Energy balance analysis for a canned motor pump used for heat supply system

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Abstract. A canned motor pump is widely used for transporting the liquids without leakage. In this work, the pump is applied for heat supply system. For this application, between the rotor and stator for cooling the canned motor there is a circulating flow, which will flow back in the pump passage with higher temperature, and bring the mixing of the liquid having different temperature between the main stream and the circulating flow. The steady state internal flow has been investigated for the pump, and the pump hydraulic performance has been compared with the experimental data. With the consideration of both mechanical energy and internal energy as the effective energy increment of the working fluid, an efficiency assessment is proposed for a canned motor pump. The proposed efficiency shows the same tendency as hydraulic efficiency, and is more understandable compared with that of the conventional method. The present results indicate that the proposed method of evaluating the efficiency of the canned motor pump can reasonably depict the energy balance inside the whole canned motor pump. Further, the introduction of circulating flow is helpful to predict the internal flow and heat transfer for the canned motor pump in numerical simulation with better accuracy. The present study is helpful for both development and application of high performance canned motor pump for various industries.

1. Introduction
Compared with other types of centrifugal pumps, the canned motor pump has advantages of low noise, low vibration, no leakage and high reliability during operation, which enables it to be applied widely for transporting various kinds of liquid. In a canned motor pump, the pump and the motor are combined together and there is a stainless mask for the rotor and stator of the motor to avoid contacting with liquid. That means there is a much larger clearance between the rotor and stator of the motor, and the large clearance makes the efficiency of a canned motor lower than that of a conventional motor. Further, a small volume of working medium is introduced from the pump outlet to cool down the canned motor, and the cooling flow returns to the impeller inlet and joins the main stream again [1-7]. In this sense, the cooling flow is also called as the circulating flow. Kong et al
studied the circulating flow by conducting the heat transfer and flow coupling analysis in the flow passage to predict the flow resistance [8-10]. The circulating flow merging with the inflow at the impeller inlet would induce extra hydraulic loss for the pump. Therefore, it is usually believed that the efficiency of a canned motor pump is much lower than a conventional pump. On the other hand, the circulating flow can get a temperature increment due to cooling the motor, and brings the extra heat compared with the inflow. The extra heat is useful for a heating system, and helpful to increase the internal energy of the main flow in the pump [11]. Traditionally, only the increment of mechanical energy is regarded as the effective work for a canned motor pump, regardless of the increment of the internal energy. This makes the efficiency of a canned motor pump much lower than that of a conventional pump. Consequently, this consideration is harmful for the application of a canned motor pump.

This study aims to propose a new method to assess the efficiency of a canned motor pump particularly used in heating supply situation, taking the energy balance analysis into account. The hydraulic performance of a canned motor pump is evaluated by experimental test and numerical simulation. The total energy including the mechanical and internal energy is discussed based on the flow analysis in the pump. Based on those results, the effect of the circulating flow on the energy transfer is investigated for the canned motor pump, and the comparison is carried out by using different assessment methods for pump performance.

2. Simulation model and experiment

2.1. Test pump and computation domain

The study focuses on a canned motor pump particularly used in heating supplying situation. Several geometrical and hydraulic parameters under design condition are listed in Table 1. Figure 1 shows that the computation domain of entire flow passage consists of suction pipe, impeller, volute casing and outlet pipe. ANSYS-ICEM is used to generate the structural hexahedral mesh for the entire domain. The pump is a centrifugal one, and has the specific speed of $65 \text{ r/min-m}^3/\text{s-m}$. That means the pump has low specific speed. For simulation, the inlet of suction pipe is set as inlet plane of computation domain, and the outlet of outlet pipe is set as outlet plane of computation domain. The interfaces of suction pipe to impeller and impeller to volute are set as Frozen Rotor.

| Parameters                          | Symbol | Value |
|------------------------------------|--------|-------|
| Impeller inlet diameter (mm)       | $D_1$  | 55    |
| Impeller outlet diameter (mm)      | $D_2$  | 158   |
| Blade number                       | Z      | 6     |
| Design flow rate ($\text{m}^3/\text{h}$) | $Q_d$ | 25    |
| Pump head (m)                      | $H$    | 32    |
| Rotational speed ($\text{r/min}$)  | $n$    | 2900  |
| Fluid temperature ($^\circ\text{C}$) | $T$  | 60    |
| Fluid                              | —      | Water |
2.2. **Numerical methods**

