Influence of Valve Core and Seat Structure of Overflow Valve on Flow Field Performance and Optimization Design

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Abstract. This article aims at the problem that the difference in the flow field performance of the valve cavity after the overflow valve is installed with the eddy current sensor. The structural optimization of the valve core and seat and a research method to improve the flow field performance of the valve cavity is proposed by analyzing the influence of the structure of the built-in eddy current sensor probe on the flow field performance of the valve cavity. First, the flow field analysis model is established, to analyze the flow field of the built-in probe structure. Then, the influence of the valve core micro modeling, valve seat flow path diameter, gradient line chamfer depth on the flow field performance of the valve cavity is analyzed. The results show that the structure of the valve core and the valve seat have been optimized and the flow field performance of the pilot valve cavity of the overflow valve can be improved to a certain extent.

1. Introduction
Pilot operated overflow valve is an important control element of hydraulic system. Its main role is to control the pressure of the entire hydraulic system to ensure that the hydraulic system is reliable and stable. The displacement and vibration signal of the overflow valve can be detected by adding a built-in eddy current sensor inside the overflow valve. However, the flow field of the valve cavity with the built-in probe of the overflow valve is easy to produce some problems, such as vibration, noise, tightening and so on [1-2]. This paper takes the pilot operated overflow valve as the research object [3-4]. On the basis of designing the online detection device of valve core displacement signal [5], first, the flow field performance of the valve cavity with the built-in probe structure is analyzed and discussed, then, the optimization design is made from two aspects of valve core micro-modeling, valve seat flow path diameter [6] and gradient line chamfer depth.

2. Detection system principle and valve cavity flow field analysis
This paper takes the YF-type overflow valve (pilot valve) online detection device as an example, as shown in Figure 1. This device consists of a pilot valve of overflow valve, a eddy current sensor and so on. The detection device can use the eddy current sensor to realize on-line detection of the valve core state. When the valve core appears tremor, jamming and other faults, it can realize quick troubleshooting and status online detection through the analysis and judgment of the detection signal.
According to the characteristics of pilot valve cavity internal flow field of overflow valve built-in probe structure, a simplified flow field analysis model is established by using the ANSYS draft module [7]. The static pressure distribution map and the velocity flow line diagram of the flow field can be obtained by the analysis and calculation of the Fluent flow field, as shown in Figure 2.

It can be seen from the figure that a large pressure difference is formed at the throttling port between the valve core and the valve seat, and a small negative pressure area appears in a position close to the valve seat gradual line on the upper side of the throttle. Negative pressure will produce cavitation, and then the occurrence cavitation damage, and even noise and vibration, seriously affecting the life of the overflow valve and stability. In order to solve these problems, this paper optimizes the structure of the valve core and the valve seat to improve the valve cavity flow field performance.

### 3. The effect of valve core micro-molding structure

When the overflow valve is working, the decrease in viscosity of the oil will cause dry friction between the valve core and the seat. This paper presents a solution to the performance optimization using micro-modeling technology. In view of the technical characteristics of micro-molding, consider the addition of lubricating belt to the original cone valve model, the number of which is m, the longitudinal section of which is 1mm square, the lubricating groove is perpendicular to the busbar, and they are evenly distributed on the cone side surface, as shown in Figure 3.
Lubricating belt number were 4, 8 and 12 when the local static pressure distribution diagram, as shown in Figure 4. The minimum static pressure of the throttle is 9.6KPa, 14.2KPa and 15.8KPa, and they all appear on the lower side of the valve core. Therefore, the application of the lubrication belt micro modeling technology can greatly improve the minimum static pressure value at the throttle, and effectively avoid the emergence of the negative pressure area of the upper side of the valve core.

![Figure 4. Local static pressure nephogram of different m.](image)

Figure 4. Local static pressure nephogram of different m.

