An Active-control digital hydraulic damper: Design, modeling and simulation

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In the paper, an active-control digital hydraulic damper is designed and modelled, which is comprised of multiple digital throttle units. The digital throttle units consist of a throttle valve and high speed on/off solenoid valve which are connected in series. A mathematical model of the digital throttle unit is provided. A control algorithm based on PID is given which is used for the active control of the variable throttle area. The working process of the active damper is simulated and analyzed in AMESim with a comparison of the working process of a passive Porous Hydraulic Damper. Pressure-Time curves, Displacement-Time curves and Velocity-Time curves are given which show thet performance differences between the active-control digital hydraulic damper and the passive Porous Hydraulic Damper. According to the simulation comparison, it is clear that the buffering stroke of the active-control digital hydraulic damper is increased by 8.9% but the peak pressure of the active-control digital hydraulic damper is reduced by 20% compared with the passive Porous Hydraulic Damper. It indicates better performance of the Digital hydraulic damper which can complete the damping process more smoothly and effectively.

Keywords: High speed on/off solenoid valve, digital throttle unit, digital hydraulic damper, active-control
Target audience: Mobile Hydraulics

1 state of art

The hydraulic damper is often used as a shock absorber to alleviate violent impacts. The basic principle of a Hydraulic damper is absorbing energy through the orifice as liquid flows, this is called flowing resistance. It is used increasingly because of its large energy density and stable damping performance. According to the control way, the hydraulic damper can be divided into these three types: passive hydraulic damper, semi-active hydraulic damper and active hydraulic damper. The damping process for a passive hydraulic damper is achieved by means of the constant throttle area. The damping process for a semi-active hydraulic damper is achieved by means of controlling the constant throttle area semi-active. The damping process for an active damper is achieved by means of controlling adjustable throttle area damper.

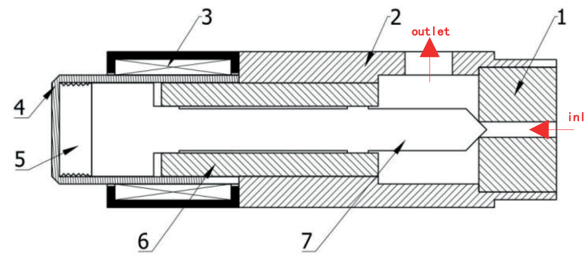
Feng Shuai analyses the influence of parameters on the performance of passive hydraulic damper used for the artillery[1].Based on the Bernoulli equation, a mathematical model of a passive hydraulic damper is given by Huang Jingfeng[2] and simulated in Simulink. Geometric modelling, numerical analysis and visualization of passive damper are done by Zheng Biao[3] by means of CFD technology. Based on Runge-Kutta method, the effect of nonlinear damping on the response of a hydraulic shock absorber is studied by Chandra, Shekhar and Hatwal [4]. As a new concept, semi-active damping control is proposed by Lord M.J.Cosby and D.C.Karnopp Davis[5], which is different from full active damping control. Liu Hui designed a semi-active damping device for landing gear by means of a parallel high-speed on-off valve, a mathematical model is set up and related experiments are carried out to verify the simulation result[6]. Based on the high-speed on-off valve, YangJianwei[7] developed a semi-active shock absorber for a high-speed vehicle. Liu Shaojun etc.[8] have given a simulation model of semi-active shock absorber based on the high-speed on-off valve and simulated its control method. An active damper for landing gear based on the hydraulic servo valve is developed by NASA[9-10].

Under uncertain conditions, satisfying the requirements of damping for a heavy impacting force, ensuring the stability and response speed of the damper with decreasing working pressure in the chamber, are big challenges for the researchers. In this paper, an Active-control digital hydraulic damper is introduced which has the features of the low cost, good stability and high response speed.

