A Final Project Report entitled

"Design and Development of Air Shock Absorbers For Two Wheelers"

Submitted

In Partial Fulfilment of Requirement For The

Bachelor of Technology

In

Mechanical Engineering

Submitted by:

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DEPARTMENT OF MECHANICAL ENGINEERING
SARDAR VALLABHBHAI NATIONAL INSTITUTE OF TECHNOLOGY

JUNE 2020

CERTIFICATE

This is to certify that B. Tech. IV (8th Semester) Final project report entitled "**Design and Development of Air Shock Absorbers for Two Wheelers**" has been submitted by

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for partial fulfilment of the requirement of the degree in Bachelor of Technology in Mechanical Engineering during the academic year 2019-20, of the Sardar Vallabhbhai National Institute of Technology, Surat.

Place: SVNIT, Surat

Date: 16/06/2020

Dr. H.J. Nagarsheth
Professor
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Dr. S.K. Budhwar
Head of the department
MED, SVNIT

EXAMINER'S APPROVAL CERTIFICATE

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The Final project report is presented before the following members of evaluation committee on 16/06/2020.

Examiners:

1) Dr. B.M. Sutaria

2) Mr. Nikunj Patel

3) Dr. Rohan Pande

Place: SVNIT, Surat

DECLARATION

We, the undersigned, registered for the Bachelor Degree at Mechanical Engineering Department, Sardar Vallabhbhai National Institute of Technology, hereby declare that B.Tech. Final Year Project entitled "Design and Development of Air Shock Absorbers for Two Wheelers" is based on our own work and has been carried out under the guidance and supervision of Dr. H.J. Nagarsheth, Professor, Mechanical Engineering Department, Sardar Vallabhbhai National Institute of Technology, Surat. The data and information which we have used from various sources have been duly acknowledged. We further declare that this work has not been previously submitted by us to any other university/institute for the award of any degree or diploma or any other purpose.

Place: SVNIT, Surat
June 2020

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ACKNOWLEDGEMENT

We would like to take this opportunity to express our gratitude and deep sense of appreciation to all those who supported us throughout the course. The valuable inputs received at all times have played a vital role in completion of this study report. We thank Dr. H.J. Nagarsheth for his continuous guidance, support and motivation throughout the semester. Also we would like to thank Dr. S.K. Budhwar (HOD, MED) for providing us an opportunity to present our work. Also we would like to thank all those who directly or indirectly helped us in making this project report a success.

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ABSTRACT

In an automobile, the suspension system is responsible to ensure traction between the tire and the road, the handling of the vehicle and also the comfort level of the passengers. For a conventional coil-spring suspension system design, higher stiffness of the system results in a lower passenger comfort while a lower stiffness gives poor road handling. A compromise has to be made between the two essential functions of a suspension system. If air dampers are used as suspension elements, the equivalent stiffness coefficient of the system could be varied by simply controlling the pressure of air. Such air suspension also facilitates the user to vary the ride height according to the need. The cost of manufacturing and maintenance for such systems is lower than the conventional systems.

Commercially, the air suspension systems used in trucks, railways and passenger cars use compressed air as working fluid and they have either an in-house compressor fitted onto the vehicle or has a reservoir filled with compressed air. The proposed suspension system uses atmospheric air as the working fluid so that such systems can be readily fitted onto two-wheelers having stringent space constraints. The study of vertical dynamics of such a suspension system needs a dynamic mathematical model. The resistance to free flow of air by an orifice results in damping characteristics of the system. In this work, a single degree of freedom model for the proposed suspension system has been derived by using the theories of fluid flow and the dynamic analysis of the forces on the piston inside the cylinder. The transfer functions are developed and the variations in the response for different design parameters are analysed.

The guidelines for an experimental test-rig to mimic single suspension unit to obtain parameters experimentally and validate the mathematical model as well as the design of the proposed system are also provided here. A piston-cylinder arrangement with an orifice controls air transfer inside the cylinder due to developed pressure changes which acts as a damper. Further, a comparative study has been performed between oil and air dampers, analytically as well as theoretically, to understand the conditions under which the air dampers can outperform the conventional suspension systems.

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NOMENCLATURE

Symbol	Meaning
P_1	Initial Pressure (P_{atm})
P_a	Upper half pressure after displacement of x
P_b	Lower half pressure after displacement of x
V_1	Initial Volume
$A\ (A_p)$	Circumferential area of piston
F_a	Force exerted on the upper side of piston
F_b	Force exerted on the lower side of piston
F_w	Force applied by road on piston $(F_w = F_0 sin(wt))$
m	Mass of piston
β	Air Damping Coefficient
ω	Time Period
F_0	Amplitude
n	Polytropic Constant (n=1.4)
k	Spring Constant

CHAPTER 1

INTRODUCTION

1.1 Air Suspension Systems

Air suspension systems essentially replace a vehicle's coil springs with air springs. The air springs are simply tough rubber and plastic bags inflated to a certain pressure and height to mimic the coil springs. But the similarities end there. By adding in an on-board air compressor, sensors and electronic controls, today's air suspension systems provide several advantages over all-metal, conventional springs, including near-instant tuning, and the ability to adapt handling to different situations and vary load capability. Moreover, they can be used to adjust the ride height of the vehicle in various road conditions. When the road is bumpy and irregular, we can adjust the ride height to a higher value which sends a signal to the compressor to pump in more pressurised air into the air springs and when the vehicle is running on highways, the ride height could be lowered to reduce the amount of drag force acting on the vehicle and hence it would help to improve the fuel economy.

Early versions of air suspension systems were relatively simple. Air bags replaced the coil springs. The bag was inflated to the correct pressure or height with an outside compressor through a valve on the bag. Changes in technology and use added more components, and control, to the system. But today's air suspension systems all have a basic stock of similar components that vary little from maker to maker. The differences come mainly in controls and ease of installation.

Air bag material has changed little over time. The bag is a composite of rubber and polyurethane, which provides structural integrity, air-tight construction, toughness against light abrasion from road debris and sand, and resistance to salt and chemical corrosion.

Although, given the advantages of the air suspension systems, it suffers from a lot of drawbacks such as:

- The possibility of air bag puncture, due to the presence of debris on the road.
- The response time is very slow for air suspension systems.
- The cost is almost three times more than that of the leaf spring suspension systems.

- The use of various components such as compressor, air bag, sleeve and drier makes the assembly heavy and complicated due to which its use is limited to 4 wheeler vehicles only.

Moreover, these systems cannot be implemented on 2 wheelers due to its large size and the high cost involved with it. Therefore, there is a need to develop such systems that are compatible with 2 wheelers. A piston and cylinder assembly is described as follows:

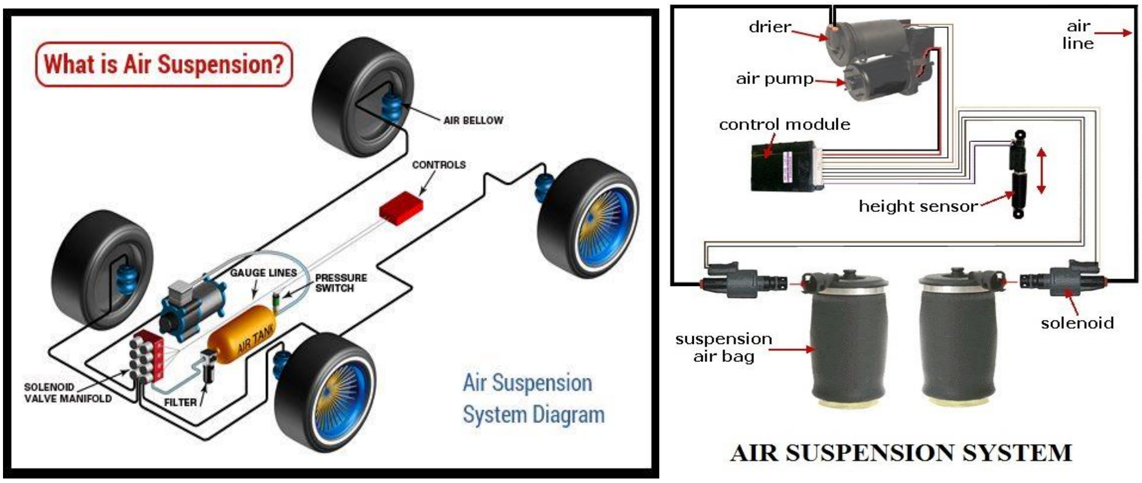


Fig 1: Air Suspension System Layout [14]

1.2 Air Shock Absorbers

Our approach to the design of air shock absorber has been influenced by the fact that the pressure required to compress a syringe whose orifice has been partially blocked is much more than the pressure that is required to compress a syringe whose orifice is open to the atmosphere. As the orifice area reduces, more amount of force is required to compress the syringe. The damping coefficient is a function of the compressibility of air.

1.3 Quarter Car models

In order to analyse the performance of the suspension systems, it becomes necessary to understand the movement of the unsprung mass with respect to the road undulations. The most basic and reliable study can be done using the quarter car models. In this type of study, the suspension analysis of only a single wheel is done. The unsprung mass is assumed to be a fourth of the whole vehicle unsprung mass. In short, the vehicle is broken down to only one wheel and that system is analysed.

The popularity of using such study for the vertical dynamics of the suspension system is because this type of model shows an accuracy very close to that of half car and full car models. The only drawback of such analysis is that it does not take into consideration the pitch and roll motion of the vehicle.

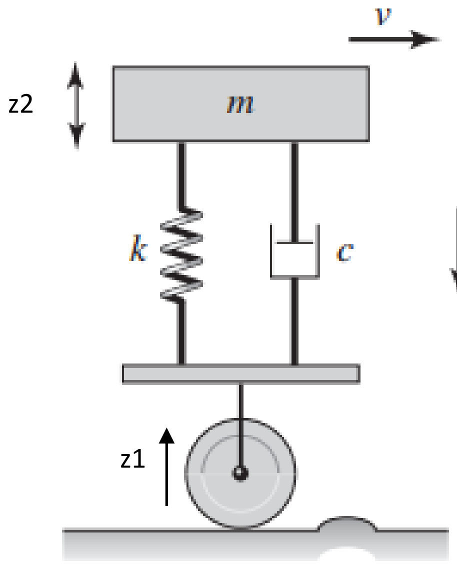


Fig 2: Conventional Quarter Car model [7]

The quarter car model is represented basically by 4 elements:

- Sprung Mass
- Unsprung Mass
- Spring element
- Shock Absorbing element (damper).

The governing equation defining the motion of the vehicle is:

$$m\ddot{z}_2 = k(z_1 - z_2) + c(\dot{z}_1 - \dot{z}_2)$$

1.4 Objective of Present work

- The objective for designing the suspension system is to find the governing equation defining the motion of the unsprung mass (vehicle body) with respect to the motion of the wheel. The simplest model with least number of degree of freedom which can be used for this analysis is the Quarter car model.

- The objective of this work is to develop the mathematical model for the piston cylinder pneumatic system to show the benefits as it replaces the conventional coil spring and viscous damper system. Hence, the damping characteristics and response charts of this system are analysed and provide guidelines to make an attempt to develop a practical setup for experimental study. This will further be used for the quarter car model using the air suspension system proposed in the present work.

- To carry out this work, the design considering dimensions and ergonomics of two wheeler "Honda Dio" were adopted for developing the model. Using an orifice in the air shock absorber, we aim to achieve equivalent damping as offered by same using hydraulic fluid. The various reasons to use air dampers are provided and parameters for good performance will be analysed.

CHAPTER 2

LITERATURE REVIEW

1. **Bachrach** and **Rivin** (1983) determined the dynamic stiffness of an air damped spring. Their work indicate the variation of spring stiffness and losses with the changes in the component parameters like the dimensions. It was shown that the dimensions of capillary affects the frequency of the damped system. [1]

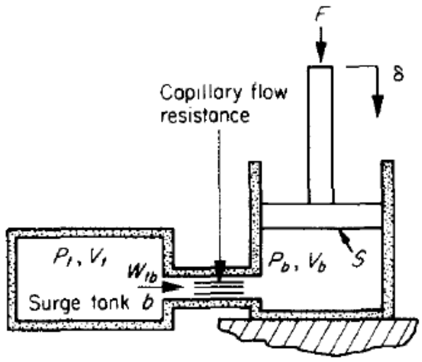


Fig 3: Model used for component effect analysis [1]

2. **Henderson** and **Raine** (1998) derived a nonlinear suspension model based on the orifice damped pneumatic system used in an ambulance stretcher. They studied the effects of variation of orifice size on the pressure and hence the system parameters using this model[2].

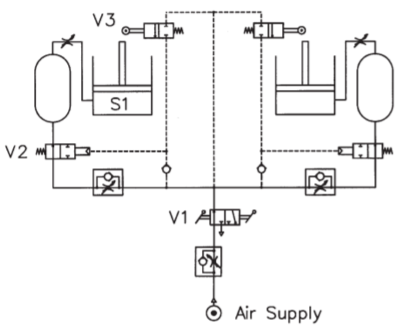


Fig 4: Air Suspension system for ambulance stretcher [2]

3. **Mats Berg** (1999) developed a three dimensional mathematical model for an air spring. This model is known as the GENSYS model. He validated the model by performing experiments and further concluded that the computational efforts for this model is moderate. This model is used extensively for simulations in air suspension analysis [3].

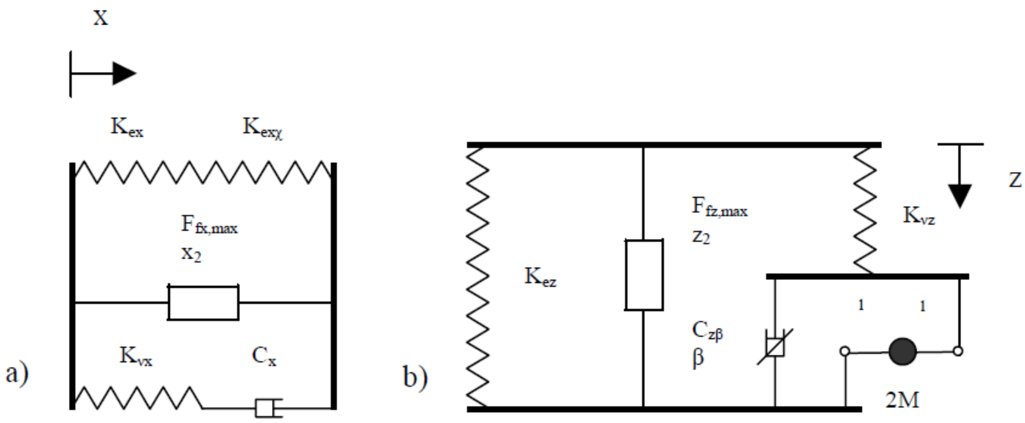


Fig 5: a) GENSYS horizontal model and b) GENSYS Vertical model. [3]

4. **Sorli et. al** (1999) did a dynamic analysis of a double acting pneumatic actuator controlled with the help of a digital valve. Two different analysis were presented, first one assumed the thermodynamic transformation and its simulation was carried out in MATLAB/Simulink, while in the second one the energy equation was introduced. The simulation results of both the analysis were compared and analyzed [4].

