Prediction and analysis of jet pump cavitation using Large Eddy Simulation

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Abstract. 3D LES numerical simulations were performed to investigate cavitation performance inside a jet pump. The results were found to match the test data most closely. The cavitation characteristics of the jet pump were then analyzed using changes in the inlet and outlet pressure to isolate its effect on cavitation. Both results shows that the increase of the inlet pressure generally increases the Reynolds number but decrease the cavitation number, thus aggravate cavitation. The closing of the outlet valve increase the outlet pressure but decrease the flowrate ratio, resulting in the increase of velocity difference and vorticity in the mixing layer. So the cavitation first declines and then grows. The cavities appear slender and extended longer in the throat with high flowrate ratio. Conversely, the cavities look short and located in the front part of the throat with low flowrate ratio. Flow analysis indicated that the turbulence behavior in the shear layer and the overall mean pressure has great influence on the local pressure in jet pump, which reveal the reason of different cavitation shape observed in experiment.

1. Introduction

A jet pump contains none of the moving mechanical components typical of other pumps. Rather, the propulsion is provided by a high velocity jet, which flows from the working inlet and is forced through a nozzle. The original jet flow and the entrained flow are mixed in the throat and both discharge from the outlet, as shown in Figure 1a. Many researchers have studied the internal flow of jet pumps using the 2-equation models based on the RANS method (e.g., ‘[1]’, ‘[2]’, ‘[3]’). The predicted 3-D cavitation evolutions based on LES, including the cavity growth, break-off and collapse downstream, and the shedding cycle as well as its frequency agree fairly well with experimental results (e.g., ‘[4]’). The use of LES to model cavitating flows is expected to give better prediction of the characteristic of cavity in jet pump.

In this paper, the LES method is used to calculate the 3D unsteady flow field. Cavitating flow field is predicted and compared with the experimental photos. The influences of different factors on the cavitation are analyzed. The performance parameters of the jet pump are defined in table 1, in which the \( Q_n \), \( Q_s \), \( P_n \), \( P_s \) and \( v_n \) are shown in Fig.1a, \( P_n \) and \( v_n \) is the mean pressure and velocity on the nozzle outlet section. \( v_c \) is the mean velocity of sucked fluid on the same section. All of them were calculated from the Bernoulli’s equation.

2. Numerical models

Unsteady simulations of cavitating flow were performed with LES model with CFX. The sub-grid scale turbulent viscosity in SGS model is closed by LES wall-adapting local eddy-viscosity (WALE) model. A homogeneous multiphase model, namely the Zwart-Gerber-Belamri cavitation model, was...
employed to describe the evolution of the cavities. The basic geometry and calculation points are summarized in Fig. 1. A structured mesh was selected with about $2.43 \times 10^6$ nodes to meet the requirement of $y^+$, which is approximately 1 in the shear layer. The velocity was set on two inlets and static pressure was set on the outlet according to the experimental data. The time step was set to $10^{-5}$s. The residual values were below $10^{-5}$.

### Table 1. Performance parameters of jet pump.

| Parameter         | Value  |
|-------------------|--------|
| Pressure ratio    | 22     |
| Flowrate ratio    | 22     |
| Efficiency        |        |
| Cavitation number |        |

\[
\begin{align*}
 h &= \frac{(p_n - \rho g (z_n - z_c))}{(p_n - \rho g (z_n - z_c))} \\
 q &= \frac{Q}{Q_n} \\
 \eta &= \frac{h}{1 - h} \\
 \sigma &= (p_n - p_e)/(\rho v^2/2) \\
 \nu &= \frac{v_n - v_e}{v_e}
\end{align*}
\]

![a) The structure diagram of the jet pump (mm)](image1)

![b) the calculation points](image2)

**Figure 1.** The structure diagram of the jet pump and the calculation points

3. **Comparison of the calculation results and experimental data**

The simulation results under 6 typical conditions (3 conditions with $P_n \approx 600$ kPa and 3 conditions with $P_n \approx 400$ kPa) were selected to compare with experimental photos. The predicted vapor fraction ISO surface is shown in Fig.2 and Fig.3. Fig.2 and Fig.3 shows the vapor fraction ISO surface is similar to the shape of the bubbles shown in experimental images. Their similarity on morphology suggests that the simulations are convincing. Fig. 1b shows that the predicted points on performance curves agree well with experimental data.