2 The working principle of the digital hydraulic damper

using high-speed on-off valve as the digital unit, active variable-throttle-control of the damper can be accomplished by means of a computer. The throttle valve and the high-speed on-off valve are connected in series so as to form the digital throttling unit shown in Figure 1, which consists of the throttle valve body, coil, valve core, gas spring, keeper and pole shoe. A number of digital throttle units are combined together in parallel to form a digital throttle valve-group. The working principle of the digital throttling unit is as follows:

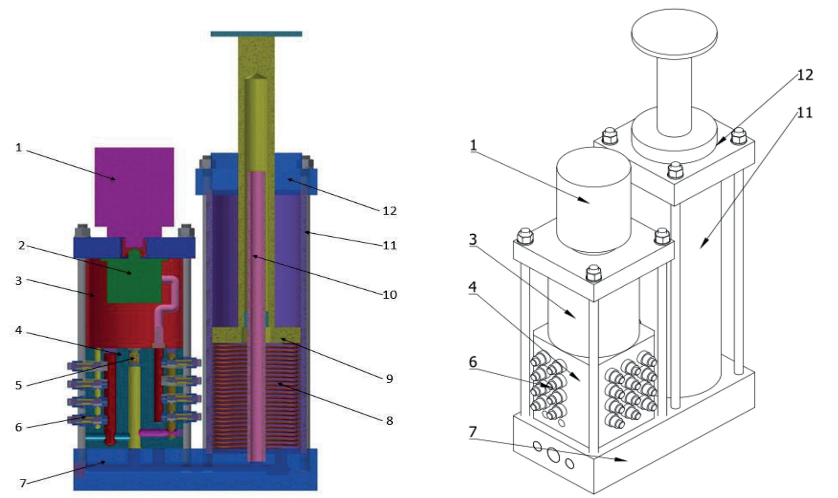
When the coil is energized, a magnetic force will be generated that drives the valve core to move, the throttle valve will be opened and the gas spring will be compressed so as to let the oil pass through. When the power of the coil is lost, the valve core will be reset by the gas spring and the throttle valve will be closed.



1.throttle valve2.valve body3.coil4. keeper 5.gas spring 6. pole shoe 7. valve core

Figure1.digital throttle unit diagram

The digital hydraulic damper, which is shown in figure 2, is mainly composed of a cushion cylinder, digital throttling unit, valve block,gear-pump, motor and tank.



1.motor2.gear pump3.tank4.valve block 5.rod cavity charging valve6.digital throttle unit 7.connection unit 8.rest spring 9.piston 10.internal guide rod 11.cylinder 12.cylinder head

Figure2.digital hydraulic damper diagram

The general configuration of the digital hydraulic damper is given in figure 3. For the digital hydraulic damper, with the help of an accelerometer and pressure sensor, real-time adjustment of the throttle area can be obtained according to the impacting force, and real-time adjustment of capacity can be obtained according to the information given by the sensors.

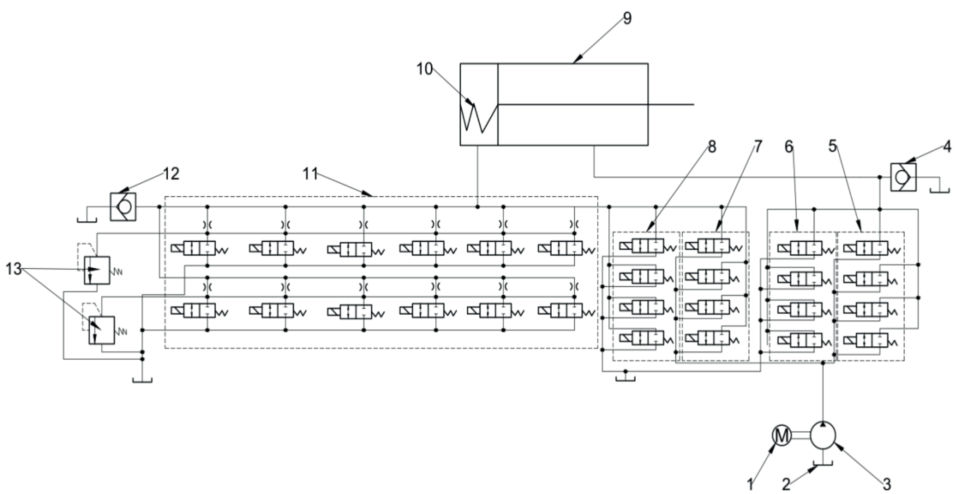
The basic working process is as follows:

