Effect of axial offset of rotor on thrust characteristics of a centrifugal pump

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Abstract. To ensure the reliability of multi-stage centrifugal pumps, the prediction of axial thrust is an important issue for the design of balancing device and selection of thrust bearing. However, it is known that the fluid forces acting on rotor could be sensitively affected by the geometrical relations between rotor and stator especially at the low flow rate. Among them, the effect of axial offset of rotor has not been studied sufficiently though the axial offset is easily caused by accumulation of manufacturing tolerances and assembling errors. In this study, experimental/numerical investigations on the effect of axial rotor offset in a final stage model of a three-stage centrifugal pump are carried out. The axial offset is intentionally given to shaft system to the both of discharge/suction side. As a result, the axial thrust is significantly affected by the axial offset in the flow rate range below 50% of the design. It is shown by CFD analysis that this effect of axial offset is caused by the interaction between the back flow from the diffuser and the main flow.

1. Introduction

In multi-stage centrifugal pumps, the balancing device such as balance drum, disk and piston is installed to cancel out the large axial fluid force generated by impellers. Consequently, the residual axial thrust acting on shaft system is reduced as small as possible. However, because such pumps are designed with increasing the number of stages and/or the rotational speed to meet the recent demand for higher energy density of turbomachines, it becomes more difficult to hold the residual axial thrust negligibly small. Besides, the steady/unsteady characteristics of axial thrust is also becoming more complicated. Hence, for ensuring the reliability of pumps, it is crucial to design the balancing device and select the thrust bearing properly. To do so, the accurate prediction method of axial thrust at design stage is essential.

Kurokawa et al. investigated the influence of impeller side leakage flow on axial thrust acting on impeller experimentally/theoretically and presented a calculation method for axial thrust [1]. Bruurs et.al recently suggested a prediction method for axial thrust in which the analytical and CFD models are combined [2]. Although these methods could be applied for stable flow condition around the design flow rate and be useful to predict axial thrust quickly at low cost, they might not be applied to the part load conditions where the flow is significantly unsteady. To ensure the reliability of pump over the wide flow rate range, it might be important to understand the mechanism of axial thrust based on the unsteady flow field.

Especially at low flow rates, the change in geometrical relations between rotor and stator is a critical matter that could affect the fluid forces acting on rotor. Among them, even though the axial offset of rotors arises inevitably from the accumulation of tolerance and assembling errors especially in multi-
stage turbomachines, the effect of that has not been understood sufficiently. It is limited to only a few number of studies [3]. In our previous studies, a series of experiments using a three-stage centrifugal pump has been conducted. In [4], the significant influence of axial rotor offset on axial thrust was observed only in the low flow rate range. In the subsequent test, it was also found that the reduction of impeller diameter including shroud walls weakened this effect of axial rotor offset [5]. At the same time, in our CFD analysis, the back flow from the diffuser getting into the side gaps was observed at the low flow rate [6]. Consequently, it is indicated that the interaction between the main flow and the leakage flow in side gaps would be the key phenomenon for the effect of axial rotor offset. However, the detailed mechanism is still not clear because of its complicated characteristics attributed to the inherent flow conditions in each stages of multi-stage pumps.

In the present study, in order to understand the mechanism of the effect of axial rotor offset on axial thrust in more detail, the experiments as well as the unsteady RANS simulations are carried out using the final stage model of multi-stage centrifugal pump. The pump is a partial model imitating the final stage of a three-stage centrifugal pump used in our previous study. In both the experiment and the CFD analysis, the axial offset is intentionally given to shaft system; in addition to the original no-offset condition, four cases with 0.5mm or 1.0mm offset to the discharge/suction side are set in experiments, and two cases with 1.0mm offset to each side are set in CFD analysis.

2. Experimental and numerical methods

2.1. Test pump

Figure 1(a) shows the schematic view of the final stage model of a three-stage centrifugal pump used in this study. This pump consists of suction casing, single impeller, a vaned diffuser, a discharge casing and a balance drum. A suction pipe is inserted in the place for the removed first and second stages. A part of working fluid (water) exiting from the impeller gets into the impeller back gap and pass through the clearance at balance drum, then returns to the suction side through the return pipe, while main flow goes directly into discharge casing through the diffuser. The primary specifications of this pump model are summarized in table 1. The vane angles are defined from circumferential direction.

