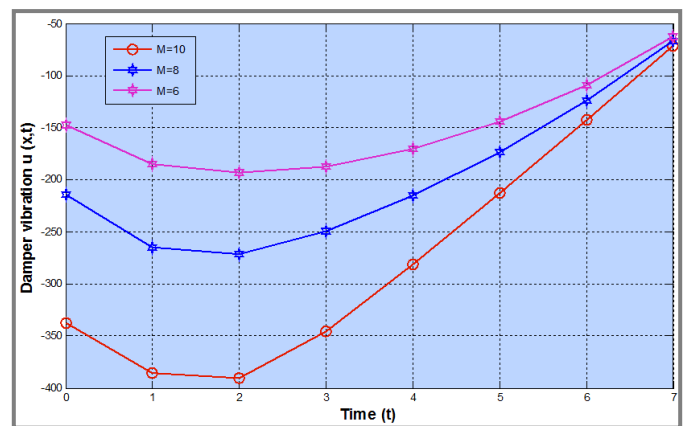


Figure 2: Damper Vibration Values for Varying Damper Mass M

Figure 2 shows a decrease in damper vibration initially then Damper vibration increases with time. When the damper mass $M = 10\text{kg}$ damper vibration is small, when the damper mass reduces to 6kg , damper vibration is more. When the damper mass increases the stiffness of the shock absorber increases hence the damper vibration reduces and when the damper reduces the stiffness also reduces hence damper vibration increases. When the vehicle starts, it vibrates more

and reduces within a short time and then it starts increasing as shown in figure 2. As the speed increases, vibration approaches the same value.

5.2. Effects of Damping Coefficient on Absorber Vibrations

Using the initial conditions we solve matrix equation (13) and get the results as in table 2 below

Table 2: Damper Vibration $u(x, t)$ Values for Varying Damping Coefficient γ at Constant Mass of Vehicle $m=1000\text{kg}$, Spring Mass $M=10\text{kg}$ and Spring Constant

Time	Damping Coefficient	Damping Coefficient	Damping Coefficient
t = 0	-167.8964	-154.075	-148.115
t = 1	-190.9384	-186.6448	-180.6724
t = 2	-196.9434	-191.622	-185.2742
t = 3	-187.8219	-179.692	-171.3095
t = 4	-169.9506	-160.495	-151.3095
t = 6	-144.219	-134.5793	-125.6802
t = 7	-109.2759	-100.7323	-93.10569
t = 8	-62.47357	-56.83753	-51.95447

As the damping coefficient increases from 4 to 8, damper vibration increases. When $t = 0$, the damper vibration is small. As time increases damper vibration increases.

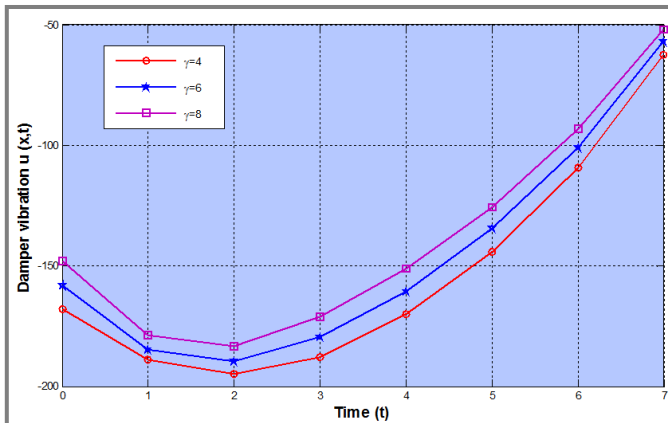


Figure 3: Damper vibration values for varying Damping coefficient

Figure 3 shows a decrease in damper vibration initially before it starts increasing with time. When damping coefficient $\gamma = 4$, damper vibration is small and when $\gamma = 8$, the damper vibration is more. When the damping coefficient increases the damping ability of the shock absorber also increases as damper vibration reduces as shown in Figure 3 above. When the vehicle starts, it vibrates more and reduces within a short time and then increases as the vehicle starts moving.

