The analysis on the cavitation performance of a steam turbine lubricating oil pump with consideration of the fluid temperature effect

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Abstract. During the operation of the pump, cavitation can cause performance drop and erosion damage to its flow parts, even can produce noise and vibration. Further, it is well known that temperature variation is of great significance for cavitation in a pump. The present paper treated the numerical analysis of cavitation in a lubricating oil pump under different temperature. In this paper, we used UGS NX to make three-dimensional model of the pump impeller and ICEM to cut the non-structural mesh of the whole flow passage. For cavitating turbulent flow simulation, a newly developed cavitation model based on Rayleigh-Plesset equation was applied. The cavitating turbulent flows in the pump impeller at different oil temperatures from 35°C to 100°C were simulated. The numerical results at different flow discharge were compared with the available experimental data. It was noted that the numerical results by the proposed cavitation model agreed fairly well with the experimental data. The results depicted that the critical cavitation number decreased with the increasing flow discharge. As the oil temperature arose, cavity volume inside the pump impeller shrank, and the depression due to cavitation at higher oil temperatures were verified by the numerical simulation. Thus, the proposed cavitation model was acceptable for the simulation of cavitating turbulent flows in the pump at different oil temperatures.

1. Introduction
The pump is a kind of machine which converts the prime mover's mechanical energy to the energy pumping the liquid, and it plays an important role in daily life and various sectors of the national economy. The pump is widely used in aviation, aerospace, energy, traffic, transportation, petroleum chemical and metallurgical industries [1]. Traditionally, vaporization occurs in two forms: one is boiling and the other is evaporation. Cavitation is a phenomenon which is different from normal vaporization. Cavitation occurs by means of gas filled or gas and vapor filled when the local pressure drops below the vapor saturated pressure in liquid flows. When the bubble flows to a higher-pressure area, the steam in the bubble will condense again and the cavity may be destroyed. A series of physical and chemical changes is also produced at this stage. The change of pressure which leads to the inception and shrinkage of cavities in the fluid flow, and a series of physical and chemical changes, is called cavitation [2, 3].

With the continuous improvement of the technology, the demand of performance indicators of pumps is high in engineering, while cavitation is the most important factor which limits further development of
pump performance \cite{4}. So far, in the field of turbo machinery, the cavitation processes are all harmful \cite{5}, so we should try to reduce the harm caused by cavitation at present.

In the process of steam turbine operation, lubricating oil supply is the important guarantee in secure and stable operation of the steam turbine shaft system. And the lubricating oil pump is the "heart" part in the steam turbine lubricating oil system. During the operation of the pump, cavitation can cause performance drop and damage to its flow parts, producing noise and vibration. When cavitation is serious, the efficiency of the pump decreases significantly and the pump does not even work properly. As for a lubricating oil pump, if cavitation occurs, the turbine will not work properly because the shaft support will damage immediately. Therefore, by cavitation performance analysis, discussing the working stability of the lubricating oil pump under different oil temperature is of great significance, which can contribute to the safety operation of steam turbine lubricating oil pump at various oil temperatures.

The present study aims to analyze cavitating turbulent flows in a lubricating oil pump by numerical simulation. The special attention will be paid for cavitation at several oil temperatures in the pump.

2. Mathematical simulation

In this paper, we applied a newly proposed cavitation model to directly reflect the impact of cavitation performance by considering the influences of physical property variations for liquid flows at different operating temperatures. The cavitation model was used to simulate the cavitating flows inside the impeller passage at the oil temperatures of 35°C, 50°C, 75°C and 100°C. The comparisons between the numerical results calculated by using the proposed model and the experimental results have been carried out, so as to validate the model.

In the cavitation model of this paper, the gas-liquid two-phase flow of lubricating oil was analysed by the homogeneous hybrid model, whose density can be described as:

\[ \rho_m = \rho \alpha_v + \rho(1 - \alpha_v) \]  \hspace{1cm} (1)

where \( \rho \) is density of a liquid. \( \alpha_v \) is the local vapor volume fraction. \( \alpha_v = 0 \) means that there is completely liquid, with no vapor. \( \alpha_v = 1 \) indicates that there is completely vapor. And the subscripts of \( m, v, l \) mean the mixture, vapor and liquid components in the mixture.

