Effect of cone throttle valve pressure on cavitation noise

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Abstract: The hydraulic cone throttle valve is prone to cavitation and cavitation noise. To address this problem, the authors studied the influence of cone throttle valve pressure and flow rate on cavitation noise. First, they used fluid–solid coupling principles to conduct a finite volume simulation of the cone throttle valve and obtain the cavitation distribution in its flow channel under different pressures. Then, they built an experimental apparatus to perform visualisation measurements on the cone throttle valve. The authors used the valve core opening to adjust the experimental pressure, used a high-speed camera to shoot the morphological changes in cavitation under different pressures, and measured the valve noise. The cavitation of the cone throttle valve was affected by the pressure. The larger the pressure difference between the front and back of the valve was, the more obvious the cavitation was and the stronger the cavitation noise was. The cavitation noise can be suppressed by reducing the pressure difference between the front and back of the valve.

1 Introduction

Cavitation is the self-vapourisation of liquid when the partial pressure of the liquid is lower than the saturated vapour pressure at that temperature. Cavitation bubble collapse can cause locally high temperature, high pressure and noise. The hydraulic valve is affected by the system structure and is often accompanied by cavitation in the working process. The collapse of a cavitation bubble generates a great amount of noise in the hydraulic valve owing to its high acoustic energy and high radiation efficiency [1]. Cavitation noise is an important factor affecting the performance of the cone throttle valve. Therefore, it is important to reduce cavitation noise to improve the performance of hydraulic valves. Consequently, we researched the factors affecting cavitation noise in a cone throttle valve.

An et al. [2] studied the cavitation and noise of the orifice of a double launcher tube throttling damper, and determined that air cavitation is an important factor in damper noise. Yang et al. [3] concluded that back pressure had a great influence on the flow characteristics of micro-damping holes. Kim et al. [4] numerically simulated the effect of viscosity on cavitation and noise. Chen et al. [5] found that setting the pressure pulsating frequency to a value different from the natural frequency effectively suppressed noise in a nozzle flapper valve. Liu [6] analysed the mechanism of servo valve howling. Zhou [7] studied cavitation effects in a partially guided jet valve. Wang [8] found the relationship between cavitation shape and noise frequency in a cone throttle valve. Shi et al. [9] experimented with suppressing the cavitation of throttle valves in water hydraulics. Banas et al. [10] studied a cone throttle valve by means of visualisation. In studying a direct-acting hydraulic relief valve, Yang et al. [11] found that the arrow-shaped compensation structure, an appropriate valve core cone angle, and the secondary valve port can effectively improve the valve performance. Zhang [12] studied cavitation in a nozzle servo valve prestage. Zheng and Quan [13] applied computational fluid dynamics (CFD) to a cone valve. Fenghua et al. [14] experimentally studied cavitation noise. Zhu et al. [15] simulated cavitation at the port of a cone valve. Using a high-speed camera to study the cavitation of a cone valve, Washio et al. [16] provided a good theoretical reference for designing such a valve. Luo et al. [17] contributed to cavitation theory by studying the collapse of two cavitation bubbles with a high-speed camera. Experiments by Takosoglu et al. [18] on a servo valve provided a good experimental reference. Yan et al. [19] provided good theoretical guidance for designing a servo valve by studying its nozzle jet mechanism. By theoretically analysing the prestage flow field of a jet deflector servo valve, Yan et al. [20] provided a reference for the pre-stage design of such a valve. Shuai et al. [21] obtained useful reference values for a high-flow high-frequency electro-hydraulic servo valve. Aung et al. [22, 23] used numerical simulation to study the cavitation phenomenon of the front stage of the nozzle-flapper servo valve, and supplemented the cavitation theory of the servo valve. Qianpeng et al. [24] used a variety of simulation software to optimise the design of a slide valve. Shen et al. [25, 26] applied valves in hydraulic system control. Jiang et al. [27] studied cavitation luminescence of a conical throttle valve under a pressure difference. Zhang et al. [28–30] studied the transient performance of a pump.