In the first stage, before the head of digital hydraulic damper is shocked by impacting workpiece, the acceleration of the impacting workpiece is detected by the accelerometer. According to the acceleration value of the impacting workpiece, the predefined stroke and the initial pressure of digital hydraulic damper are given. Through the gear pump, a group of control valves for the non-rod chamber and a group of control valves for the rod chamber, the stroke and the initial pressure can be adjusted in real-time.

In the second stage, after the head of digital hydraulic damper is shocked by the impacting workpiece, the acceleration of the impacting workpiece will be detected in real-time by the accelerometer and the pressure in buffering chamber will be detected in real-time by the pressure sensor. By means of the average value processing, the processed value will be inputted to the digital throttling unit which is mounted on the valve block as shown in Figure 2, so that the optimal throttling area can be given with real-time control.

With the control valve for the non-rod chamber, the oil can be supplied from the tank quickly during the buffering process, avoiding suction or the effect of negative pressure. With the control valve for the rod chamber, the rapid reset of the digital hydraulic damper can be realized in the resetting process.

Using a non-rod cavity control valve, oil can be provided quickly from the tank during the buffering process, avoiding the suction or influence of negative pressure. With the rod cavity control valve, the digital hydraulic damper can be reset quickly during the reset process.



1.motor 2.tank3.gear pump 4.check valve for rod chamber 5. group of oil inlet control valves for rod chamber 6. group of oil outlet control valves for the rod chamber 7. group of oil inlet control valves for the rodless chamber

8. group of oil outlet control valves for the rodless chamber 9.cylinder10.Reset spring11.digital throttle unit

12. check valve for the rodless chamber 13 safety valve

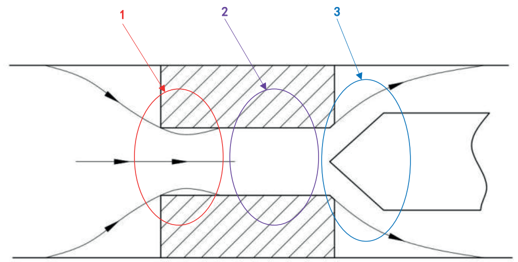
Figure3.digital hydraulic damper Hydraulic system schematic diagram

There are two safety valves in the damper. These two safety valves can help to prevent accidents if the buffering process cannot be achieved in the case of an electrical system failure. In addition, these two safety valves can provide a security guarantee for effectively buffering in unexpected circumstances.

3 Calculation model of the digital throttle unit

A digital hydraulic damper consists of several digital throttle units which are made up of the high-speed on-off valve and the throttle valve. When the oil passes through the unit, there is a local pressure loss and frictional pressure loss, a damping force is created so as to buffer the impacting object until it stops. For the fluid in the damper, the following assumptions are made:

- The fluid is incompressible
- Quality of the fluid is negligible
- The fluid is Newtonian fluid



1. local pressure loss abrupt contraction in cross-section 2. friction pressure loss through the orifice

3. local pressure loss through cone valve port Figure4.energy loss in digital throttle unit

As shown in Figure 4, when the fluid flows through the digital throttling unit, there are three kinds of energy losses which are local pressure loss for an abrupt contraction in cross-section, frictional pressure loss through the orifice and local pressure loss through the cone valve port. A mathematical model for the energy loss of the unit is analyzed according to the above three factors.