The governing equations for cavitating flow are the continuity, momentum and energy conservation equations of the mixture fluid (marked with subscript \( m \)) listed as follows:

\[ \frac{\partial \rho_m}{\partial t} + \nabla \cdot (\rho_m U) = 0 \]  \hspace{1cm} (2)

\[ \frac{\partial (\rho_m U)}{\partial t} + \nabla \cdot (\rho_m U \times U) = -\nabla p + \nabla \cdot [(\mu_m + \mu_v) \nabla U] + \frac{1}{3} \nabla \cdot [(\mu_m + \mu_v) \nabla U] \]  \hspace{1cm} (3)

\[ \frac{\partial (\rho_m \lambda_m)}{\partial t} + \nabla \cdot (\rho_m \lambda_m U - \lambda_m \nabla T_m) = (m^+ - m^-) h_l \]  \hspace{1cm} (4)

where \( U \) is the velocity at a local point of the cavitating flow field, and \( h_l \) is the latent heat. \( p, \mu \) represent static pressure, and dynamic viscosity. \( m^+ \) and \( m^- \) represent condensation and evaporation rates, respectively.

When calculating cavitation flow, the mass transfer equation is also introduced:

\[ \frac{\partial (\rho \alpha_v, U)}{\partial t} + \nabla \cdot (\rho \alpha_v, U) = m^+ - m^- \]  \hspace{1cm} (5)

Based on the Rayleigh–Plesset equation and the concept of thermal boundary, a thermodynamic cavitation model modified by Zhang Yao \cite{6,7} was proposed to show the effects of temperature difference on the growth of the vapor. In the model, the source terms, \( m^+ \) and \( m^- \) are given as follows:

\[ m^+ = C_{prod} \frac{3 \rho \alpha_v}{R_b} \left[ \frac{16 \mu_v^2}{9 \rho_l R_b^2} + \frac{2 \max(p - p_v, 0)}{\rho_l} \right]^{\frac{1}{2}} + C_{th}(T - T_i) \]  \hspace{1cm} (6)

\[ \frac{4 \alpha_v \mu_v}{\rho R_b} \]
\[ m^+ = C_{\text{dest}} \frac{3\rho_0 \alpha_v}{R_b} \left[ \left\{ \frac{16 \mu^2}{9 \rho_0^2 R_b^2} + \frac{2}{3} \frac{\max(p_v - p, 0)}{\rho_0} \right\}^2 + C_{\text{th}}(T_i - T) \right] - \frac{4\alpha_v \mu \rho_v}{\rho R_b^2} \]  

(7)

where \( C_{\text{prod}} \) and \( C_{\text{dest}} \) are the empirical parameters calibrated by the experimental data. \( R_b \) is the bubble radius.

Though the above thermodynamic cavitation model \([6, 7]\) seems suitable for simulating cavitating flow, the physics of the model is somehow entangled due to the fact that both the bubble dynamics and the concept of thermal boundary are involved. Basically, the cavitation process is decided by bubble dynamics, consequently a cavitation model based on the Rayleigh-Plesset equation is preferable. Equation (8) shows the Rayleigh-Plesset equation with the surface tension term omitted.

\[ \frac{3}{2} \frac{(dR_b)}{dt}^2 + \frac{4\mu}{\rho R_b} \frac{dR_b}{dt} = \frac{p_v - p}{\rho} \]  

(8)

The vaporization and condensation rates for equation (5) can be shown as:

\[ m^+ = C_{\text{prod}} \frac{3\rho_0 \alpha_v}{R_b} \left[ \frac{16 \mu^2}{9 \rho_0^2 R_b^2} + \frac{2}{3} \frac{\max(p_v - p, 0)}{\rho_0} \right]^2 - \frac{4\alpha_v \mu \rho_v}{\rho R_b^2} \]  