Figure 5 is the static pressure streamlined diagram on both sides of the valve core at different m. The static pressure of lubricating belt model on the upper side of the valve core is higher than that on the bottom side of the valve core. The maximum pressure drop before and after the throttle orifice at different m is 22.9MPa, 24.2MPa and 26.0MPa respectively. The maximum pressure drop on the upper and lower sides of the valve plug is -5.2MPa, -3.7MPa and -2.1MPa respectively. The maximum pressure drop rates (located at the lower side of the valve core) is 4.6 MPa/mm, 2.2 MPa/mm and 1.4 MPa/mm. Therefore, the more the number of lubrication belts, the smaller the possibility of producing cavitation in the flow field. Among them, the flow field performance comparison of different m models in Table 1.

| Table 1. Comparison of model flow field performance at different m. |
|---------------------------------------------------------------|
| Number of lubricating belts | 4   | 8   | 12  |
| Minimum static pressure(KPa) | 9.6 | 14.2| 15.8|
| Maximum flow rate(m/s) | 217 | 232 | 227 |
| Throttle static pressure difference(MPa) | 22.9 | 24.2| 26.0|
| The maximum pressure drop rate of change(MPa/mm) | 4.6 | 2.2 | 1.4 |
| The maximum pressure on both sides(MPa) | -5.2 | -3.7 | -2.1 |

![Figure 5. Static pressure curves on both sides of the valve core of different m.](image)

(a) 4 lubricating belts  (b) 8 lubricating belts  (c) 12 lubricating belts

Figure 5. Static pressure curves on both sides of the valve core of different m.
It can be known that the more the number of lubrication belts, the greater the static pressure of the cone core surface is, the more obvious the lubrication effect. But the too dense lubrication belts will weaken the mechanical properties of the valve core.

4. Effect of valve seat structure on flow field performance
Combined with the structural features of the valve cavity, this section studies the influence of valve seat flow path diameter D and gradient line chamfer depth d on flow field performance. Among them, the angle between the valve core busbar and the valve seat gradient line is constant at 6°, as shown in Figure 6.

![Figure 6. Valve seat optimization model diagram.](image)

4.1. Effect of valve seat flow path structure
Figure 7 shows a partial pressure diagram of the choke orifice with respective valve seat flow path diameter D of 2 mm, 3 mm and 4 mm. Among them, the negative pressure area of the throttle negative pressure maximum value is -4.3KPa, -16.3KPa and -12.5KPa, which shows that increasing and decreasing the flow path diameter can help inhibit the negative pressure, and reduce the flow path diameter effect is more prominent.

![Figure 7. Throttle local pressure nephogram of different D.](image)

Figure 8 is the static pressure distribution curve of different flow path diameters on the upper and lower sides of the valve core. It can be seen from the figure, the maximum static pressure difference before and after the throttle is 17.8MPa, 16.4MPa and 14.6MPa, the upper and lower valve core on the maximum static pressure is 9.1MPa, 8.7MPa and -8.5MPa. Therefore, when the flow path diameter increases, the racial imbalance force gradually decreases, but when the diameter increases to a certain extent, the racial imbalance force will increase in the opposite direction. Table 2 shows the comparison of the flow field performance of the valve cavity when the flow path diameter is different.
Table 2. Model comparison of flow field performance of different D.

| Flow path diameter | 2mm | 3mm | 4mm |
|--------------------|-----|-----|-----|
| Maximum negative pressure (KPa) | -4.3 | -16.7 | -12.5 |
| Maximum flow rate (m/s) | 252 | 248 | 254 |
| Throttle static pressure difference (MPa) | 17.8 | 16.4 | 14.6 |
| The maximum pressure drop rate of change (MPa/mm) | 4.2 | 3.8 | 8.4 |
| The maximum pressure on both sides (MPa) | 9.1 | 5.3 | -8.5 |

Figure 8. On both sides of the valve core static pressure distribution curve of different D

4.2. Effect of valve seat gradient line chamfer structure

Figure 9 is a nephogram that represents the local pressure of the throttle of valve seat gradient line chamfer depth d for 0.2mm, 0.8mm, 1.4m and 2.0mm. It can be seen from the figure, the maximum negative pressure value in the negative pressure area of the orifice is -26.8KPa, -14.2KPa, -15.0KPa and 0.8KPa. Increasing d can effectively suppress the generation of holes.

Figure 9. Local pressure nephogram of the throttle of different d.

Figure 10 is a flow chart of the valve cavity at different d, it can be seen from the figure, with the increase of d, the maximum orifice velocity is 304 m/s, 265 m/s, 254 m/s and 231 m/s. Therefore, increasing d can reduce the maximum flow velocity of the flow field.
Figure 10. Valve cavity flow field flow line diagram of different d.