3.1 local pressure loss for an abrupt contraction in cross section

In Figure5, local pressure loss for an abrupt contraction in cross section is illustrated when fluid flows into the orifice of the digital throttle unit. Local pressure loss $_m$es for an abrupt contraction in a cross-section can be described as follows. C_c is obtained from the experiment by J.Weishach[11], which is shown in Table 1.

$$h_m = \left(1 + \frac{1}{C_v^2 C_c^2} - \frac{2}{C_c}\right)\frac{v_2^2}{2g} = \zeta \frac{v_2^2}{2g} \qquad (1)$$

3.2 frictional pressure loss through the orifice

According to the Navier-Stokes equation, the pressure lossΔp[12]flow through the distance of L is as follows:

$$\Delta p = \frac{32\mu l v_2}{d^2} \qquad (2)$$

3.3 local pressure loss through the Cone valve port

Flow through the orifice can be expressed as follows：

$$Q = C_q A_2 \sqrt{\frac{2\Delta p}{\rho}} \qquad (3)$$

The theoretical formula of flow coefficient C_q[13] is as follows：

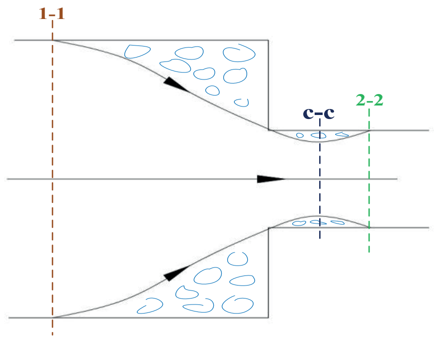
$$C_q = \left\{\frac{24}{\sin\theta}\left(\ln\frac{d_1}{d_2}\right)\frac{1}{Re\left(\frac{2h}{d_m}\right)} + 0.18\left(\frac{d_m}{d_2}\right)^2 + \frac{54}{35}\left(\frac{d_m}{d_1}\right)^2\right\} \tag{4}$$

In figure 6, if the number of opened digital throttle units is n at the moment, then the flow Q which is flowing through the port of the cone spool is as follows:

$$Q = \frac{Av}{n} \tag{5}$$

The pressure loss caused by the cone spool of the digital throttle unit is as follows:

$$\Delta p = \frac{\rho}{2}\left(\frac{Av}{nC_qA_2}\right)^2 \tag{6}$$



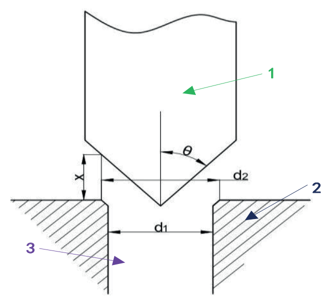


Figure5. local pressure loss for an abrupt contraction in cross-section *Figure6. spool Schematic of the digital throttle unit*

A_2/A_1	0.01	0.10	0.20	0.30	0.40	0.50	0.60	0.70	0.80
C_c	0.618	0.624	0.632	0.643	0.659	0.681	0.712	0.755	0.813
C_v	0.98	0.982	0.984	0.986	0.988	0.990	0.992	0.994	0.996
ζ	0.49	0.458	0.396	0.364	0.317	0.264	0.197	0.126	0.197

Table1.C_c、$C_v$and ζ of abrupt contraction in cross section

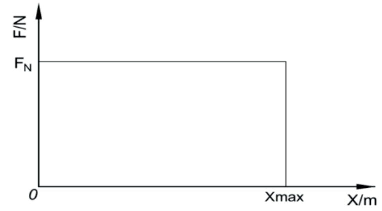
4 Control algorithm

As mentioned in part 2, there are two working stages for the damper. During the first stage, the value of buffering stroke is calculated according to the measured acceleration value. In this case, the safety coefficient is set as 1.3 to prevent accidents.The initial pressure is determined according to the stiffness of the reset spring and 10% of the total buffering energy.

During the second stage, a PID control algorithm is adopted, in which the control goal is to get a minimum peak buffering force during the shocking process. In the ideal case, the force-displacement curve of the damper is shown in Figure7. When the force-displacement is approximately trapezoid, the maximum buffering force of the damper is a constant value, the buffering efficiency is the highest, and the impact energy absorbed by the damper is the largest. It can be described in equation 7.

$$F = F_N \tag{7}$$

When the mass and the velocity of the moving object impacting the damper is known, the energy absorbed by the damper in each stroke is described in equation 8.



