© 2016 IJSRSET | Volume 2 | Issue 5 | Print ISSN: 2395-1990 | Online ISSN : 2394-4099
Themed Section: Engineering and Technology

Experimental Characterization of Gas Filled Hydraulic Damper Using Ramp Excitation

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ABSTRACT

The automotive damper is one of the important components of a vehicle suspension system. It controls spring motion by damping energy from the spring, and it is used for the optimization of driving comfort and driving safety. Many automotive dampers have non-linear asymmetric characteristics to accommodate the incompatible requirements between ride comfort and road handling, the engineer requires techniques that can characterize this non-linear behavior and provide models of the dampers for use in ride performance simulations of the full suspension system. In this paper, experimental characterization is done by developing a mathematical model of the front wheel gas-filled hydraulic damper of Maruti Suzuki swift corresponding to road conditions. The dampers have tested under both sinusoidal and ramp excitations on the dynamic material test platform. To accurately predict damping force experimental data is used to fit the equation of curve using the least square curve fitting method in Matlab software interface and components of the mathematical force model is identified. Finally, results of a mathematical model are verified with experimental results. The results of the mathematical model show good correlation with experimental data with precision above 90%. Though some error has been found. It is mainly due to hysteresis effect which is present because of damper compliance.

Keywords: Damping, Experimental characterization, Hydraulic shock absorber, Nonlinear modelling, Ramp excitation, Sine excitation, Vibration.

I. INTRODUCTION

A key element in any vehicle suspension system is the damper. It plays a vital role in the vertical and horizontal motion of the vehicle. The suspension is needed to guarantee handling and comfort, for a good braking, to maintain constant tire-road contact. Many automotive dampers have non-linear asymmetric characteristics to accommodate the incompatible requirements between ride comfort and road handling, the engineer requires techniques that can characterize this non-linear behavior and provide models of the dampers for use in ride performance simulations of the full suspension system. The accuracy of the vehicle model is highly dependent on the accuracy of the damper model. The damper is also one of the most non-linear and complex suspension system elements to model. In order to design damper for the dual demand of resisting violent impact and attenuating vibration in vibration-impact safety, the accurate characterization of the damper is of paramount importance. Indeed, the characterization makes it possible to define a sufficiently precise mathematical model of a damper for design purposes. Some authors had presented this concept in their various publications which are presented further, Stefaan Duym et. al Presented a short view of several damper models and the question, how well the models match experimental data is answered. Non-parametric model is presented to describe in a satisfactory way the fully nonlinear and dynamical behavior of damper [1]. R. Basso investigated experimental characterization of the hydraulic shock absorber to predict damping force using constant velocity excitation which allows various advantages over conventional method [2]. Stefaan Duym et. al showed development and identification of damper model that predicts the damper force as a function of damper displacement and velocity for purpose of simulation are shown. The physical model is developed and implemented in several software. Damper model structure is shortly elaborated together with some measurement and estimation techniques to retrieve the model parameters [3]. Y. Ping studied the dynamic behavior of an oil-air coupling shock absorber. The mathematical model developed to describe the non-linear phenomena occurring within the shock absorber is discussed [4]. C Surace et al. presented an experimental study of a number of shock absorbers, also a new physical model for the absorber is presented which incorporates effects due to the compressibility of the fluid in the shock absorber is demonstrated which provides a more realistic representation of the stiffness characteristics than previous simple models [5]. X. C. Akutain et al. presented the design and experimental validation of an explicit parametric model for monotube dampers. The aim is to develop a model with few physical parameters, in order to make it both easy to manage and computationally light. An explicit and parametric model for monotube shock absorbers is shown. It is intended to achieve an accurate and manageable model in order to make it suitable for full vehicle simulations and for routines of a semi-active suspension [6]. A. M. Salem et. al Presented identification of characteristics and damping coefficient. Dynamic behavior is studied by using experimental and simulation methods. Simulation is done on ADAMS. Predicted characteristics are compared with results from both methods [7]. Yan Cui et. al presented a new testing and analysis methodology to obtain nonlinear characteristics of an automobile shock absorber three shock absorber models that can be quickly fit experimental data and used for vehicle simulation. These models are based on the understanding that the shock absorber is predominantly a velocity-dependent device. Further a single-post shaker test bench is introduced and the experimental procedure for the shock absorber testing is described. Also, the influence of the shock absorber models on vehicle dynamics in the vertical direction was analyzed [8]. Yongjie Lu et. al. presented structural features of the hydraulic shock absorber. The detailed scheme of testing is proposed. Tests carried out under sinusoidal and random displacement excitation. Besinger model is chosen to describe nonlinearity of shock absorber. Finally, a least square method is used to identify parameters of Besinger model. The virtual prototype model is used to fit shock absorber model effectively [9].