In the pump, the flow is regarded as incompressible. The continuity, momentum and energy equations are described in Cartesian coordinates as follows [12]:

\[
\frac{\partial u_i}{\partial x_i} = 0 \quad (1)
\]

\[
\frac{\partial (\rho u_i)}{\partial t} + \frac{\partial (\rho u_i u_j)}{\partial x_j} = -\frac{\partial p}{\partial x_i} + \frac{\partial}{\partial x_i} \left[ \mu \left( \frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right) \right] \quad (2)
\]

\[
\frac{\partial (\rho T)}{\partial t} + \frac{\partial (\rho T u_i)}{\partial x_j} = \frac{\partial}{\partial x_j} \left[ k \frac{\partial T}{\partial x_j} \right] + S_T \quad (3)
\]

where \( u_i \) (\( i, j = 1, 2, 3 \)) is the component of velocity vector, \( x_j \) (\( i, j = 1, 2, 3 \)) is the coordinate component, \( p \) is the static pressure, \( \rho \) is the fluid density, and \( \mu \) is the dynamic viscosity, \( T \) is the fluid temperature, \( k \) is the heat transfer coefficient, \( c \) is the fluid specific heat capacity, and \( S_T \) is the internal heat source caused by fluid viscosity.

Note that the commercial CFD code ANSYS-CFX is used to solve the steady Reynolds-averaged Naiver-Stokes equations. Additionally, the \( k-\varepsilon \) turbulence model is selected for its high robustness and low computation cost. The maximum nondimensional wall distance value \( y^+ \) near the critical surfaces like blades of impeller is under 300 and the mean value is below 100, which will meet the requirement of the turbulence model. The total pressure is set as inlet boundary condition and the mass flow rate is set as outlet boundary condition. The solid walls are set as no-slip, and are assumed to be adiabatic. The temperature of the fluid at domain inlet is set as 60°C. The fluid density is set as the constant value measured at 60°C because the temperature increment is limited below 3°C, while the viscosity of the fluid is set as the function of the temperature according to the Helmholtz formula.

The mesh independence test is carried out by varying seven different mesh densities to calculate the steady flow at the mass flow rate of 6.66 kg/s, nearly equivalent to the operation condition of the design point.

In Figure 2, it is shown that the head and efficiency fluctuate slightly as the number of grid increases, and after reaching the number of 4.7million, the head and efficiency nearly remains stable and no longer varies according to the mesh number. Therefore, based on the balancing of the time against accuracy, the 6th scheme for mesh generation is selected as the computational mesh for the following study.
3. Results and discussions

3.1. Pump performance

The comparison between experiment results and numerical results of the test pump head is shown in Figure 3, where \( H_{\text{exp}} \) is the pump head measured by experiment, \( H_{\text{cal}} \) is the calculated pump head, \( \eta_{h,\text{cal}} \) is the calculated hydraulic efficiency of the pump, and \( \eta_{\text{exp}} \) is the pump efficiency by experiment. Note that the electric power input to the motor is used as the input power for the canned pump.

As shown in Figure 3, pump head is over-estimated by the simulation because the circulating flow as well as disc friction loss is not included. There is a remarkable efficiency discrepancy between the experiment and simulation for the pump. Besides the circulating flow, the pump efficiency measured by experiment is much lower than the predicted hydraulic efficiency due to the low efficiency of the canned motor. The significant difference for the efficiency like Figure 3(b) seems to be the great obstacle to the application of the canned motor pump.

3.2. Proposed assessment for pump efficiency

To evaluate the circulating flow influence on the main flow, the model of the impeller part and outlet pipe part should be modified according to the experimental arrangement. The modified model is shown in Figure 4, where a circulating outlet and circulating inlet are designated for the flow passage of the circulating flow. Note that the circulating inlet includes six balance holes at the impeller hub. For the purpose of convenience, the flow passage between the circulating outlet and circulating inlet is not considered for the circulating flow. In the experiment, the circulating flow i.e. \( Q_c \) is measured by...
an electro-magnetic flow meter. For simulation, the circulating outlet is the extra outlet boundary, and the circulating inlet is the extra inlet boundary. Because the energy equation should be solved, the temperatures at different planes are necessary. At the inlet of computation domain, the water temperature is 60°C as mentioned above. For the circulating flow, the temperature at the circulating outlet is $T_{c,\text{out}}$, and that at the circulating inlet is $T_{c,\text{in}}$.