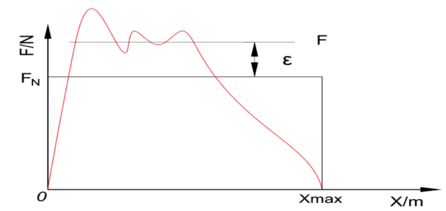


Figure7. force- displacement in ideal state *Figure8.Control target diagram*

$$W = \frac{1}{2}mv_1^2 \tag{8}$$

Therefore, in the ideal case, the ideal maximum buffering force of the damper is described as follows:

$$F_N = \frac{W}{X_{max}} \tag{9}$$

When the real working condition shown in red in Figure 8 is taken into consideration, the control objective function proposed in this paper is as follows:

$$\varepsilon = F - F_N \tag{10}$$

PID control flow chart of the damper is shown in Figure 9.

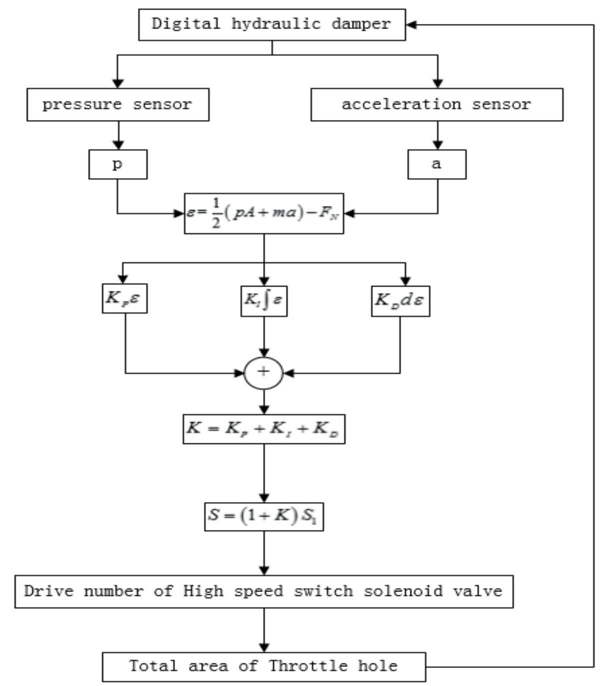
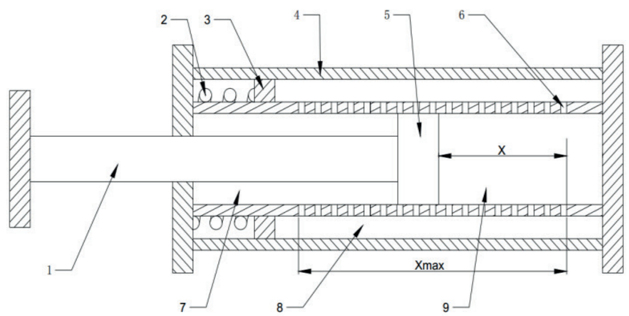


Figure 9. PID control flow chart

5 Simulation analysis with a comparison of the active-control digital hydraulic damper and the passive Porous Hydraulic Damper.

At the same time, a passive Porous Hydraulic Damper is also designed, the 2D assembly drawing of the passive Porous Hydraulic Damper is shown in Figure 10. The passive Porous Hydraulic Damper has the same maximum orifice area as the Digital hydraulic damper. During the working process, the buffering capacity and buffering force cannot be changed in the passive Porous Hydraulic Damper.

The same simulation parameter is applied to the passive Porous Hydraulic Damper and the Digital hydraulic damper.The pre-defined simulation parameter is given in table 2. A simulation model of a passive Porous Hydraulic Damper is given in Figure 11. A simulation model of the Digital hydraulic damper is given in Figure 12. According to the simulation results shown in Figure 13 to Figure 15, different performance characters are shown, which illustrates the advantages of the Digital hydraulic damper.