In this paper, experimental characterization is done by developing a mathematical force model of the front wheel gas-filled hydraulic damper of Maruti Suzuki swift corresponding to road conditions by using nonparametric modelling approach. The developed model is intended to quickly fit experimental data and used for vehicle simulation. This model is based on the understanding that the damper is predominantly a velocity-dependent device. The components of the model are identified by fitting experimental data using the least square method and results of model and experiment are compared in graphical form to verify the accuracy of the developed model.

II. METHODS AND MATERIAL

1. Experimentation

Experimental testing of the gas-filled hydraulic damper was performed in order to fit with mathematical force model stated for a damper that has been developed to accurately predict the force output as a function of velocity. The servo-hydraulic dynamic testing system was used to collect time, force and displacement data of the test specimen. The physical parameters of the damper were carefully measured.

Figure 1: Test Specimen -Front Damper

A. Experimental Test setup and Apparatus.

Left-hand side front wheel gas-filled hydraulic damper of Maruti-Suzuki Swift (2012) model, has been tested on standard test rig (MYTIDYNE servo-hydraulic test rig) located at Automotive Research Association of India (ARAI), Pune. This hardware is used in conjunction with controller MTS Flextest and MTS Flextest series 793 software interface in a computer so that the user can specify the amplitude, frequency, sampling frequency and type of excitation. The test system contains servo control system, signal acquisition system, function generator, test bench security monitoring system, analogue to digital converter interface system, servo-valve driver. The loading unit is the main part of the test rig, it consists of load frame, cross-head lift and lock, force transducer, actuator, three stage servo-hydraulic valve, hydraulic manifolds, and accumulators. The test specimen, test setup with equipped test specimen and schematic diagram of test setup is shown in fig.1, fig.2, and fig.3 respectively,

Figure 2: Experimental Test Setup

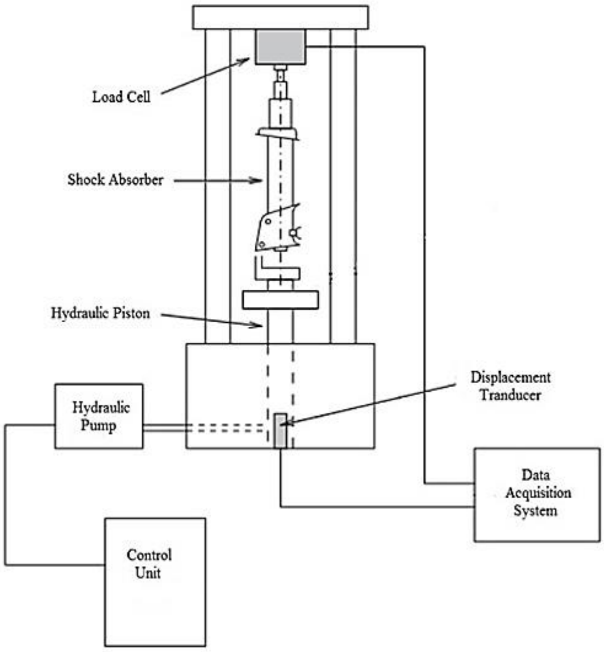


Figure 3: Schematic Diagram of Test Setup

B. Input Data for Experimentation

To obtain damping characteristics of dampers, which generally nonlinear in nature, the experimental test, is performed under sinusoidal and ramp excitation at various amplitude and frequency? The displacement of dynamometer under sinusoidal and ramp excitation can be calculated from the amplitude and frequency and velocity. It is then determined by taking the first derivatives of the displacement. These equations are restated as (1) and (2). frequencies of excitation and amplitudes as input for experimental tests as given below in table I and table II for sinusoidal and ramp excitation respectively, this combination of amplitudes are taken based on the stroke of dampers and the frequencies are chosen depends on mainly two factors dynamometer maximum velocity that was 0.7 m/s and vehicle suspension resonance frequencies.

$$x(t) = X.\sin.(2.\pi.f.t) \quad (1)$$

$$\dot{x}(t) = \frac{dx}{dt} \quad (2)$$

TABLE I
INPUT DATA FOR SINE EXCITATION

Sr. No	Amplitude (mm)	Frequency of Excitation (Hz)
1	10	0.05,5,8,11
2	30	0.1,0.5,1.2,2,3.5
3	63	0.05,1,1.5

TABLE II
INPUT DATA FOR RAMP EXCITATION

Sr. No	Amplitude (mm)	Frequency of Excitation (Hz)
1	63	0.2,0.4,1.2,2,2.7

C. Experimental results and discussion

The data recorded from the series of experiments for damper is time, displacement and the force generated across damper in the form of an array, this data is further analyzed and various plots are drawn in Microsoft Excel and Matlab software for analysis. Time-Displacement diagram, the damping characteristics curves (V-F Diagram) and work diagrams (Force-Displacement or F–X Diagram), are obtained from experimental test results of the damper, it is presented below in this section. Fig. 4 and 5 represents time-displacement plot for sinusoidal and ramp input respectively, which shows a variation of the displacement w.r.t time, fig. 6 shows the characteristic curve. Which shows nonlinear behavior in both compression and extension region with some effect of hysteresis, it is also seen from this diagram that damping is kept more in extension than that of compression section and variation of force w.r.t velocity. As we go towards a higher frequency of excitation the damping force become predominant as it depends on upon velocity. Fig. 7 and 8 shows (F-X) Diagram allow us to understand the energy dissipated by damper during its cyclic operation under sine input whereas fig. 9 shows energy dissipated using ramp input. In this way, experimental results and plots obtained from it will help us to understand and visualize characteristic of the damper. These various plots obtained for both damper shows nearly similar nature which is presented below.

