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Fluid flow modeling of a four-stage damping adjustable shock absorber and its experimental research

Fangwei Xie and Erming Ding
School of Mechanical Engineering, Jiangsu University, Zhenjiang, China
Heng Dai
Shenhua Group Zhungeer Energy Group Co. Ltd, Erdos, China
Qingsong Gao
Xishan Coal Electrictity Group Co. Ltd, Taiyuan, China, and
Jin Zhang, Yixian Feng and Hongtuo Liu
School of Mechanical Engineering, Jiangsu University, Zhenjiang, China

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Abstract

Purpose – In order to improve the ride comfort of vehicle suspension, this paper first proposed a shock absorber with four-stage adjustable damping forces. The purpose of this paper is to validate its modeling and characteristics, indicator diagrams and velocity diagrams, which are the main research points.

Design/methodology/approach – In order to validate the fluid flow modeling, a series of mathematical modeling is established and solved by using Matlab/Simulink. An experiment rig based on electro-hydraulic loading servo system is designed to test the prototype. Finally, indicator diagram and velocity diagram are obtained and compared both in simulation and experiments.

Findings – Results indicate that at the same damping position, damping force will increase with the rise of rod's velocity: if the rod's velocity is fixed, the damping force changes apparently by altering the damping position. The shock absorber is softest at damping position 1, and it is hardest at damping position 4; although there is no any badly empty stroke and skewness in indicator diagram by simulation, a temporary empty stroke happens at maximum displacement of piston rob, both in rebound and compression strokes.

Research limitations/implications – Compared with results of the simulation and experiments, the design of a four-stage damping adjustable shock absorber (FDASA) is validated correctly in application, and may improve the overall dynamic performance of vehicle.

Originality/value – This paper is mainly focused on the design and testing of an FDASA, which may obtain four-stages damping characteristics, that totally has a vital importance to improve the performance of vehicle suspension.

Keywords Damping adjustable, Damping characteristics, Electro-hydraulic servo, Four-stage, Indicator diagram, Velocity characteristics

Paper type Research paper

1. Introduction

In recent decade, with the innovation and development of the modern technology, vehicle suspension technologies based on active control and semi-active control systems have become the mainstream research directions in this field. The active suspension system which is capable to provide the best performance in driving process has its special advantage, but it is restricted to use widely because of its high price (Pellegrini *et al.*, 2010; Reybrouck, 1994). Compared to the active suspension system, the semi-active suspension system has the same dynamic performance as the active system, but has a cheaper price.

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Adjustable damping shock absorber, which is the main part and component of semi-active suspension systems, has a great influence on the damping properties of the automobile suspension system (Simms and Crolla, 2002).

Adjustable damping shock absorbers have two types: multistage adjustable damping (MAD) and continuously variable damping (CVD) (Chen and Zong, 2014). MAD shock absorber can satisfy the driving performance requirement in different road conditions by switching to different damping positions (Zeng and Zhang, 1992). While CVD shock absorber realizes its continuous variable damping characteristics by changing the intensity of electric field or magnetic field to control oil viscosity (Koh, 2001; Yazid *et al.*, 2014; Milecki and Hauke, 2012). For example, American Delphi Corp. has explored a Magneride magnetorheological fluid (Xuan, 2016), which has been applied to some high-grade automobile suspension systems.