1. rod 2.resetting spring 3.resetting piston 4.outer cylinder 5.piston 6. orifice 7. chamber with a rod 8. resetting chamber 9. non-rod chamber

Figure 10. 2D assembly drawing of the passive Porous Hydraulic Damper

Parameter	value
the weight of impact object, M	200Kg
Velocity of impact object, V	4.427m/s
Working pressure of Hydraulic Damper, Pw	80Bar
Maximum Working pressure of Hydraulic Damper, Pmax	100Bar
Diameter of piston, D	63 mm
the wall thickness of the outer cylinder, Toc	7.5 mm
piston stroke, ps	150 mm
Diameter of rod, Dr	30mm
Diameter of inner cylinder, Dic	63mm
Spring constant	23.4 N/mm

Table 2. simulation parameter

In Figure 13 it shows that the maximum pressure in the non-rod chamber of the passive damper is 84 bar and the maximum pressure in the non-rod chamber of the active damper is 63 bar, the peak pressure is reduced approximately by 20%. In Figure 14 it shows that the damping displacement of the passive damper is 120mm and the damping displacement of the active damper is 130mm, it also shows that the damping displacement is increased by 8.3% and the change of rate of displacement is relatively stable in the initial stage.

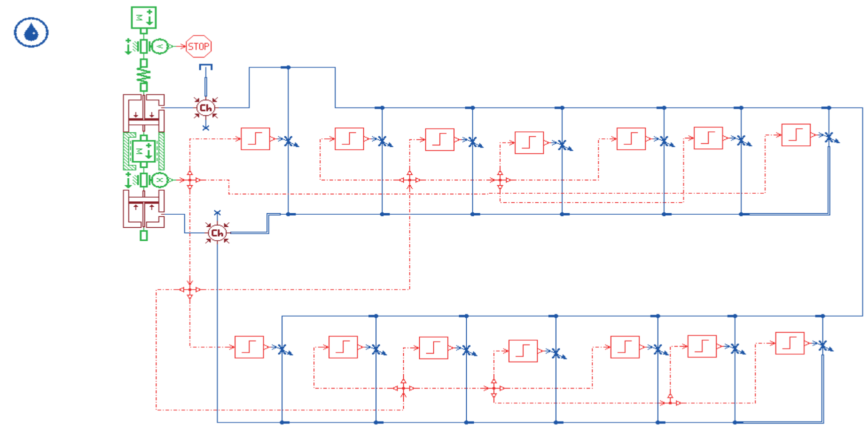


Figure 11. Simulation model of a passive Porous Hydraulic Damper

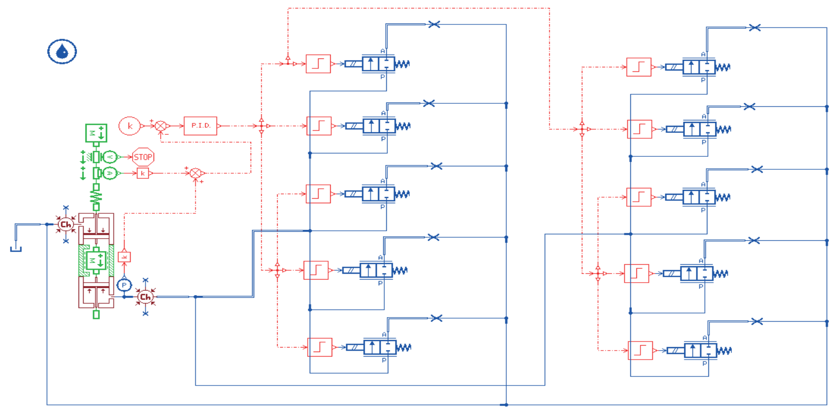


Figure 12. Simulation model of the Digital hydraulic damper.