In summary, the current research on hydraulic valves and cavitation mainly includes numerical simulation and experiment. There is some research on cavitation and noise of valve elements including cone valves, but the mechanism of cavitation noise of hydraulic valves is complicated. The cavitation noise of cone throttle valves needs to be further studied. As the fluid flow in the hydraulic valve is a typical multi-physical coupling problem, it is difficult to obtain the analytical solution of the flow. The finite volume method can be used to simulate the flow in the valve runner. However, because the finite volume method needs to simplify the model to simulate the fluid flow in the valve runner, it cannot fully reflect the true flow state of the fluid in the runner. The development of high-speed camera technology makes it possible to visualise the hydraulic valve. Through a visualisation experiment, the actual flow field can be directly observed and the accuracy of the finite volume simulation can be verified. The finite volume simulation and visualisation experiment can support each other and provide a reliable method for researching hydraulic valves. We study the relationship between the flow field of the cone throttle valve and cavitation noise by combining numerical simulation with experiment, and explore the main factors affecting cavitation noise to provide theoretical and experimental reference for optimising the design of such valves.
2 Numerical simulation

2.1 Simulation method

The numerical simulation in this study adopted fluid–solid coupling principles because the flow field of the cone throttle valve involves such coupling. Commercial CFD software has become highly developed, providing a very effective platform for simulating fluid–structure coupling. The commonly used simulation methods are the finite element and finite volume methods. Fluid–solid coupling simulation is based on the finite volume method. This study used the non-linear finite element software ADINA, which achieves more accurate simulations because it has the most developed fluid–solid coupling computation. The structural characteristics of the cone throttle valve lead to a large mutation in the flow area of the valve port. Therefore, hydraulic oil flow at the valve port easily forms turbulence. The simulation of the cone throttle valve applied a turbulence model to ensure accuracy.

The structure of the cone throttle valve is shown in Fig. 1. The throttle valve structure was imported into ADINA to establish a solid model, and the fluid model in the throttle channel was established according to the solid structure, as shown in Fig. 2. The fluid model had a one-to-one correspondence to the solid model. A tetrahedral grid was adopted. As the valve is axially symmetric, only half of the model was simulated to save computing time. Fluid–solid coupling boundary conditions were set for the interface between the fluid and the solid, and a gas–liquid phase algorithm was used to simulate cavitation of the flow field. The structural and fluid models were built when the fluid volume simulation model was built. The mesh densities of the solid and fluid models were basically the same. The $k$–$\varepsilon$ turbulence model was used for the fluid because the large changes in the flow of hydraulic oil through the throttle hole of the valve lead to turbulence.

2.2 Governing equations

2.2.1 Cavitation model: The volume fractions of vapour and liquid, $f_v$ and $f_l$, are defined as

$$ f_v = \frac{V_v}{V_v + V_l}, \quad f_l = 1 - f_v, $$

where $V_v$ is the vapour volume and $V_l$ is the liquid volume. We then assume the density of the mixture of vapour and liquid is

$$ \rho_m = f_f \rho_f + f_l \rho_l, $$

where $\rho_f$ is the vapour density and $\rho_l$ is the liquid density.

The continuity equation of mixed fluids is

$$ \frac{\partial \rho_m}{\partial t} + \nabla \cdot (\rho_m \mathbf{v}) = 0, $$

where $t$ is the elapsed time and $\mathbf{v}$ is the velocity vector.

2.2.2 Turbulence model: As the hydraulic oil flows through the throttle valve orifice, there is a sudden narrowing of the flow passage. The oil flow rate therefore increases sharply to form a turbulent flow. We assume that the oil is incompressible, and hence the standard $k$–$\varepsilon$ model equations become

$$ \frac{\partial (\rho_k)}{\partial t} + \text{div}(\rho_k \mathbf{v}) = \frac{\partial}{\partial x_i} \left[ \frac{\mu}{\sigma_k} \frac{\partial k}{\partial x_i} \right] + 2 \rho_S \mathbf{S}_{ij} \cdot \mathbf{S}_{ij} - \rho_k \varepsilon, $$

$$ \frac{\partial (\rho \varepsilon)}{\partial t} + \text{div}(\rho \varepsilon \mathbf{v}) = \frac{\partial}{\partial x_i} \left[ \frac{\rho \mu_v}{\sigma_\varepsilon} \frac{\partial \varepsilon}{\partial x_i} \right] + C_{1\varepsilon} \rho \varepsilon \frac{\partial k}{\partial x_i} \cdot \frac{\partial k}{\partial x_i} - C_{2\varepsilon} \rho \varepsilon^2 $$

