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DESIGN AND SIMULATION OF A CONTINUOUSLY CONTROLLED DAMPER USING AN ELECTROMAGNETIC ACTUATOR

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Abstract:- This paper presents the design, analysis, and simulation of a continuously controlled damping setup using an electromagnetic actuator. Our approach uses a method of creating an virtual experimental setup and simulating the results on MATLAB. The emphasis of our paper is on the damping ratio and damping coefficient which supposedly is to remain constant hence providing validation on our concept of an equivalent pull force to the applied force on the piston. A system which works fully independent of the driver. It works in accordance with the terrain it's traveling on and controls damping depending on the amount of force coming on the tires. The success of this project industrially will enable us to get better ride comfort while maintaining weight to power ratio of the vehicle.

Keywords:-*Adaptive air suspension, continuously controlled damping, variable stiffness, Electromagnetic actuator***.**

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I .INTRODUCTION

Suspension plays an important role while designing a vehicle. In recent years, with the development of travel requirements, handling, stability and ride comfort are very important features of automotive driving. Thus, to make the vehicle compatible to road conditions it is necessary to design a suspension system that can handle the roughest of terrain and endure extreme force conditions without affecting the vehicle's stability and at the same time also provide a smooth ride to the driver.

Suspension is defined as isolation of two masses. In automobile sector suspension deals with isolation of the wheels and the entire body with the help of a spring and damper arrangement. The wheels are termed as unsprung mass. The remaining body of the vehicle is termed as sprung mass.

A major component of the suspension system is the damper. A shock absorber or damper is a mechanical or hydraulic device designed to absorb and damp shock impulses. It does this by converting the kinetic energy of the shock into another form of energy (typically heat) which is then dissipated. Hence, our design of a damper is aimed to uncover problems of controlled damping with commercial vehicles and then use these findings, coupled with appropriate research, to create a damper that possesses improved performance.

In common with carriages and railway locomotives, most early motor vehicles used leaf springs. Motorcycle front suspension adopted coil sprung Druid forks from about 1906, and similar designs later added rotary friction dampers, which damped both ways - but they were adjustable. These friction disk shock absorbers were also fitted to many cars. The first production hydraulic dampers to act on the main leaf spring movement were probably those based on an original concept by Maurice Houdaille patented in 1908 and 1909. These used a lever arm which moved hydraulically damped vanes inside the unit.

II METHODOLOGY

Step 1: Conduct an extensive study in our domain through research papers and such relevant study material obtained from the internet.

Step 2: Conceptualize the idea of a continuously controlled damper and begin research over it.

Step 3: Form a report and approach various companies for sponsorship.

Step 4: Design of setup and run analysis of force distribution to achieve successful FOS.

Step 5: Validate the design and run simulation on MATLAB SIMULINK

Step 6: Record the results and make a final report of it.

III LITERATURE REVIEW

1. Performance Analysis of Directional Control Valve
Author: N. Kumar, Jayanta Das, Ajit Kumar
Description: Valve plays an important role in the modern electro-hydraulic systems, which integrates the versatility of advanced electronic control with the high power density of hydraulic actuation. However, the slow valve response often becomes the holdup of whole system performance and even causes instability of the resulting closed-loop system if not

proper addressed. Although fast valves like high-bandwidth servo valves are available, they are far more ex-pensive than slow valves such as the proportional directional control (PDC) valves due to the much more stringent manufacturing tolerance requirements. Directional control valves start, stop or change the direction of flow in compressed air applications.

2. Study of Hydraulic Directional Control Valve with Cylinder Performance using Matlab Simulink
Author: Kishor S Joshi, C.S.Rajshekar
Description: The use of hydraulic systems for control and regulation has made possible important fields for automation. However, in spite of the specialized field has not yet been circulated to a high enough degree. As a result of this, the application of hydraulic systems has been restricted. The objective of a fluid power system is to do useful work. This is accomplished in three fundamental steps. First, a mechanical energy input is converted into fluid energy by a hydraulic pump. Next, this fluid energy is transmitted through fluid conduits and any necessary control devices. Last, the fluid energy is reconverted into mechanical energy by an output device – usually a hydraulic cylinder or a hydraulic motor.