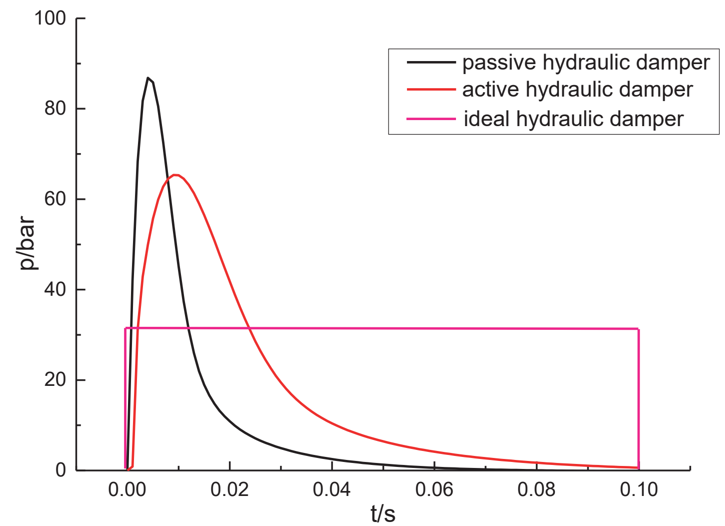


Figure13. Pressure-Time curves of non-rod cavity

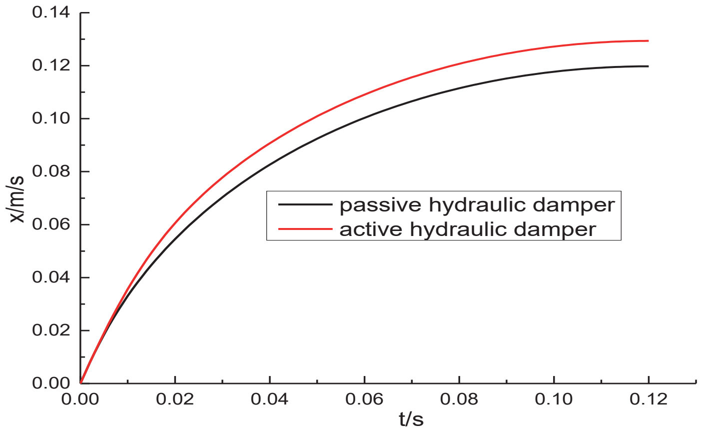


Figure14. Displacement-Time curves

In Figure 15 it shows that the speed of the active damper is decreased, which is slower than the speed of the passive damper so that severe shock pressure and higher pressure in the non-rod chamber can be avoided when the speed of the impact object is relatively high. The working pressure in the non-rod chamber is decreased obviously and the performance of the active damper is stable.

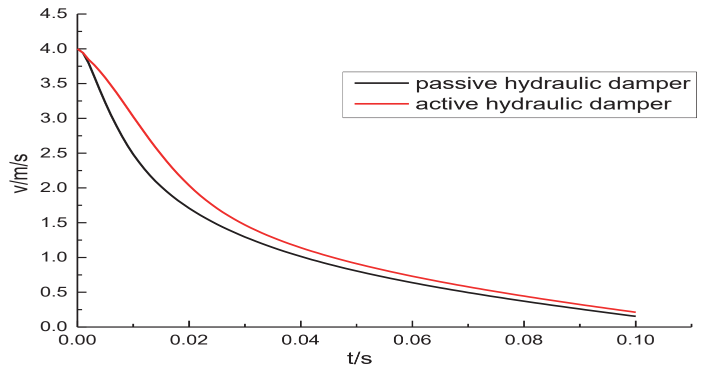


Figure15. Velocity -Time curves

6 Conclusion and future works

In the paper an active-control digital hydraulic damper is designed and modelled, a mathematical model of the digital throttle unit model is provided. Control algorithm based on PID is given to achieve active control of variable throttle area. According to the simulation, it is clear that the damping stroke of the active-control digital hydraulic damper is increased by 8.9% but the peak pressure of the active-control digital hydraulic damper is reduced by 20% compared with the passive Porous Hydraulic Damper. It demonstrates that the Digital hydraulic damper can complete the damping process more smoothly and effectively.

In the future, the physical prototype will be tested in the performance testing system so as to test the relevant parameters that affect the performance of the hydraulic damper. The model will be verified and validated according to the testing data.