$$ \mu_t = \rho C_f \frac{k^2}{\varepsilon}, $$

$$ k = 1.5 \times ((0.01 \sim 0.1) \nu_{\infty})^2, $$

$$ \varepsilon = k \nu^{1/3}/(0.3L), $$

where $k$ is the turbulent kinetic energy, $\varepsilon$ is the turbulent dissipation rate, $\mu_t$ is the turbulent viscosity, and $S$ is the mean strain rate tensor. The coefficients $C_{1\varepsilon}$, $\sigma_k$, $\sigma_\varepsilon$, $C_{1\varepsilon}$, and $C_{2\varepsilon}$ are adjusting constants; their values are $C_{1\varepsilon} = 0.09$, $\sigma_k = 1$, $\sigma_\varepsilon = 1.3$, $C_{1\varepsilon} = 1.44$, and $C_{2\varepsilon} = 1.92$.

The transport equation of the liquid in terms of volume fraction is

$$ \frac{\partial f_l}{\partial t} + \mathbf{v} \cdot \nabla (f_l \mathbf{v}) = - \frac{1}{\rho_l} \frac{\partial}{\partial t} \left[ (f_l \rho_l) \mathbf{v} \right], $$

where $m$ is the mass conversion rate between vapour and liquid and is given by

$$ m = \begin{cases} C_{\text{dest}} \rho_l \frac{3 f_{\text{dest}}}{2} \left( \frac{2 p - p_c}{p_l} \right)^{3} & \text{if } p \leq p_c, \\ -C_{\text{prod}} \rho_l \frac{3 f}{} \left( \frac{2 p - p_n}{p_l} \right)^{3} & \text{if } p > p_n, \end{cases} $$

where $p$ denotes the liquid pressure, $R$ is the cavitation bubble radius, $f_{\text{dest}}$ is the gas nuclei volume fraction, $C_{\text{dest}}$ is the coefficient of vapourisation of the liquid, and $C_{\text{prod}}$ is the coefficient of gas condensation; $C_{\text{dest}}$ and $C_{\text{prod}}$ take the values 50 and 0.01, respectively.

For cavitation flow, the compressibility of the mixture near the cavitation interface needs to be considered in the turbulence model. The density of mixed fluids is corrected to

$$ \rho_m = \rho_l - \left( \frac{\rho_c - \rho_{\text{m}}}{\rho_l - \rho_{\text{m}}} \right)^{n} (\rho_l - \rho_f), $$

where the constant exponent $n$ needs to satisfy $n \geq 1$; for large cavitation number, $n$ is set to 1. The value of $n$ increases with decreasing cavitation number and increasing Reynolds number.

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where the constant exponent $n$ needs to satisfy $n \geq 1$; for large cavitation number, $n$ is set to 1. The value of $n$ increases with decreasing cavitation number and increasing Reynolds number.
2.3 Analysis of simulation results

In the simulation of this study, the inlet pressure of the valve was 4 MPa; the outlet pressures were 0, 0.2, 0.4, 0.6, 0.8 and 1 MPa; the flow rate was set as 5 m/s; and the valve core opening was set as 8 mm. The fluid medium selected was No. 32 hydraulic oil, whose parameters are shown in Table 1. The simulation did not consider the effect of temperature.

Figs. 3–8 show the pressure distributions for an inlet pressure of 4 MPa and outlet back pressures of 0, 0.2, 0.4, 0.6, 0.8 and 1 MPa. We see that the flow channel pressure is low mainly around the valve core, and the greater the pressure difference between the inlet and outlet of the valve, the lower the local pressure at the valve core. That is, increasing the back pressure reduces the degree to which the throttle valve lowers the local pressure. This is due to increased back pressure, which reduces the pressure differential in the valve channel. The development of cavitation is affected by the pressure; the lower the local pressure in the valve runner, the more easily cavitation is produced. Cavitation bubbles are more fully developed owing to the low local pressure and large pressure difference, resulting in slow pressure recovery. Therefore, the strength of the cavitation increases as the pressure in the low-pressure zone decreases, which causes an increase in cavitation noise.