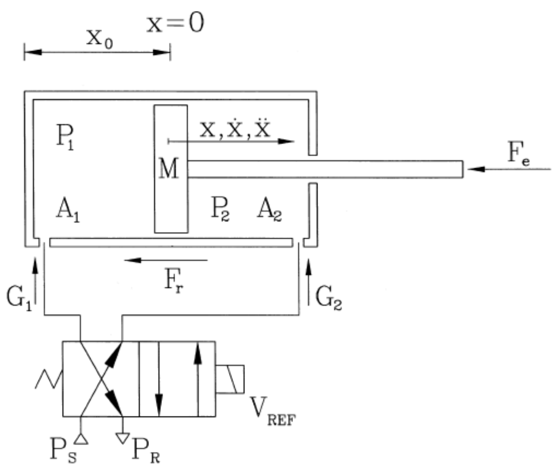


Fig 6: Pneumatic Actuator using digital valve control [4]

5. **Quaglia and Sorli** (2001) presents a dimensionless model of a pneumatic suspension with an external tank and also provides its design consideration. The model could describe the axial force and effective area apart from its geometric size [5].

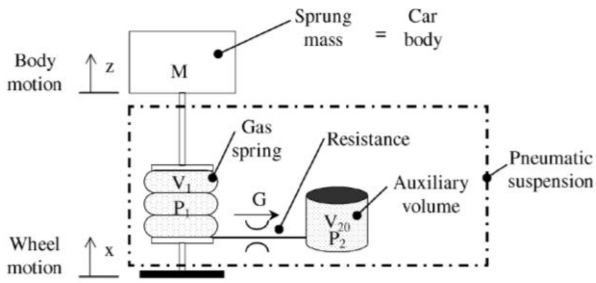


Fig 7: The air spring model used for dimensionless model [5]

6. **Ranjit Todkar** (2011) analytically studied the Maxwell's model of an air damper which used an air-cylinder arrangement coupled with a tank as a reservoir. He used the model to develop a quarter-car SDOF model of suspension with the air damper in parallel to conventional quarter-car model. He carried out experiments to find the spring rate and damping constants for the air damper and used it in the obtained values in the study. The results showed a decrease in the motion transmissibility with an increase in the spring rate and air damping ratio [6].

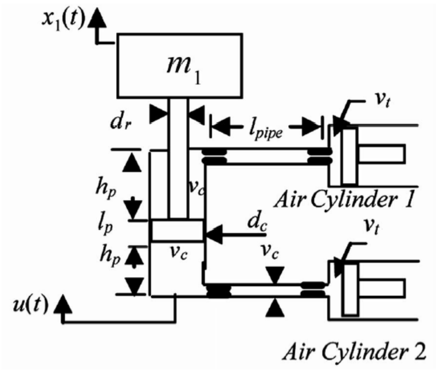


Fig 8: Air-cylinder based air suspension model [6]

7. **Harris and Piersol** showed in their book that air has a larger energy storage capacity per unit weight than metals and hence the potential energy storage of suspensions made of air is more efficient than of conventional systems. They showed that, in vibration-isolation

systems, the conventional systems have a higher natural frequency than the systems using air suspension [7].

8. **Nieto et al**. (2008) developed an analytical model for an auxiliary tank air suspension system which was characterized experimentally. The spring rate, damping coefficient and motion transmissibility were modeled. Both, the nonlinear and its linear model were given[8].

9. **Chang and Lu** (2008) developed a dynamic air spring model with the incorporation of the heat transfer phenomenon of the airspring. Their developed dynamic model of air spring was incorporated into multi body vehicle dynamics model in order to study the vehicle dynamics characteristics using ADAMS and MATLAB/Simulink environments. Their analytical model was validated from the experimental results too [9].

10. **Sreedhar et al.** (2013) built a simplified air suspension multi body system model and the leveling valve model with a customized function in ADAMS software. A full commercial passenger vehicle was developed to evaluate the ride performance and in further to obtain force excitation. They created the model and evaluated using experimental tests [10].

11. **Haider J. Abid et al.** (2015) developed an air suspension model equivalent to a conventional passive suspension model using Simulink and OptiY optimization tool. They used the GENSYS model with an auxiliary chamber. The optimized and derived the parameters of the air suspension system like the dimensions of the system components and initial pressures. They matched the responses of the passive suspension model and the proposed GENSYS model [11].

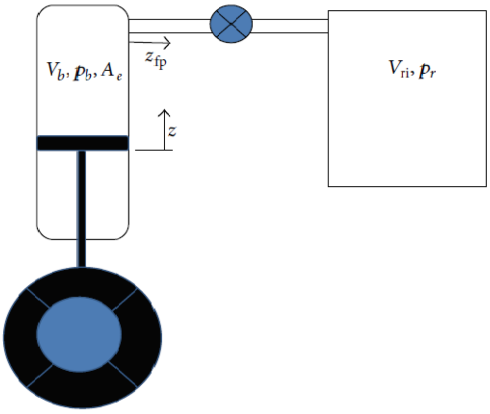


Fig 9: GENSYS Air suspension model [11]

12. **Zeng et. al** (2017) experimentally analyzed the variation in the damping effect for different orifice for a dual chamber air spring. It was found out that the damping coefficient of the system decreases with an increase in the ratio of area/(perimeter)^2. The variation with orifice number and tightness factor was also obtained. Formulas were derived for calculating the damping coefficient for square and circular shaped orifice [12].

CHAPTER 3

MATHEMATICAL MODELLING

3.1 Assumptions

- We have assumed the process to be polytropic.
- Friction between the piston and the cylinder is negligible.
- We have considered the road irregularities as a sinusoidal wave.
- Thermal effects due to compression and expansion of air are negligible and hence have not been taken into account.
- Air is considered to be as perfect gas.
- Ambient conditions remain constant throughout.
- Air leakage through the clearance between cylinder and piston is negligible.
- Resistive, inductive, capacitive and propagation effects in pipes neglected.

3.2 Closed Cylinder System (without spring)

The proposed system consists of an internally bored cylinder and a reciprocating piston which is given a sinusoidal wave input which resembles the bumps and pits on the road. The cylinder is plain without orifice and the input is given to the piston using a cam and follower mechanism with zero dwell. The bumps on the road produce vibrations in the arrangement, which are damped by the air present in the cylinder.

Using the equation:

$$P_1 {V_1}^n = P_2 {V_2}^n$$

Assuming, both the sides opposite to cylinder piston is symmetric and pressure on both the sides is assumed to be equal to P_{atm}.

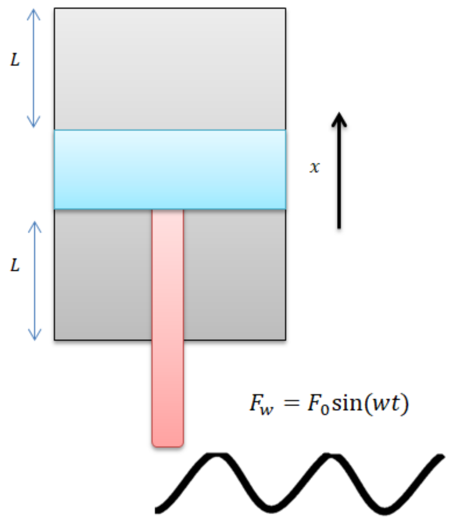


Fig 10: Closed system without spring model

Assuming polytropic process,

$$\therefore P_{atm} {V_1}^n = P_a (V_1 - Ax)^n$$

Where V_1=A.L

$$\therefore P_a = \frac{P_{atm} {V_1}^n}{(V_1 - Ax)^n}$$

$$= \frac{P_{atm}}{\left(1 - \frac{x}{L}\right)^n}$$

Using Binomial Expansion (x<<L)

$$= P_{atm}\left(1 + \frac{nx}{L}\right)$$

Similarly,

$$P_b = \frac{P_{atm} {V_1}^n}{(V_1 + Ax)^n}$$

$$= P_{atm}\left(1 + \frac{x}{L}\right)^{-n}$$

$$= P_{atm}\left(1 - \frac{nx}{L}\right)$$

<u>Assuming piston in equilibrium in dynamic condition:</u>

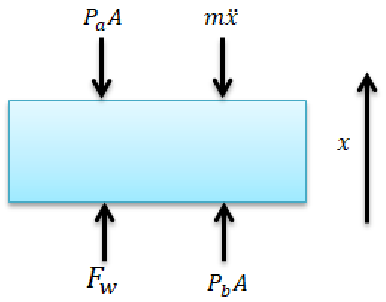


Fig 11: Free Body Diagram of Piston

$$m\ddot{x} = -F_a + F_w + F_b$$

$$m\ddot{x} = P_a.A + F_w + P_b.A$$

$$m\ddot{x} = -P_{atm}.A\left(1+\frac{nx}{L}\right) + F_w + P_{atm}.A\left(1-\frac{nx}{L}\right)$$

$$m\ddot{x} = -P_{atm}\frac{A.n.2.x}{L} + \frac{F_w}{m}$$

$$\frac{d^2x}{dt^2} = \frac{-P_{atm}.A.n.2.x}{mL} + \frac{F_w}{m}$$

This is a second order differential equation and its solution can be obtained by:

$$\left(D^2 + \frac{2.P_{atm}.A.n}{m.L}\right)x = \frac{F_w}{m}$$

Assuming:

$$\frac{2P_{atm}.A.n}{m.L} = \zeta^2$$

So, Solution is given by:

$$x = CF + PI$$

$$D^2 + \zeta^2 = 0 \rightarrow D = \pm\zeta$$

$$CF = c_1 cos(\zeta t) + C_2 sin(\zeta t)$$

$$CF = C_1.cos\sqrt{\frac{2.P_{atm}.A.n}{m.L}}t + C_2.sin\sqrt{\frac{2.P_{atm}.A.n}{m.L}}t$$

$$for\,\frac{1}{D^2+\zeta^2}\left(\frac{F_w}{m}\right)$$

$$PI = \frac{F_o.sin(\omega t)}{m.(\zeta^2-\omega^2)}$$

Hence, the required solution for the above differential equation is:

$$x = c_1.cos\sqrt{\frac{2.P_{atm}.A.n}{m.L}}t + c_2.sin\sqrt{\frac{2.P_{atm}.A.n}{m.L}}t + \frac{F_o sin(\omega t)}{m.(\zeta^2-\omega^2)}$$

The above equation determines the deflection of spring with respect to the given input which in this case is a sinusoidal function.

The above derivation does not take into consideration the damping effect of air.

If we consider damping then air friction co-efficientβ;

$$m\ddot{x} + \beta\dot{x} = -F_a + F_w + F_b$$

$$\ddot{x} + \left(\frac{P.A_p.2n}{mL}\right)x + \beta\frac{\dot{x}}{m} = \frac{F_w}{m}$$

CF Solution:

$$D^2 + \beta\frac{D}{m} + \left(\frac{2.P.A.n}{mL}\right)x = \frac{F_w}{m}$$

$$D = \frac{-\frac{\beta}{m} \pm \sqrt{\frac{\beta^2}{m^2} - 4.\left(\frac{P.A.2.n}{m.L}\right)}}{2}$$

Here, for calculating critical damping coefficient:

$$\frac{{\beta_{critical}}^2}{m^2} = 4.\left(\frac{P.A.2.n}{m.L}\right)$$

Now there are 3 cases:

- Over Damping ($\beta > \beta_{critical}$)
- Critical Damping ($\beta = \beta_{critical}$)
- Under Damping ($\beta < \beta_{critical}$)

Using the critical damping condition we determined the value of damping coefficient for the following parameters:

Table 1: Data (Honda Dio)

Parameters	Value
Spring Stiffness(k)	81370 N/m
Mass of Piston + Mass of Tire(m)	5 kg
Pressure(atmospheric)	101325 Pa
Cylinder Diameter(D)	22 mm
Total length of piston (2L)	144 mm
Polytropic Constant	1.3

Using the above value the value of $\beta_{critical}$ is 166.78 N.s/m.

- The β value is unknown and it is to be determined experimentally or by assuming a suitable damping ratio.
- Due to the complexity of the above equation, the further solution is developed using MATLAB Coding Environment.

Later in the report, we have analysed the responses for the above proposed model.

3.3 Closed Cylinder System (with spring)

This system consists of a spring loaded piston through which an extra spring force is applied on to the piston. The system used here is very much similar to the one used in the previous case except for the presence of the spring.

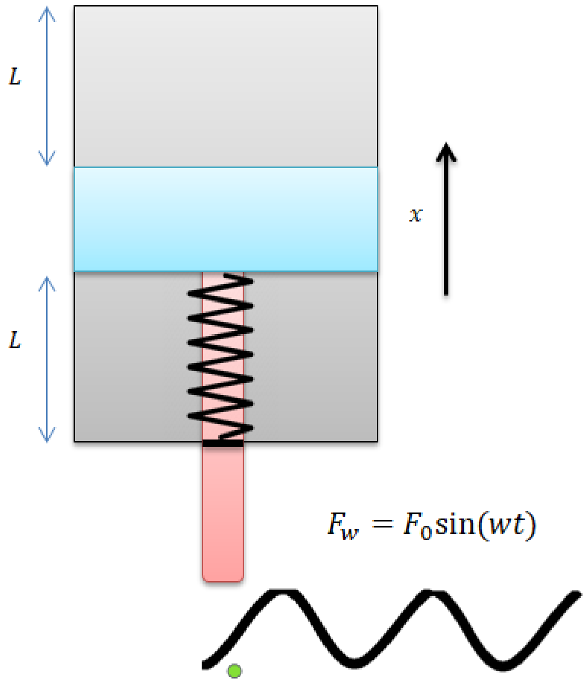


Fig 12: Closed System with spring

Using the equation:

$$P_1 {V_1}^n = P_2 {V_2}^n$$

Assuming, both the sides opposite to cylinder piston is symmetric and pressure on both the sides is assumed to be equal to P_{atm}.