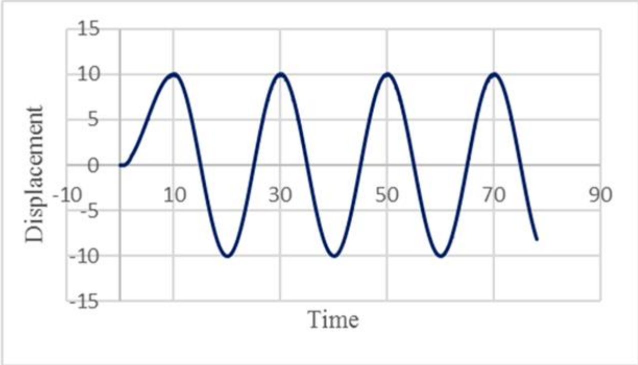


Figure 4: Variation of Displacement with Time using Sinusoidal Excitation

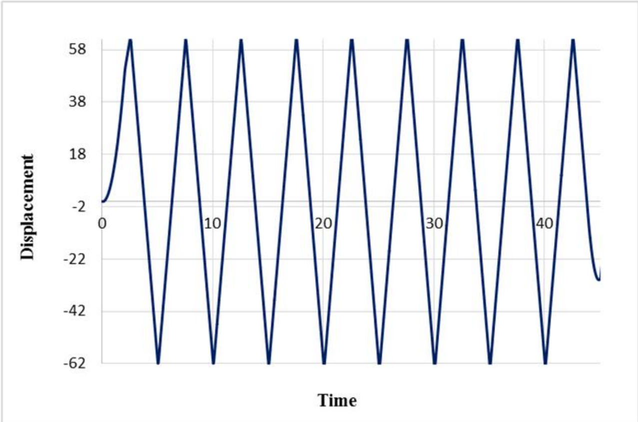


Figure 5: Variation of Displacement with Time using Constant Velocity Excitation Frequency 0.2 Hz and 60 mm Amplitude.

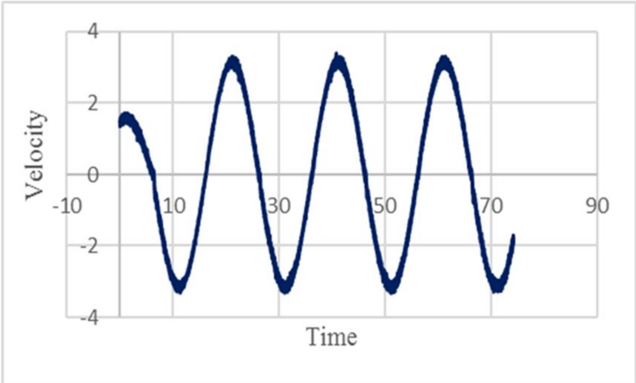
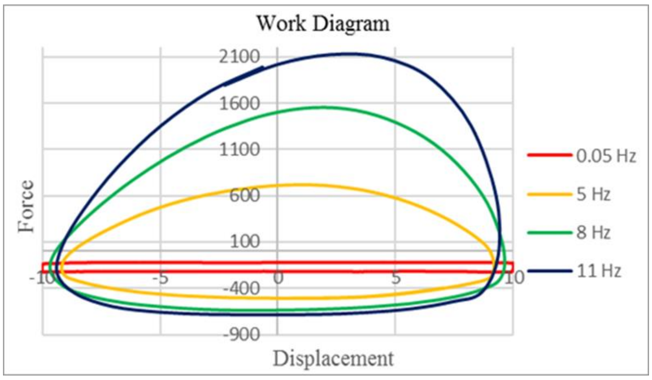
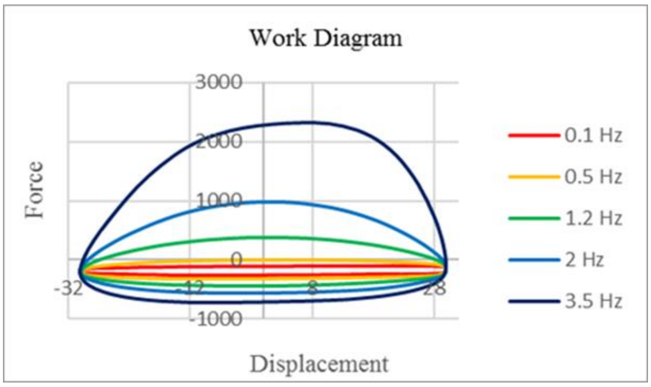