To date, many studies have worked on MAD shock absorber and their results showed a great improvement in damping force control. Margolis *et al.* (1975) developed a semi-active shock absorber controlled by a switch valve, established its accuracy modeling by using state-space equations, and, finally, they validated it through experiment tests, the result proved that it has a good driving performance. In 1983, Toyota Motor Corp. (Yokoya, 1984) developed a two-stage damping adjustable shock absorber, which could be set to low damping position while in normal driving condition, however, it may adjust to high damping position when at car starting, acceleration, deceleration and turning to keep the car body in safety and balance sates. Shen (2005) studied a damping adjustable shock absorber controlled by a solenoid valve to solve the fixed damping question in compression stroke. He introduced the solenoid valve's code, structure, parameters, performance and working principle in detail, and described operation process of damping changing. Aubouet *et al.* (2008) have developed a nonlinear model of a semi-active damper in a quarter vehicle model; in addition, he studied the comfort and road-holding level of this suspension by comparing with the passive ones using simulations, and results indicated the performance improvement by the control of this damper. Chen *et al.* (2015) has designed a MAD shock absorber; to acquire the damping characteristics, he built and solved out the mathematical modeling of damping force by using combination fluid dynamic and thermodynamic theories. In further, he analyzed the influence of rod's stroke and moving frequency on the damping force by simulating in Matlab and Simulink, and proven its validity by testing. It is not difficult to find out that many present researches are mainly focused on one to three-stage damping adjustable shock absorbers. Damping adjustable shock absorber with four-stage is able to improve the performance of vehicle suspension, in theoretical. However, there is little research studied on its characteristics till now.

The contribution of this paper is first to focus on the design and draft of a four-stage damping adjustable shock absorber (FDASA), and exploit its sample prototype. To test its damping performance, an experiment rig has been built by using electro-hydraulic servo system, and testing data have been obtained by using this experiment rig. Finally, by modeling and simulating the FDASA, the comparison can help to validate the damping characteristics of this FDASA.

2. Working principle

The shock absorber mentioned in this paper has four-stage damping positions, which can support drivers to choose different damping characteristics according to road condition and driving velocity. It is a kind of valve-controlled shock absorber, which is configured as a damping adjuster outside a typical two-tube fixed damping shock absorber. The damping adjuster has a similar structure as the main body, which is to connect the rebound chamber and compression chamber and to cause different pressure drops, so that to realize different combinations of the damping characteristics.

The diagram of the FDASA is vividly shown in Figure 1. As mentioned above, the main body is two-tube cylinder structure, whose piston has throttles designed and one-way damping valve slices mounted on both sides. The damping adjuster, which consists of two solenoid valves and four slice-type check valves, is fastened outside the main body. It has two oil flow ports on the damping adjuster: one is connected to rebound chamber, and another one is connected to compression chamber. As shown in Table I, when in working states, four-stage damping characteristics can gain by charging or discharging of solenoid valve Y1 or Y2 to meet different requirements of road conditions, and to obtain the optimal damping characteristics of ride comfort and road-holding stability. Figure 2 is the prototype of this FDASA developed for experiment.

3. Experiment design and set up

3.1 Experiment design

As per the government standard QC/T545-1999 (1999 QC/T, 1999), an experimental rig for automobile shock absorbers is designed according to the requirements of performance testing. Figures 3 and 4 are used to illustrate the schematic and physical diagrams of this experiment rig, respectively. Obviously, it composes of three key parts of electro-hydraulics servo system: mechanical structure, hydraulics unit and electronic control units. The mechanical structure, which is the foundation of this experiment rig, mainly consists of one bracket and some connector; moreover, the bracket is made up of one body support, four guiderails, and one integrated sliding sleeve. The piston rod of shock

