International Conference on Advances in Computational Modeling and Simulation

The CFD analysis of main valve flow field and structural optimization for double-nozzle flapper servo valve

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Abstract

This paper uses INVENTER software to establish the model of main valve and the servo valve is simulated by FLUENT software. Then, the flow field model of the main valve spool on certain position is established and the possible effects on the hydraulic valve are analyzed due to the shape of the flow field. Finally, we make some improvement of the main valve core, and give some conclusions by comparing the flow field characteristics of main valve with the unimproved valve.

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Keywords: double-nozzle flapper servo valve, fluent software, analysis of flow field, valve spool improvement

1. Introduction

Electro-hydraulic servo valve is electro-hydraulic converter component. Meantime it is also a power amplifier component. As a bridge connecting electrical parts and hydraulic parts, electro-hydraulic servo valve is the core component of the servo control system. There are many representative products of electro-hydraulic servo valve, such as jet-pipe electro-hydraulic servo valve, moving coil rotary piston servo valve and so on. The double-nozzle flapper servo valve has an advantage of small volume, high power magnification and rapid response speed. So, in electro-hydraulic servo system of metallurgical industry it has been used widely [1]. Electro-hydraulic servo valve is carried out by manipulating fluid flow. The drop of valve pressure also dues to energy loss when fluid flow goes through the valve channel. Therefore, the state of fluid flow in the valve becomes the most important factor in determining the

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operational aspects of the valve.

This paper uses Fluent software to analyze the flow field for double-nozzle flapper servo valve on the main valve and uses the visual graphics simulation results to analyze characteristics of electro-hydraulic servo valve. Through the flow field analysis, the structure is proposed for the improvement of main spool.

1.1. Double-nozzle flapper valve structure and working principle

The structure of double-nozzle flapper servo valve was shown in figure 1. As can be seen from figure 1, servo valve is made up of move iron type torque motor, double-nozzle flapper valve and guide valve. Its working principle is: the armature in the main valve is in the middle position between up and down magnetizer supported by bourdon tube if there are no current flows. The flapper is in the middle position between these two nozzles. Main valve spool is in neutral position under the constraints of ball in the lever. At this time there is no output pressure from main valve. When there are differentiating control current flows, it will produce counterclockwise electromagnetic torque on the main valve armature, which makes bit baffle components deflect anticlockwise around the center of spring to move the main valve spool. When the fluid pressure on the spool, the reacting force caused by the deformation of feedback rod and the hydrodynamic force are equals, the spool stops motion. When the load pressure difference is constant, the output flow from double-nozzle flapper servo valve and control current is in proportion [2].

![Double-nozzle flapper force feedback electric-hydraulic servo valve schematic diagram](image)

1—permanent magnet; 2—bottom guide magnet; 3—armature; 4—coil; 5—bourdon tube; 6—top guide magnet; 7—nozzle; 8—main spool valve; 9—fixed orifice

Figure 1 Double-nozzle flapper force feedback electric-hydraulic servo valve schematic diagram

1.2. The flow field analysis of main valve

1.2.1. The flow field modeling of main valve

According to the structure of double-nozzle flapper valve, we use INVENTER software to establish the three-dimensional model of main valve, which is shown in figure 2. In the picture the model is in the situation of fully open valve.
1.2.2. Flow assumed condition

There are many influence factors in the actual environment, which is difficult to simulate the real work state. Now assumptions are as follows:

(1) The hydraulic oil compressibility is very small. When add one project atmospheric pressure, its volume is reduced less than a millionth. So in the process of computation we can ignore the influence of compressibility for hydraulic oil.

(2) Because the Reynolds number is $Re = \frac{V(D-d)}{\nu} = 3000$, which far outweigh the gap critical Reynolds number $Re_c = 1000$. So in the valve flow state belongs to turbulence and k-epsilon model turbulence equation is used to solve the calculation.

(3) The flow of fluid is single-phase flow. There is no heat transfer fluid phenomenon in the system.