![Figure 1. Final stage model of three-stage centrifugal pump](image)

Table 1. Specifications of target pump

| Design specification |              |
|----------------------|--------------|
| Rotational speed, \(N\) | 1600min\(^{-1}\) |
| Flow rate, \(Q_d\)      | 1.5 \(m^3/min\) |
| Head, \(H_d\)          | 38m          |
| Specific speed, \(n\)  | 128[\(m, m^3/min min^{-1}\)] |

| Impeller              |
|-----------------------|
| Number of blades      | 7            |
| Outer diameter, \(D_2\) | 318mm         |
| Outlet angle          | 25deg        |
| Outlet width, \(b_2\) | 13mm         |

| Diffuser              |
|-----------------------|
| Number of vanes       | 10           |
| Inlet diameter, \(D_3\) | 325.5mm      |
| Inlet angle           | 6.2deg       |
| Inlet width, \(b_3\)  | 18mm         |

We set five cases of the axial position of rotor system as below; The “Normal” case is the original condition where the rotor is set in the design position. The condition of the cases with axial rotor offset
is expressed by \( t/b_2 \), which is the ratio of the amount of axial rotor offset \( t \) and impeller outlet width \( b_2 \). The positive amount of \( t \) means the offset is set to the suction side. Hence in the cases with 0.5mm and 1.0mm axial offset to the suction side, \( t/b_2 = +0.038 \) and +0.077 respectively, while \( t/b_2 = -0.038 \) and -0.077 with that to the discharge side. The cases with 0.5mm offset is considered only in the experiments. The axial overlap between impeller shroud and casing wall is 1.5mm in the “Normal” case.

2.2. Experimental method

The hydraulic performance of the pump is measured as follows according to JIS B 8301:2000. The discharge flow rate is measured by an electromagnetic flow meter installed at discharge pipe. The total head rise of the pump is calculated from the difference between outlet and inlet pressure \( P_{out} \) and \( P_{in} \) measured by pressure transducers with considering the dynamic pressure based on the area-averaged flow velocity at corresponding locations. The shaft torque \( T \) is measured by a torque meter installed between the pump shaft and the motor shaft, and the mechanical loss estimated by measurement \( T_0 \) is subtracted from \( T \) for the comparisons with CFD result. The mechanical torque \( T_0 \) is estimated in the form of linear function of pump rotational speed \( N \), which has been obtained as a deviation of the torque from the similarity law \( (T \propto N^2) \) under the constant operation with the design flow coefficient. The pump performance is summarized in non-dimensional forms, the flow coefficient \( \phi \), the head coefficient \( \psi \) and efficiency \( \eta \), which are defined as follows;

\[
\phi = \frac{Q}{AU}, \quad \psi = \frac{gH/3}{U^2}, \quad \eta = \frac{\rho gQH}{(T - T_0)\omega}
\]

where \( A \) is the area of impeller outlet passage, \( Q \) is the discharge flow rate, \( \omega = 2\pi N/60 \) and \( U(=D_2\omega/2) \) are the impeller angular and peripheral speeds, \( g \) is the gravitational acceleration and \( \rho \) is the density of water. Stationary part of all bearings, one thrust and two radial bearings, are supported by load cells, from which the radial and residual axial thrust acting on the shaft system can be measured. The residual axial thrust is summarized using the axial thrust coefficient defined as follows;

\[
C_{FA} = \frac{F_A}{\rho AU^2/2}
\]

where \( F_A \) is the measured axial thrust for the whole rotor.

In order to investigate the mechanism of change in impeller axial thrust due to the axial offset, the static pressure is measured by pressure transducers installed at three locations in the front gap \( (R/R_2=0.09, \ 0.79 \ 0.61) \) and two locations in the back gap \( (R/R_2=0.69, \ 0.56) \) as indicated by the red points in figure 1 (b), where \( R \) is the radial location of measuring position and \( R_2 \) is the outer diameter of impeller. The measured static pressure \( P_s \), are summarized using static pressure coefficient \( \psi_s \) defined as follows;

\[
\psi_s = \frac{P_s - P_{in}}{\rho U^2/2}
\]