Figs. 9–14 show the cavitation at back pressures of 0, 0.2, 0.4, 0.6, 0.8 and 1 MPa. We see that the cavitation is mainly generated in the valve core, which is consistent with the pressure distribution. From the simulation results, the pressure change greatly affects the cavitation of the flow field. As the back pressure increases, the cavitation gradually decreases. This indicates that increasing the back pressure has the effect of suppressing the cavitation, and then suppressing cavitation noise. The simulation results also show that the maximum cavitation noise is generated at the throttle valve port, and the noise in other areas of the channel is much lower than near the valve port.

The change in pressure causes change in flow. The decrease in pressure difference increases the hydraulic fluid flow resistance

### Table 1: Hydraulic oil parameters

| Parameter                        | Value       |
|----------------------------------|-------------|
| Density \( \rho \) kg/m\(^3\)     | 870         |
| Surface tension coefficient \( N \), N/m | 3.14 \( \times 10^{-2} \) |
| Dynamic viscosity \( \mu \), Pa s | 0.02784     |
| Latent heat of vapourisation \( L \), J/kg | 3.6758 \( \times 10^5 \) |
| Saturated vapour pressure \( \rho_v \), Pa | 2000       |
| Specific heat at constant pressure \( c_{pl} \), J/(kg·K) | 1880       |

Fig. 3 Pressure distribution for back pressure of 0 MPa

Fig. 4 Pressure distribution for back pressure of 0.2 MPa

Fig. 5 Pressure distribution for back pressure of 0.5 MPa

Fig. 6 Pressure distribution for back pressure of 0.6 MPa

Fig. 7 Pressure distribution for back pressure of 0.8 MPa

Fig. 8 Pressure distribution for back pressure of 1 MPa

Fig. 9 Cavitation at back pressure of 0 MPa

Fig. 10 Cavitation at back pressure of 0.2 MPa

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and reduces the flow rate. According to the Bernoulli equation, the decrease in flow rate inevitably causes the pressure to rise. This causes the pressure in the low-pressure zone to rise, which suppresses cavitation and then reduces the cavitation noise. Therefore, cavitation noise can be suppressed in the cone throttle valve by controlling the pressure distribution in the channel, making the locally low pressure in the channel as high as possible. Pressure directly affects the cavitation distribution, so carefully designing the valve structure to control the pressure distribution can improve the overall valve performance.

3 Experimental study

An experimental system was built to verify the simulation results and further obtain the main factors affecting the cavitation noise of the cone throttle valve. This part mainly describes the experiment and the analysis of its results.

3.1 Experimental system

The principle of the experimental apparatus is shown in Fig. 15. The relief valve (1) set the outlet pressure of the hydraulic pump (2), which was the inlet pressure of the model valve (4). The hydraulic pump (2) consisted of a double-acting quantitative vane pump to reduce the pressure and flow pulsation of the pump itself and improve experimental accuracy. To visualise the cone throttle valve, the model valve (4) was made of highly transparent PMMA. From a safety standpoint, the thickness of the valve was designed to withstand a pressure of 10 MPa. The function of the throttle valve (6) was to adjust the outlet pressure of the model valve. The sensors (3, 5) measured the pressure and temperature of the inlet and outlet of the model valve. The flow sensor (7) measured the system flow rate. The tank (8) provided an oil source for the experimental system. The system had a rated pressure of 6.3 MPa and rated flow of 25 L/min, and used No. 32 hydraulic oil. The apparatus is shown in Fig. 16. An audiometer was placed near the model valve to measure its noise. The audiometer measured the frequency range from 10 Hz to 20 kHz. A high-speed camera was used to capture the flow field changes in the valve flow channel. The camera had a maximum shooting rate of 10,000 frames per second. The motor was a major noise source, so it was isolated with a sound-insulating hood to prevent its noise from affecting the measurement of the throttle valve noise, as shown in Fig. 17.