3. Directional Control Spool Valve Performance Criteria and Analysis of Flow - Reaction Forces
Author: B V Hubballi, Dr V B Sondur
Description: The control valves either meters the fluid into the actuator as a spool traverses within the valve body (i.e. flow control), or changing the direction of oil flow (i.e. direction control), or make an action according to the increase (or decrease) of the system pressure (i.e. pressure control). Flow control in a hydraulic system is commonly used to control the velocity of linear actuators or the rotary speed of hydraulic motors. There are three ways to accomplish flow control. One is to vary the speed of a fixed-displacement pump; another is to regulate the displacement of a variable-displacement pump. The third way is with the use of flow control valves. Flow control valves may vary from a simple orifice to restrict the flow to a complex pressure-compensated flow control valve and to flow dividers. Pressure control valves protect the system against overpressure, which may occur due to excessive actuator loads or due to closing of a valve.

4. Experimental investigation of the check valve behavior when the flow is reversing
Author: D. Himra, V. Haban, M. Hudec, and V. Pavlik
Description: The check valve is an important part of hydraulic systems and allows flow in one direction and prevents the reverse flow through the pump. The check valve can be also installed in the system to limit the pressure surge induced by the pump failure. The basic requirements on the check valve are the low pressure loss in the positive direction, no flow in the opposite direction; it means good sealing when the valve is closed.

5. Analysis of an electromagnetic actuator with permanent magnet.
Author: Daniel Mayer
Description: Electromechanical actuators are devices that convert effects of electric currents on mechanical, force effect. They are widely used in various industrial and transport applications and in technological processes of automated production systems. The function of an actuator may be based on various physical principles. Frequently used are particularly:

· Ferromagnetic actuators that employ magnetic force of an electromagnet and provide relatively high forces at shifts up to about 20 mm.

· Linear electromagnetic actuators working on the principle of a three-phase electromotor that have relatively high lift but their shifts are rather small.

· Thermoelastic actuators – using thermal dilatability of metals during their heating or unequal dilatability of two various metals that are mechanically connected and electrically heated. these actuators produce extremely high forces, but their shifts are of the order of only tenth of mm.

IV DESIGN

The test rig consists of an interconnected arrangement of an electromagnetic actuator, a piston-cylinder damper, an oil reservoir/ air tank, and FRV with a pressure gauge and a DCV connected as shown in the Fig.1.

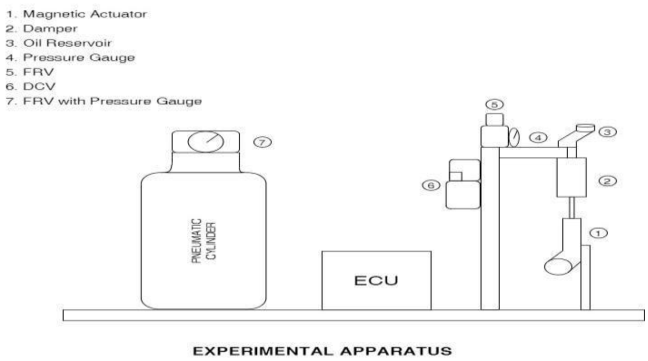


Fig.1

The base and frame network is made using MS pipes. We used MS as it is ductile and can be easily be machined. Generation of heat is less than that of stainless steel. So, the tool can have better life and you can do more machining. Mild Steel has high toughness and high strength than other steel. A pneumatic cylinder attached with a FRV and pressure gauge is used to send controlled amount of air in the damper for the compression stroke of the piston.

An ECU consisting of a power supply, an Arduino, two LCDs and a relay is fabricated whose block diagram is shown in Fig. 2.

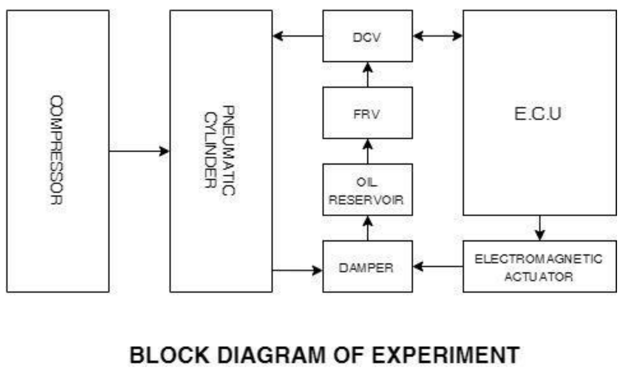
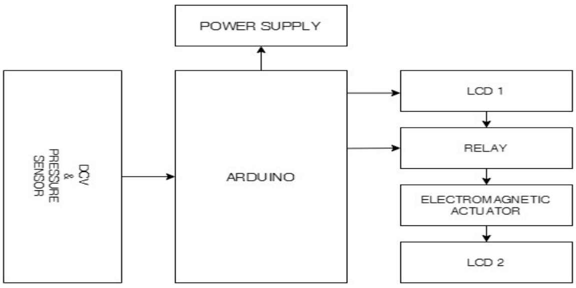