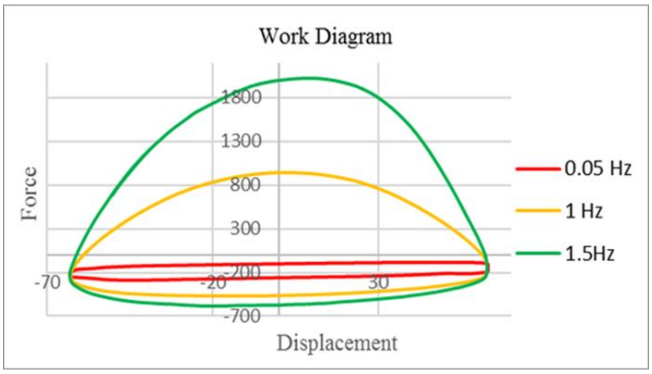
Figure 6: Variation in Velocity with Time using Sinusoidal Excitation for Frequency 0.05 Hz.



(a)

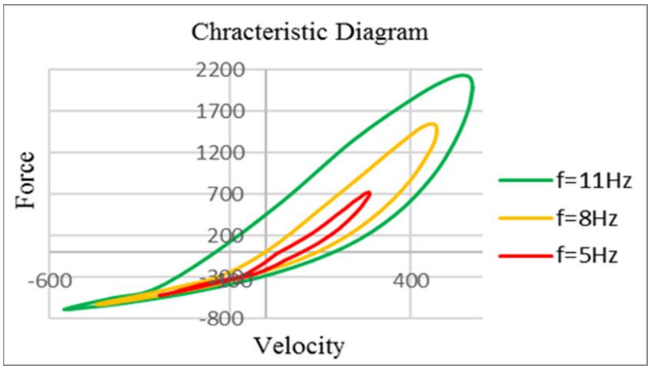


(b)

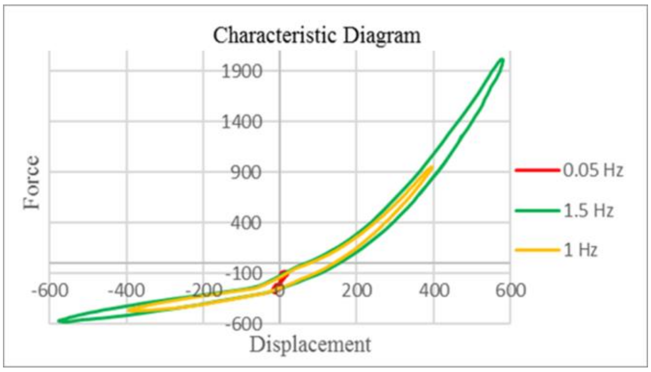


(c)

Figure 7: (a),(b),(c) are Work Diagram using Sine Excitation for 10 mm, 30 mm, and 63 mm Amplitude respectively.



(a)



(b)

Figure 8: (a) and (b) are Characteristic Diagram using Sine Excitation for 10 mm, and 63 mm Amplitude respectively.

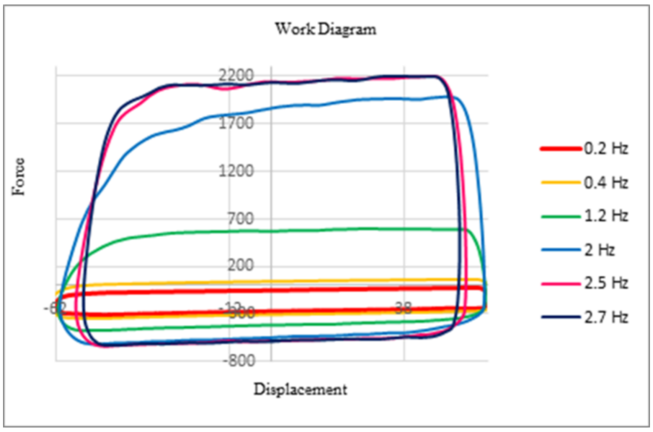


Figure 9: Work Diagram using Ramp Excitation for 63 mm Amplitude.

2. Characterization of Damper

Automotive design engineers have long been using various mathematical models and computer simulations to aid the development of new vehicle models and to understand vehicle performance limitations. This model requires accurate parameter characterization for the different suspension components in order to predict a vehicle's dynamics. A common approach for these systems is to take the vehicle in static positions and apply forces in specific ways to record the deflection of the component to be characterized. As dampers are velocity dependent; therefore, any static test will not incorporate any forces affected by the damper. The force-velocity curve is often used to characterize the damper in simulation models. The operating conditions are derived from the nature of excitation of the vehicle hull. This excitation is depending on the configuration of the road that the vehicle is moving on and on the vehicle velocity. Dampers are generally characterized through experimental testing that generates a characteristic diagram, expressing the damping force as a function of the damper velocity. While these tests allow the effect of the damper on the vehicle response to be determined, they give a little insight to how and where the actual forces in the damper are generated.

