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Simulation research on the four-nozzle flapper valve based on GMA

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Abstract. Nozzle-flapper type electro-hydraulic servo valve operated by torque motor has been widely used in industrial applications. As their bandwidths are limited, they are not suitable for high-speed applications. This paper presents a novel nozzle flapper valve driven by the giant magnetostrictive actuator, which has been designed and integrated into the four-nozzle flapper valve to replace the torque motor. And the influence of involved structural parameters on the dynamics of the actuator and the nozzle flapper valve is analyzed by AMESim. The simulation results can provide an important reference and basis for the optimization and design of the four-nozzle flapper valve.

Introduction

Giant magnetostrictive material (GMM) is a kind of new function material with big magnetostrictive strain, fast response speed, high energy density and large output force etc, which can effectively realize the reciprocal transformation of electromagnetic energy and mechanical energy. In recent years, many applications[1,2] such as linear motors, sensors, fluid control valves micro-robots and high accuracy actuators etc, have been reported. An application of GMM for the novel nozzle flapper valve with dual fixed orifices and four-nozzles is presented dealt with in the paper. And then the simulation models of the giant magnetostrictive actuator (GMA) and the nozzle flapper valve are separately established by AMESim[3,4,5]. Incorporating the dynamic of the GMA, the dynamic performance of the valve is simulated. The simulation results show that it could be a very useful guidance on the further optimization design of the four-nozzle flapper valve with GMA.

The structure and working principle of four-nozzle flapper valve

The basic structure of the four-nozzle flapper valve with GMA which is designed based on driving principle of giant magnetostrictive material is shown in Fig.1.

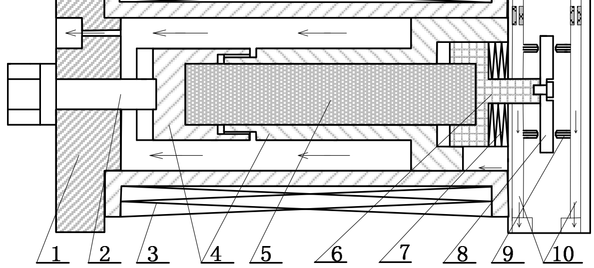


Fig. 1 Structure of four-nozzle flapper valve with GMA (1. Front-end housing, 2. Adjusting screw, 3. Driving coil, 4. Thermal compensation module, 5. GMM rod, 6. Output rod, 7. Spring, 8.Flapper, 9. Nozzle, 10. Rear-end housing)

As Fig.1 shows, the GMA employs spring and output rod to achieve mechanical pre-stress which can be adjusted by adjusting screw according to the demand. The thermal deformation of the thermal compensating module could balance out the thermal deformation generated by GMM rod. The driving coil wound around the GMM rod is employed as the excited magnetic field to cause the GMM rod to expand in magnetic field direction, which provide a force to the output rod and accordingly results in the movements of a flapper between the two nozzles. Due to the movement of the flapper, there is a pressure difference between the nozzle-flapper (Fig.2), which is communicated to the load such as spool end chambers, thereby causing proportional load movement.

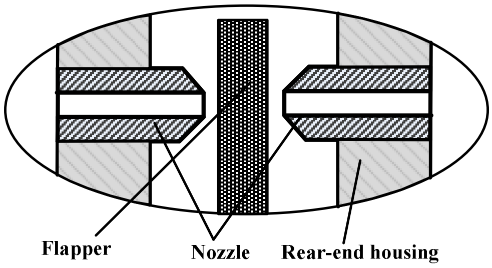


Fig. 2 Detail View of nozzle-flapper structure

AMESim modeling

AMESim model of GMA. Some assumptions are required before building the mathematical model.

- GMM works under non-resonance condition.
- The inductance of the driving coil remains constant.
- The magnetic force generated by the surface of GMA output rod is ignored.
- The influence of eddy current on the driving current is ignored.

According to the model assumptions above, the process of magnetic-mechanic coupling can be expressed by Fig.3.

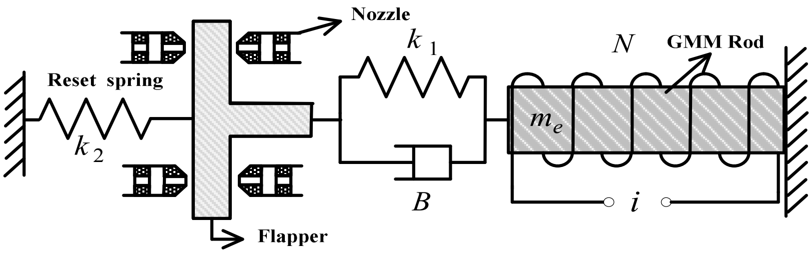


Fig.3 Sketch of GMA magnetic-mechanic coupling

When the coil was driven with sine unit step current i, the magnetic flux ϕ is defined as:

$$\phi = (Ni + xd_{33})/R_m \tag{1}$$

where, N is coil turns; x is displacement of flapper; d_{33} is piezomagnetic coefficient; R_m is the equivalent reluctance of GMM rod which ignored the reluctance of magnetizer and thermal compensation module. In that case, R_m can be calculated by:

$$R_m = l/(\mu_0 \mu_r A) \tag{2}$$

where, l is the length of GMM rod; μ_0 is the permeability of vacuum; μ_r is relative permeability; A is cross-sectional area of the GMM rod.