Fig. 2



BLOCK DIAGRAM OF ECU CIRCUIT

Fig. 3

Components Manufactured

I. Main Frame and Base

Fig 4.1

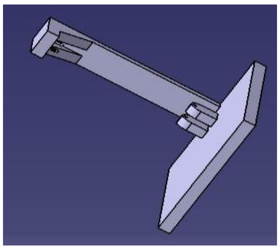


Fig 4.2

The base is 500 x 300 with a thickness of 25.

The total height of the support frame is 490.

II. OIL RESERVOIR/ AIR TANK

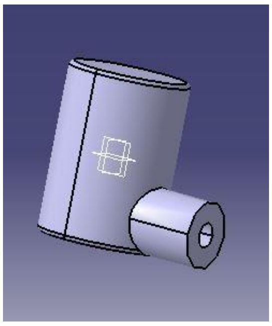


Fig 5

Fig 6

Material: AISI 1018 MS (Low carbon steel).
Pressure inside, $P_{or} = P$ (neglecting losses)
$P_{or} = 10$ bar
Hoop stress is given by, σ_{hoop}
$\sigma_{hoop} = (P \times r_{mean})/t$ (For thin-walled cylinder)
Longitudinal Stress is given, σ_l
$\sigma_l = (P \times r_{mean})/2 \times$ Axial length (For thin-walled cylinder)
Assuming the following dimensions:

- OD: 40
- ID: 30
- Height/ Axial length: 27 Tensile strength of material is 440 MPa.

The bursting stress can be referred to as the amounts of hoop stress and longitudinal (axial) stress that are produced in the wall of the cylinder when subjected to internal and external pressures that may cause the material which the cylinder is made from to fail. This happens if the hoops stress exceeds the tensile strength of the material.

$\sigma_{hoop} = 7$ MPa $\sigma_l = 0.65$ MPa

Hence, these values are quite less than the tensile strength of the material and the dimensions selected for the design are safe to use.

Therefore, design of the Oil reservoir is;

Main tank/reservoir

- OD: 40
- ID: 30
- Height: 30
- Total Volume: 19085.2 mm^2
- Swept Volume: 17318.0 mm^2

Orifice
OD: 8
ID: 3
Length: 10

Oil used is Engine Oil, SAE 15W-40 with a density of 872.5 kilogram per cubic meter at 30°C (86°F or 303.15K) at standard atmospheric pressure.

Components Purchased

I. Electromagnetic Actuator

Actuator Gun
5 Wire,
Power: DC 12V
Outer material: ABS plastic
Colour: Black
Included: Motor, captchur X1, screws X6
Item Weight: 141 g
Actuating Force: 5.5 Kgs
Stroke: 21-22mm
No load and Load current: 1.5A±5% & 3A±10%

Electromagnetic actuators convert electrical and mechanical energy into one another.

These are used in many applications from precise control using small actuators to the quite large powerful units using electrical drives.

II. Piston-Cylinder Damper

Fig 7.1

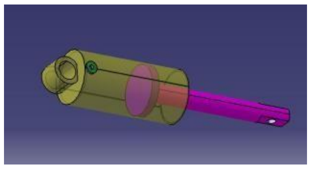


Fig 7.2

- Cylinder bore diameter is 32 and stroke length is 50.
- Total length of cylinder is 75.
- Max pressure is 1.0MPa
- Piston travel is kept at 21.
- The piston-cylinder damper

has two valves on two ends of the cylinder.