5.3. Effects of Fluid Viscosity on Damper Shock Absorber Temperature

Using the initial conditions 0, we solve matrix equation (16) and get the results as in table 3 below.

Table 3: Damper Temperature Values for Varying Damping Coefficient γ at Constant Mass of Vehicle $m=1000\text{kg}$ and Spring Constant

Time	Damper Fluid Viscosity	Damper Fluid Viscosity	Damper Fluid Viscosity
t = 0	-0.001163586	-0.08337443	-0.143021
t = 1	1.383917	0.8984096	0.4043614
t = 2	0.7519727	0.4705725	0.1836569
t = 3	2.682886	1.851645	1.015541
t = 4	0.8056278	0.5450549	0.2822089
t = 6	4.0003484	2.812625	1.618418

In table 3, as damper fluid viscosity is increases, the temperature of the fluid reduces. When $t = 0$, the damper fluid viscosity is low, and it increases with increase with time. As the temperature of the vehicle increases the viscosity also reduces.

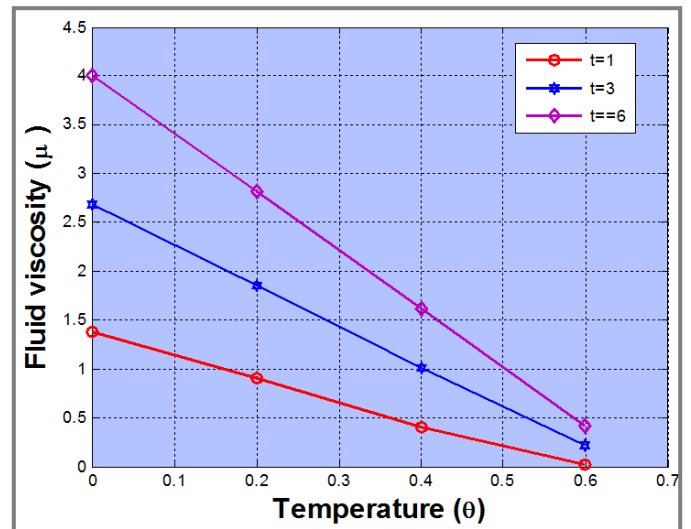


Figure 4a: Damper Temperature Values for Varying Damping Coefficient

Figure 4a shows that as the temperature increases, the damper fluid viscosity reduces with time. This is because as the temperature of the vehicle increases the density of fluid reduces and hence the viscosity. At $t=1$, temperature is low and hence the viscosity and at $t = 6$, the temperature is higher and hence the viscosity.

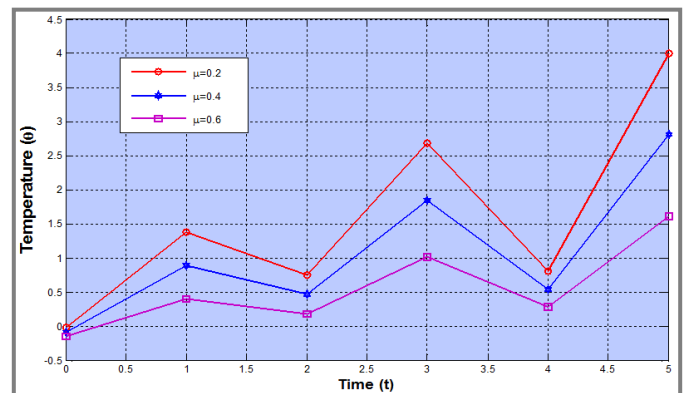


Figure 4b: Damper Temperature Values for Varying Damping Coefficient

Figure 4b shows that temperature increases with time as the vehicle moves. Due to external cooling of the vehicle using water and fan, the temperature is brought down. As the vehicle moves, the temperature increases again. So the thread repeats itself- temperature increasing due to engine heating and reducing due to external cooling. Although there is external cooling, the final temperature in the damper fluid keeps increasing with time as shown in figure 4b above. When the viscosity of the fluid is high the fluid density is low hence the temperature is low. Therefore, temperature decreases with increase of fluid viscosity.