2.3. Numerical method

For the CFD simulations, a commercial code, SCRYU/Tetra V13 [7] developed by Software Cradle is used. The numerical model is composed of the entire flow passage of the test pump, including the impeller side gaps and the leakage paths at annular seals such as liner ring and balancing channel. Figure 2 shows the sectional view of the computational grid for the model. Tetra-prism mesh is used for the suction and discharge volute casing (green), and hexahedral mesh is used for impellers, diffusers, impeller side clearances and annular seals domains (red). The number of computational grid is totally about 18 million nodes. We have compared the results with those using a finer grid (28 million nodes) in some cases, and the present configuration reproduces similar results quantitatively as stated later. The unsteady RANS simulation is carried out using the \( k-\omega \) SST model for turbulent closure. The discretization schemes for finite volume method used in SCRYU/Tetra are summarized in table 2. For
the inlet and outlet boundaries, the fixed mass flow rate condition and constant static pressure condition is applied respectively. The ALE (Arbitrary Lagrangian Eulerian) method is utilized to connect the rotating domains and stationary domains for unsteady analysis. The time step for unsteady simulations $\Delta t$ is set to be 1/360 of impeller rotational duration, i.e. $\Delta t=1.04\times 10^{-4}$ s. As an initial condition, the result of steady flow simulation with frozen rotor interfaces between rotationally and stationary domains is used. As mentioned above, two cases with 1.0mm axial rotor offset are considered in addition to the normal case. The computations are made for the flow coefficient of $\phi_d=1.0$ and 0.20. The time-averaged flow field in the last two impeller rotation periods are used for the performance evaluation. The residual axial thrust force is calculated by integrating the wall pressure and the wall shear stress over the all surfaces of rotating part.

3. Results and discussions

3.1. Pump performance

The measured and computed pump performance is shown in figure 3, where the open and closed symbols correspond to the experimental and the CFD result respectively and color of them means the cases of axial rotor offset. The error bar is presented for the experimental results, whose one side means the double of standard deviation of ten measured samples. In the experimental results, it is clear that the head is almost not affected by the offset. However, only in the case with 0.5mm offset to suction side, the efficiency is slightly decreased; actually, the head is slightly decreased and the torque is increased in this case. Unfortunately, we have no reasonable explanation about this reduction of efficiency.

It is confirmed that the computed head agrees well with the measured one throughout the flow rate. However, the efficiency is noticeably underestimated at the part flow rates, which is due to the overestimation of the calculated torque. Comparing with the CFD result using finer grid depicted by orange symbols, it seems that the resolution of grid is sufficient in the present configuration for qualitative discussion. About the effect of axial offset, the computed head as well as the efficiency are slightly decreased by giving the axial offset at both of $\phi_d=1.0$ and 0.20, indicating that the effect of axial offset seems to be a little over-estimated in the CFD simulations.
3.2. Axial thrust
Figure 4 shows the characteristic of residual axial thrust against the flow rate. Please note that the positive amount means that the axial thrust acts in the direction toward the suction side. Likewise the performance, there exist almost no variation of predicted axial thrust against the number of computational grid nodes. In the normal case, the predicted axial thrust qualitatively agrees with measured one over the whole flow range while a noticeably over-estimated at $\phi/\phi_d=0.6$.

Looking at the result of experiment indicated by the open circle symbols, the effect of axial rotor offset is remarkably observed in the low flow rate range. The axial thrust is increased toward the same direction as the given axial offset, and the variation of axial thrust from the normal condition changes depending on the amount of the axial rotor offset ($r=0.5\text{mm or } 1.0\text{mm}$). In the CFD result, although the effect of axial offset on axial thrust is quantitatively over-estimated in comparison with the experiment, the tendency of that is well simulated in CFD; the alteration of axial thrust depends on the direction of axial rotor offset and it is more significant at the low flow rate.

![Figure 5](image1.png)  
**Figure 5.** Measured static pressure distribution in side gaps ($\phi/\phi_d=0.20$, effect of axial offset to suction side)

![Figure 6](image2.png)  
**Figure 6.** Measured static pressure distribution in side gaps ($\phi/\phi_d=0.20$, effect of axial offset to discharge side)

![Figure 7](image3.png)  
**Figure 7.** Computed Static pressure distribution in side gaps by CFD ($\phi/\phi_d=0.20$, time-averaged)

The static pressure distributions measured in impeller side gaps at $\phi/\phi_d=0.20$ are shown in figures 5 and 6, focusing on the effect of axial offset to the suction or discharge side respectively. From these figures, it is found that the slope of pressure distribution becomes steeper in the gap located at the same side as the given offset, while more gradual at the opposite side. At the same side as the given offset, the 1.0mm offset seems to have almost no farther impact more than 0.5mm offset on the side gap pressure distribution. However, at the opposite side to the given offset, the slope of static pressure gradient is more gradual in the cases with 1.0mm offset than those in the cases with 0.5mm offset. Therefore, it seems that the influence on axial thrust due to axial offset is more attributed to the flow condition in the gap located at the opposite side to given offset, rather than the same side.

Figure 7 shows the computed static pressure distribution in side gaps. Although the predicted static pressure is not quantitatively agree with the measured one, the influence of axial offset is substantially simulated also in the CFD result. Hence in the following, the detailed mechanism of the effect of axial rotor offset is discussed on the basis of the flow field obtained from the CFD simulations.