1.2.3. boundary conditions of flow field

After the grids, it is also important to define the type of grid model boundary. It will produce very different results to set up different boundary type, or even inconsistent with the actual situation. According to the practical work situation, we set inflow port boundary types as velocity-inlet and orifice boundary types as pressure-outlet. For other faces which are not set, the default set is fixed wall. [3]

1.2.4. The flow field calculation mesh division

Servo valve flow field is in GAMBIT meshing process for the production of the grid, using its powerful ability to automatically generate the mesh. Then according to structural characteristics of flow in complex flow field, the mesh is refined. Ultimately we achieve the purpose of precise and rapid solution. According to established geometric model, we set that baffle displacement is $0 \text{ mm}$ and inlet boundary pressure is $21 \text{ MPa}$, outlet boundary pressure is atmospheric pressure. In the meshing process, the initial set is tetrahedron TGrid grid, all of the static meshes, and Equal interval is 0.8 in order to use the least number of nodes to get as much grid as possible. Finally we obtain the grids shown in figure 3.
In several iterations, calculation results tend to converge and its simulation accuracy is close to the theoretical value of the CFD model. Figure 4 is the main valve static pressure cloud diagram. Figure 5 is the main valve turbulent energy dissipation cloud diagram. Figure 8 is the main valve turbulent energy dissipation cloud diagram. This paper uses $x = 0, y = 0$ axisymmetric surface observation to illustrate which is shown in figure 6 and 7.

From figure 4, figure 6 a) and figure 7 a) we can learn that when the fluid goes into the servo valve, the pressure is the highest. Then the pressure has a decrease when the fluid goes into left cavity of main valve.
Meanwhile there is pressure fluctuation. Energy dissipation is also apparent in the valve opening. Fluid will produce vortex when the direction of fluid changes in the channel. But the dissipation energy of these vortexes is much smaller than the vortex at valve port. Therefore, when the valve port opens, the flow will have a significant impact on the spool and valve.

2. Improvement of main spool

There is steady-state fluid force caused by fluid movement, when servo valve is opened, which has a great impact on the spool and valve. According to optimal design method of Electro-hydraulic servo valve, in order to smooth the flow field flow lines and reduce the occurrence of excessive pressure drop and cavitation. It has been found that we can use some special structure in servo valve like ball, parabolic, chamfering to improve the performance of the valve and extend its life, which are very effective measures. The ends of the spool are improved that is shown in figure 8. The flow field model was established. In the same condition these two different situations are compared and then flow movement improved is got.

This situation is established when the valve port opens. Parameters are same with the above set. Then we can get the simulation results. Figure 9 a) and figure 10 a) are pressure cloud pictures of main valve improved. Figure 9 b) and figure 10 b) are turbulent energy dissipation map of main valve improved.

Figure 8 Fluid motion model for the improved structure of valve chamber

Figure 9 Improved pressure diagram

Figure 10 Improved turbulent energy dissipation diagram

X = 0 plane
Pressure distribution and turbulent energy dissipation in the valve are analyzed and compared based on flow field simulation. From Figure 4 and figure 9, we can learn that valve static pressure minimum value is larger after structure optimization. If the pressure is smaller, it is prone to cavitation. So the optimized structure is beneficial to suppress cavitation. Because minimum static pressure is in the middle chamber, the cavitation will be there. Since the cavitation has relationship with the pressure difference at valve port. The small pressure difference helps to reduce cavitation. Middle cavity structure helps to reduce cavitation. Through the comparison of Figure 11 b) and figure 6 b), we can learn that energy loss is reduced approximately 10% after the improvement of valve chamber, which has some guidance on the analysis of energy loss and valve chamber improvement.

3. Conclusion

In this paper, the internal flow field of servo valve was simulated by fluent software. Then, through the comparison of improvement and not improvement cloud map we can learn that:

(1) When flows go through the valve port, pressure at the inlet is maximal. When flows go out the valve port, the pressure reduces further along the direction of flow, while the pressure is minimal at the outlet. It is prone to produce cavitation at sharp edge of the valve port. If we adopt curve or chamfer it is easy to reduce cavitation.

(2) In the opening it still has a whirlpool. But the diameter of whirlpool is larger, which means energy dissipation of servo valve is smaller and the energy of flow field losses small. The insufficient place is
that the improved body cavity surface is curved surface, which will have an impact on the surface processing quality. Moreover this will affect the performance of its sealing.

References