![Figure 4. Circulating flow involved physical model.](image)

To reasonably evaluate the efficiency of the canned motor pump, a new assessment is proposed by using the energy balance of both the mechanical energy and internal energy increment of the working fluid. The total efficiency of the pump unit is composed of four parts, hydraulic efficiency, volumetric efficiency, mechanical efficiency and motor efficiency. In the present study, the volumetric efficiency and mechanical efficiency are high and nearly 100%. The total efficiency of the test pump is dependent on two efficiencies: hydraulic efficiency and the motor efficiency.

Assuming that the energy loss of the canned motor mainly transfers to the heat generation in the motor, the power for heat generation i.e. $P_h$ can be calculated as follows:

$$P_h = \left(1 - \eta_g\right)P_{\text{tot}}$$  \hspace{1cm} (4)

where $P_{\text{tot}}$ is the total electric power supplied to the canned motor, and $\eta_g$ is the efficiency of the canned motor.

If all energy loss is transformed to the heat in the canned motor, and the heat is fully absorbed by the circulating flow, the temperature increment of the circulate flow can be calculated by the following equation:

$$\Delta T = T_{c,\text{in}} - T_{c,\text{out}} = \frac{P_h}{c \times \rho \times Q_c}$$  \hspace{1cm} (5)

where $c$ is the specific heat capacity of the water, and $\Delta T$ is the temperature increment of the circulating flow between the circulating inlet and circulating outlet. In circulating flow involved simulation, $\Delta T$ and $Q_c$ are set as the extra inflow and outflow boundaries as shown in Figure 4.

Because the mechanical energy increment per time of the pump i.e. $P_m$ can be calculated using the formula of $P_m=\rho g Q H$, the total power increment $P_e$ is the sum of the powers for the mechanical energy and internal energy increment of the fluid:

$$P_e = P_m + P_h$$  \hspace{1cm} (6)

Therefore, the proposed unit efficiency of the canned motor pump $\eta_{\text{pro}}$ can be calculated as below:

$$\eta_{\text{pro}} = \frac{P_e}{P_{\text{tot}}} = \frac{P_m + P_h}{P_{\text{tot}}} = \frac{\rho g Q H + (1 - \eta_g)P_{\text{tot}}}{P_{\text{tot}}}$$  \hspace{1cm} (7)

In this case, the motor efficiency can be derived from the ratio between experimental efficiency and numerical efficiency stated above and the temperature increment can be obtained according to Equation (5). The flow rate of the circulating flow and the total electric power under different
operational conditions are obtained from the experiment. The parameters of the circulate flow measured by the experiment are shown at Table 2.

### Table 2. Parameters of circulating flow under different operating conditions.

| Q (m³/h) | Q_c (m³/h) | P_tot (kW) | ΔT (℃) |
|----------|------------|------------|--------|
| 15       | 0.89       | 3.84       | 1.77   |
| 17.5     | 0.89       | 3.88       | 1.78   |
| 22       | 0.87       | 4.04       | 1.82   |
| 25       | 0.86       | 4.08       | 1.71   |
| 30       | 0.85       | 4.29       | 1.68   |
| 35       | 0.84       | 4.45       | 1.52   |
| 40       | 0.82       | 4.57       | 1.31   |
| 45       | 0.78       | 4.73       | 1.14   |

Figure 5 shows the pump head and hydraulic efficiency under the circulating flow involved condition and none circulating flow involved condition based on the numerical results. Note that \( H_{cal,c} \) and \( \eta_{h,cal,c} \) are the head and efficiency calculated under the circulating flow involved condition respectively. It is clear that pump performance such as pump head and hydraulic efficiency decrease due to the circulating flow. Under the condition for the main flow rate of 45 m³/h and 35 m³/h, the decrease caused by the circulating flow reaches its maximum value, 2.44% for pump head and 2.71% for hydraulic efficiency respectively. For comparison, the pump head measured by the experiment is also plotted in the figure. The results indicate that the predicted pump heads with the consideration of the circulating flow are closer to the experimental data.