Figure 11 is a static pressure curve distribution diagram of upper and lower sides of the valve core of different d. The figure shows that the maximum static pressure before and after the throttle orifice is 14.5MPa, 14.3MPa, 14.7MPa and 14.0MPa, indicating that the size of d before and after the throttle orifice pressure difference is not significant; second, the maximum hydrostatic pressure difference between the two sides of the valve core is 12.8MPa, 7.6MPa, 5.3MPa and 4.1MPa respectively, that is, increasing d helps to eliminate the radial imbalance force; in addition, the maximum rate of pressure drop changes is 8.6MPa/mm, 7.0MPa/mm, 4.8MPa/mm and 2.4MPa/mm, which shows that increasing the d can effectively inhibit cavitation. Table 3 is a comparison of valve cavity flow field performance of different d.

Table 3. Flow field performance comparison of model with different d.

| Valve seat chamfer depth(mm) | 0.2  | 0.8  | 1.4  | 2.0  |
|-----------------------------|------|------|------|------|
| Minimum static pressure(KPa)| -26.8| -14.2| -15.0| 0.8  |
| Maximum flow rate(m/s)     | 304  | 265  | 254  | 231  |
| Throttle static pressure difference(MPa) | 14.5 | 14.7 | 14.3 | 14.0 |
| The maximum pressure drop rate of change(MPa/mm) | 8.6  | 7.0  | 4.8  | 2.4  |
| The maximum pressure on both sides(MPa) | 12.8 | 7.6  | 5.3  | 4.1  |
Figure 11. Static pressure curve diagram of the valve core on upper and lower sides of the valve core of different d.

5. Conclusion
Through analyzing the valve cavity flow field of the built-in eddy current sensor probe structure, the valve cavity flow field of the built-in probe structure of the overflow valve is prone to vibration, noise and jamming. In order to improve the performance of the flow field, a method of optimizing the structure of the valve core and the valve seat is proposed. The results show that the use of micro modeling technology can effectively avoid the negative pressure on the upper side of the valve core and reduce the possibility of cavitation. Valve seat flow path diameter D and gradient line chamfer depth d have great influence on the flow field performance of valve cavity, and the effect is varied. That is to improve the performance of a certain index and decrease the performance of another index. Therefore, the performance of each index should be considered comprehensively when the valve seat is optimized. In conclusion, through the micro-modeling of valve core and the optimization of valve seat flow path diameter D and gradient line chamfer depth d, we can solve the problems of noise, vibration and clamping in the valve chamber, enhance the flow field performance of the valve cavity, and ensure the working stability of the overflow valve.

Acknowledgments
First author’s name. Yang Xu (1993-), Male, Master Degree Candidate. Corresponding author’s name. Feng Yong-bao (1971-).

References
[1] Zhang Tianxiao, Liu Xinhui, Zhang Nong. Vibration analysis of hydraulic pressure overflow valve [J]. Journal of Jilin University, 2014, 44 (1): 91-94.
[2] Ma Wei, Ma Fei, Zhou Zhihong, et al. Instability analysis and experimental study of hydraulic pressure overflow valve [J]. Journal of Engineering Science, 2016, 38 (1): 135-142.
[3] Jadwisienzczak W M, Wang J, Tanaka H, et al. Optical and magnetic properties of GaNepilayers implanted with ytterbium [J]. Journal of Rare Earths, 2010, 28 (6): 931-935.
[4] Si G L, Yang F Y, Wang W J, et al. Study on relief valve with permanent magnetic spring [J]. Advanced Materials Research, 2011, 328/329/330: 224-227.
[5] Li Y S, Lian H Y, Li Y F. Research and development of eddy current testing system suitable for on-line detection [J]. Process Automation Instrumentation, 2008, 29 (7):60-62.
[6] Ji Hong, Fu Xin, Yang Huayong. Influence of inner flow channel shape on cavitation noise of overflow valve [J]. Chinese Journal of Mechanical Engineering, 2002, 38 (8): 19-22.
[7] Zhu Hongjun. Fluent15.0 flow field actual guide [M]. Beijing: People's Posts and Telecommunications Press, 2015: 108-138.