III. Filter Regulator Lubricator (FRL)

- Flow Rate: 500 L/min
- Filter Element Size: 40 micron
- Port size: 1/4 Inch
- Working Pressure: 10 Kg/c

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Fig 8

Fig 9

IV. Pressure Sensor
Pressure Measurement Type: Gauge
Operating Pressure Max: 10bar
Voltage Rating: 5V
Sensor Output: Analogue
Port Style: Manifold
Supply Current: 2mA
Product Range: SSC Series
V. Directional Control Valve (DCV-Solenoid controlled)

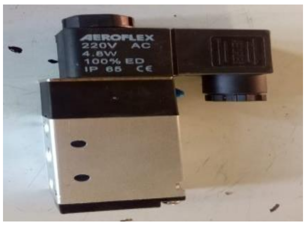

Fig 10.1

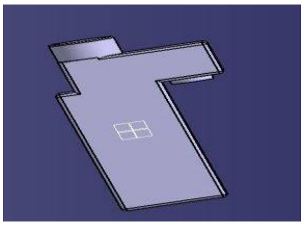

Fig 10.2

- Coil Voltage: 220V AC/24V DC 110V
- ACBSP Thread Solenoid Valve
- Number of ports: 3X2
- Port size: ¼ inch

V WORKING

- Controlled air enters the piston-cylinder damper from the pneumatic cylinder via the DCV.
- The ECU signals the DCV to cutoff the supply of air.
- The piston moves upwards to TDC, compressing the oil which displaces the air in the oil reservoir/air tank and hence, applies pressure on the pressure switch.
- The pressure switch signals the ECU to activate the electromagnetic actuator which in turn pulls the piston back to its original position i.e. to BDC
- The ECU signals the DCV to supply the air again to the damper and the cycle repeats itself.

VI EXPERIMENTAL PROCEDURE

1. Calibrating Supply Air Pressure
The pressure switch located at the oil reservoir/air tank is responsible of converting applied air pressure to equivalent electrical signal. But the supply air pressure is NOT the same as pressure coming on the pressure switch.
Hence, calibration of supply air pressure is necessary for accurate analysis.
Steps involved:
1. Set a random pressure value using the FRV1 on the pneumatic cylinder and allow air to enter the damper.
2. The piston displaces the air in the air tank that generates some pressure on the pressure switch.
3. Note this pressure on the FRV2 attached to the support frame.
4. At different pressure values note the readings on FRV2 and calibrate.

2. Experimental Procedure
1.After calibration of supply air, at a particular pressure reading adjust the pressure via the FRV1 on the pneumatic cylinder.
2.Signal the ECU to activate the DCV which allows air to enter the damper.
3.Mark the displacement of the needle on the scale attached to the frame.
4.Note time and number of oscillations made by the piston at that applied input pressure.
5.Repeat the procedure for different input pressure values.

ANALYSIS

A. Theoretical Equations:

1. Input and Output force.
a. Force on Effective Rod End Area (EREA) [Input]
$F_i = P \times EREA$
$= P \times$ (Piston head area - area of rod)
$= P \times \pi(D^2 - d^2)/4$
b. Force on Blind End Area (BEA) [Output]
$F_o = P \times \pi D^2/4$

2. Force applied by oil pressure
$F = \int P \times da$
$= \int \rho gh \times 2\pi r dh$
$= 2\pi\rho gr \int hdh$
$= \pi r\rho gh^2$

3. Pressure in Oil reservoir (P_{or})
Neglecting pressure losses due to pipe, eddies etc.
$P_{or} = P$

4. Kinetic Energy of Piston (KE_p) $F_o = m_p \times a_p$
Stroke Length $= u_{pt} + ½ (a_p t^2)$ But $u_{pt} = 0$
Therefore, $t = (2 \times \text{Stroke length}/a_p)^{½}$ $v_p = a_p t$
$KE_p = ½ (m_p \times v_p^2)$

5. Power Consumption by actuator
In case of DC, power consumption can be calculated by simply multiplying voltage and current. The standard units for voltage and current are volt and ampere.
Therefore, Power = V x I (Watts)

6. Retraction Speed and Time
a. Time
Net Force on Piston during retraction, F_{ret}
F_{ret} = Pull force by actuator, F_{pull} + Self-weight of piston, W_p + Pressure from oil, P
$F_{ret} = m_p \times a_p$
$t_{ret} = (2 \times \text{Stroke length}/a_p)^{½}$
b. Speed, v_{ret}
Cylinder Speed (ft/min) = (19.25 x GPM)/EREA
GPM = (EREA x Stroke length x 0.26)/t_{ret}
Therefore,
$v_{ret} = (\text{Stroke length} \times 5.005)/t_{ret}$

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A. ANSYS ANALYSIS

I. Piston Rod Analysis and Design

The piston rod of a hydraulic cylinder is highly stressed, and therefore it should be able to resist the bending, tensile and compressive forces that it may encounter during the operation without buckling.