5.4. Effects of Fluid Density on Damper Shock Absorber Temperature

The result in table 4 shows the effects of changing fluid density on temperature values.

Table 4: Temperature Values for Fluid Density ρ at Constant Damper Fluid Viscosity and Fluid Thermal Conductivity

Time	Fluid Density	Fluid Density	Fluid Density
t = 0	0.2074995	0.1076995	0.0087995
t = 1	0.4437043	0.3439843	0.2457043
t = 2	0.5806606	0.4806846	0.3876606
t = 3	0.6398893	0.5399693	0.4398773
t = 4	0.6985007	0.5885227	0.4995007
t = 5	0.7360024	0.6366724	0.53605524

In table 4, if the damper fluid density reduces, the vehicle’s damper fluid temperature increases. The fluid density increases with time as the vehicle moves. When t = 0, fluid temperature is small and as time increases, fluid temperature also increases.

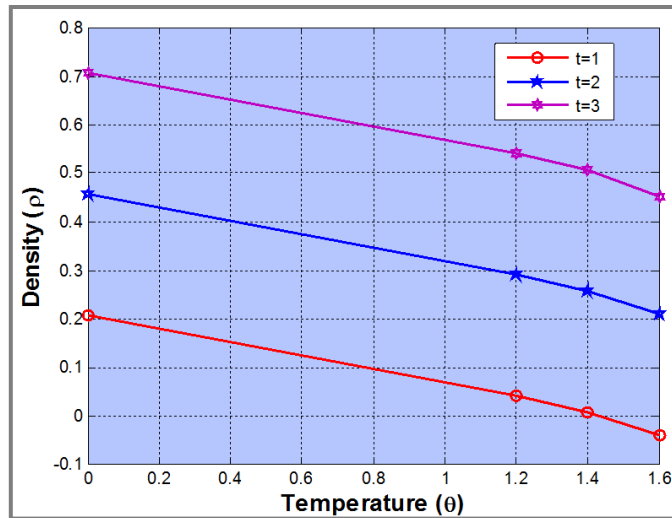


Figure 5a: Temperature Values for Fluid Density

Figure 5a; Shows that as the vehicle moves the temperature of the damper fluid increases this leads to reduction in the fluid density. As the time increases, temperature increases and correspondingly the density reduces as shown in figure 5a above. Temperature and density increase reduces fluid viscosity of the shock absorber damper.

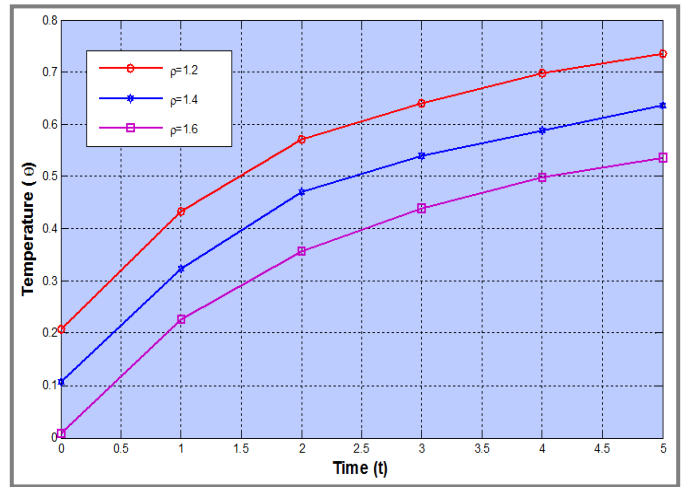


Figure 5b: Temperature Values for Varying Time

Figure 5b shows that temperature of the damper fluid increases as the vehicle moves. Initially the rate of temperature change is higher but it smoothens with time due to external cooling. Damper fluid with higher density takes more time for temperature change as compared to a less dense fluid as shown in figure 5b.