3.3. Mechanism of axial rotor offset on residual axial thrust
The influence of axial offset on each axial thrust generated by the impeller and the balance piston is separately shown in figure 8. It is clear that the alteration of residual axial thrust due to the axial rotor offset is primarily caused by the axial thrust acting on impeller rather than that on the balance piston, which suggests us to focus on the flow mechanism leading to the change in the impeller axial thrust.
Figure 9 shows the tangential velocity distribution in side gaps at $\phi/\phi_d=0.20$ calculated by CFD in the all cases of axial rotor offset. From this figure, it is clear that the fluid rotation is enhanced in the gap at the same side as the given offset, while reduced at the opposite side. Consequently, the remarkable difference of fluid rotation in the front and back gaps is produced by the axial rotor offset, which should leads to the influence on the axial thrust.

In order to investigate how the angular momentum is brought into side gaps, which results in the change in the tangential velocity distribution shown in figure 9, the circumferential distributions of axial
and tangential velocity components acquired at the inlet of side gaps are respectively shown in figures 10 and 11 for each offset case. In these figures, solid and dashed curves show the velocity component at the inlet of front and back gaps respectively, which are evaluated from the time-averaged flow field in the last two impeller rotation. The position of diffuser vane leading edge is indicated by the green vertical lines. Please note that the axial velocity getting into side gaps is defined as positive direction. In figure 10, regardless of the cases of axial offset, the axial velocity is positive only in the regions within 10–15 degrees upstream of the leading edge of diffuser vanes. The angular momentum should be brought into side gaps through these regions. Focusing on the difference due to axial offset shown in figures 10 and 11, the tangential velocity in this region is obviously larger at the same side as the given axial offset than that at the opposite side. Moreover, probably since the axial offset alters the overlap between impeller shroud and casing wall, the axial velocity is increased at the opposite side to the given offset, while decreased at the same side. However, considering the fact that the fluid rotation is enhanced in the same side gap as the given offset as shown in figure 9, the effect of axial offset on the axial thrust is more attributed to the change of tangential velocity rather than the change of axial velocity.

For the sake of the flow field analysis near the leading edge of diffuser vanes, the time-averaged streamline passing through the inlet of side gaps is presented in figures 12 (a) and (b) for the cases with 1mm offset to discharge and suction side respectively. The color of streamline indicates the time-averaged tangential velocity. At the both sides, it is observed that the flow getting into side gaps comes from the backflow occurring near the diffuser walls, which has small rotational component. At the same side as the given offset, the impeller exit is near the diffuser wall as well as the side gap inlet, thus the backflow from diffuser is more easily mixed with the impeller exiting flow which has high rotation before entering the side gap. On the other hand, at the opposite side to the given offset, probably because the shroud wall is located in front of the region where the back flow is observed, the backflow directly enters side gaps without sufficiently receiving the angular momentum from the main flow. This seems to be the reason why the axial offset causes the difference of fluid rotation in side gaps between front and back side, leading to the remarkable influence on the axial thrust at the low flow rate.

![Figure 12. Streamline across inlet of side gaps around leading edge of diffuser vane](image)
4. Conclusion
In this study, the effect of axial rotor offset on residual axial thrust in the final stage model of a three-stage centrifugal pump is investigated by experiment as well as unsteady RANS simulation. The main results are summarized as follows.

(1) Pump performance is hardly affected by the axial rotor offset.
(2) In the experiment, residual axial thrust is significantly affected by the axial offset in the low flow rate range, depending on the direction and amount of axial offset. At the same side as the given offset, there is almost no difference of side gap static pressure between the cases with 0.5mm offset and 1.0mm offset. However, at the opposite side, the stronger influence is observed in the case with 1.0mm offset than those in the case with 0.5mm offset. Therefore the effect of axial offset on axial thrust seems to be more attributed to the flow condition at the opposite side to the given offset than the same side.

(3) CFD results qualitatively simulate the tendency of the effect of axial offset on the axial thrust, and the axial thrust change is due to the difference of fluid rotation in side gaps between the front and back side. The detailed analysis of flow field shows that the backflow from diffuser enters the side gaps and brings the angular momentum there. Since the impeller exit is slid axially by giving the axial rotor offset, the backflow getting into side gaps tends to be mixed with the main flow at one side, while flows directly into side gaps without receiving the angular momentum at the other (opposite) side. This seems to be the mechanism leading the change in the flow in side gaps, resulting in the axial thrust change due to the axial rotor offset.

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