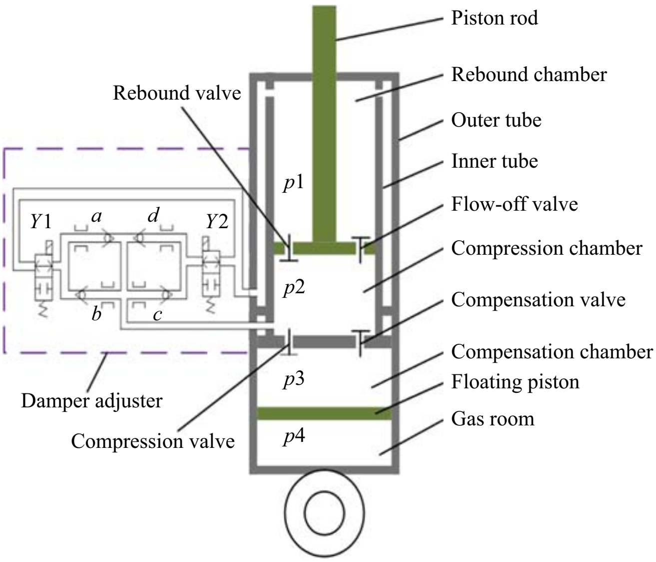
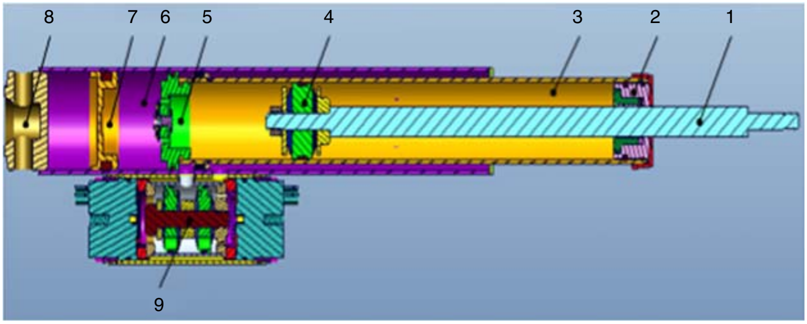


Figure 1. Diagram of the FDASA

Table I. Opening stations at four stages of shock absorber

Damping position	Working valves	Compression stroke	Rebound stroke	Damping
1	Y1 and Y2	a and d	b and c	Soft
2	Y1	a	b	Medium
3	Y2	d	c	Hard
4	None	None	None	Harder



Notes: 1 – rod; 2 – guide seal; 3 – main cylinder; 4 – piston valve; 5 – body valve; 6 – reservoir chamber; 7 – floating piston; 8 – joint; and 9 – damping adjuster

Figure 2. Structural of adjustable damping shock absorber

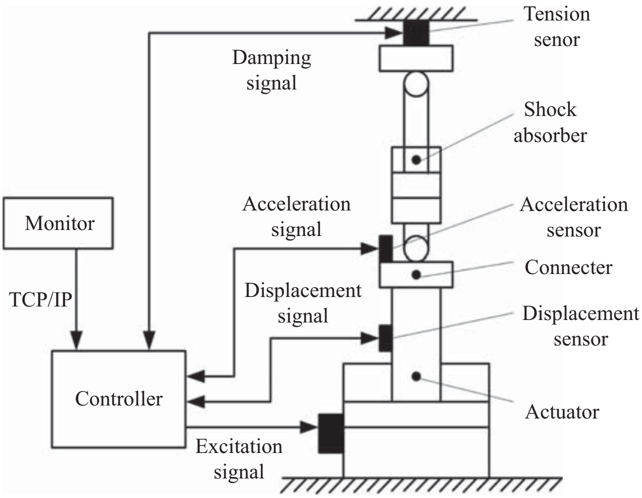


Figure 3. Schematic diagram of experiment rig

absorber connects to the tension sensor by one connector, and it is fixed to the bracket. The tube end of shock absorber connects to actuator, whose displacement can be measured by the displacement sensor. The hydraulics unit is composed of oil power unit, electro-hydraulic servo valves and hydraulic servo cylinder; this part is the core of driving control in the experiment rig. The actuator is the excitation modular of a servo cylinder controlled by servo valve. Thereinto, hydraulic cylinder is a single piston rod hydraulic servo cylinder. This servo cylinder's stroke is 120 mm in maximum, and its highest velocity can be up to 1.0 m/s. Electrical control unit is mainly constituted by monitor, control unit and corresponding sensors, and the acquisition of damping force, acceleration and displacement signal is accomplished by data acquisition and driving board. Driving board is an electronic unit used to drive the servo valve, which is to force the hydraulic cylinder motion. This motion works as the corresponding excitation for shock absorber.