Considering the effects of eddy current, the magnetostrictive force generated by GMM rod can be defined as:

$$F = \phi k / d_{33} \tag{3}$$

where, k is eddy current coefficient, here $k=1$ in particular, which ignored the effects of eddy current.

The force balance equation of GMA is:

$$F = m_e \frac{d^2x}{dt^2} + B_a \frac{dx}{dt} + k_1 x + F_L \tag{4}$$

where, m_e is equivalent mass of GMM rod; k_1 is the stiffness coefficient for GMM rod, which comes from equation $k_1 = AE/l$, here E is the effective bulk modulus of GMM rod; B_a is damping coefficient; F_L is the hydraulic force exerted on the flapper.

According to the equation above, the AMESim model will be shown as Fig.4. In addition, super-component is used to simplify the overall model of the actuator, and the ports of the actuator's model are expressed numerically correspond to the ports in super-component.

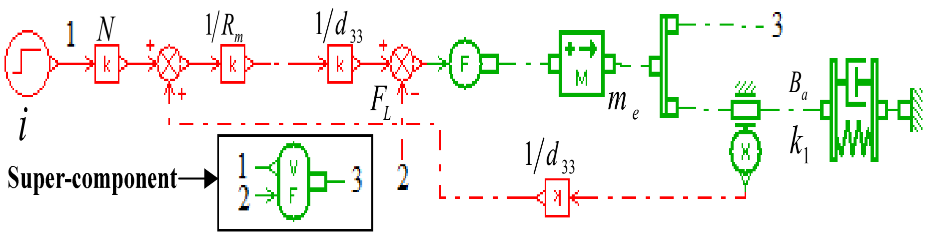


Fig .4 AMESim model of GMM Actuator

AMESim model of four-nozzle flapper valve. Fig.5 is a four-nozzle flapper valve of physical model diagram[6,7]. As the diagram shows, the main parameters of the nozzle flapper valve consist of the diameter of nozzle D_N, diameter of fixed orifice D_0, zero clearance x_{f0} etc. Here only illustrates the design principle of fixed orifice D_0.

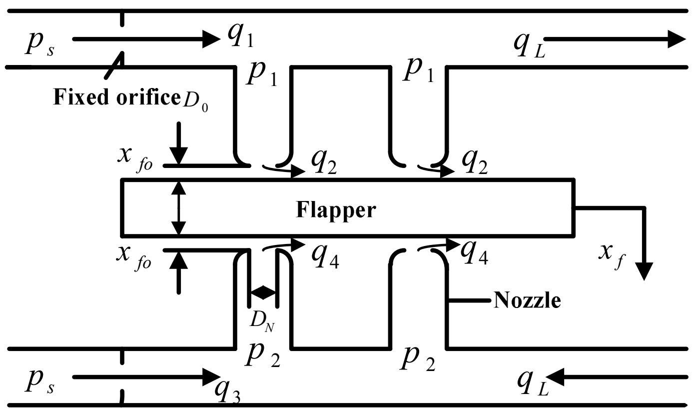


Fig .5 Sketch of four-nozzle flapper valve

According to the continuum equation of flow:

$$q_1 = 2q_2 + q_L \tag{5}$$

where q_L is the load flow. q_1, q_2 can be defined by Eqs. (8) and (9):

$$q_1 = C_{d0}\pi \frac{D_0^2}{4}\sqrt{\frac{2}{\rho}(p_s - p_1)} \tag{6}$$

$$q_2 = C_{df}\pi D_N\left(x_{f0} + x_f\right)\sqrt{\frac{2}{\rho}p_1} \tag{7}$$

where C_{df}, C_{d0} are flow coefficient of variable orifice and fixed orifice respectively. p_s is oil pressure, p_1 is outlet pressure of fixed orifice, x_f is flapper displacement.

In case of $p_1 = 0.5p_s$ and $q_L = x_f = 0$ we can further obtain:

$$2C_{df}\pi D_N x_{f0} = C_{d0}\pi\frac{D_0^2}{4} \tag{8}$$

From this, if $C_{df}/C_{d0} = 0.8$ and $x_{f0} = D_N/8$, then Eq. (10) can be simplified as:

$$D_0 = 0.9D_N \tag{9}$$

Eq.5~Eq.9 shows that the diameter of the fixed orifice varies with the diameter of nozzle, that is, there is a relationship of some kind under certain parameters.

