Critical cavitation coefficient analysis of a space low specific centrifugal pump with micro gravity

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Abstract. Centrifugal pump was used in the loop as a baselined unit. The flow rate of the pump was very small, while the head was high. This space pump must work stable for a long time (more than a year), so the performance of the pump attracted public attention. The rotational speed of the impeller was limited for stability, so the pump belonged to low specific centrifugal pump. In this paper, a single-phase centrifugal pump, which was designed for single-phase fluid loops in satellites, was modeled for numerical simulation. The hydraulic region of the pump was discretized by structured mesh. Three dimensional (3-D) flow in the pump was studied by the use of computational fluid dynamics. Partially-Averaged Navier-Stokes (PANS) model based on RNG k-ε turbulence model was developed for the simulation of the unsteady flow. Velocity inlet and pressure outlet was used as the boundary conditions. Interface was used between the impeller and the casing, as well as the impeller and inlet pipe. Performances and pressure fluctuation of the pump were investigated. The dominant frequency of the pressure fluctuation is blade passing frequency at the region close to the tongue of the casing, while it is twice of blade passing frequency at the other region.

1. Introduction

Single-phase mechanically pumped fluid loop (MPFL) is one of active thermal control technology which is used in the field of thermal control in a spacecraft. Precise control of the temperature in the cabin of the spacecraft and key devices can be realized with the help of MPFL. The MPFL has advantages including strong capability of heat exchange, safe and reliable structure and simple fluid mechanism [1]. Since the 1990 s, the MPFL technology has been used in many international spacecraft. The key component of the MPFL is the circulating pump, which should have long life, high reliability and stability, as well as high efficiency. The energy loss of the pump will be change into heat in the spacecraft, so the high efficiency is one of the most important factors for the designation of the pump.

In manned spacecraft, such as skylab space station, “Gemini” spacecraft, “Alliance” spacecraft and “Salute” space station, the MPFL has played an important role in the development of the technology of aerospace [2]. The MPFL also has been used in the field of Mars exploration program.

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since the 1990s. Most of pumps in MPFL for NASA space program including Mars Pathfinder in 1997, Mars Exploration Rovers in 2003, Mars Science Laboratory in 2011, were all developed by Jet Propulsion Laboratory (JPL)[3,4].

ESA started to develop their MPFL technology in 2004. The MPFL technology was used in large communications satellite for the demand of heat exchange, and the pump used in MPFL were designed inheriting to the pump developed by ESA in 1980s and 1990s[5]. Rapid development has come true in the past few years with the corporation of Bradford Engineering, Realtechnologie and Netherlands National space lab. The pump developed by Realtechnologie has small diameter with lone life for 5 years [6]. In China, the study of MPFL technology started from 1980s and it has been used in the Shenzhou manned spaceship. Although China had ripe experience of the MPFL technology in manned spacecraft, there hasn’t have a report on the MPFL for the heat control used in the satellite [7].

In this paper, a pump used in large satellite for MPFL technology were designed and analyzed. Performances of three kinds of impeller were compared and flows in the impeller were analyzed.

2. **Nonlinear PANS model**

For incompressible flow, $V_i$ is partitioned into resolved and unresolved parts in the instantaneous velocity field, using an arbitrary homogeneous filter.

$$V_i = U_i + u_i$$

where $U_i$ is the resolved velocity field; $u_i$ is the unresolved field. It is used by equation 4 instead of the resolved field.

$$U_i = \langle V_i \rangle$$

The additional non-linear term $\tau(V_i, V_j)$, which is the generalized central second moment, is defined as:

$$\tau(V_i, V_j) = (\langle V_i V_j \rangle - \langle V_i \rangle \langle V_j \rangle)$$

For PANS methods,

$$f_k = \frac{k_0}{k}$$

$$\varepsilon_k = \frac{\varepsilon_0}{\varepsilon}$$

Then the PANS model by the modification of RNG $k$-$\varepsilon$ turbulence model are:

$$\frac{\partial (\rho k)}{\partial t} + \frac{\partial (\rho U_j k_j)}{\partial x_j} = \frac{\partial}{\partial x_j} \left[ \alpha_k \left( \mu + \frac{\mu\varepsilon}{\sigma_k} \right) \frac{\partial k}{\partial x_j} \right] + \rho P - \rho \varepsilon_u$$

$$\frac{\partial (\rho \varepsilon)}{\partial t} + \frac{\partial (\rho U_j \varepsilon_j)}{\partial x_j} = \frac{\partial}{\partial x_j} \left[ \alpha_\varepsilon \left( \mu + \frac{\mu \varepsilon}{\sigma_\varepsilon} \right) \frac{\partial \varepsilon}{\partial x_j} \right] + C_\varepsilon^* P_e - C_{\varepsilon^*} \rho \frac{\varepsilon^2_u}{k_u}$$

thus,

$$\mu_u = \rho C_\mu \frac{k^2}{\varepsilon_u}$$

$$\sigma_u = \frac{f_k^2}{f_k}, \quad \sigma_u = \frac{f_k^2}{f_k}$$

$$C_{\varepsilon^*} = C_{\varepsilon^*} + \frac{f_k}{f_k - C_{\varepsilon^*}} (C_{\varepsilon^*} - C_{\varepsilon})$$

$$C_{\varepsilon^*} = C_{\varepsilon^*} - \frac{\eta (1 - \eta / \eta_b)}{1 + \beta \eta}$$

where $C_\mu = 0.0845$, $k_0 = \alpha = 1.39$, $C_{\varepsilon^*} = 1.42$, $\eta_b = 4.377$, $\beta = 0.012$, $C_{\varepsilon^*} = 1.68$
Evaluating the source terms, $P_k$ in RNG $k$–-$\varepsilon$ turbulence mode has a relationship with $P_{ku}$ in PANS model shown in equation 12.