(9)

\[ m^- = C_{\text{dest}} \frac{3\rho_0 \alpha_v}{R_b} \left[ \frac{16 \mu^2}{9 \rho_0^2 R_b^2} + \frac{2}{3} \frac{\max(p_v - p, 0)}{\rho_0} \right]^2 - \frac{4\alpha_v \mu \rho_v}{\rho R_b^2} \]  

(10)

3. 3D model and mesh generation

The present study aimed to analyze cavitation performance for a steam turbine lubricating oil pump. Since a pump casing did not affect cavitation in the pump impeller directly, the simulation model established in this paper included the impeller, the inlet pipe and extending passage, as shown in figure 1. The impeller model was generated by using the software named UGS NX \([8]\). After the model was established, ICEM CFD was used to generated the non-structural mesh of the flow passage \([9]\). In consideration of the accuracy of calculation results, the mesh number of the whole domain was 260164, while the mesh number of the impeller part was 117948. The mesh for the pump is shown in figure 2.

Figure 1. The 3D model for numerical calculation
4. Numerical simulation and analysis

This paper mainly discussed cavitation performance of the pump. First, this paper changed the fluid flow and simulated flow situation at different flow discharges so as to discuss the influence of incoming flow on cavitation performance of the lubricating oil pump. Second, this paper changed the temperature of the lubricating oil and analyzed the steady cavitating turbulent flow so as to discuss the effect of temperature variation on cavitation performance of the lubricating oil pump.

The parameters at the design point condition, which are shown in table 1, are as follows: rotational speed \( n \) of 5135 r/min, flow discharge \( Q_{\text{bep}} \) of 85 \( \text{m}^3/\text{h} \), and oil temperature \( t_{\text{oil}} \) of 35°C. At 35°C, the density of the lubricating oil is 868 kg/m\(^3\), and the dynamic viscosity is 0.1078 Pa·s. The boundary conditions are as follows: uniform flow velocity, temperature, and turbulence intensity were given at the inlet plane, the pressure was assigned at the outlet, and the non-slip wall condition was applied for the solid surfaces. The simulation was achieved by using the commercial CFD code CFX with the \( k-\varepsilon \) turbulence model\(^{[10]}\).

| Rotational speed | flow discharge | oil temperature | density of the lubricating oil | Dynamic viscosity |
|------------------|----------------|-----------------|-------------------------------|--------------------|
| \( n \) (r/min)  | \( Q_{\text{bep}} \) (m\(^3\)/h) | \( t_{\text{oil}} \) (°C) | \( \rho_{11} \) (kg/m\(^3\)) | \( \mu_{11} \) (Pa·s) |
| 5135             | 85             | 35              | 868                           | 0.1078             |

4.1. The effect of flow discharge

It shows the comparison of the flow discharge-head curve among the numerical results and experimental data in cavitating flow for water at 35°C in the figure 3.
It is noted that the numerical result using the present mesh and proposed cavitation model agrees fairly well with the experiment at the design point.

It shows the relation between head and inlet total pressure i.e. $H_t$ at design condition ($Q_{bep}=85$ m$^3$/h, and $t_{oil}=35^\circ$C) in figure 4. Both the calculation and experimental data are included in the figure 4. With the total pressure decreasing near $H_t=6.46$ m, the head begins to change. When $H_t=5.94$ m, the head drops dramatically. This performance breakdown must be caused by cavitation development inside the impeller. The required net positive suction head i.e. $NPSH_r$, predicted by the numerical simulation can be calculated as follows:

$$NPSH_r = 5.94 - p_v = 5.93 \text{m}$$

The value of $NPSH_r$ by the experimental test is 6.05 m, near the predicted result. Though there is some difference between them and the head drop is a little underestimated by the simulation, the relative error is limited within 2% for $NPSH_r$, and the prediction accuracy is acceptable.