VI. CONCLUSION

The purpose of this study has been to find the model of a shock absorber that is variable. This is achieved by using a variable spring constant done by using a helical spring with variable wire diameter and coil diameter. Variable damper dissipates the energy to avoid oscillations. A vehicle varies its mass when its loaded or unloaded or when passengers get into it or alight. Also the absorption of a shock by shock absorber depends the extent of the shock and the speed of the vehicle. For the shock absorber to absorb the varying shock for increased or decreased mass effectively, the shock absorber should also vary. This is achieved by making a shock absorber spring using a wire of varying coil diameter and varying wire thickness. As the mass increases, the compression and stretching reduces with time due to varying coil thickness as shown by figure 3. The fluid viscosity and temperature also changes with time although heat is controlled by external cooling. Temperature increase reduces viscosity of the damper fluid and density as shown by figures 4, 5 and 6. For the damper to work effectively, room should be left for temperature changes in the engine.

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- Master of science in applied mathematics
- Units: Complex Analysis I and II, Numerical Analysis,

Fluid Mechanics, Ordinary Differential Equations, Partial Differentials Equations and Riemannian geometry Supervision of postgraduate students:

- Doctor of philosophy: thesis (3 completed, 5 ongoing)
- Masters of science in Applied Aathematics: (13 completed, 8 ongoing)



Dr. Okwoyo James Mariita holds a Bachelor of Education degree in Mathematics and Physics from Moi University, Kenya, Master Science degree in Applied Mathematics from the University of Nairobi and PhD in applied mathematics from Jomo Kenyatta University of Agriculture and Technology, Kenya. James holds a Bachelor of Education degree in Mathematics and Physics from Moi University,

Kenya, Master Science degree in Applied Mathematics from the University of Nairobi and PhD in applied mathematics from Jomo Kenyatta University of Agriculture and Technology, Kenya. Affiliation: University of Nairobi, Chiromo Campus School of Mathematics P.O. 30197-00100 Nairobi, Kenya. He is currently a lecturer at the University of Nairobi (November 2011 – Present) responsible for carrying out teaching and research duties. He plays a key role in the implementation of University research projects and involved in its publication. He was an assistant lecturer at the University of Nairobi (January 2009 – November 2011). He has published 7 papers on heat transfer in respected journals. Supervision of postgraduate students.

- Masters of science in applied mathematics: (8 completed and 8 ongoing)

Dr. Okelo Jeconia Abonyo holds a PhD in Applied Mathematics from Jomo Kenyatta University of Agriculture and Technology as well as a Master of science degree in Mathematics and first class honors in Bachelor of Education, Science; specialized in Mathematics with option in Physics, both from Kenyatta University. I have dependable background in Applied Mathematics in particular fluid dynamics, analyzing the interaction between velocity field, electric field and magnetic field. Has a hand on experience in implementation of curriculum at secondary and university level. He has demonstrated sound leadership skills and has the ability to work on new initiatives as well as facilitating teams to achieve set objectives. He has good analytical, design and problem solving skills. Affiliation: Jomo Kenyatta University of Agriculture and Technology, (JKUAT), Kenya. 2011-To date Deputy Director, School of Open learning and Distance e Learning SODEL Examination, Admission &Records (JKUAT), Senior lecturer Department of Pure and Applied Mathematics and Assistant Supervisor at Jomo Kenyatta University of Agriculture and Technology. Work involves teaching research methods and assisting in supervision of undergraduate and postgraduate students in the area of Applied Mathematics. He has published 10 papers on heat transfer in respected journals. Supervision of postgraduate students; Doctor of philosophy: thesis (3 completed)

- Masters of science in applied mathematics: (13 completed, 8 ongoing)

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