Assuming polytropic process,

Initially,

$$P.(A.L)^n = P_b.\left(A.(L+x)\right)^n$$

$$P_b = \frac{P.L^n}{(L+x)^n}$$

$$P.L^n.L^{-n}.\left(1+\frac{x}{n}\right)^{-n} = P\left(1-\frac{nx}{n}\right)$$

$$P(A.L)^n = P_a\big(A(L-x)\big)^n$$

$$P_a = \frac{P.L^n}{(L-x)^n} = P.L^n L^{-n}\left(1-\frac{x}{L}\right)^{-n} = P\left(1+\frac{nx}{L}\right)$$

$$P_b = \frac{P.L^n}{(L+x)^n} = P.L^n L^{-n}\left(1+\frac{x}{L}\right)^{-n} = P\left(1-\frac{nx}{L}\right)$$

Assuming piston in equilibrium in dynamic condition:

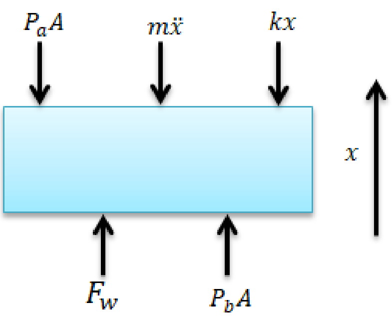


Fig 13: Free body diagram of piston

$$F_o sin(\omega t) + P_b.A_P = P_a.A_p + K.x + m.\dot{x}$$

$$F_w + P\left(1-\frac{nx}{L}\right).A_p = P\left(1+\frac{nx}{L}\right)A_p + K.x + m\ddot{x}$$

$$m\ddot{x} + K.x + \frac{P.A_p.2.n.x}{L} = F_w$$

Solution without considering damping:

$$m\ddot{x} + K.x + P.A_p 2\frac{n}{L}.x = F_w \quad Where: \left(F_w = F_o sin(\omega t)\right)$$

$$\ddot{x} + \left(\frac{K}{m} + \frac{P.A_p.2n}{mL}\right)x = \frac{F_w}{m}$$

CF Solution:

$$D^2 + \left(\frac{K}{m} + \frac{P.A_p.2.n}{m.L}\right) = 0$$

$$CF = c_1 cos\sqrt{\frac{K}{m} + \frac{P.A_p.2n}{m.L}}t + c_2 sin\sqrt{\frac{K}{m} + \frac{P.A_p.2n}{m.L}}t$$

PI Solution:

$$PI = \frac{F_w/m}{D^2 + \frac{K}{m} + \frac{P.A_p.2n}{mL}}$$

$$PI = \frac{F_o.sin(\omega t)}{m\left(-\omega^2 + \frac{K.P.A_p.2n}{m.m.L}\right)}$$

$\text{Total Solution} = \text{CF} + \text{PI}$

$$x = c_1 cos\sqrt{\frac{K}{m} + \frac{P.A_p.2n}{m.L}}t + c_2 sin\sqrt{\frac{K}{m} + \frac{P.A_p.2n}{m.L}}t + \frac{F_o sin(\omega t)}{m\left(-\omega^2 + \frac{K}{m}\frac{P.A_p.2n}{m.L}\right)}$$

- The above equation determines the deflection of spring with respect to the given input which in this case is a sinusoidal function.
- The above derivation does not take into consideration the damping effect of air.

If we consider damping then air friction co-efficientβ;

$$m\ddot{x} + \beta\dot{x} = -F_a + F_w + F_b$$

$$\ddot{x} + \left(\frac{K}{m} + \frac{P.A_p.2n}{mL}\right)x + \beta\frac{\dot{x}}{m} = \frac{F_w}{m}$$

CF Solution:

$$D^2 + \beta\frac{D}{m} + \left(\frac{K}{m} + \frac{2.P.A.n}{mL}\right)x = \frac{F_w}{m}$$

$$D = \frac{-\frac{\beta}{m} \pm \sqrt{\frac{\beta^2}{m^2} - 4.\left(\frac{k}{m} + \frac{P.A.2.n}{m.L}\right)}}{2}$$

So, for critical damping coefficient:

$$\frac{\beta_{critical}{}^{2}}{m^2} = 4.\left(\frac{k}{m} + \frac{P.A.2.n}{m.L}\right)$$

Now there are 3 cases:

1. Over Damping (β>$\beta_{critical}$)
2. Critical Damping (β=$\beta_{critical}$)
3. Under Damping (β<$\beta_{critical}$)

Using the critical damping condition we determined the value of damping coefficient for the parameters assumed in the previous model.

Using the above equations, the value of $\beta_{critical}$ is 1286.55 Ks/m.

- The β value is unknown and it is to be determined experimentally or by assuming a suitable damping ratio.
- Due to the complexity of the above equation, the further solution is developed using MATLAB coding environment.
- Later in the report, we have analysed the responses for the above proposed model.

3.4 Cylinder System with Orifice

Additional assumptions:

The flow here is considered compressible flow. The process is isothermal in nature. Friction is assumed as constant for simplicity and further assumptions are mentioned whenever required in the following model since it is comparatively more practical and complex model.

Case 1: The excitation provided is a function of time x = f(t) ; (For analytical purpose)

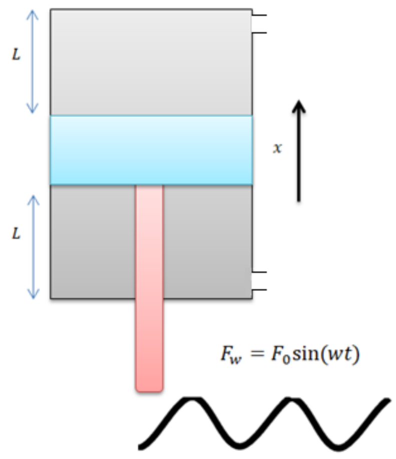


Fig 14: Cylinder system with orifice

For j = 1, 2 for respective chambers (Top and Bottom)

$$V_1 = A_P(x_0 - x)$$
$$V_2 = A_P(x_0 + x)$$
$$V_j = A_P(x_0 \pm x) \to 1$$

Since we are considering a polytropic process,

P = Pressure and ρ = Density

$$\frac{P}{(\rho)^n} = Const\,an\,t$$
$$\frac{P_1}{(\rho_1)^n} = \frac{P_2}{(\rho_2)^n} = \frac{P_i}{(\rho_i)^n}$$
$$\rho_j = (\frac{P_j}{P_i})^{\frac{1}{n}}(\rho_i)$$
$$\rho_j = (\frac{P_j}{P_i})^{\frac{1}{n}}(\frac{P_i}{RT}) \to 2$$

For finding the mass flow:

$$d(m) = d(\rho V)$$
$$d(m) = \rho * d(V) + V * d(\rho)$$
$$\dot{m} = \rho * \dot{V} + V * \dot{\rho}$$

Hence for each chamber:

$$\dot{m} = \rho_j * \dot{V}_j + V_j * \dot{\rho}_j \rightarrow 3$$

Using equations 1, 2 and 3:

$$\dot{m}_j = (\frac{P_j}{P_i})^{\frac{1}{n}} (\frac{P_i}{RT})[\frac{d}{dt}\{A_P(x_0 \pm x)\}] + A_P(x_0 \pm x)[\frac{d}{dt}\{(\frac{P_j}{P_i})^{\frac{1}{n}}(\frac{P_i}{RT})\}]$$
$$\dot{m}_j = (\frac{P_j}{P_i})^{\frac{1}{n}} (\frac{P_i}{RT})[\pm A_P \dot{x}] + A_P(x_0 \pm x)\{(\frac{1}{P_i})^{\frac{1}{n}}(\frac{P_i}{RT})\}[(\frac{1}{n})(P_j)^{\frac{1}{n}-1}\frac{dP_j}{dt}]$$

Here, we can observe that mass flow rate depends on pressure inside the respective chamber of the cylinder and the displacement of the piston.

Rearranging the above equation to find the pressure variation with respect to time:

$$\frac{dP_j}{dt} = \frac{nRT}{A_p(\mathrm{x}_0 \pm \mathrm{x})\{\frac{P_j}{P_i}\}^{\frac{1}{n}-1}}\dot{m}_j \pm \frac{\mathrm{nP_j}}{(\mathrm{x}_0 \pm \mathrm{x})}\dot{x}$$

Case 2: The excitation provided to the piston is through a crank (For experimental purpose)

Assuming that excitation is provided by rotating crank

N = RPM of the crank

Vd = Displacement volume without clearance = Apx

Q = Volume flow rate in m3/sec

ηv = Volumetric efficiency

$$Q = \frac{\eta_v V_d N}{2 * 60}$$

Mass flow rate can also be given as:

$$\dot{m} = \rho Q$$
$$\dot{m} = \frac{\eta_v V_d N \rho}{2 * 60}$$

For each chamber:

$$\dot{m}_j = \frac{\eta_v V_d N \rho_j}{2*60}$$

$$\dot{m}_j = \frac{\eta_v V_d N P_j}{120RT}$$

To find the discharge velocity:

$$Q = uA_o$$

$$\frac{\eta_v V_d N}{2*60} = u\frac{\pi}{4}d^2$$

$$u = \frac{4\eta_v V_d N}{2\pi * 60 * d^2}$$

Where,

d = diameter of the orifice

u = Velocity of air at orifice discharge

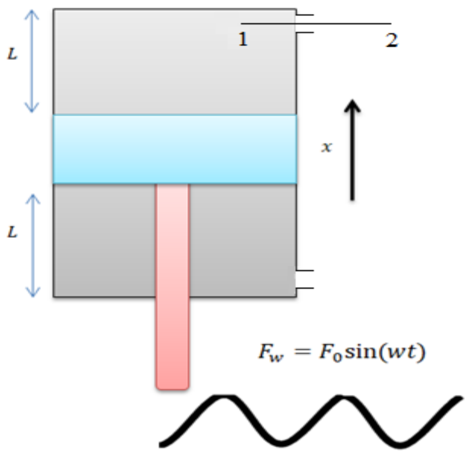


Fig 15: Bernoulli's analysis across orifice

Applying Bernoulli Principle for compressible flows:

$$\int \frac{d(P)}{\rho} + \int v d(v) + \int g d(z) = Cons\tan t$$

$$\int \frac{d(P)}{\rho} + \frac{v^2}{2} + gz = Cons\tan t$$

$$RT \int \frac{d(P)}{P} + \frac{v^2}{2} + gz = Cons\tan t$$

$$RT \ln(P) + \frac{v^2}{2} + gz = Cons\tan t$$

Applying at two points on the same horizontal level, just outside and inside the orifice:

$$RT \ln(P_1) + \frac{v_1^{\ 2}}{2} + gz_1 = RT \ln(P_2) + \frac{v_2^{\ 2}}{2} + gz_2$$

Consider,

- Point 1 inside the cylinder, so v_1 = u (discharge velocity) and $P_1 = P_j$ (Instantaneous pressure in the chamber.
- Point 2 outside in the atmosphere, so v_2 = Velocity of the vehicle and P_2 = atmospheric pressure

Solving for pressure variation inside the chamber:

$$RT \ln(P_j) + \frac{u^2}{2} + 0 = RT \ln(P_{atm}) + \frac{v_{vehicle}^{\ \ 2}}{2} + 0$$

$$\ln(\frac{P_j}{P_{atm}}) = \frac{1}{2RT}(v^2{}_{vehicle} - u^2)$$

$$P_j = e^{\frac{1}{2RT}(v^2{}_{vehicle} - u^2)} \quad P_{atm}$$

It is derived that the pressure of the gas inside the cylinder is dependent on velocity of gas.

To derive the variation of velocity (u) with respect to piston displacement (x):

$$\frac{dP_j}{dt} = \frac{nRT}{A_p(\mathrm{x}_0 \pm \mathrm{x})\{\frac{P_j}{P_i}\}^{\frac{1}{n}-1}} \dot{m}_j \pm \frac{\mathrm{nP_j}}{(\mathrm{x}_0 \pm \mathrm{x})} \dot{x}$$

..... (A)

$$P_j = P_{atm} e^{\frac{(v^2{}_{vehicle} - u^2)}{2RT}} \text{ ...... (B)}$$

$$\dot{m} = \frac{P_j}{RT} A_{orifice}\, u \text{ ....(C)}$$

Substituting equation (B) and (C) in equation (A), we get:

$$[\, P_{atm} e^{\frac{(v^2{}_{vehicle} - u^2)}{2RT}} \frac{(-u)}{RT}\frac{du}{dt}\,] = \left[\frac{nRT}{A_p(\, x_0 \pm x)} \frac{P_{atm} e^{\frac{(v^2{}_{vehicle} - u^2)}{2RT}}}{RT} A_{orifice}\, u\, \frac{P_{atm}}{P_{atm} e^{\frac{(v^2{}_{vehicle} - u^2)}{2RT}}} \}^{\left(\frac{1}{n}-1\right)} \right] \pm \left[\frac{n}{(\, x_0 \pm x)}\, P_{atm} e^{\frac{(v^2{}_{vehicle} - u^2)}{2RT}} \frac{dx}{dt} \right]$$

After simplification:

$$\frac{(-u)}{RT}\frac{du}{dt} = \left[\frac{n}{A_p(\, x_0 \pm x)} A_{orifice}\, u \left\{ \frac{1}{e^{\frac{(v^2{}_{vehicle} - u^2)}{2RT}}} \right\}^{\left(\frac{1}{n}-1\right)} \right] \pm \left[\frac{n}{(\, x_0 \pm x)} \frac{dx}{dt} \right]$$

Now, if we assume $\left\{\frac{A_{orifice}}{A_P}\right\}$ is very very small, so we can neglect the middle term.

$$\frac{(-u)}{RT}\frac{du}{dt} = \pm \frac{n}{(\, x_0 \pm x)}\frac{dx}{dt}$$

Assuming zero clearance for simplicity:

$$\frac{(-u)}{RT}\frac{du}{dt} = \pm \frac{n}{(x)}\frac{dx}{dt}$$

Integrating the above equation:

$$\int \frac{(-u)}{RT}\, du = \pm \int \frac{n}{(x)}\, dx$$

$$\frac{(-1)}{2RT} u^2 = \pm\, n\, ln(x) + C$$

Boundary condition:

U (x = L) = 0 && U (x = 0) = 0

$$- C = \pm\ n\, ln(L)$$

Hence, final variation of velocity of gas near the orifice entrance, inside the cylinder:

$$\frac{(-1)}{2RT} u^2 = \pm\ n\, ln(x) \mp n\, ln(L)$$

$$\frac{(1)}{2RT} u^2 = \mp n\, ln(x) \pm n\, ln(L)$$

Hence for all x > 0:

For compressing gas:

$$u^2 = 2RTn \ln(\frac{x}{L})$$

For expanding gas:

$$u^2 = 2RTn \ln(\frac{L}{x})$$

The equations are not valid for x =0, at x = 0 we assume u = 0;

This formulation is not yet verified.