A. Development of Force Model

A damper model could be used to limit the need for extensive testing to determine the range and resolution of the damper adjustment. It could also be used to predict what valving is needed to produce a desired characteristic diagram, without actually revalving and testing the damper. This could significantly reduce the testing and revalving time required for the damper. In this study, the focus is concentrated on developing a nonparametric model of a commercially available automotive twin tube damper (Maruti Suzuki Swift 2012 model Front Damper) experimentally, and validate that model by comparing it to experimental results. The model developed will generate characteristic diagrams from the various parameters of a damper. These parameters include the dimensions of the damper internals, and the initial gas pressure in the damper, position dependent gas pressure happens mainly due to insertion of the rod, mechanical friction happens due to friction between rod and piston seals. The model had to be efficient and meticulous but at the same time not so heavy as to embarrass simulation or process time. An examination of the internal parameters of the damper has been avoided for the characterization of the dynamic response of the damper, as this paper is dealing with the development of the nonparametric model and hence inner mechanics are considered unknown. The model consists of a total of 3 parameters. The force through the damper is a combination of hydraulic force, frictional force and gas force.

$$F = F_d + F_f + F_g \tag{3}$$

Where,

F = Total Force Measured Across Damper
F_d = Damping Force
F_f = Frictional Force
F_g = Gas Force

$$(F_d) = c\dot{x} + c_n\,|\dot{x}|^n.\,\mathrm{sign}\,(\dot{x}) \tag{4}$$

$$(F_f) = F_f.\,\mathrm{Sign}\,(\dot{x}) \tag{5}$$

$$(F_g) = F_{gst} + F_g(x) \tag{6}$$

In equation (3) F denotes the force recorded by load cell while experimentation which is a total force generated across the damper. And Equation (5) denotes the friction force and which is generated in damper while in operation due to the friction between seals and piston rod assembly. Equation (6) denotes the gas force both static gas force and rod position dependent gas force which is present in damper to avoid cavitation in the fluid. Both equation (5) and (6) is being identified by experimental results. Then the goal is to find F_d the damping force which is velocity dependent force and desirable force in damper to provide a damping effect. This is achieved by fitting experimental data to an equation of curve using least square method then verify that model by giving input of amplitude and frequency in simulation software which will give a corresponding plot of work diagram or characteristic diagram with values of force and displacements. If this is matched with the experimental results, then suggested force model is precise.

B. Static Forces

Although the main damper forces are related to velocity, there are also some static forces present in the damper and needs to be considered. When damper traversed very slowly, to eliminate fluid dynamic forces, a pressurized damper will exert,

i. A force produced by pressurization times rod area (F_{gst});
ii. A stiffness from pressure rise due to rod insertion (K_{gas});
iii. A static (Coulomb-type) friction arising from rod and piston friction (F_f).

These will all appear on a very-low-speed F(X) curve,

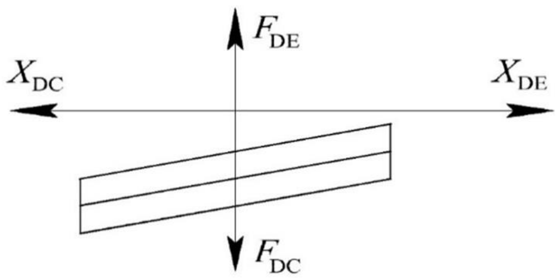


Figure 10: The Force Loop for Gas Pressure Force and Friction Force.

Fig. 10 the force loop for creeping motion reveals the gas pressure force and friction forces only. The damper parameters related to the above are static (compression) force F_{gst} at the central position, stiffness Kg through the range, and coulomb static damper friction force F_f. creeping the damper in and out at the central position requires damper 'static compression and extension forces.

$$F_f = \frac{1}{2}(F_{in} - F_{out}) \quad (7)$$

$$F_g = \frac{1}{2}(F_{in} + F_{out}) \quad (8)$$

$$K_{gas} = \frac{dF_{gst}}{dX} \quad (9)$$

The equation (7), (8) and (9) are used to evaluate static forces by using experimental data.

C. Coefficient Identification of Damping Force

Identification of coefficient and exponent of the front damper is done using the least square curve fitting technique to know the degree of a polynomial of the characteristic curve of damper during its extension and compression region. This is achieved by using Microsoft excel and Matlab software interface using Ramp excitation input data. The very first velocity versus damping force curve is plotted fig. 11 by minimizing static forces from the force generated across the damper in experimentation.