Figure 6 shows three efficiencies: hydraulic efficiency under the circulating flow involved condition \( \eta_{h,cal,c} \), pump unit efficiency based on the proposed assessment \( \eta_{pro} \), and the tested efficiency for the canned motor pump by the conventional method \( \eta_{exp} \). Based on the conventional method, the efficiency of the pump unit is very low, and the maximum efficiency is 62%, which is lower than the pump efficiency of many ordinary centrifugal pumps. Though the maximum hydraulic efficiency \( \eta_{h,cal,c} \) is around 85%, the pump unit efficiency cannot reach the equivalent value to the ordinary pump unit because of the low efficiency for the canned motor. However, for the application of the heating supply system, the heat generation during cooling the canned motor is useful and cannot be neglected. The results shown in Fig.6 depict that the proposed efficiency \( \eta_{pro} \) for the canned motor pump is larger than that of hydraulic efficiency. The maximum efficiency of the pump unit is near 90%, much higher than that assessed by the conventional method.

![Figure 5. Pump performance with or without circulating flow.](image)

![Figure 6. Efficiency comparison by different assessment.](image)
It is also noted that two curves for $\eta_{pro}$ and $\eta_{calc}$ have the similar tendency. The maximum efficiency is near 30 m$^3$/h for both curves, while the curve for $\eta_{exp}$ has the maximum efficiency at 40 m$^3$/h, which is far from the designed flow condition i.e. $Q_d=25$ m$^3$/h. This discrepancy is due to the unfavorable match between the canned motor and the pump part. If the proposed assessment is applied, the concept of $\eta_{pro}$ is more understandable. Therefore, for the canned motor pump particularly used in the heating supply situation where the internal energy is also a useful energy, the proposed efficiency can depict the energy conversion of the whole system more precisely.

Figure 7 shows the temperature distribution in the impeller, and reveals the heat brought from the motor mixing with the main stream during three operation conditions. The temperature difference is larger at the operation for smaller flow rate between the circulating flow and the main stream. That also means there will be more heat generation at low flow rate condition. Because the heat can be effectively transferred to the main stream, the pump unit efficiency is also not low at part-load operation as shown in Figure 6. Under the design condition and larger flow rate conditions, it can be seen that before reaching the impeller exit, the circulate flow and the main stream almost have the uniform temperature.

![Temperature distribution under different flow rate conditions](image)

**Figure 7.** Temperature distribution under different flow rate conditions.

In order to investigate the influence exerted by the circulating flow in the impeller, velocity distribution at design condition is shown in Figure 8. With the introduction of circulating flow, the flow separation along the pressure surface of impeller blade becomes clear, and results in larger hydraulic loss in the impeller. It seems the setting angle at blade inlet is smaller than the flow angle, and the design improvement is necessary in the future.

![Velocity distribution under design condition](image)

**Figure 8.** Velocity distribution under design condition.

4. Conclusions
In this study, the energy balance analysis on the influence of circulating flow has been carried out focusing on the canned motor pump used in high temperature situation. The comparison of hydraulic performance shows fairly good agreement between the numerical results and experimental data. On the basis of the simulation, further study on the influence of the circulating flow has been conducted. For this study, the following conclusions can be drawn:
1) With the consideration of both mechanical energy and internal energy as the effective energy increment of the working fluid, an efficiency assessment is proposed for a canned motor pump. The proposed efficiency shows the same tendency as hydraulic efficiency, and is more understandable compared with that of the conventional method. The results indicate that the proposed method of evaluating the efficiency of the canned motor pump can reasonably depict the energy balance inside the whole canned motor pump system.

2) The introduction of circulating flow is helpful to predict the internal flow and heat transfer for the canned motor pump in numerical simulation.

Acknowledgement
This work was financially supported by the National Natural Science Foundation of China (Grant Nos. 91852103 and 51776102), Beijing Natural Science Foundation (3182014), a grant from the Institute for Guo Qiang, Tsinghua University, and the Tsinghua National Laboratory for Information Science and Technology.

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