3.2 Experiment set up

The experiment rig of electro hydraulic servo control is used to test the designed shock absorber. Parameters which are set up are as follows: the piston rod stroke is ±37.5 mm in

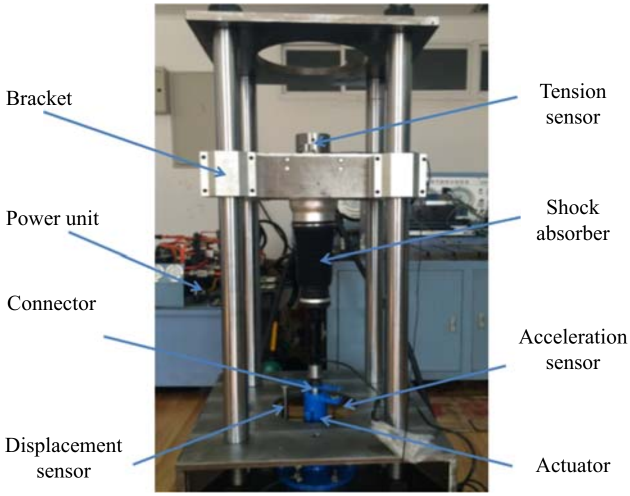


Figure 4.
Experiment rig

maximum; a sine wave signal is used as the input excitation of this shock absorber; a power of 24 V DC is used to charge solenoid valves to make it in different damping positions; and piston rod initial velocities are 0.052, 0.131, 0.262 and 0.524 m/s. By changing the frequency of actuator, the velocity may vary in difference, respectively. And then, the corresponding damper force would be gained at each time.

4. Mathematical model

According to the diagram of the FDASA, the damper components are piston valve assembly, body valve assembly and damper adjuster. Based on the hydraulic principle, the mathematical model of the FDASA can be established through each component's models, as it is same working process at each damping position. For simplification, this paper only takes the damping position 3 into consideration.

Continuity equations of hydraulic principle are used for compression and rebound chambers. Without considering the compressibility effect of oil fluid, when in compression stroke, flow rate Q_1 caused by the movement of piston rod in rebound chamber can be expressed as:

$$Q_1 = (A_p - A_r)\dot{x}_{y1} \tag{1}$$

where x_{y1} is the displacement of piston rod, A_p the area of piston, and A_r the area of piston rod.

And the flow rate in compression chamber can be expressed as:

$$Q_2 = A_p\dot{x}_{y1} \tag{2}$$

The flow rate Q_1 can be divided into two parts: Q_l is the flow rate through flow-off valve, and Q_{zd} is the flow rate through check valve d:

$$Q_1 = Q_l + Q_{zd} \tag{3}$$

And the flow rate Q_2 also can be divided into two parts: Q_1 is the flow rate through flow-off valve, and Q_y is the flow rate through compression valve:

$$Q_2 = Q_1 + Q_y \tag{4}$$

Also, continuity equations are used in different orifices, and the functions between flow rate Q and pressure drops can be obtained:

$$Q_l = \frac{\pi\delta_l^3(p_2-p_1)}{6\mu \ln(r_{bl}/r_{kl})} \tag{5}$$

$$Q_{zd} = \frac{\pi\delta_{zd}^3(p_2-p_1)}{6\mu \ln(r_{bzd}/r_{kzd})} \tag{6}$$

$$Q_y = \frac{\pi\delta_y^3(p_2-p_3)}{6\mu \ln(r_{by}/r_{ky})} \tag{7}$$

where p_1, p_2, p_3 are the pressure in rebound chamber, compression chamber and compensation chamber, respectively; δ_l, δ_{zd}, δ_y is the deformation of valve slices mounted on flow-off valve, check valve d and compression valve, respectively; r_{bl} r_{bzd}, r_{by} are the outer radius of each valve slice, and r_{kl}, r_{kzd}, r_{ky} are the radius of each orifice configured on flow-off valve, check valve d and compression valve. The term u is the viscous of oil.