In practice, the rod is more likely to fail by buckling under the compressive load than by bending. In this case, the rod behaves like a column and is subjected to buckling. The rod diameter can be related to critical load.
Therefore Euler's formula in the equation below for long column can be used to obtain the piston rod diameter.
$P_b = (\pi^2.E.I)/(L^2.K^2)$
Where,
P_b = Buckling load (N)
L = Stroke length (m)
I = Moment of inertia (m^4)
E = Young's modulus of the material used in this design calculation is 69 GPa for A6061 Aluminum alloy.
K = the end fixing factor which is 1 as both ends free.
Hence, P = 10237.41N

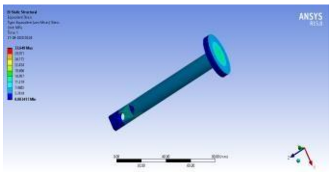

Fig 11.1

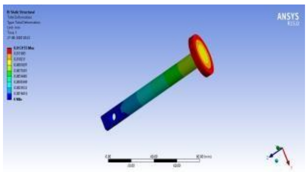

Fig 11.2

Max. Permissible Stress, σ_p = 33.65MPa
$F_p = \sigma_p$ x Area of Piston Rod
= 3805.73N
FOS = P/F_p
= 2.69 ≈ 3
From the above values of force, it is clear that the FOS is under safe limits meaning our design is safe enough from any type of failure.
Also, it is clear from the above images that min. amount of stress is being applied on mostly the entire region of the piston.
I. Piston head Analysis and Design
Piston head is also made of A6061 Aluminium alloy.
The hydraulic piston design must not be complicated. It must be designed for ease of assembly and disassembly for maintenance purposes. The main failure point was the edges of the piston and kind of seals used at tolerances between the piston and the cylinder wall.
F_{ph} = P x (Area of piston head - Area of Rod)
= 880.00N
$F_{max} = \sigma_p$ x (Area of piston head - Area of Rod)
= 1100.21N
FOS = F_{max}/F_{ph}
= 1.25
Hence, it is clear from the FOS that our design is safe enough from any type of failure.
Also, it is clear from the above images that min. amount of stress is being applied on mostly the entire region of the piston.

II. Cylinder Analysis and Design

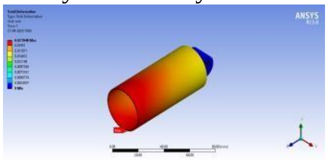

Fig 12.1

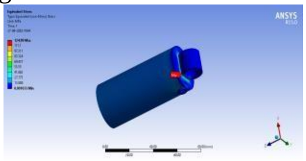

Fig 12.2

A. Failure due to working stress.
Max. Permissible Working Stress, σ_{wp} = 125MPa

Since the cylinder is also of A6061 Aluminium alloy. The tensile stress is 150MPa.
FOS = Tensile Stress of material/σ_{wp} FOS = 1.2
From the above values of stresses, it is clear that the FOS is under safe limits meaning our design is safe enough from any type of failure.
Also, it is clear from the above images that min. amount of stress is being applied on mostly the entire region of the cylinder.

B. Failure due to bursting Stress
The bursting stress can be referred to as the amounts of hoop stress and longitudinal (axial) stress that are produced in the wall of the cylinder when subjected to internal and external pressures that may cause the material which the cylinder is made from to fail. This happens if the hoops stress exceeds the tensile strength of the material.
Hoop Stress
$\sigma_h = P \times [(d_0^2 + d_i^2)/(d_0^2 - d_i^2)]$ Where:
P = supply pressure, 10 bar
do = outer diameter of cylinder, 42mm
di = inner diameter of cylinder, 32mm
σ_h = 3.76 MPa
Longitudinal Stress
$\sigma_l = (P_1R_1^2 - P_2R_2^2)/(R_2^2 - R_1^2)$
Where
P1 = Internal pressure (10 bar)
P2 = External pressure (atmospheric pressure = 1.0135 bar)
R1 = Internal radius
R2 = External radius
σ_l = 11.42Mpa
Hence, the safety against bursting is also guaranteed as the hoop stress, σ_h and longitudinal stress, σ_l developed inside the cylinder is quite less than the tensile stress.