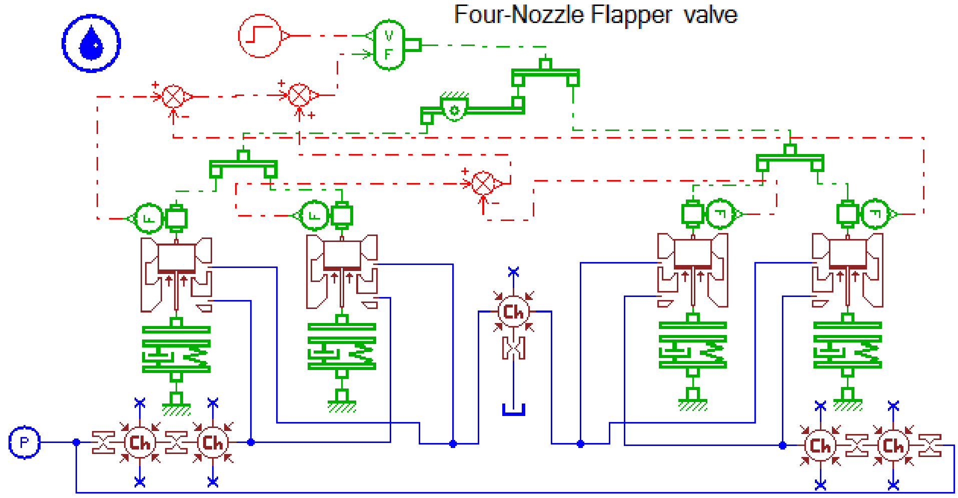


Fig. 6 Simulation model of four-nozzle flapper valve with GMA

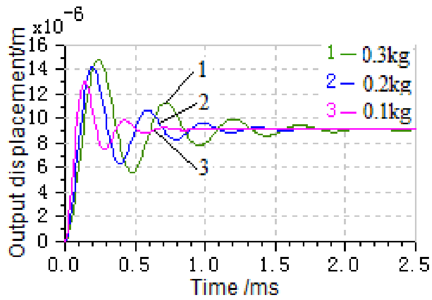
Fig.6 shows the AMESim model of four-nozzle flapper valve with GMA under unit step signal according to the designed parameters which are listed in the Table 1.

Table 1 Main structure parameters of four-nozzle flapper valve with GMA

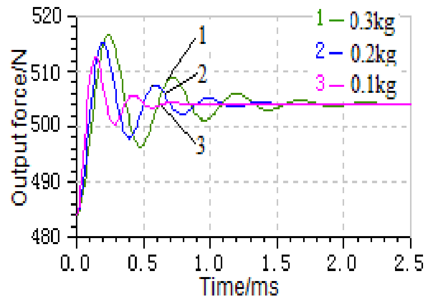
μ_0	Permeability of vacuum (H/m)	$4\pi\times10^{-7}$	k_1	Stiffness coefficient (N/m)	5.5×10^{7}
μ_r	Relative permeability	10	B_a	Damping coefficient (Ns/m)	1200
N	Turns of driving coil (za)	2200	P_s	Oil pressure (bar)	100
d_{33}	Coupling coefficient (m/A)	1×10^{-7}	D_N	Diameter of nozzle(mm)	0.4
m_e	Equivalent mass (kg)	0.2	x_{f0}	Zero clearance (mm)	0.04
R_m	Magnetic resistance (H)	4.79×10^{-7}	D_N	Diameter of Fixed orifice (mm)	0.36

Simulation analysis

According to the overall simulation model and given parameters, the step response characterization of GMA has been done for output force and output displacement shown by Fig.7 and Fig.8. By comparing Fig.7(a) with Fig.7(b), Fig.8(a) with Fig.8(b), the oscillation amplitude and the steady-state adjustment time of GMA decrease while the damping coefficient increase, which can improve its dynamic response significantly. On the contrary, the dynamic performance will become deteriorate while equivalent mass increases.