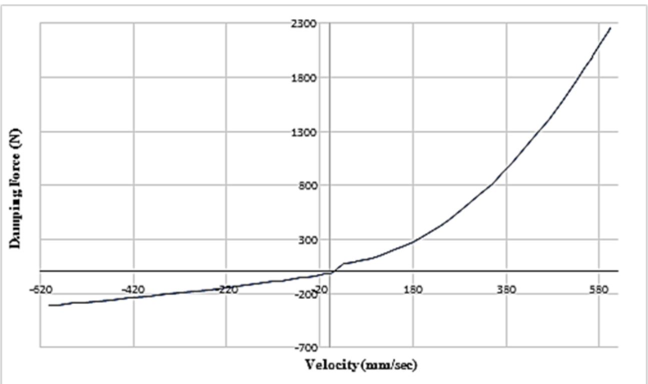


Figure 11: Variation of Damping Force with velocity using Ramp Input

This is then subdivided into two parts according to compression and extension region. And then curve fitting technique is used to evaluate the coefficient and exponent in both compression and extension region. As presented in fig. 12 and 13.

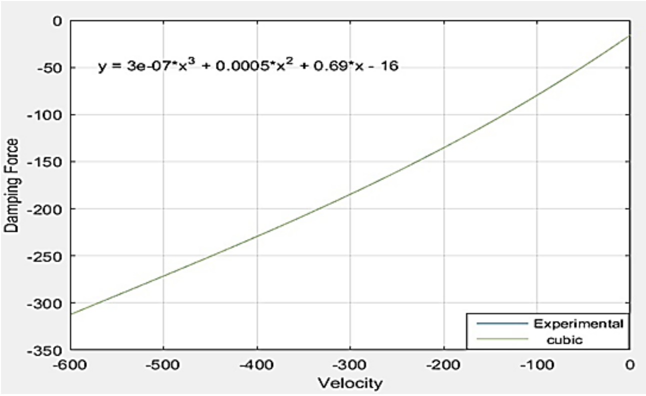


Figure 12: Curve Fitting to Compression Region

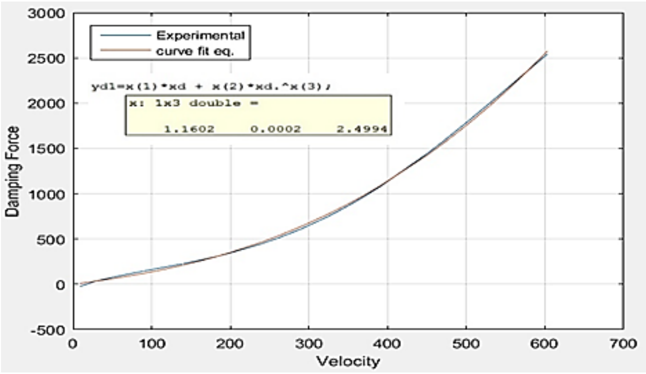


Figure 13: Curve Fitting to Extension Region.

The fig. 12 and 13 shows the equation of fitted curve to experimental data for compression and extension region respectively. In compression region, linear part of the equation is seen predominant, while in extension region due to nonlinearity behavior the curve varies in more than 2 degree of a polynomial with some linear part near to the zero. These identified components are tabulated in table III by putting these components in Force model that is in equation (4) the damping force is evaluated.

The identified values of the various component in force model are given in table III.

TABLE III
IDENTIFIED VALUES OF VARIOUS FORCES OF MATHEMATICAL FORCE MODEL

Sr. No.	Force Components		Numerical Values	
1	Friction Force (F_f) = F_f. $Sign$ ($\dot{x}$)		47 N	
2	Static Gas Force (F_{gst})		178 N	
3	Position Dependent Gas Force Fg.(x)		K_{gas} = 0.49 N/mm	
4	Damping Force (f_d) = $C_1\dot{x}$ + $C_n|\dot{x}|^n$. $Sign$ ($\dot{x}$)		Compression region	Extension Region
		C_1	0.6100	1.1602
		C_n	0.0002	0.0002
		n	00002	2.4993

III. RESULTS AND DISCUSSION

The fig. 14 to 19 represents the comparison of work diagram and characteristic diagram plotted using experimentally recorded force and force evaluated by a mathematical model for same input condition. The results have plotted for all displacement amplitudes and for some frequencies as taken as an input parameter in order to verify the precision of force model suggested. The graphical representation of results allows easy interpretation of data.