The pressure and velocity variations are exponential functions. Also, solving linear differential equation with dependent parameter in the exponential form is very complicated to implement in coding environment. So the general solution for this case has not been obtained yet.

Balancing all the forces on the piston:

$$m\ddot{x} + \beta\dot{x} + (P_1 - P_2)A + F_r = F_{excitation}$$

The friction force is a strong non-linearity of the system. In general, it can be a function of the relative speed, working pressures, displacement, type of load (radial or axial) and of time, as it can change during the actuator life:

Fr = f (P1, P2, x, t)

Although in the present analysis the friction force is assumed to be constant.

The value of n – polytropic process coefficient can also vary with the stroke and the working conditions. For simplicity, we take n as constant.

Hence, this model is mathematically complete but solution is complex to calculate. The second order differential equation becomes non-linear, when all the parameters are substituted from the above derived equations, so response graphs for this case and validation is pending for future.

3.5 Matlab Response Analysis:

A Simulink model is as shown in figure below. This analysis is for step input.

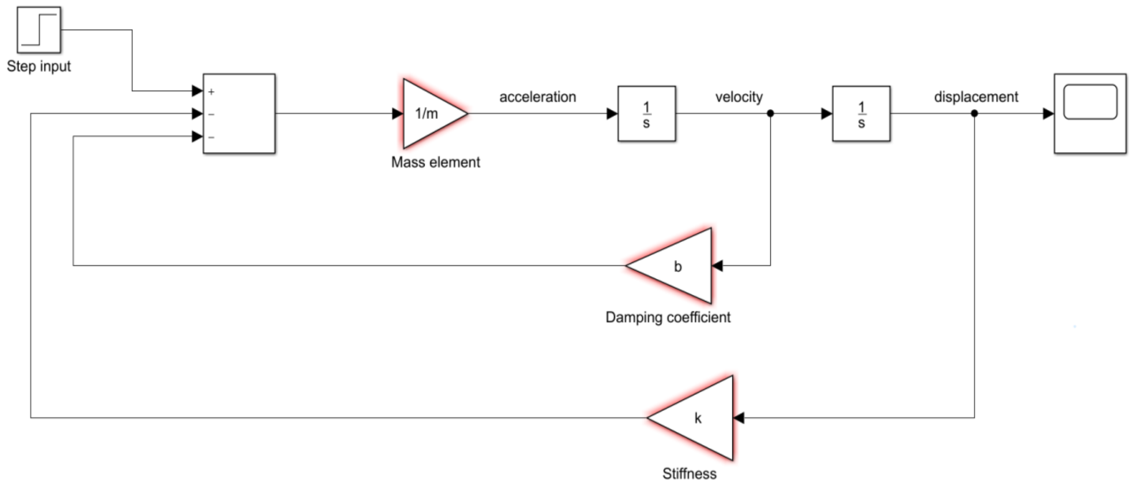


Fig 16:1 MATLAB Simulink Model for the proposed system

A single degree of freedom model for the proposed system is shown. It incorporates the damping coefficient, the stiffness and the mass of the piston. A **step input** was provided and the response to the step input is observed through the scope.

Response for closed system without spring:

Considering the critical damping coefficient as 203.42 Ns/m and the damping coefficient ratio as 0.3. The damping coefficient thus obtained was 61.2 Ns/m. The responses at critical damping coefficient and at the obtained damping coefficient are as follows:

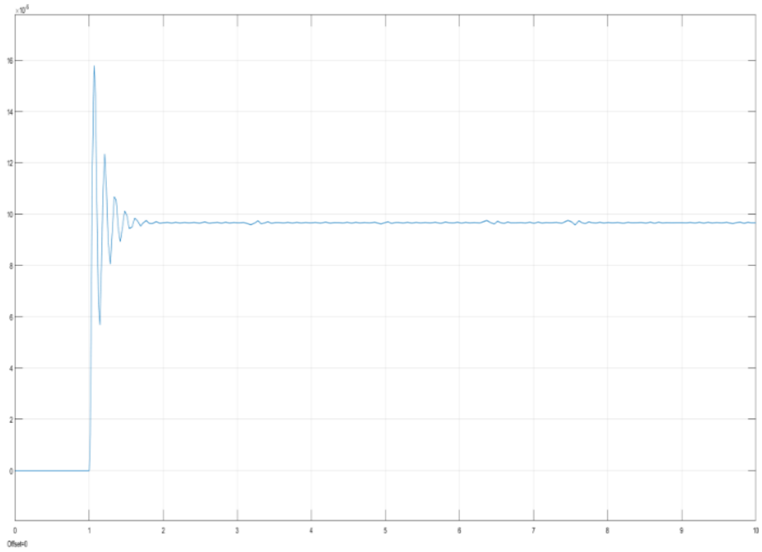


Fig 17: Step Response of the closed model at the assumed damping ratio

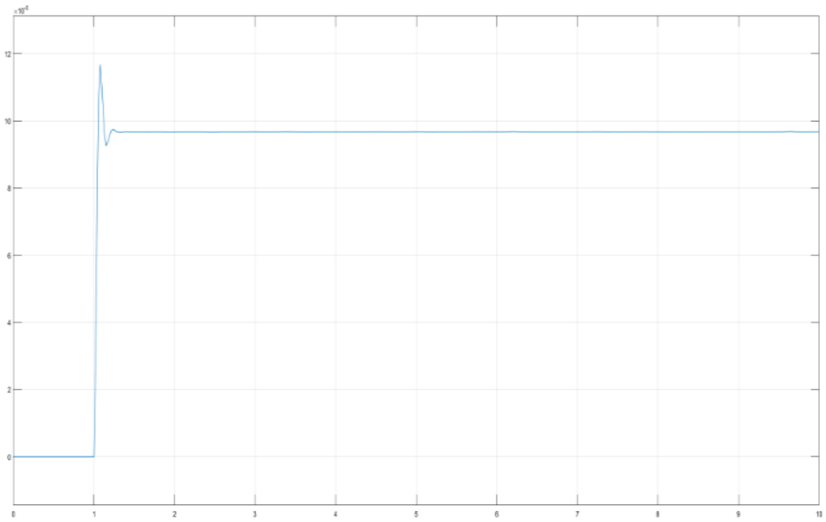
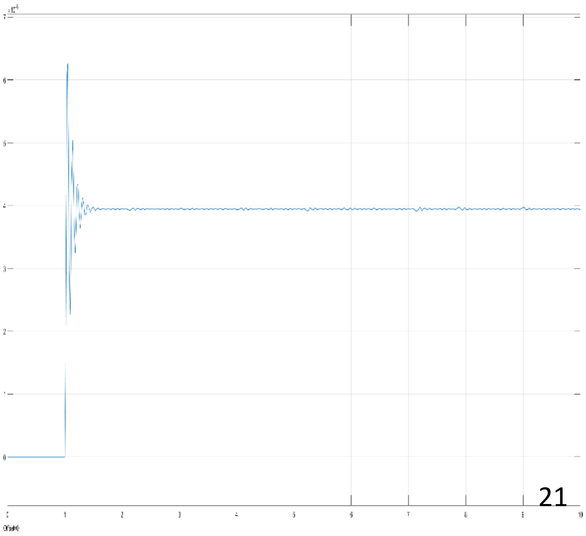


Fig 18: Step Response of the closed model at critical damping coefficient

Response for closed system with spring:

Considering the critical damping as 318 Ns/m and the damping coefficient ratio as 0.3. The damping coefficient thus obtained was 95.4 Ns/m. The responses at critical damping coefficient and at the obtained damping coefficient are as follows:



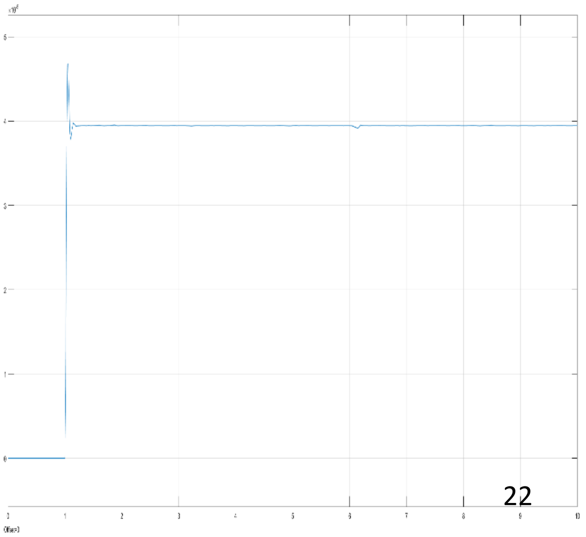


Fig 19: Step Response of the spring model at the assumed damping ratio

Fig 20: 2 Step Response of the spring model at critical damping coefficient

Interpretation

As it can be seen from the responses, the damping is better at the critical coefficient. The frequency of damping is higher for the model with the spring while the frequency is lower for the closed system. But, it can be observed that the amplitude of vibration is lower for the spring system than with the closed one.

Using Matlab Simulink, we could produce response graphs for critical damping coefficient.

We can also perform response analysis by developing our own Matlab code using the 'dsolve' function. We cannot determine the exact value of air damping coefficient for our design without experimentation. Hence, the following variation in response charts of our design suspension with respect to air damping coefficient can give some idea about the range of values that can be considered for comfortable air shock absorption.

MODEL – I

For closed system without spring:

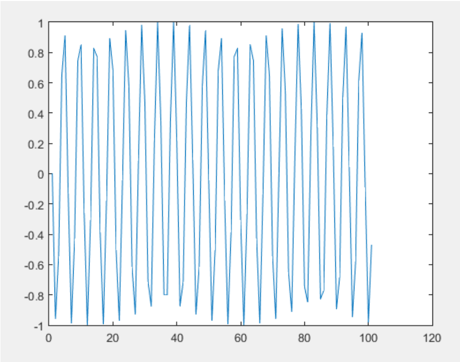
Matlab Code based on mathematical model: -

```
eqn1 = 'D2x + ((P*A*2*n)/(m*L))*x + (Dx)*(b/m) = (F/m)';
ini1 = 'x(0) = 0, Dx(0) = 5.67';
x = dsolve(eqn1,ini1);  % ( b = Air damping coefficient)
```

The general solution obtained is as follows:

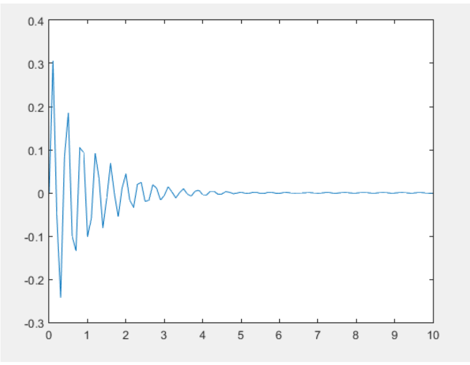
x = (F*L)/(2*A*P*n) - (L*exp(-(t*(L*b - (L*(L*b^2 - 8*A*P*m*n))^(1/2)))/(2*L*m))*(25*F*(L*(L*b^2 - 8*A*P*m*n))^(1/2) + 25*F*L*b - 567*A*P*m*n))/(100*A*P*n*(L*(L*b^2 - 8*A*P*m*n))^(1/2)) - (L*exp(-(t*(L*b + (L*(L*b^2 - 8*A*P*m*n))^(1/2)))/(2*L*m))*(25*F*(L*(L*b^2 - 8*A*P*m*n))^(1/2) - 25*F*L*b + 567*A*P*m*n))/(100*A*P*n*(L*(L*b^2 - 8*A*P*m*n))^(1/2))

<u>Input excitation force:</u> (F – t graph) { SI units }



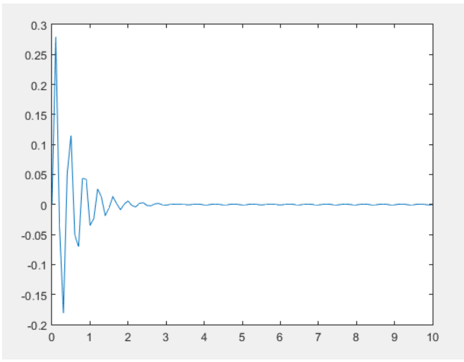
(a)

<u>Response at b = 10 Ns/m:</u> (x- t graph)



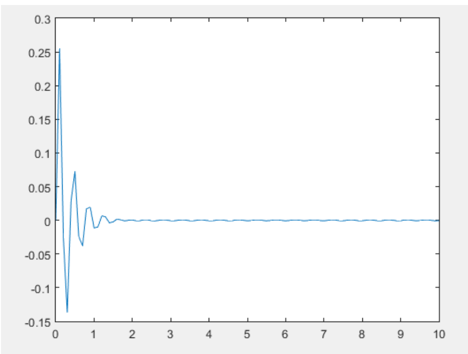
(b)

<u>Response at b = 20:</u>



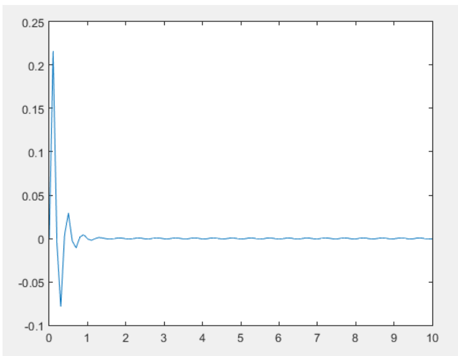
(c)

<u>Response at b = 30:</u>



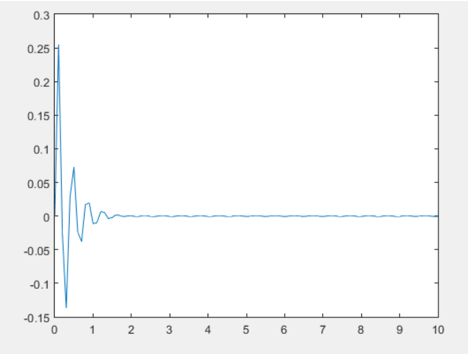
(d)