$$P_k = \frac{1}{f_k} (P_{ku} - \varepsilon_k) + \frac{e_{\varepsilon}}{f_k}$$  \hspace{1cm} (12)

Considering the nonlinear turbulence flow in the pump-turbine, the shear stress was solved by nonlinear turbulence model which was proposed by Ehrhard [15].

$$P_k = - \rho \frac{\partial U_i}{\partial x_j} \frac{\partial U_j}{\partial x_i}$$  \hspace{1cm} (13)

$$ \frac{\partial U_i}{\partial x_i} + \frac{\partial U_j}{\partial x_j} = \frac{2}{3} k \delta_j - 2C_{\mu} \mu \Omega S_j + C_{\mu} \mu \Omega T^2 \left( S_{ij} S_{ij} - \frac{1}{3} S_{ij} S_{ij} \delta_j \right)$$

$$+ C_2 C_{\mu} \mu \Omega S_j \left( \Omega_i S_{ij} - \Omega_j S_{ij} \right) + C_3 C_{\mu} \mu \Omega T \left( S_{ij} S_{ij} - \frac{1}{3} S_{ij} S_{ij} \delta_j \right)$$

$$C_{\mu} = \min \left( \frac{1}{0.9 \mu + 0.15} \right)$$  \hspace{1cm} (23)

$$\Omega_j = \frac{1}{2} \left( \frac{\partial U_i}{\partial x_i} + \frac{\partial U_j}{\partial x_j} \right)$$  \hspace{1cm} (24)

$$S = k \frac{1}{\varepsilon} \sqrt{2S_j S_j}$$  \hspace{1cm} (25)

$$\Omega = \frac{1}{k} \sqrt{2\Omega_j \Omega_j}$$  \hspace{1cm} (26)

where $C_{\mu} = C_{\mu}^2 \varepsilon_k$, $C_1 = -0.2$, $C_2 = 0.3$, $C_3 = 2.0 - \exp \left( - (S - \Omega)^2 \right)$, $C_4 = -32.0 C_{\mu}^2$, $C_5 = -16.0 C_{\mu}^2$, $C_6 = 16.0 C_{\mu}^2$.

3. Rated conditions of the pump

The rated parameters of the pump are shown in Tab.1. $H_r$ denotes the rated head; $n$ denotes the rotational speed of the impeller; $Q_r$ denotes the rated discharge. $n_s$ denotes the specific speed of the impeller; $\rho$ is the density of the transmission medium. $D_o$ is the outlet diameter of the impeller. $D_i$ is the diameter of the impeller inlet. The structure of the pump is shown in Fig.1.

| Table 1. Parameters of the pump |
|---------------------------------|
| $H_r$ (m) | 12 |
| $Q_r$ (L/min) | 1 |
| $n$ (r/min) | 8000 |
| $n_s$ | 18 |
| Transmission medium | ethylene glycol |
| $D_o$ (mm) | 82 |
| $D_i$ (mm) | 10 |
| $\rho$ (kg/m$^3$) | 1030 |
| No. of blades | 4 |
4. Simulation conditions
Mesh of the pump was created by hexahedron using ICEM. The total number of mesh of the pump was 5 million.
The runner’s hydraulic region was set to rotate at a speed of 8000 rpm. Velocity at pipe inlet was specified. The outlet condition was chosen and the value of static pressure was used. The outlet pressure was specified according to the experimental data. No slip wall boundary condition on the solid walls was specified. The upwind discretization scheme was used to discretize the advection term. The time step was 0.0000208 s. Thus, the converged turbulent flow solutions were obtained by rotating the mesh in the runner region by 1° per time step. The unsteady solutions are formed by the converged solutions at all times. For the unsteady calculation, the pressure and velocity change with time in the whole flow passage.

5. Results and discussions
5.1 Performance analysis
The external characteristic of the pump with different rotational speeds is shown in Fig.2. Hump characteristic can’t be seen in the head curve. The head of the pump maintain in a small range as the flow rate changes in the computational range. Constant pressure can be provided by the pump.
In order to predict the critical cavitation coefficient, the pressure value of the pressure outlet is reduced and the performance of the pump is calculated. The performance of the pump with different cavitation coefficient is shown in Fig.3. The critical cavitation coefficient of the pump is 0.065.

![Figure 3. Performance of the pump with different cavitation coefficients](image)

**Figure 3.** Performance of the pump with different cavitation coefficients

5.2 *Cavitating flow in the pump*

Figure 4 shows the cavitating flow in the pump with different cavitation coefficients. When the pump operates at critical cavitation coefficient condition, cavitation occurs at the inlet of the blade in the runner and the volume of the cavitating flow is small. The cavitating flow is gradually developed along the suction side of the blade. The cavitating flow almost block the passage of the runner when the cavitation coefficient is 0.01.
5.2 Cavitating flow in the pump

Pressure fluctuation in the casing close to the outlet of the blade is shown in Fig. 5. The amplitude of the pressure fluctuation is 20.2%. The dominant frequency of the pressure fluctuation is blade passing frequency at the region close to the tongue of the casing, while it is twice of blade passing frequency at the other region.

6. Conclusions

In this paper, a PANS model was used to analyze the instability of a centrifugal pump used in space. The external characteristic of the pump with different rotational speeds is calculated and is could be found that constant pressure can be provided by the pump as the flow rate changes in the
computational range. The critical cavitation coefficient of the pump is 0.065. The amplitude of the pressure fluctuation is 20.2%. The dominant frequency of the pressure fluctuation has a relationship with the blade passing frequency.

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