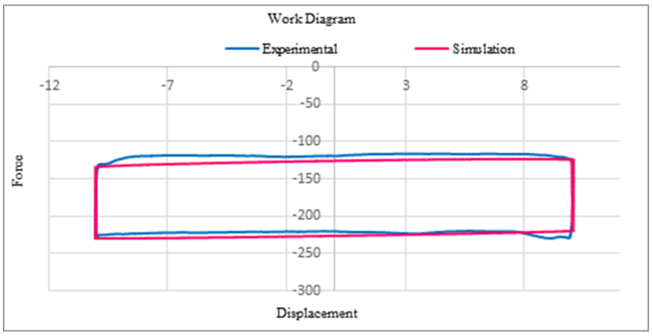
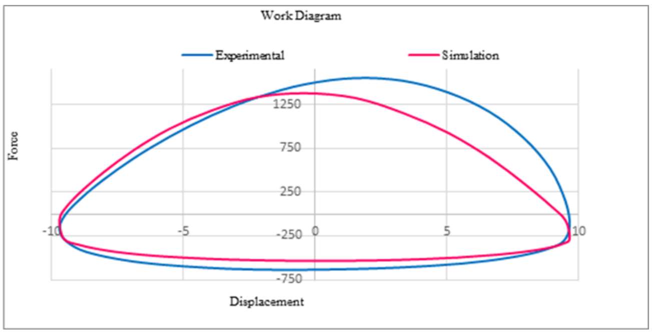
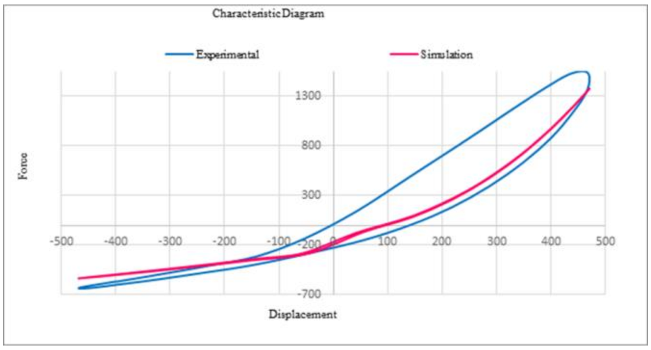


Figure 14: Comparison of Work Diagram for 10 mm Amplitude and Frequency 0.05 Hz



(a)



(b)

Figure 15: (a) and (b) Comparison of Work Diagram and Characteristic Diagram for 10 mm Amplitude and Frequency 8 Hz respectively.

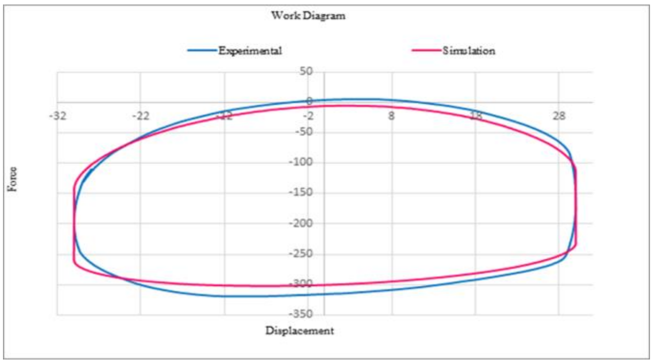
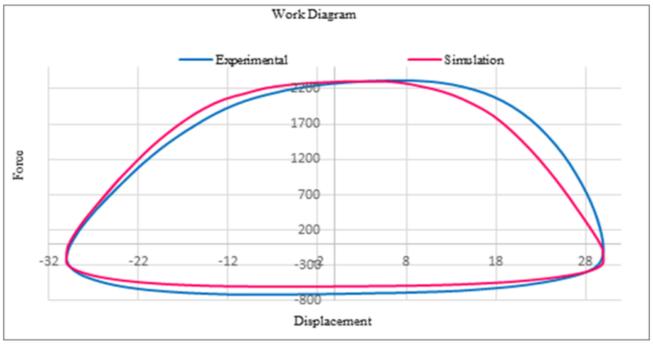
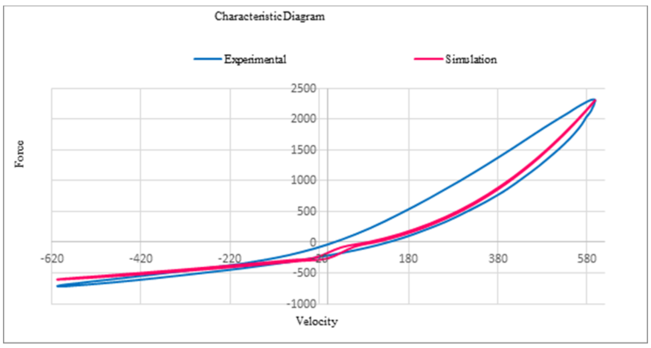


Figure 16: Comparison of Work Diagram for 30 mm Amplitude and Frequency 0.5 Hz

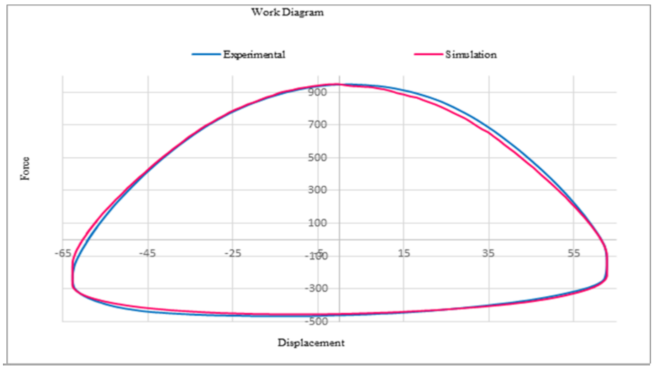


(a)