7 Acknowledgements

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Nomenclature

Variable	*Description*	*Unit*
p_F	Surface Pressure	[bar]
T	Temperature	[K]
v	Velocity	[m/s]
α_D	Discharge Coefficient	[-]
ε	Angled	[rad]
C_c	Contraction coefficient of stream	[-]
A_c	Area of stream in cross-section c-c	[m^2]
A_2	Area of throttle orifice in cross-section 2-2	[m^2]
C_c	Contraction coefficient of stream	[-]
A_c	Area of stream in cross-section c-c	[m^2]
A_2	Area of throttle orifice in cross-section 2-2	[m^2]
C_v	Coefficient of flow rate, $C_v = \frac{v_c}{v_0}$	[-]
V_c	Average flow rate in cross-section c-c	[m/s]
V_0	Average flow rate in cross-section 1-1	[m/s]
V_2	average flow rate in cross-section 2-2	[m/s]
Z	Coefficient of local pressure loss in cross section	[-]
G	Gravitational acceleration	[m/s^2]
M	Kinematic viscosity of fluid	[m^2/s]
D	The diameter of orifice	[m]
P	Fluid density	[kg/m^3]
C_q	Flow coefficient	[-]
Δp	Pressure loss between inlet port and outlet port	[bar]
X	Rising height of the cone spool	[m]
Θ	The half angle of the cone spool	[rad]
H	xsinθ	[m]
Re	$\frac{v_m d_m \rho}{2\mu}$	[-]
v_m	Average flow rate in section d_m	[m/s]

d_1	Chamfering diameter in the outlet port	[m]
A	Area of piston in cross section	[m^2]
v	Piston velocity at the moment	[m/s]
F	Buffering force at any time of the damper	[N]
F_N	Ideal maximum buffering force	[N]
v_1	Initial velocity of the moving object before contacting with the damper	[m/s]
W	Energy absorbed by damper in one stroke	[N*m]
X_{max}	Maximum buffering stroke	[m]
E	Difference between F and F_N	[N]

References

/1/ FENG Shuai.Design and Research of 60mm Automatic Light Mortar Recoil Buffer and Direction [D]. Shenyang:Dongbei University,2013.

/2/ HUANG Jing-Feng, DONGNan, LIU Sheng-Jie.Dynamic Characteristic Analysis of Hydraulic Buffer Based on Simulink[J]. Machine Tool and Hydraulics,2015, 43(13):145-148.

/3/ ZHENG Biao.Computational Analysis and Experimental Study of a New Type of Liquid gas Buffer Based on CFD Technology[D].Dalian: Dalian University of Technology,2014.

/4/ Liu Hui.Research on Property and Semi-active control of Landing Gear Shock Absorption System[D].Nanjing: Nanjing University of space and Aeronautics and Astronautics,2007.

/5/ M.J. Cosby, D.C vibration control, Karnopp, The active damper-a new concept for shock and Shock and Vibrations Bulletin, 1973, 43(4): 119-133.

/6/ YANG Jian-Wei.Study on Semi-Active Control System for Lateral Vibration of High-Speed Vehicle[D].Beijing: Academy of Railway Sciences.

/7/ LIU Shao-jun.Simulation of the Lateral Semi-Active Damper of High-Speed Train with Sky-hook Control[J]. Machine tools and hydraulics,2006(10),70-72.

/8/ CHANDRA SHEKHAR N, H.HATWAL.The response of Non-linear Dissipative Shock Isolators[J].Journal of Sound and Vibration,1998,20(04):15-30.

/9/ CBrecher , A Schulz ,M Week.Electrohydraulic Active Damping System[J].Manufacturing Technology , 2005 , 54 (1) :389-392.

/10/ J.N. Daniels, A method for landing gear modeling and simulation with experimental validation[J].NASA Contractor Report201601,1996.

/11/ SHENG Jing-Chao.Hydraulic fluid mechanics，Beijing：Machinery Industry Press，1980.

/12/ WeisbachJ.,Die Experimental-Hydraulik, Engelhardt, Freiburg,1855.

/13/ 市川常雄,清水孝.ポペット弁の流量係数について,机论 31 卷 222 号，1965.