<u>Response at b = 50:</u>



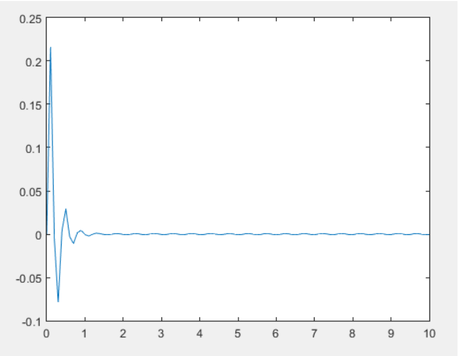
(e)

__Response at b = 30:__



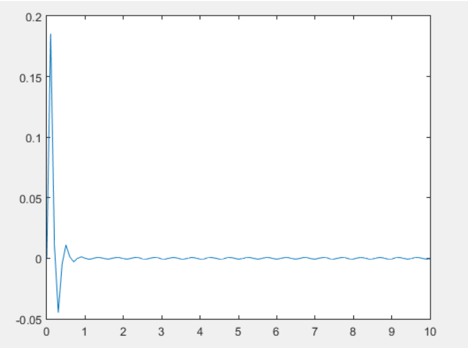
(d)

__Response at b = 50:__



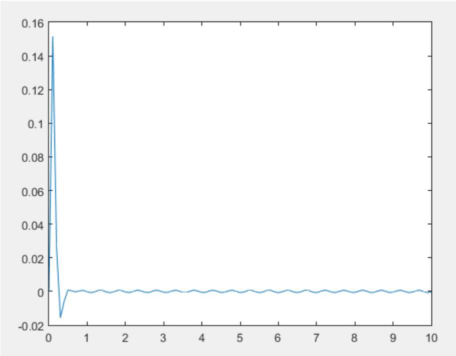
(e)

__Response at b = 70:__



(f)

__Response at b = 100:__



(g)

Fig 21: (a) Input force; (b) to (g) Responses for model I

Hence, we observe that the initial excitation amplitude is larger than the cylinder half length (L), but the suspension response anplitude gets lower by increasing the value of air damping coeffecient (b) as the system shifts from underdamp to overdamp state. In this analysis, no spring is considered to support the system, which means the suspension is completely working on air. Leakage of air is not considered, friction or temperature effects are neglected.

MODEL - II

For closed system with spring:

Matlab Code based on mathematical model: -

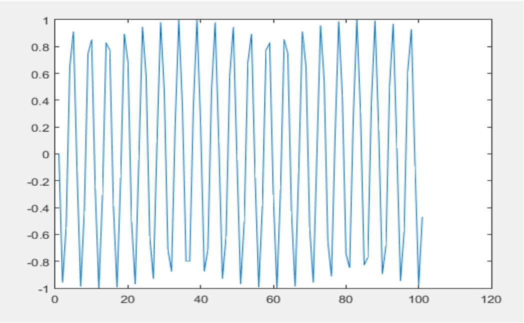
```
eqn1 = 'D2x + (((P*A*2*n)/(m*L))+(k/m))*x + (Dx)*(b/m) = (F/m)';
ini1 = 'x(0) = 0, Dx(0) = 5.67';
x = dsolve(eqn1,ini1);
```

b = Air damping coefficient (Ns/m)

The general solution obtained is as follows:

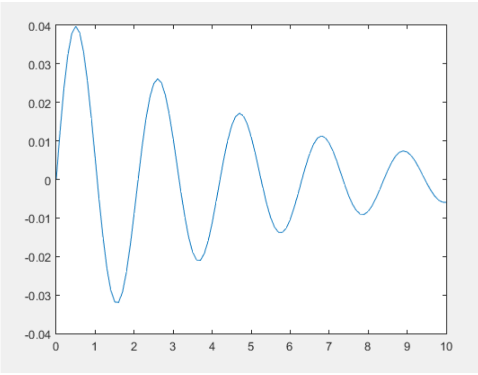
x = F/(m*(k/m + (2*A*P*n)/(L*m))) - (L*exp(-(t*(L*b + (-L*(- L*b^2 + 4*L*k*m + 8*A*P*m*n))^(1/2)))/(2*L*m))*(50*F*(-L*(- L*b^2 + 4*L*k*m + 8*A*P*m*n))^(1/2) - 50*F*L*b + 567*L*k*m + 1134*A*P*m*n))/(100*(-L*(- L*b^2 + 4*L*k*m + 8*A*P*m*n))^(1/2)*(L*k + 2*A*P*n)) - (L*exp(-(t*(L*b - (- L*(- L*b^2 + 4*L*k*m + 8*A*P*m*n))^(1/2)))/(2*L*m))*(50*F*(-L*(- L*b^2 + 4*L*k*m + 8*A*P*m*n))^(1/2) + 50*F*L*b - 567*L*k*m - 1134*A*P*m*n))/(100*(-L*(- L*b^2 + 4*L*k*m + 8*A*P*m*n))^(1/2)*(L*k + 2*A*P*n))

Input excitation force: (F – t graph)



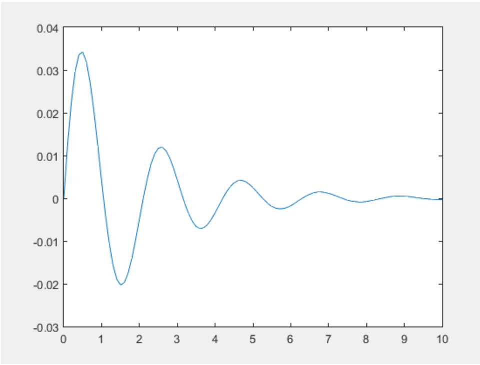
(a)

Response at b = 2 Ns/m:(x- t graph)



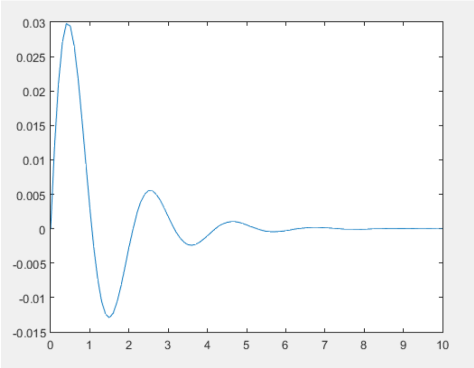
(b)

Response at b = 5:



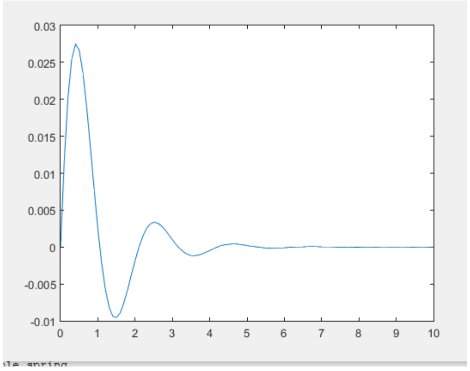
(c)

Response at b = 8:



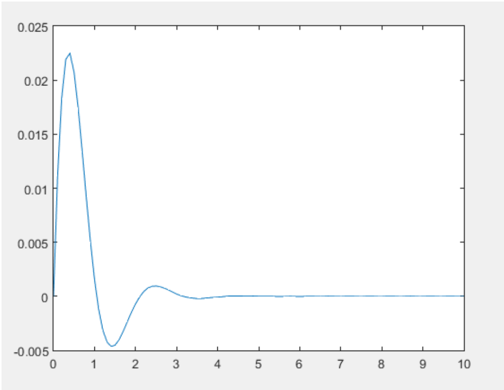
(d)

Response at b = 10:



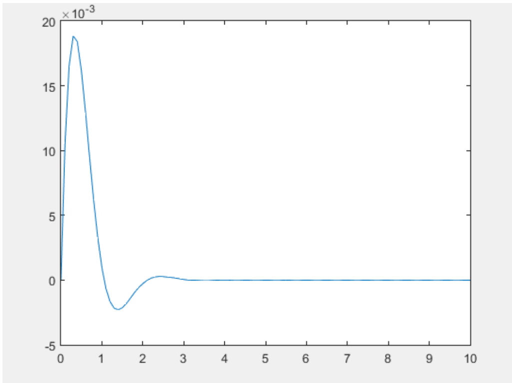
(e)

Response at b = 15:



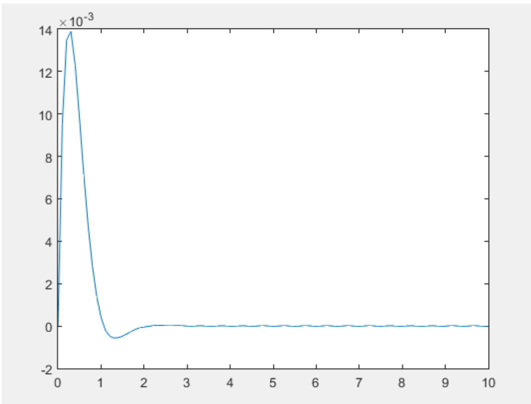
(f)

Response at b = 20:



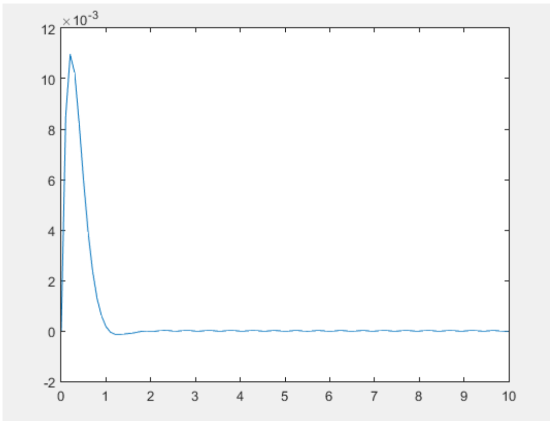
(g)

Response at b = 30:



(h)

Response at b = 40:



(i)

Fig 22: (a) Input force; (b) to (i) Responses for model II

- Here, it can be noted that because of addition of spring as a support system and shock absorber, the response displacement amplitude has been lowered. Also, the value of damping coefficient as well as critical damping coefficient gets reduced since the spring also acts against the excitation in series with the air damper. Again, the friciton losses, air leakage have been neglected and analysis is isothermal in nature.

- Further, in the report, a design is proposed for the air suspension system based on the mathematical models described in previous chapters. The analytical calculations and manufacturiing steps are given for the proposed design with stress simulations wherever necessary.

- The response charts shown above are obtained using the dimensions from the proposed design in the next chapter.

Based on the response graph in this chapter, we can consider damping coefficient of air ; b = 25 Ns/m, for the further analysis since this value gives the best comfort among all. Actually, the value can only be obtained experimentally, but since we cannot perform experiment due to COVID 19 situation, so we take this value from the best graph obtained through matlab.

CHAPTER 4

PROPOSED DESIGN

4.1 Introduction

The shock absorber was developed according to the dimensions and ergonomics of the two wheeler "Honda Dio". Fig shows the CAD model of the modified damper that is used in the design. The important aspects of this damper are:

Annular Orifice: The idea of our project is to replace the hydraulic fluid inside the damper with atmospheric air so that there are no problems of fluid leak and the shock-absorber to become stiff and obsolete. An orifice is provided to intake the air from the atmosphere. If the same had not been provided, the air might get leaked out and the damping would be nullified. Hence an orifice is utmost necessary.

Instead of a conventional orifice, the annular area around the piston rod is used as orifice. In order to obtain that in the commercially available air cylinders, the bottom part of the cylinder is cut and is replace by a cap with a whole inside it. This hole has a diameter marginally larger than the piston rod in order to provide the annular area for the orifice. The cap is threaded from inside and the bottom portiion of the cylinder is threadeon the outside to proivde a threaded joint between the cap and the cylinder, making the cap replaceable in he event of wear and tear.

The total length that can be swept by the piston is 72mm making the wheel travel of 72mm on the vehicle suspension. The maximum distance between the chassis end and the wheel end is equal to 288.55mm.

Cylinder: The metallic cylinder consists an inner bore of diameter 22mm and the total length of 144mm. The thickness of the cylinder is derived using the principles of Machine Design.

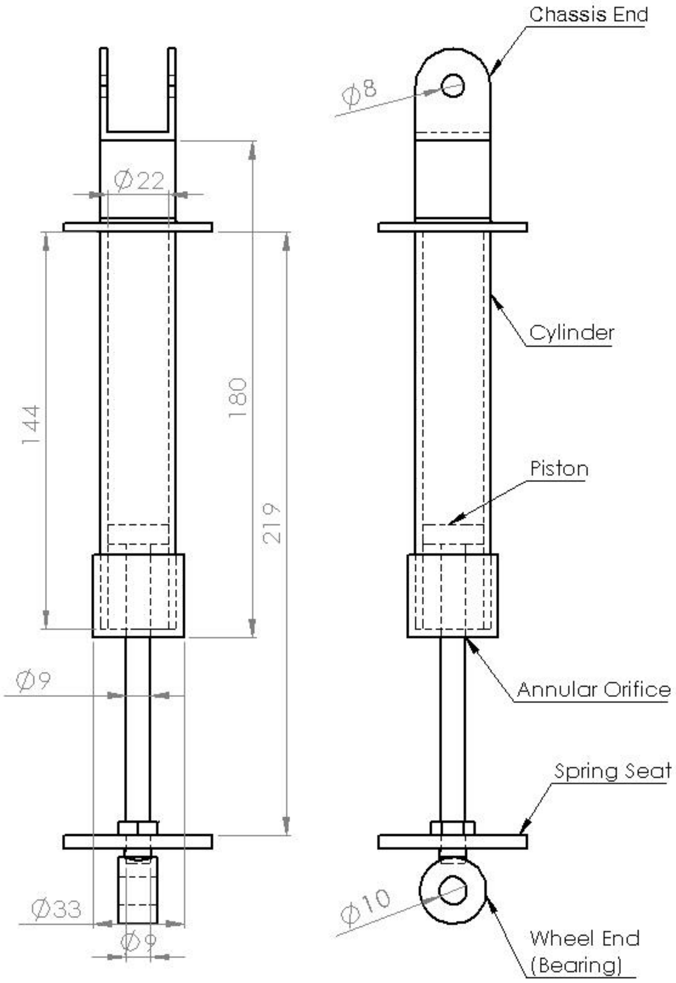


Fig 23: CAD Drawing of damper

4.2 Cylinder, Piston & Spring Design:

- **Thickness of cylinder wall**

The cylinder in shock absorber system has two functions:

a. To retain the atmospheric air or working fluid.

b. To guide the piston during its motion.

c. To dissipate heat released as the working fluid is being compressed.