It shows the relation between cavity volume in the impeller at different flow discharge in figure 5. The inlet total pressure i.e. $H_i$ is 23.5 m, and $t_{oil}=35^\circ$C. The results indicate the tendency that there is a larger cavity at a larger flow discharge. The experimental results also show the same tendency, e.g. $NPSH_i=5.64$ m at 60 m$^3$/h, and $NPSH_i=7.11$ m at 110 m$^3$/h. By using the tested required net positive suction head and the pump head at the design point ($H=87.9$ m at 85 m$^3$/h), the corresponding critical cavitation number, i.e. $\sigma_c$ can be calculated as follows: 0.0642 at 60 m$^3$/h, 0.0675 at 85 m$^3$/h, and 0.0809 at 110 m$^3$/h. That means the critical cavitation number increases with the increasing flow discharge.
Figure 5. Relation between cavity volume and flow discharge at 35°C.

It shows the comparison of the static pressure distributions at different flow discharge in the pump impeller at 35°C in figure 6, figure 7 and figure 8. In the same range of pressure, the low-pressure area at large flow discharge condition is larger than that at small fluid flow condition and design point condition, which shows that the greater the flow discharge is, the larger the cavitation area in the impeller passage would be. What’s more, from the lowest pressure value in the flow passage, it is noted that the higher the flow discharge is, the lower the minimum pressure inside the flow passage is.

Figure 6. Pressure distributed along the shroud and hub at 35°C, $Q=60\text{m}^3/\text{h}$: (a) The hub surface, (b) The shroud surface.

Figure 7. Pressure distributed along the shroud and hub at 35°C, $Q=85\text{m}^3/\text{h}$: (a) The hub surface, (b) The shroud surface.
Figure 8. Pressure distributed along the shroud and hub at 35°C, \( Q = 110 \text{m}^3/\text{h} \): (a) The hub surface, (b) The shroud surface.

4.2. The effect of temperature

In this paper, four temperature points (35°C, 50°C, 75°C and 100°C) are selected for numerical simulation at the flow discharge of 110 m³/h. According to the results, it is convenient to obtain the influence of temperature on the cavitation performance of the lubricating oil pump.

The comparison of the static pressure distribution among different oil temperature at 35°C, 50°C, 75°C and 100°C is shown in figure 9, figure 10, figure 11 and figure 12. It is noted that, the low-pressure area in low temperature condition is larger than that in high temperature, which indicates that in the higher temperature, cavitation phenomenon inside the impeller passage is restrained. This must be because the effect of thermodynamic cavitation is inevitable in the actual process. When the cavitation bubbles are generated, the bubbles will absorb latent heat of vaporization from the flow field, thus causing the local temperature of the cavitation to decrease. The vaporization pressure also decreases, thereby cavitation is restrained.

Figure 9. Pressure distributed along the hub and shroud at 35°C: (a) The hub surface, (b) The shroud surface.
Figure 10. Pressure distributed along the hub and shroud at 50°C: (a) The hub surface, (b) The shroud surface.

Figure 11. Pressure distributed along the hub and shroud at 75°C: (a) The hub surface, (b) The shroud surface

Figure 12. Pressure distributed along the hub and shroud at 100°C: (a) The hub surface, (b) The shroud surface

The relation between cavity volume at various temperature and at the flow discharge of 110 m³/h is shown in figure 13. It is noted that the curve of temperature – cavity volume has a declining tendency. With the oil temperature arises, the cavity inside the pump impeller shrinks remarkably.
5. Conclusion
In summary, a newly proposed cavitation model with the consideration of the viscous effect based on bubble dynamics has been applied for the numerical simulation of cavitating turbulent flows in a steam turbine lubricating oil pump. Based on the above discussion, the following can be drawn:

The present numerical treatment is suitable for the numerical simulation of thermodynamic cavitation in a pump. The prediction accuracy is acceptable for this kind of study.

Both numerical and experimental results depict that the critical cavitation number decreased with the increase of flow discharge.

As the oil temperature arises, cavity volume inside the pump impeller shrinks, and the depression due to cavitation at higher oil temperatures are verified by the numerical simulation.

Due to the fact that the surface tension effect is not included in the model, future study is necessary to improve the prediction accuracy further, though cavitation model used in this paper can reflect the effects of cavitation depression at higher temperatures.

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