3.2 Analysis of results

Fig. 18 shows the cavitation shapes in the throttle valve for an inlet pressure of 2 MPa and back pressures of 0, 0.2, 0.4 and 0.6 MPa with a constant valve opening, and Figs. 19 and 20 show the corresponding cavitation shapes for inlet pressures of 3 and 4 MPa, respectively. We see that the greater the valve inlet pressure is, the greater the cavitation intensity is, and that the cavitation strength in the valve flow channel gradually weakens with increasing back pressure.
pressure. This means that reducing the pressure difference between the front and the back of the valve inhibits the development of cavitation. The cavitation in the flow channel is mainly generated near the valve core, which is consistent with the simulation results. This verifies the validity of the simulation. The experimental results also show that the cavitation in the cone throttle valve is cloudy at higher strength. As the cavitation strength gradually decreases, the shape of the cavitation becomes flaky.

Figs. 21–23 show the change in noise with back pressure for inlet pressures of 2, 3 and 4 MPa, where \( A \), \( C \) and \( Z \) denote \( A \)-weighted, \( C \)-weighted and \( Z \)-weighted pressures. As the back pressure increases, the noise at the valve port is basically reduced, which indicates that the cavitation has a direct influence on the noise of the throttle valve. The lower the cavitation strength is, the lower the noise of the throttle valve is. We see from Figs. 21–23 that the noise at the valve port increases as the inlet pressure increases. This indicates that the pressure has a great influence on the cavitation noise, and high pressure is more conducive to generating cavitation noise.

Figs. 24–26 show the noise peaks under different back pressures for inlet pressures of 2, 3, and 4 MPa. We see from Figs. 24–26 that the cavitation noise peak basically decreases as the back pressure increases, which indicates that the noise peak is affected by the cavitation strength. The greater the cavitation strength is, the greater is the cavitation noise peak.
The experimental results show that the noise of the cone throttle valve is closely related to the cavitation. When the cavitation is flaky, the noise is relatively low and when the cavitation is cloudy, the noise is relatively high. This further shows that the noise of the cone throttle valve is mostly cavitation noise.

### Conclusion

Although the mechanism of cavitation noise is not very clear at present, cavitation noise is closely related to the generation of cavitation. The cavitation of the cone throttle valve is affected by pressure, so the noise generated by cavitation is also affected by pressure. We investigated the relationship between pressure and cavitation noise through simulation and experiment. The following conclusions can be drawn from this study:

**Fig. 20** Cavitation shape for an import pressure of 4 MPa
(a) Back pressure of 0 MPa, (b) Back pressure of 0.2 MPa, (c) Back pressure of 0.4 MPa, (d) Back pressure of 0.6 MPa

**Fig. 21** Noise change with back pressure for an inlet pressure of 2 MPa

**Fig. 22** Noise change with back pressure for an inlet pressure of 3 MPa

**Fig. 23** Noise change with back pressure for an inlet pressure of 4 MPa

**Fig. 24** Noise peak at an inlet pressure of 2 MPa

**Fig. 25** Noise peak at an inlet pressure of 3 MPa
Our study has determined that cavitation contributes most performance between the front and rear of the valve effectively reduces the load and improves the efficiency of the valve. Reducing the pressure difference in the valve can be achieved by suppressing the cavitation area. The noise of the throttle valve can be reduced by suppressing the cavitation area. The pressure difference between the front and rear of the valve greatly impacts the cavitation. Reducing the pressure difference between the front and rear of the valve effectively reduces the cavitation strength, thereby reducing the cavitation noise. Therefore, reducing the pressure difference of the throttle valve through structural design can effectively improve the valve performance.

Our study has determined that cavitation contributes most significantly to the noise of a cone throttle valve and has obtained the main factors influencing the noise. This provides an important reference for designing a cone throttle valve. Our study has also provided experimental reference for theoretical research on cavitation noise. The relationship between cavitation morphology and cavitation noise will be further studied in follow-up research.

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Fig. 26 Noise peak at an inlet pressure of 4 MPa

(i) There is obvious cavitation in the valve runner when the cone throttle valve works, and cavitation bubbles can fill the whole runner.
(ii) The cavitation of the cone throttle valve is mainly produced near the valve core, so the cavitation noise is mainly generated at the valve port. The noise of the throttle valve can be reduced by suppressing the cavitation area.
(iii) The pressure difference between the front and rear of the valve greatly impacts the cavitation. Reducing the pressure difference between the front and rear of the valve effectively reduces the cavitation strength, thereby reducing the cavitation noise. Therefore, reducing the pressure difference of the throttle valve through structural design can effectively improve the valve performance.

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