Material used:

The cylinder is generally made of cast iron or cast steel, alloyed or unalloyed or mild steel. The material of cylinder should be such that it can retain sufficient strength at high pressure and temperature. In the design of shock absorber aluminium alloy is considered to fulfil the purpose.

Following are the properties of the material:

Syt: 276 Mpa

Design methods:

The cylinder is to be designed for strength, thermal stress, wear etc. Under the strength consideration the cylinder may be designed by treating it as a thick or thin cylinder. It depends upon the ratio of mean diameter to thickness.

Assuming it to be thin cylinder, the wall thickness t may be determined from the Hoop stress criterion. The equation for the stress can be written as:

$$t = \left(\frac{P_{max} * D}{2 * \sigma_c}\right)$$

Here;

D=22mm

σ_c = Permissible hoops stress= 100Mpa (FOS=3)

To find P_{max}, using adiabatic relations,

$$Pi*(Vi)^n=Pmax*(Vf)^n$$

Where, n=1.4

Vi=Internal Area*(Length of cylinder)/2

Vf=Internal Area*(Length of cylinder –length of stroke)/2

Pi= 165*9.81/(Piston Area) = 4.26 Mpa

Therefore, Pmax=4.26*(72/36)^1.4= 11.242 Mpa t=11.242 *22 / 100

=2.47 mm

- **Thickness of cylinder head**

$$t_h = D\sqrt{\frac{K\, p_{max}}{\sigma_c}}$$

K= 0.162 (constant)

t_h = 22root(0.162*11.242/100)

=2.96mm

- **Thickness of Piston head**

$$t_h = D\sqrt{\frac{3\, p_{max}}{16\, \sigma_b}}$$

Permissible bending stress for Aluminium alloy=70Mpa

t_{ph}=22*root(3*11.242/16/70)

=3.82mm

- **Design of spring**

The Shock absorber contains a spring made of spring steel and the final product after coiling the spring around the damper is as follows:

The spring is one of the most essential parts of a suspension system. It determines the stiffness of the suspension system which in turn decides the performance of the vehicle. The following section shows the spring design considerations and the analytical calculation for finding the stiffness of the spring. The analytical solution is followed by the ANSYS Structural simulation of the suspension spring.

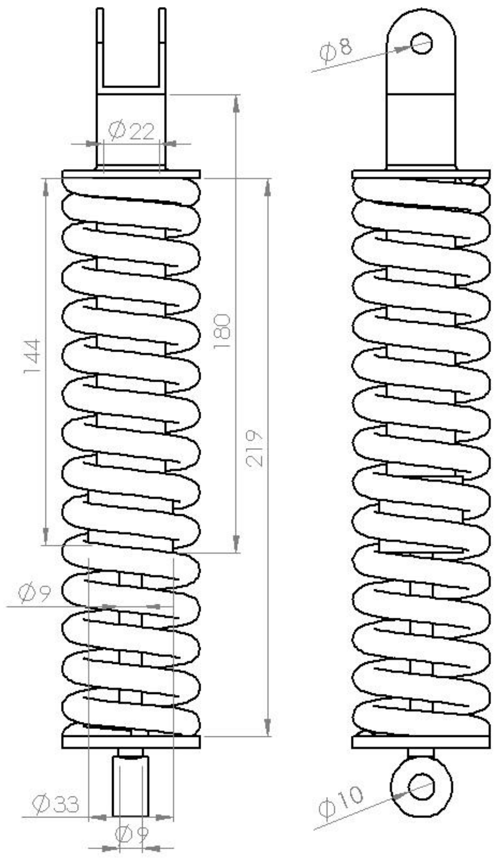


Fig 24: CAD Drawing of Shock Absorber

The material being considered is Spring Steel. Some benefits of spring steel are listed below:

1. Spring Steel is typically used because of its high yield strengths, resistance to deformation and its ability to return to its original shape.

2. Spring steel is advantageous for its flatness while It has not been practical to formulate flatness tolerances for cold rolled carbon spring steel to represent the range of widths and thicknesses and variety of properties produced in coils and cut lengths.[1]

3. It has the unique ability to be formed, shaped, and post heat treated.[2]

These physical characteristics are what allow spring steel to be a general use steel.

Calculations for stiffness and maximum shear stress:

- **Material properties:**

Modulus of rigidity (G) = 78000N/mm2

Ultimate Tensile Strength= 2800 Mpa

- **Dimensions of spring:**

Mean diameter of a coil, D=44 mm

Diameter of wire, d = 10mm

Spring Index (C) = D/d = 4.4

Total no of coils, n1= 16

Outer diameter of spring coil, D0 = D+d =54 mm

No of active turns, n= 14

- **Loading Parameters:**

Weight of Scooty (Dio) = 105 kg

Let weight of 1person = 75 kg

Weight of 2 persons = 75×2=150 kg

Weight of bike + persons = 255 kg

Since the analysis is for rear suspension, the overall load distribution is 65% to the rear side and 35% to the front side.

Load on Rear Suspension = 255*0.65=165.75kg

Considering dynamic loads it will be 1.5 times the load value.

W = 248.625kg= 2439.01 N

Now stiffness of spring is given by:

$$K = \frac{G \times d^4}{8 \times D^3 \times n}$$

$$K = \frac{78000 \times 10^4}{8 \times 44^3 \times 14}$$

$$K = 81.755\ N/mm = 81755\ N/m$$

Now maximum shear stress is given by:

$$\tau_{max} = \frac{8 \times W \times D}{\Pi \times d^3} \times K_w$$

Where, K_w is the Wahl stress Factor and is given by:

$$K_w = \frac{4C - 1}{4C - 4} + \frac{0.615}{C}$$

$$K_w = \frac{4 \times 4.4 - 1}{4 \times 4.4 - 4} + \frac{0.615}{4.4}$$

$$K_w = 1.36$$

Substituting the Wahl stress factor in shear stress equation:

$$\tau_{max} = \frac{8 \times 2439.01 \times 44}{\Pi \times 10^3} \times 1.36$$

$$\tau_{max} = 371.659\ Mpa$$

Now allowable shear stress is given by:

$$\tau_{allowable} = 0.5 \times Ultimate\ Tensile\ Strength/FOS$$

$$\tau_{allowable} = 0.5 \times 2800/3$$

$$\tau_{allowable} = 466.66\ Mpa$$

Since$\tau_{max} < \tau_{allowable}$, hence the design is safe and will not fail.

4.3 Finite Element Analysis

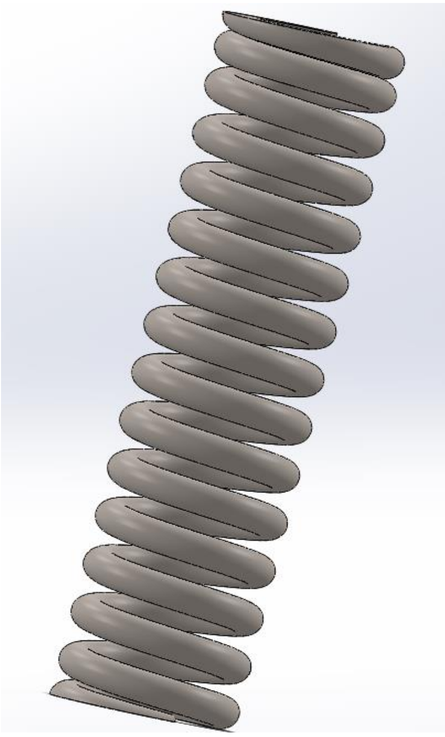
Static FEA was carried out using Ansys workbench to find out the stiffness and to check weather the spring will fail or not. The specifications of the spring is as mentioned above.

Step 1: Structural simulation.

The first step in the static structural simulation is the modelling of the object. Three Dimensional model of the spring was made using Solidworks Software which is presented below.

Step 2: Assigning the Material

The material used for analysis was structural steel with the modified values of Shear modulus and poisson ratio. The modified values simulated the material properties of Spring Steel with G = 78000MPa and Poisson Ratio of 0.3.



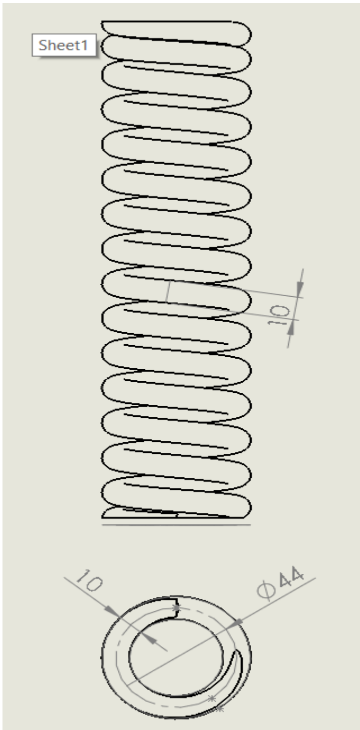


Fig 25: 3D Model of coil spring **Fig 26: Coil spring dimension**

Step 3: Meshing

Meshing plays a very important role in the results of a simulation. ANSYS is known for its ability to mesh complicated geometries and give satisfactory results. ANSYS Workbench default program controlled mesh type was used with fine sized elements. The mesh of the model looks as shown in Fig.

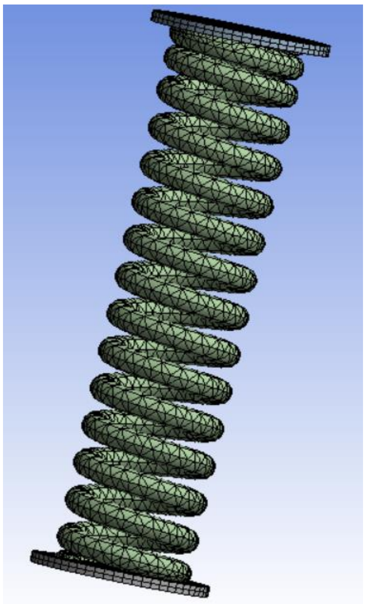


Fig 27: Quadratic Mesh

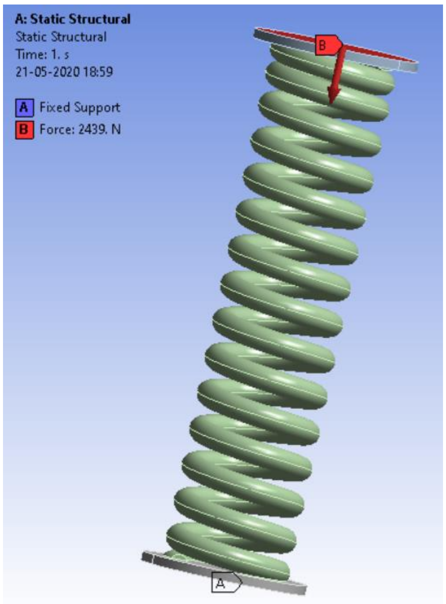


Fig 28: Problem Formulation

Step 4: Problem Formulation

Problem formulation includes the assignment of boundary conditions and loading conditions according to real life situations. For finding the spring stiffness, a load, as calculated in analytical spring design, 2439N was applied on the top end and the bottom end was fixed. Two end caps are created on the top and bottom of the spring for the application of uniform load and providing a better fixed geometry.

Step 6: Post-Processing

Post Processing is the step where the results are analyzed and the important data is obtained. The maximum deflection in the spring as shown in figure is 0.96985mm at a load of 15N. Hence the spring stiffness according to FEA is obtained as:

$K = F/\delta$

Hence, K = 2439/30.506 = 79.951N/mm

For the Failure Analysis, the FEA has to be carried out at the maximum load allowed on the shock absorber. This was calculated earlier as dynamic loading which is equal to 248.625kg or 2439N. The Maximum Shear Stress Analysis was carried out at that load and the results obtained are shown in Fig.

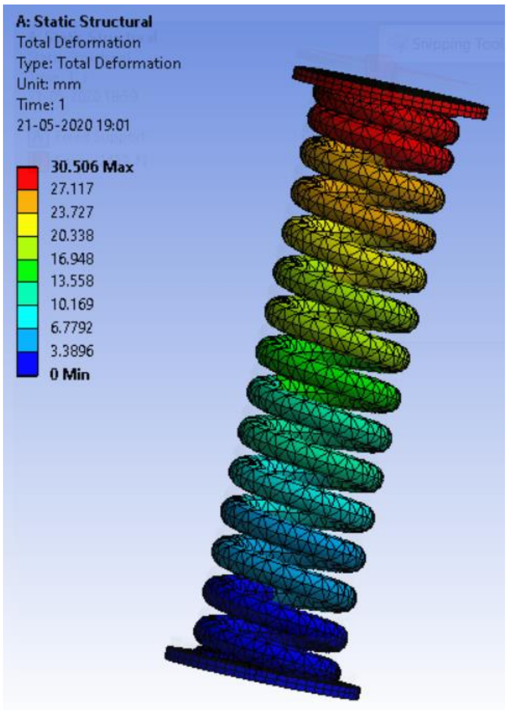


Fig 29: Total Deformation

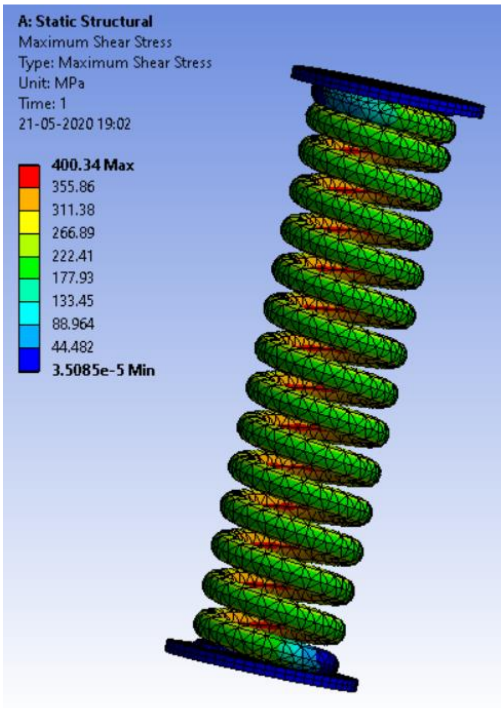


Fig 30: Maximum Shear Stress

The maximum shear stress obtained through simulation is 400.34 MPa which is under the permissible limit of 466.66MPa. Hence the spring is safe from failure.