According to the movement of floating piston, following function can be obtained:

$$(p_3-p_4)A_c = m_2\ddot{x}_{y2} \tag{8}$$

where x_{y2} is the displacement of floating piston, A_c the area of compensation chamber, and p_3, p_4 are the pressure in compensation chamber and gas room, respectively.

And the state equation of gas can be expressed as:

$$p_{40}V_{40}^n = p_4V_4^n \tag{9}$$

where p_{40}, V_{40} are the values of pressure and volume in gas room when at the initial state, V_4 is the value of volume in gas room when at the optional position, and n is the constant of air, normally, $n = 1.4$.

Solving all the formulas listed above, the pressures values of p_1, p_2, p_3, p_4 can be worked out according to the initial conditions. Furthermore, after force balance analysis of piston rod, damper force in compression and rebound stroke may be calculated during to the follow function:

$$F_c = p_2A_p - p_1(A_p - A_r) \tag{10}$$

The model of the damper force can be simulated in Matlab and Simulink, and results are plotted out finally.

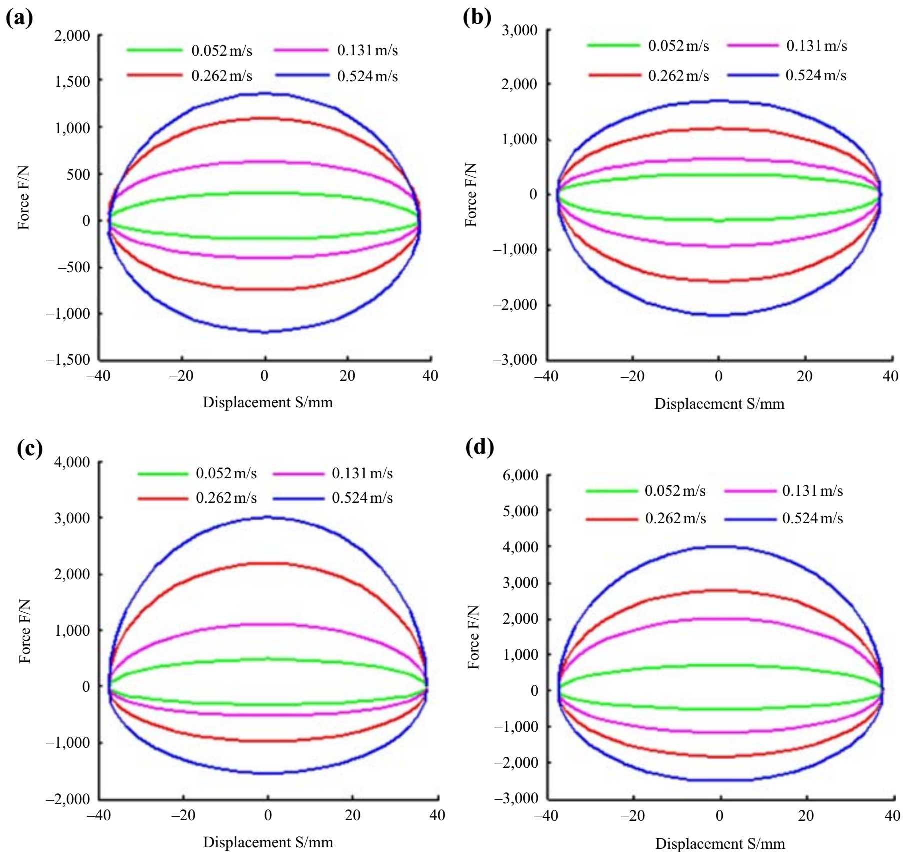
5. Result analysis

As we all know, there are mainly two parts of performance tests and experiments for shock absorber: indicator diagram characteristics and velocity characteristics. The former is about damping force vs displacement curves, which shows the work damping force has done in a cycle displacement, and the latter is about damping force vs velocity curves, which shows the nonlinear change of damping force under different velocities.