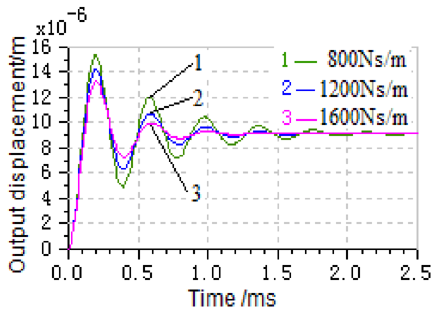


(a)

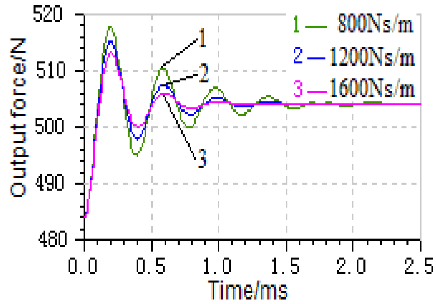


(b)

Fig.7 Step response of output displacement (a) and output force (b) with different equivalent mass



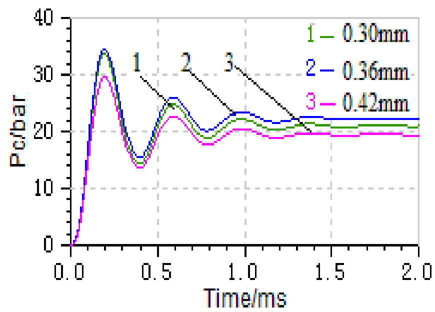
(a)



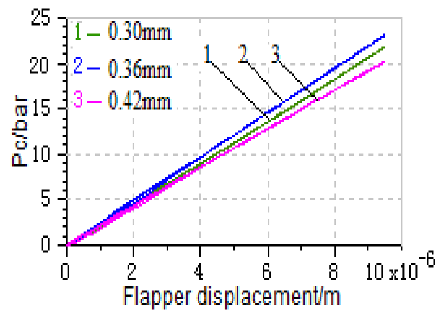
(b)

Fig.8 Step response of output displacement (a) and output force (b) with different damping coefficient

In order to research the characterize of four-nozzle flapper valve with GMA in case of different structural parameters, the dynamic and static of control pressure have been made and shown in Fig.9

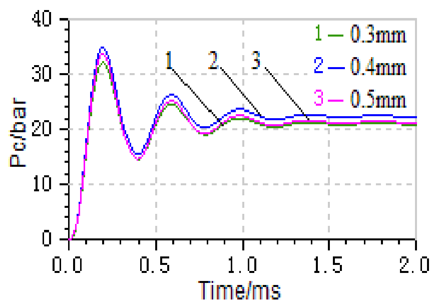


(a)

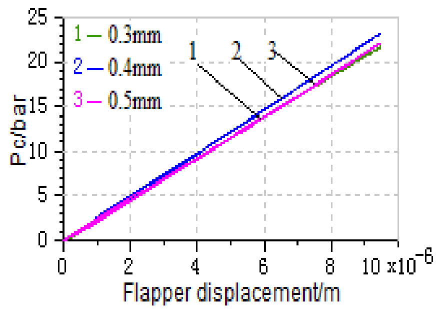


(b)

Fig. 9 Dynamic (a) and static (b) performance of control pressure with different fixed orifice

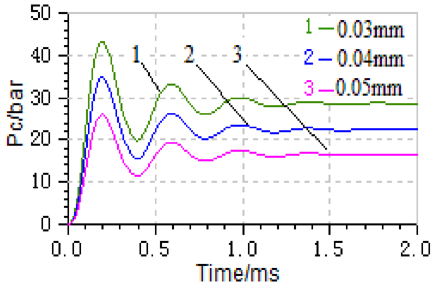


(a)

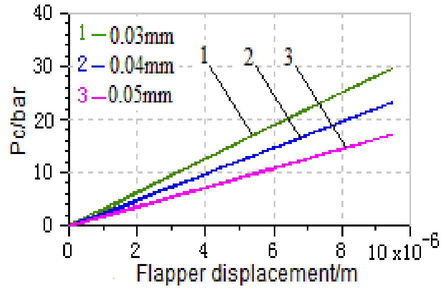


(b)

Fig. 10 Dynamic (a) and static (b) performance of control pressure with different nozzle orifice



(a)



(b)

Fig. 11 Dynamic (a) and static (b) performance of control pressure with different zero clearance

(a) and (b), Fig.10(a) and (b), Fig.11(a) and (b), respectively. Based on the graphs above, when the diameter of the fixed orifice and nozzle varies in a small range, the control pressure changes in a relatively slight manner. However, a proper size of fixed orifice and the nozzle can make the control pressure achieve a larger value. The zero clearance should be as small as possible, which demands higher level of oil filtration and assembling process simultaneously. The simulation results have shown that optimal control pressure characteristics could be gained by appropriately setting the parameters of zero clearance, nozzle and fixed orifice's diameter.

Conclusion

This paper has presented a novel nozzle flapper valve based on giant magnetostrictive material. Both the dynamics of the actuator and the valve are simulated by AMESim software. The simulation results have shown that the optimal control pressure characteristics could be gained by appropriately setting the parameters of zero clearance, nozzle and fixed orifice's diameter, which lays a foundation of further optimization and design of the four-nozzle flapper valve.

Acknowledgements

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