4.4 Results & Discussion for the FEA analysis:

The stiffness value obtained through analytical calculations: 81.755 N/mm

The stiffness value obtained through FEA: 79.951 N/mm

Hence the percentage error,

$$e = \frac{Simulation - analytical}{analytical}$$

$$e = \frac{79.951 - 81.755}{81.755} * 100$$

$$e = -2.206\%$$

The Max Shear Stress obtained through analytical calculations: 371.659MPa

The Max Shear Stress obtained through FEA: 400 MPa

Hence the percentage error,

$$e = \frac{Simulation - analytical}{analytical}$$

$$e = \frac{400 - 371.659}{371.659} * 100$$

$$e = 7.62\%$$

4.5 Basic Manufacturing Guidelines:

The following steps can be followed to manufacture the proposed air damper system. The system would be an assembly of 3 main parts, namely piston, cylinder and the cylinder cap.

Step 1: A cylinder of required dimensions is produced by performing a set of turning and facing operations on a lathe machine. This will generate the outer profile of the cylinder.

Step 2: The cylinder is to be bored out using a boring tool of 22mm diameter for a length of 144mm.

Step 3: The inner lining of the cylinder can be finished by lapping to clear out any surface irregularities that might have generated during or after the material removal process.

Step 4: External threads are to be generated over the cylindrical surface at one of the ends over which a cylinder cap can be attached which itself has internal threads which can be generated by tapping operation. It should be ensured that the cap provides a partial seal so that air can escape through the cylinder and provide the required damping. This partial seal acts as an orifice that helps in the damping.

Step 5: Piston can be produced by turning a cylindrical stock to the required dimensions. The piston head can be turned to a diameter of 22mm while the stem is turned to a 9mm diameter.

External threads are to be produced at the stem side of the piston so that the spring seat can be bolted at this end.

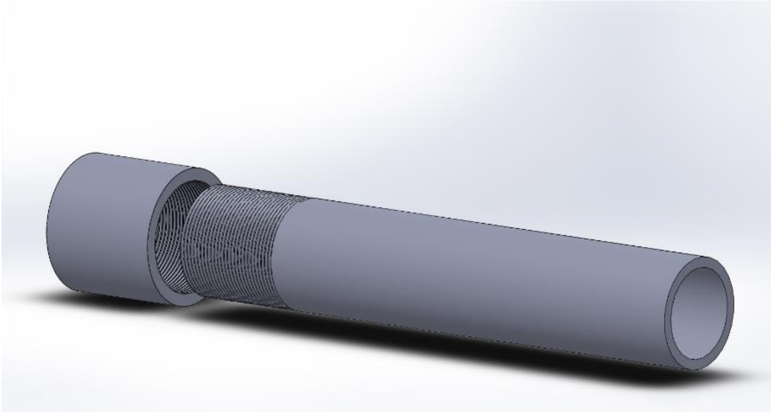


Fig 31: Threaded cylinder with end cap

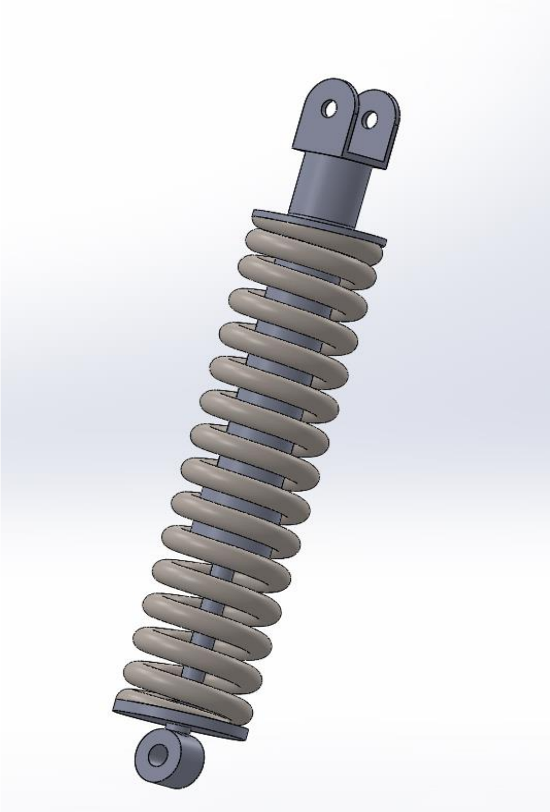


Fig 32: Piston **Fig 33: Damper Assembly**

Step 6: The fork end which is to be joined at the chassis frame can be fixed to the cylinder using arc welding process along the interface. The wheel end will be connected to the damper system using a bearing.

CHAPTER 5

Air and Oil – A comparative study

5.1 Analytical Response for proposed design

Assuming, no leakage for simplicity, it behave like closed sealed cylinder model described in math-models chapter. The design dimensions are taken from proposed design. These assumptions are considered to draw the comparison for performance between air and oil as damping fluid.

For Air Damping:

Forces on the piston:

For a piston displacement x,

Dead weight of bike – (165/2)*g

Spring Force – Kx

Pressure force from air – PA

Inertia Force – m$\ddot{x}$

Air Damping force - $\beta\dot{x}$

Excitation – F(t)

Initial Conditions: P_i = initial pressure of air at no load

$V_i = \pi r^2 2L$

At any point x: P_x = P

$V_x = \pi r^2(2L - x)$

Considering polytropic process – P = $P_i(1 + \frac{nx}{2L})$

Differential Equation:

$$m\ddot{x} + \beta\dot{x} + Kx + P_i\left(1 + \frac{nx}{2L}\right)A + \left(\frac{165}{2}\right)g = F(t)$$

For Oil Damping:

For a piston displacement x,

Dead weight of bike – (165/2)*g

Spring Force – Kx

Pressure force from Oil – PA

Inertia Force – m$\ddot{x}$

Oil Damping force - $c\dot{x}$

Excitation – F(t)

Initial Conditions: P_i = (Weight of Oil above piston / A) at no load

= (ρVg/A) = ρg (2L)

$V_i = \pi r^2 2L$

At any point x: P_x = P

$V_x = \pi r^2 (2L - x)$

Since, density is changing with compression and expansion of oil. So, it is difficult to represent instantaneous pressure P with density. We consider Bulk modulus (G) of oil as constant, since we have neglected temperature effects in the entire analysis in this report.

$$G = \left\{ \frac{-\,dP}{\frac{dV}{V}} \right\}$$

By integrating

$$\int_{P_i}^{P} dP = -G \int_0^x \frac{d(\pi r^2 x)}{V_i}$$

$P = P_i + \frac{Gx}{2L}$, represents instantaneos pressure at any given displacement x of piston.

Differential Equation:

$$m\ddot{x} + c\dot{x} + Kx + \left(P_i + \frac{Gx}{2L}\right) A + \left(\frac{165}{2}\right) g = F(t)$$

Now, considering the SAE 20W as the oil for our analysis. This is generally used in engine oil, damping oil, motor oil etc.

Density (ρ) = 860 kg/m^3

Acceleration due to gravity (g) = 9.81 m/s^2

Length of cylinder (2L) = 144 mm

Bulk Modulus of Oil = 1.66 * 10^9 pascal

Clearly, it can be observed that the damping effects for air and oil depend upon the damping coeffecient (β *for air* ; c *for oil*) and the initial pressure of the fluid (P_i) under no load. The initial pressure can be set manually based on the design of the damper. The damping coeffecient of the fluid depends on the design dimensions. Hence, for a given design, it can only be obtained experimentally.

Therefore, we will compare air and oil dampers using different values of damping coeffecient and initial pressure of fluid. The accurate value of damping coefficient for the proposed design shall be obtained experimentally as future work.

Case I: If Air is at atmospheric pressure at no load

The P_i of air = Atmospheric pressure = 101325 Pascal.

- For a given positive sinusoidal test input: F(t) = | sin(50*t) |
- These are the suspension responses (x – t graphs): {SI units}

Oil Damper:

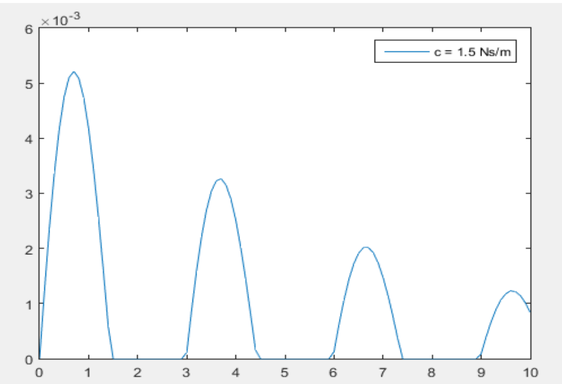


Fig 34: Oil Damper response

Air damper:

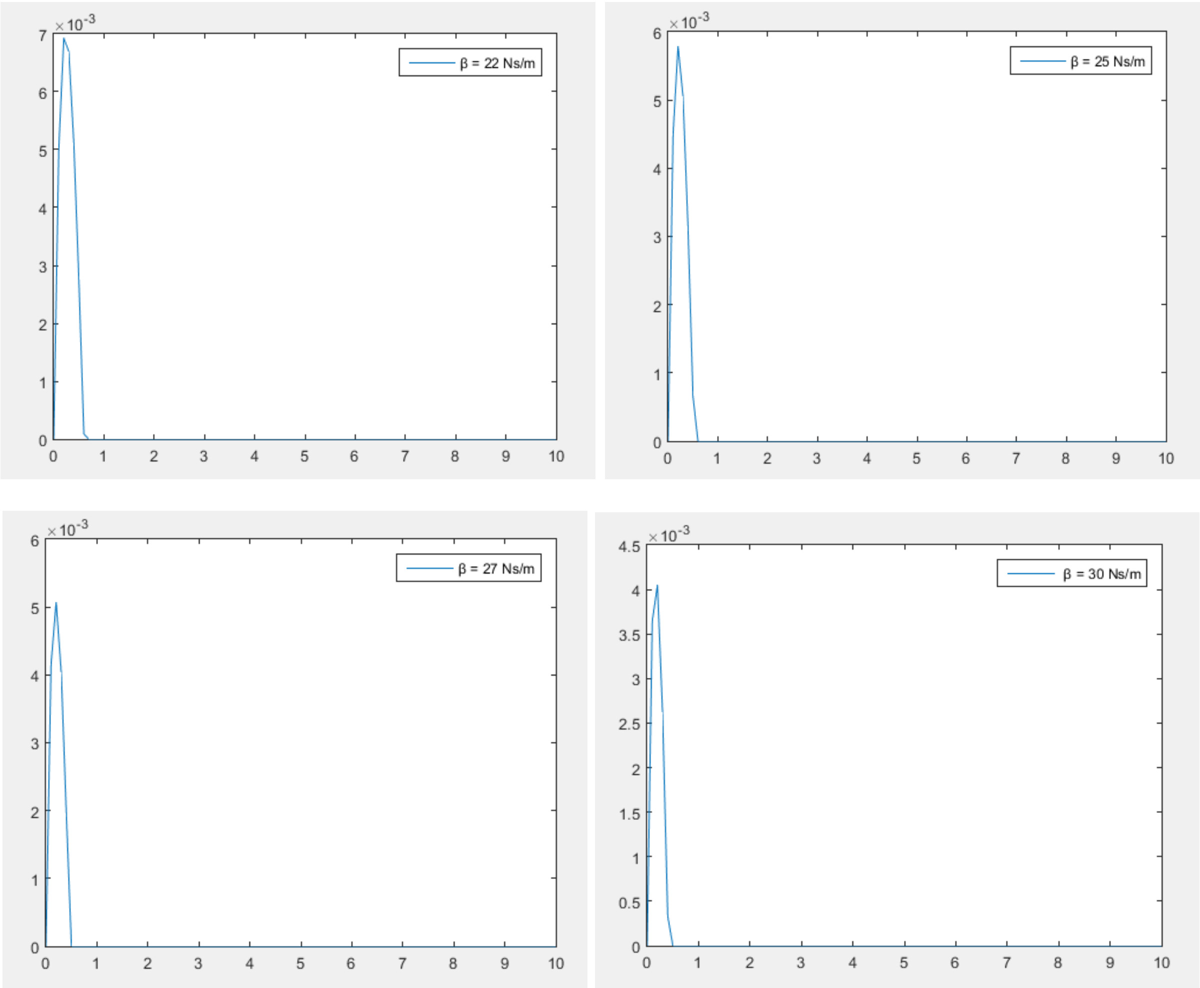


Fig 35: Air damper responses at atmospheric pressure

From analysis of above graphs,

- At c = 1.5 Ns/m [15] and $\beta \geq 25\ Ns/m$, the damping of air and oil is almost same but the oil damper gives a more smooth damping effect, whereas the air damper giver sharp and faster response.

- It is observed that for c = 1.5 Ns/m [15] and $\beta \geq 27\ Ns/m$, the Air damping effect is better compared to Oil damping.

Case II: If Compressed Air is filled inside cylinder at no load

- For a given positive sinusoidal test input: F(t) = | sin(50*t) |

These are the suspension responses (x – t graphs): {SI units everywhere}

From above graphs, it is clear that for $\beta \leq 27\ Ns/m$, Air damper does not outperform the conventional damper. So, we can improve damping by using compressed gas. Let's analyze how much initial pressure is required.

Air Damper:

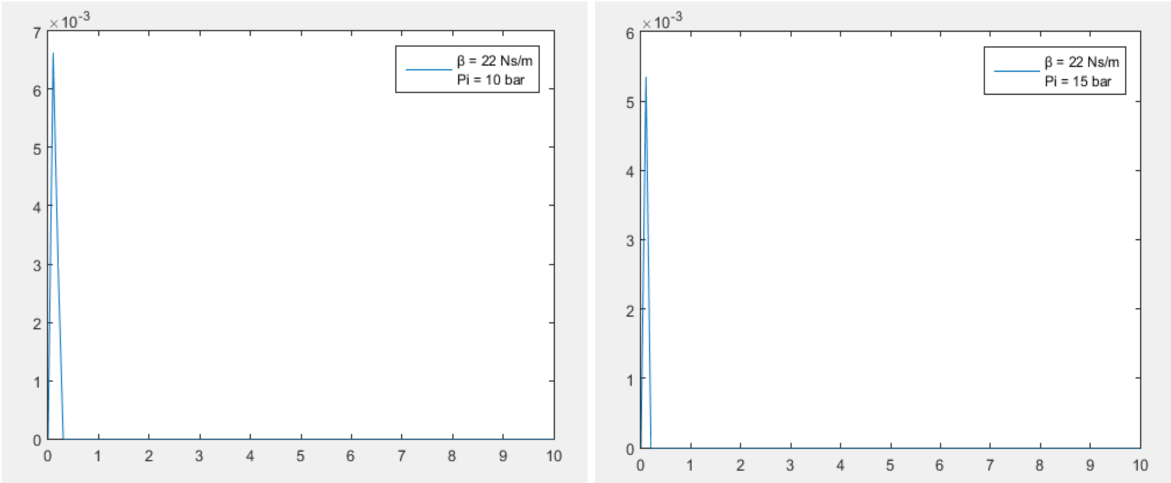


Fig 36: Air Damper responses at higher pressure at β=22 Ns/m

From analysis of above graphs,

- At c = 1.5 Ns/m [15] and $\beta = 22\ Ns/m$, the damping of air can be better than oil only if compressed air is used at $P_i \geq 15\ bar$.