5.1 Indicator diagram

Assuming displacement, velocity and damping force are negative in compression stroke. As a result, each damper force at different damping positions is figured out after simulation and prototype experiments. Figure 5 shows results of indicator diagrams through theoretical analysis and simulation. Figure 6 shows results of indicator diagrams through the experiments.

We can get important information from these indicator diagrams. First, each curve of indicator diagram is in a close loop continuously at each damping position and each velocity, calculated by simulation or measured by experiments. Coincidently, curves indicate that the same damper force occurs at the same damping position and velocity, which states that this FDASA has good-quality damping characteristics. Second, when at the same damping position, increasing the piston rod's velocity makes the damping force higher; apparently, when in the same velocity, changing different damping positions makes a significant change in the damping force. Finally, the damping force is softest and hardest at the damping positions 1 and 4, respectively. When at the damping position 2, the damping force is harder in compression stroke than that in rebound stroke; on the contrary, when at the damping position 3, it is softer in compression stroke than that in rebound stroke. The sentence "at the same damping position and velocity" means comparison in simulation and measured. For example, at the damping position 1 in Figure 5 and 6,



Notes: (a) Damping position 1; (b) damping position 2; (c) damping position 3; (d) damping position 4

Figure 5. Indicator diagrams by simulation

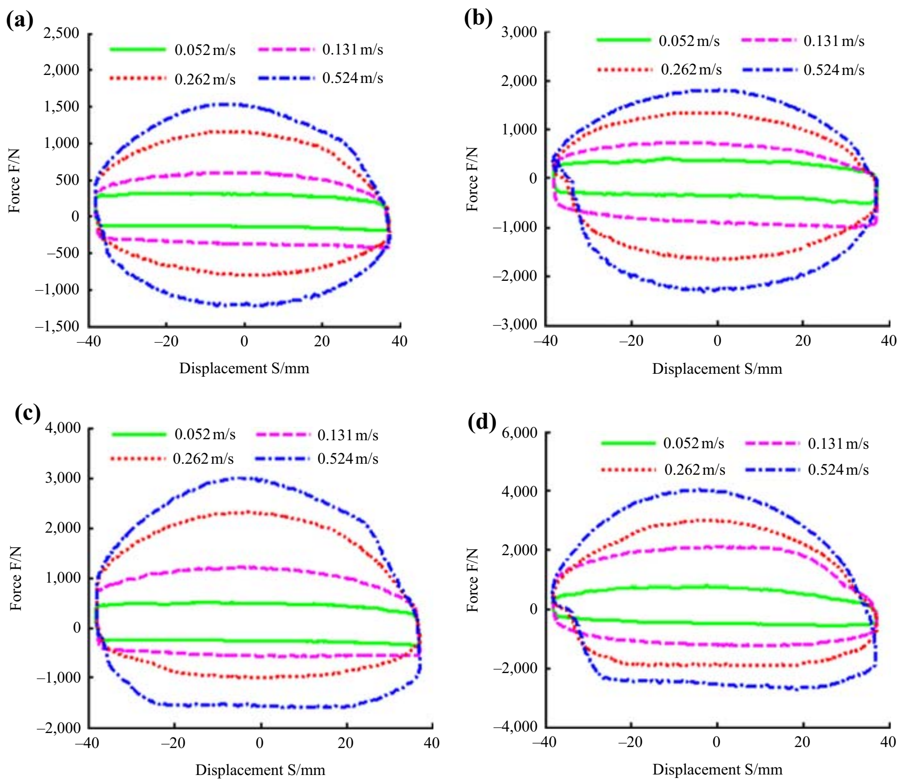


Figure 6.
Indicator diagrams by measurement

Notes: (a) Damping position 1; (b) damping position 2; (c) damping position 3; (d) damping position 4

if velocity is in 0.052 m/s, the maximum damper force would be about 1,500 N. So we call it "curves indicate that the same damper force occurs".