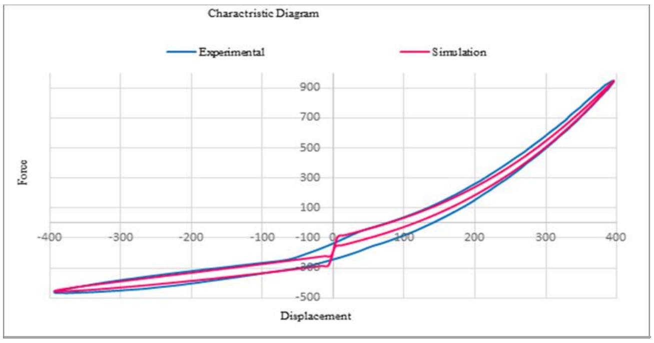


(b)

Figure 17: (a) and (b) Comparison of Work Diagram and Characteristic Diagram for 30 mm Amplitude and Frequency 3.5 Hz respectively.

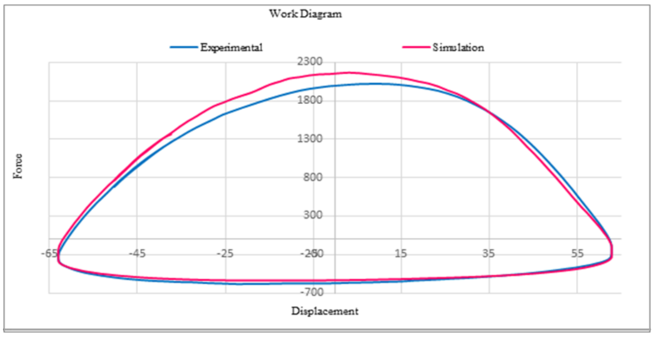


(a)



(b)

Figure 18: (a) and (b) Comparison of Work Diagram and Characteristic Diagram for 63 mm Amplitude and Frequency 1 Hz respectively.



(a)

(b)

Figure 19: (a) and (b) Comparison of Work Diagram and Characteristic Diagram for 63 mm Amplitude and Frequency 1.5 Hz.

Analysis of fig. 14 to 19 shows that the model replicates the behavior of the damper quite good. However, the measurements show some distortions that the model doesn't capture, it is mainly due to hysteresis. The hysteresis effect is found more during higher frequencies due to oil compressibility phenomenon. The simulation model and experimental results are almost unanimous when the damper is excited at a low to moderate frequency. The deviation present can be minimized by even complex analysis. The damping is found almost linear in compression region whereas in rebound section it is found in the degree of polynomial more than 2. The model is capable of predicting nonlinear behavior of damping force which is clearly seen from the characteristic diagram. Therefore, the model can be used to forecast the performance of the hydraulic damper in simulation at the design stage.

TABLE IV

COMPARISON OF ENERGY CALCULATED USING EXPERIMENTALLY RECORDED FORCE AND MODELLED FORCE FOR SAME INPUT CONDITION

Sr. No	**Ampli-tude**	**Frequency of Excitation (Hz)**	**Work Done Calculated Using Experimental Force (N.m)**	**Work Done Calculated Using Modelled Force (N.m)**
1	10	0.0500	1.9828	0.0878
2		8.000	25.5000	5.7240
3	30	0.500	15.0370	1.2220
4	63	3.500	127.0100	13.8000
5		1.000	128.6200	1.8200
6		1.500	243.5700	-5.6700

Table IV shows by comparison of energy dissipation, which is evaluated by experimentally recorded force and modelled force for same input condition. It can be said that proposed model predicts damping force accurately; the precision is near to 90%. Therefore, the model can be used to forecast the performance of the damper when design. Which can fulfil the engineering requirement?

IV. CONCLUSION

The proposed mathematical model is simulated by using Microsoft excel. The simulation results fit the experiment data very well, this shows that the proposed force is able to reproduce force generated across damper similar to the experimental case and hence this model can be used to forecast the performance of hydraulic damper preferably at its design stage. The model matches imposed criteria relative to simplicity, accuracy, and speed of calculation. Following major conclusions have been drawn from the outcomes of this paper. Comparison of the characteristic diagram shows that the force model is capable of predicting nonlinear behavior. Which is a better replacement for the linear model for accurate prediction of damper characteristics. The model replicates the behavior of the damper quite good. However, at some frequencies error is seen and this error is encountered due hysteresis effect. The hysteresis effect is recorded in experimental results mainly because of oil compressibility and all type of damper compliance. It is also seen that during higher frequencies the hysteresis effect is more. The comparison between energy dissipation calculated using experimentally recorded force and force evaluated from the model simulation for same amplitude of excitation and frequency the percentage variation found less than 10%. Which can suggest stated force model gives satisfactory precise results and can be suggested in the simulation of vehicle dynamics.

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