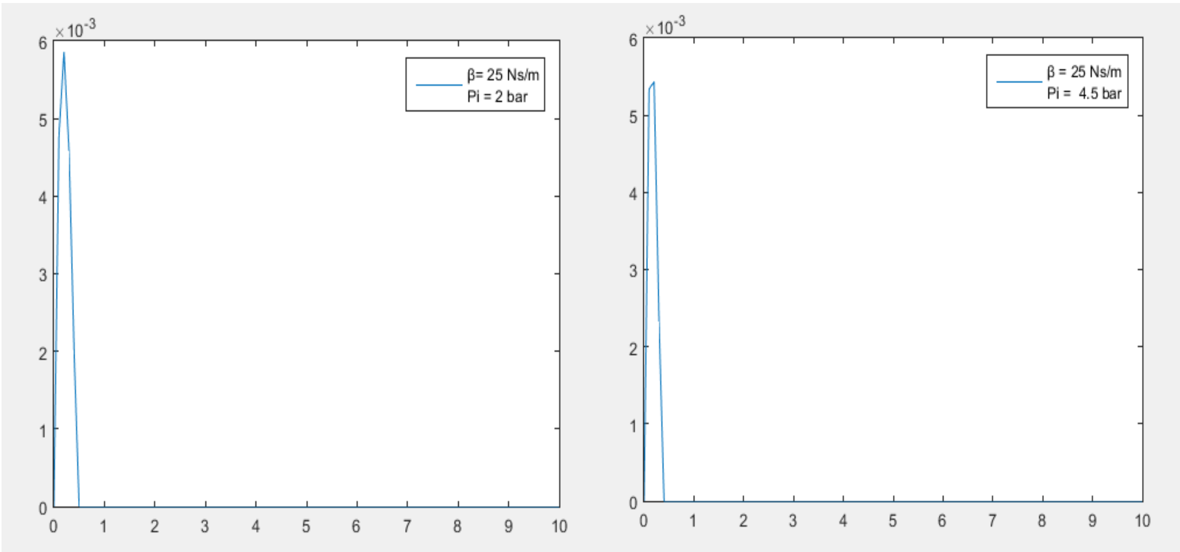


Fig 37: Air Damper responses at higher pressure at β=25 Ns/m

From analysis of above graphs,

- At c = 1.5 Ns/m [15] and $\beta = 25\ Ns/m$, the damping of air can be better than oil only if compressed air is used at $P_i \geq 5\ bar$.
- Also, it is observed that for very small change in the damping coefficient of air, the compressed pressure requirement falls significantly.

With these observations, we would like to conclude the analytical comparison of this report between oil and air damper for the basic model proposed in the report. The behaviour of air gets better, which is clearly potrayed here.

5.2 Theoretical Comparison:

Construction and Working of Oil damper:

Figure below shows the exploded view of a conventional hydraulic damper that uses oil as the working fluid. This conventional hydraulic damper uses viscous oil as a working fluid and this structure comes with certain difficulties. To understand this, let's first understand the working of the conventional oil damper.

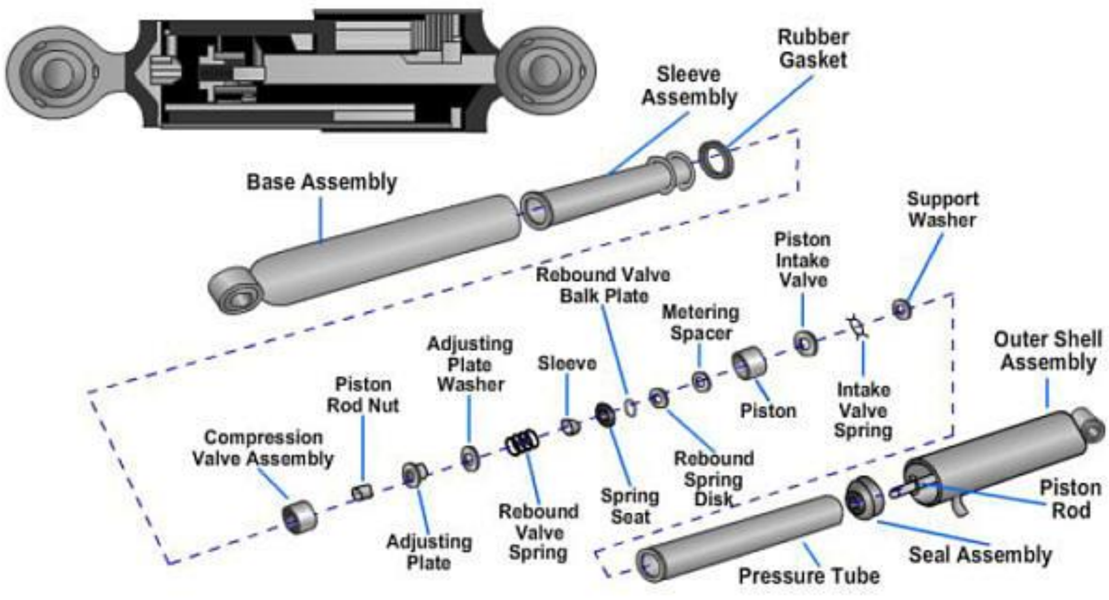


Fig 38: Oil Shock absorber [16]

The shock absorber is placed to absorb the road impacts and decrease the transmission of the wheel motion to the passenger by absorbing the impacts. As the wheel moves due to disturbance, the piston inside the damper also moves. This movement is resisted by the oil that is filled inside the tube of the shock absorber. As the piston moves with high energy, this energy is transmitted to the oil that is present inside the shock-absorber and this piston movement in turn increase the pressure inside the pressure tube. This increase in pressure resists the movement of the piston and hence the vehicle oscillations start to fade away instead of the ever longing oscillations due to the spring alone. Hence the failure of this shock absorber can be attributed to the failure of the viscous oil to damp the oscillations and make the system stiff without damping effect. This problem can occur if there is no oil inside and the system starts moving without the hydraulic fluid. The piston consists of a set of orifice distributed over its surface such that the oil passes through these orifice as it gets compressed. This transfer of oil to low pressure part of the cylinder through the orifice leads to release of heat. Hence, the absorbed energy is released in the form of heat. Also, this reduces leakage of oil.

Construction and Working of Air damper:

The air damping system consists of various components such as hollow cylinder, piston, orifice (variable and non-variable), orifice area controlling mechanism, compression rings, and cam and follower mechanism to provide reciprocating motion to the piston.

The whole suspension system is to be replaced with the air shock absorber. This system is basically a pneumatic cylinder. Orifice is provided on the cylinder on either side of the piston. A rod is extended from piston which can be connected to the wheel hub and hence to the tyre. The undulations from the road will be transmitted to the shock absorber via this rod to the piston. The previous air shock absorber systems use a compressor or a reservoir for the air flow to the cylinder space. But our proposed model uses only atmospheric air and hence there is no additional components for air storage. Different system can be developed by incorporating a spring in the lower part of the cylinder and removing the orifice of this bottom part. Hence the motion will be governed by the air pressure on one side of the piston while a spring on the other side.

The damping is done by using the damping properties of air. During rest, the piston is positioned in such a way that the piston rests on the bottom, inside the cylinder supporting the dead weight of the car. When the car encounters any undulation, the motion is transmitted via the connecting rod to the piston. As the piston moves upwards, the air inside the cylinder in upper chamber gets compressed. The extremely small diameter of the orifice does not let the air to move out freely. Hence the air compresses. The compressibility of the air starts to decrease because of this and thus a force acts in the downward direction. This force damps the motion of the piston until the compressibility of the air becomes zero and the system becomes stiff stopping the further motion of the piston.

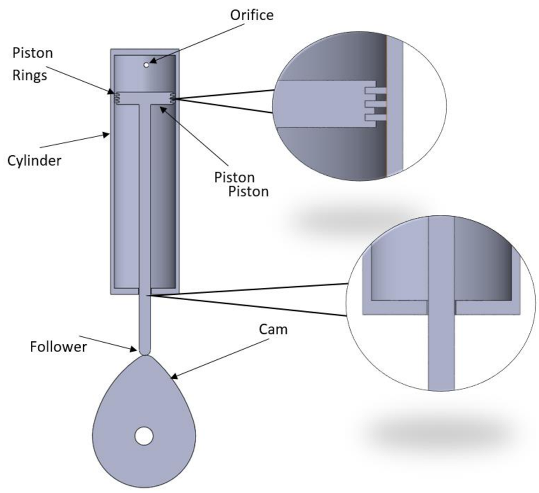


Fig 39: Construction of cam driven piston cylinder arrangement

During this upward motion, some of the air gets out from the orifice. The downward force acting on the piston due to the increased pressure in upper chamber pushes the piston downwards. During this downward motion of the piston, the same phenomena occurs in the lower chamber. While in the upper chamber, the air gets rushed in from the orifice, occupying the entire volume of the chamber. Hence it can be assumed that the volumetric efficiency of the orifice is unity. The compression rings help the sealing of the chambers. This does not allow air to leak into the suspension system. The orifice in the lower chamber is provided in the form of clearance between the piston and the cylinder or can be considered as account of leakage. The area of this clearance is equal to the area of the orifice and hence this clearance area acts as an orifice. Hence the leakage of air through this region can be used to an advantage.

5.3 Advantages of Air damper over Oil dampers:

Oil Leak:

Oil leak is one of the biggest issues faced by a suspension system. This oil leak may occur due to wearing of the seal/washer on one end of the pressure tube. This wear is bound to occur in a long run due to fretting fatigue and continuous usage [17]. Even other small parts inside the damper may wear out leading to the dissipation of the fluid into the cracks. Even a small decrease in the quantity of the fluid would result in a much lower damping and hence deterioration of the suspension performance. This deterioration cannot be rectified and the only solution is the replacement of the shock absorber. This replacement is a costly solution to the problem and this replacement is frequently seen in the life cycle of a vehicle.

This problem can be tackled by using the air suspension. The problem of oil leak is fully subsided by using atmospheric air instead of viscous oil. As the orifice is provided in the form of the annular opening, the air that is lost during the process can be easily inhaled during the upward stroke. This oil seeps to the upper chamber through the holes provided on the piston. Hence there is no need of components like oil seals, washers etc. Hence even the weight of the shock absorber decreases by using air instead of oil for damping. The manufacturing cost of the system decreases and hence the overall maintenance cost of the suspension system decreases.

Comfort Level:

The function of suspension component is two-fold: the ride comfort and road handling. Ride comfort can be referred as the ability of a vehicle to insulate the occupants of the road irregularities like bumps and pits. A vehicle can be said to have a good ride quality when the passengers inside the car do not feel sudden rise and fall due to road undulations. Road handling is the ability of the automobile to react or respond to the driver input and how well the vehicle follows the track or road.

A stiffer suspension system leads to a better road holding ability while it seems to compromise the ride comfort. As the stiffness decreases, the road holding ability starts decreasing and the ride comfort increases. Hence both the functions of the system contradicts each other and a compromise has to be made. It is generally observed that the commercial passenger vehicle tends to focus on increasing the comfort level as the speeds of travel are relatively lower. Hence in such cases the suspension system is designed to be of lower stiffness compared to performance/sports vehicles.

Hence, one of the goals of designing this air suspension system is to increase the ride quality of the two wheeler. When a conventional oil shock absorber is used for suspension purposes, the shocks experienced by the wheel is resisted by the hydraulic fluid. Hence, the motion of the piston is relatively slower and hence the disturbance gets transmitted to the sprung mass. This leads to a higher displacement of the rider.

While in case of air, the compression of air is relatively faster than that of the viscous oil. Hence the shock is absorbed by the damper before transmitting it to the rider. This decreases the relative stiffness of the system as compared to the conventional damper. Hence the air shock absorber gives a "softer" system and improves the ride quality.

5.4 CONCLUSION

1. The air damper can behave better than oil damper due to the benefits associated with leakage which is an asset. The mechanical parts are also minimized and the variable damping can be achieved, which can be controlled by air pressure inside the cylinder.

2. The oil dampers are better than oil dampers under certain conditions as described in this report, but overall, the air damper has more advantages and is the future of vehicle suspension systems.

CHAPTER 6

SCOPE OF THE FUTURE WORK

- Experimental study to obtain accurate values of all the governing parameters and to analyse experimental response of the suspension based on the provided design.
- Comparison as well as validation of the performed analytical work in this report with experimental data from setup manufactured based on the design and manufacturing guidelines given in this report.

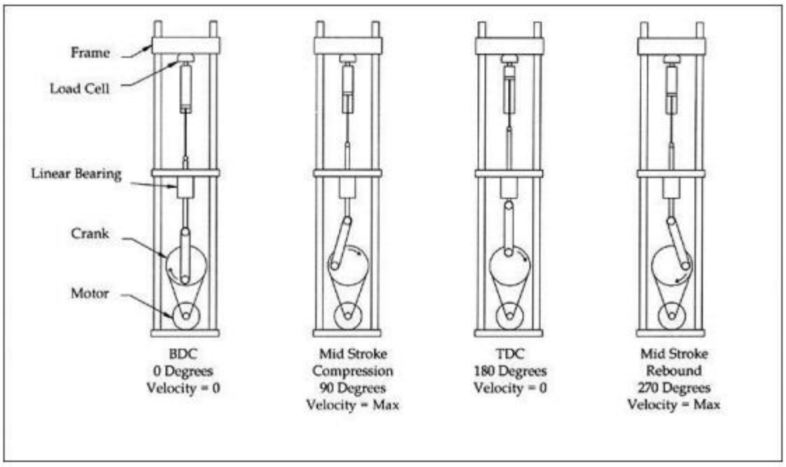


Fig 40: Glimpse of the experimental test rig

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