4. **Analysis of the calculation results**

4.1. **The influence of different parameters on the cavitation performance**

The velocity in cavitation number is not the velocity at the reference point but the velocity difference before mixing $v = v_n - v_e$. The cavitation number is listed in Fig.2 and Fig.3 for different conditions. Generally the cavitation is severe with low cavitation number. However, the cavity vapor volume fraction does not increase linearly with the cavitation number. Since the jet pump has two inlets and one outlet, the pressures on each of the 3 section can influence both the overall pressure and mean velocity in the throat. This means that the pressures affected the cavitation in a more complicate way.

With the decrease of $P_n$, the outlet pressure and the flowrate on both inlets also decrease as expected. But generally the cavitation number is high and the Reynolds number is low, thus the cavitation on the throat is weak. In fact when $P_n$ is below 350kPa, no cavitation was observed during experiments since the outlet pressure was limited by the test system.

The outlet valve was closing from condition A to C or D to F with the pressure on the working inlet $P_n$ keeping almost constant. With the closing of the valve, the outlet pressure increased but the flowrate ratio decrease, which means the velocity difference decrease. Thus the cavitation number increase first then decrease. Both experimental and calculation results show that the cavitation first declined and then grew as expected. It is especially obvious for high pressure case (condition A to C,
\( P_n \approx 600\text{kPa} \). Figure 2 and 3 show the cavitation variation in this process. Cai’s experiment (Cai, 2005) also reported the same cavitation phenomenon in jet pump.

| Condition  | Flowrate \( q \) | Reynolds Number \( R_e \) | Outlet Pressure \( P_o \) | Overall Mean Pressure \( \sigma \) |
|------------|-----------------|------------------|----------------|------------------|
| A          | 1.04            | 52.46 \( \times 10^3 \) | 164kPa         | 0.171            |
| B          | 0.68            | 51.97 \( \times 10^3 \) | 208kPa         | 0.219            |
| C          | 0.10            | 51.30 \( \times 10^3 \) | 252kPa         | 0.191            |
| D          | 0.99            | 51.85 \( \times 10^3 \) | 142kPa         | 0.378            |
| E          | 0.88            | 51.73 \( \times 10^3 \) | 151kPa         | 0.393            |
| F          | 0.01            | 50.899 \( \times 10^3 \) | 196kPa         | 0.332            |

Figure 2. Comparison of experimental photo (up) and the predicted vapor fraction ISO surface (bottom, Vapor fraction=0.1), \( P_n \approx 600\text{kPa} \)

4.2. The vorticity in the mixing layer

Above results show that the cavitation in jet pump is different from the wall attached cavitation since cavitation usually appear in the mixing layer. The turbulence behaviour in the shear layer has great influence on the local pressure, thus on cavitation. The pressure and contours of the vorticity in the throat are shown in Fig.4 for Condition A to C.

The ISO surfaces of pressure and vapor fraction form ring near the nozzle outlet. With the increase of the distance to the nozzle, both become fragmented and distributed in a random way. Obviously the high vorticity in the mixing region cause dramatic local pressure decrease. Meanwhile the overall mean pressure also influences local pressure level. Thus, with condition A, the overall mean pressure is the lowest in the throat because the outlet pressure is the lowest. The cavitation region extends much longer to the downstream than with Condition B and C. When the flowrate ratio decrease to condition B, the vorticity magnitude varies a little but the overall mean pressure level increases a lot. The cavitation thus decline. On the other hand, with condition C, even the overall mean pressure is the highest, local pressure decrease is much dramatic than with other 2 cases because the velocity difference and vorticity are high. Thus the cavity is short and the bubble is limited to the region near the nozzle. In Cai’s experiments (Cai, 2005), the cavitation on condition A was referred as pressure-leading, while on Condition C velocity-leading.
5. Conclusions
In this work, the LES WALE method coupled with ZGB cavitation model has been employed to predict cavitation flows in jet pump. The results agreed well with experimental data. The numerical methods in this study are convincing. Both results show that the increase of the inlet pressure generally increases the Reynolds number but decreases the cavitation number, thus aggravate cavitation. The closing of the outlet valve increases the outlet pressure but decreases the flowrate ratio, resulting in the increase of velocity difference and vorticity in the mixing layer. So the cavitation first declines and then grows. The cavities appear slender and extended longer in the throat with high flowrate ratio. Conversely, the cavities look short and located in the front part of the throat with low flowrate ratio. Flow analysis indicated that the pressure field in the shearing layer is dependent on both overall mean pressure and local shear strain rate, which reveals the reason of different cavitation shape observed in experiment.

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