VII RESULTS

Experimental results are obtained using MATLAB. The equations mentioned above are used to calculate certain parameters which are then used as variables for the simulation.
These results are stated in a tabular format.
A graphical representation of relations between input force and retraction speed of piston, damping coefficient and number of cycles, input force and damping coefficient have also been mentioned.
Given parameter:
Mass of piston, m_p = 0.07kg
Stroke length = 0.021m
Values of retraction forces against corresponding values of current generated on ECU are given as:
Hence, the tabulated form of retraction force is given as:

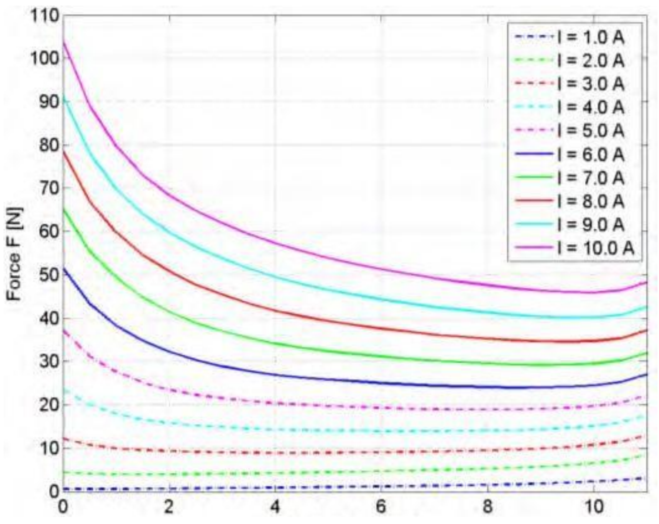

Fig 13

CURRENT (Ampere)	RETRACTION FORCE (N)
1	2.031
2	6.031
3	13.031
4	23.045
5	40.023
6	53.056
7	66.031
8	80.084
9	92.055
10	105.027

Using MATLAB, values of retraction time, tret, retraction velocity, vret and piston retraction acceleration, ap are simulated as follows:

Retraction Force (N)	Retraction Time (seconds)	Retraction Velocity (m/s)	Retraction acceleration x 10^3 (m/s^2)
2.031	0.0380	2.762	0.029
6.031	0.0221	4.760	0.086
13.031	0.0150	6.997	0.186
23.045	0.0113	9.305	0.392
40.023	0.0086	12.263	0.572
53.056	0.0074	14.119	0.758
66.031	0.0067	15.752	0.943
80.084	0.0061	17.347	1.144
92.055	0.0057	18.598	1.315
105.027	0.0052	19.866	1.500

Damping coefficient, c is also calculated against the input force and a graphical representation of the same is given alongside its table.

FORCE(N)	DAMPING COEFFICIENT (N-s/m)
2.031	25.0207
6.031	29.0395
13.031	29.6337
23.045	29.7116
40.023	28.1818
53.056	29.3723
66.031	30.7172
80.084	31.2356
92.055	33.4482
105.03	34.7939

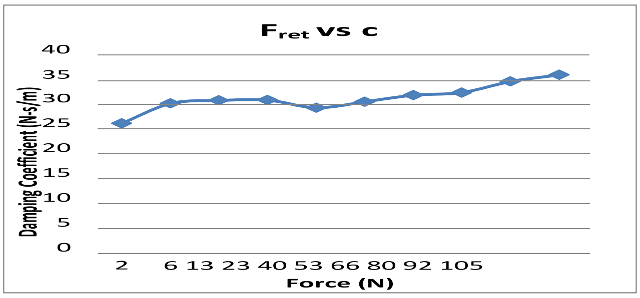

Fig 14

VIII CONCLUSION

This paper represents the design, analysis, and simulation of a continuously controlled damping setup using an electromagnetic actuator. The setup works as a prospective type of device providing relatively high forces with constant retraction time.

- The FEA analysis carried out on the piston cylinder provides credible validation for the reliability, functionality, and safety of the hydraulic cylinder designed.
- The MATLAB software simulation has been done to theoretically Calculate and Validate the retraction time on different pressures.

The following table represents the pressure and retraction time values:

Pressure (Bar)	Retraction Time (s)
1	0.0380
2	0.0221
3	0.0150
4	0.0113
5	0.0086
6	0.0074
7	0.0067
8	0.0061
9	0.0057
10	0.0052

- The bursting pressure, longitudinal stress piston rod and piston diameter, barbell thickness was determined and analyzed.
- The designed continuously controlled damping setup can be effectively employed when manufactured for industrial automation.

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