The sentences "Second, when at the same damping position…" results are compared in each figure. For example, at the damping position 1, increasing the velocity, damper force would rise.

So, those other results would be showed in comparison between respective figures.

Obviously, there is no badly empty stroke and skewness in indicator diagram by simulation. However, when the piston rod reaches near the lowest and highest displacement, there are some temporary empty strokes at the damping positions 2 and 4 by measurements at speed of 0.262 and 0.524 m/s. This phenomenon may be caused by some oil leakage comes from the joint part of outer tube and damping adjuster, which are fastened by iron hoops, and lead to its seal is not very well at high speed.

5.2 Velocity characteristics

Velocity characteristics under different damping positions also have computed out according to simulation and test data. Results of simulation are plotted in Figure 7 apparently. Figure 8 shows that same velocity characteristics are occurring reproductively at same damping position in measurements. Simultaneously, each curve is changing very continuously and steady, and there are no distortion points both in simulation and in measurement.

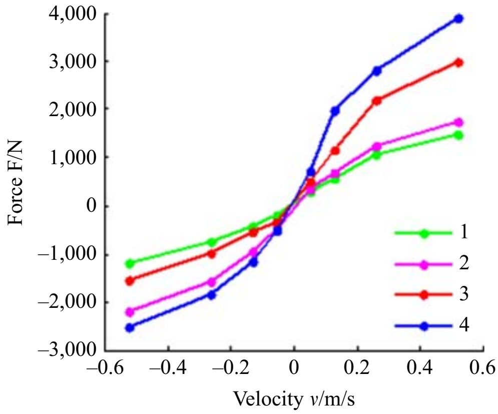


Figure 7. Velocity characteristics by simulation

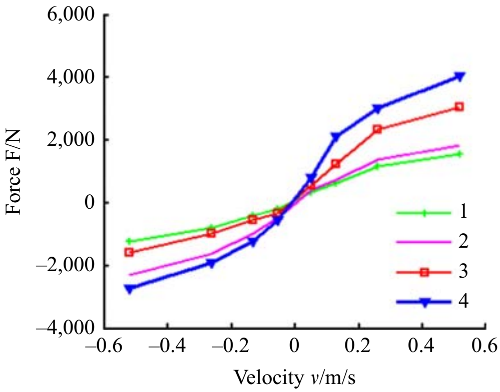


Figure 8. Velocity characteristics by measurement

6. Conclusion

This paper has developed an FDASA, and introduced its working principle in detail. Then, the product sample prototype and experiment rig have developed out. Finally, it focused on the damping characteristics test of this prototype. Results are as follows:

(1) When at the same damping position, increasing the rod's velocity makes damping force higher; apparently, when in the same velocity, changing different damping positions makes a significant difference in the damping force.

(2) Damping force is softest and hardest at the damping positions 1 and 4, respectively. When at the damping position 2, the damping force is harder in compression stroke than that in rebound stroke; on the contrary, when at the damping position 3, it is softer in compression stroke than that in rebound stroke.

(3) A temporary empty stroke of damping force occurred near the highest or lowest position both at the damping positions 2 and 4. This phenomenon may be caused by some oil leakage coming from the joint part of outer tube and damping adjuster at high speed.

(4) Each curve of force-velocity diagrams is changing very continuously and steady, and there are no distortion points. That means the design of this shock absorber could meet the performance requirements as expected. All studies and tests worked on this FDASA can extremely improve the ride comfort and safety for automobile suspensions.

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Corresponding author
Fangwei Xie can be contacted at: xiefangwei@ujs.edu.cn