

ACTIVE THRUST BALANCING CONTROL OF A TURBOPUMP WITH A SWIRL BREAKER

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Abstract

Turbopumps are responsible for supplying high-pressure propellant to aircraft and rocket engines. As the demand for higher-performance engines increases, the turbopumps are required to achieve higher discharge pressures. This increased discharge pressure leads to an augmentation of the axial thrust within the turbopump, consequently imposing excessive loads on the supporting bearings. Therefore, the application of an active thrust balancing system is necessary to control the load exerted on the bearings. The active thrust balancing system can be configured using a balance disk and control gaps, and it is crucial to secure sufficient control performance to manage an adequate range of thrust load. In this study, a parameter analysis was conducted using an in-house developed reduced-order computer program to enhance the performance of the active thrust balancing system. The analysis results demonstrated that the load control capability can be effectively increased by incorporating a swirl breaker into the balancing system configuration.

Keywords: Turbopump, Thrust control, Active balancing, Swirl breaker

1. Introduction

As engine performance demands increase, turbopumps are required to generate higher pressures, and these escalating discharge pressure requirements present significant challenges in turbopump development. The high axial thrust generated by the high-pressure turbopump discharge is one of the challenges that must be overcome, as this increased thrust imparts excessive loads on the bearings supporting the rotating assembly. In particular, when operating conditions change due to engine thrust modulation requirements, the turbopump's operating environment also changes, necessitating an appropriate response to the resulting variations in axial thrust. Active thrust balancing systems can respond to unexpected axial thrust changes by adjusting the control gap. In an active thrust balancing system, the difference between the forward and backward axial loads applied to the balance disk causes the rotor system to move axially. The control gap, which forms part of the balance chamber, is then automatically adjusted according to the movement of the rotor system [1, 2]. The static characteristics of active thrust balancing systems have been understood and analyzed through 1D analysis and Computational Fluid Dynamics (CFD) [3–5]. The axial thrust was calculated by considering the effect of the leakage flow entering the internal cavity between the rotor and the casing, determining the differential pressure characteristics of each component in the balancing system, and obtaining a converged solution for the pressure distribution and leakage flow through iterative numerical analysis. By calculating the axial thrust corresponding to the change in the control gap, the controllable range of the balancing system can be confirmed. In this paper, an analytical investigation was conducted whether the load control capability of the balancing system can be improved by adding a swirl breaker to the system to enhance the axial thrust control range.

2. Active thrust balancing control

Active thrust balancing control is achieved by controlling the axial gap that regulates the leakage flow rate in the balance chamber. Figure 1 illustrates the active thrust balancing system utilizing a two-gap control method. For system simplification, the impeller is utilized as the balance disk, and two control gaps are configured at the outer and inner sections of the impeller's rear shroud. The space enclosed by the rear shroud of the impeller, the inner surface of the casing, the outer control gap, and the inner control gap forms the balance chamber. The internal pressure of this chamber changes according to the clearance variation of the control gaps, thereby altering the load applied to the impeller's rear shroud. The swirl breaker is installed on the inner surface of the stationary casing to suppress the fluid's swirl. If the axial load applied to the impeller's front shroud is greater than the axial load applied to the impeller's rear shroud, the rotor system, including the impeller, moves rearward, and the inner control gap's clearance narrows. The narrowing of the inner control gap increases the pressure within the balance chamber. This pressure rise consequently increases the axial load applied to the rear shroud, causing the rotor system to stop its movement at the point where the axial thrust is balanced. When the clearance of the inner control gap becomes narrower, causing the force acting on the impeller rear shroud to exceed the force acting on the front shroud, the rotor system moves forward. As a result, the inner control gap widens while the outer control gap narrows. This narrowing of the outer control gap leads to a decrease in balance chamber pressure, and the movement of the rotor system ceases at the point where the forces acting on the front and rear shrouds of the impeller reach equilibrium.

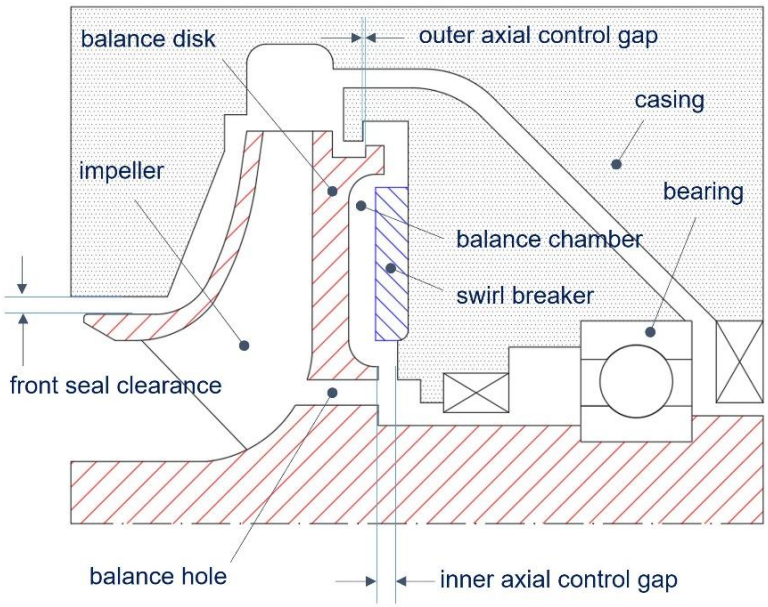


Figure 1 - Active thrust balancing system.

The internal pressure $p(r)$ of the balance chamber can be obtained by subtracting the fluid centrifugal term in the radial direction from the impeller outlet pressure $P_{r_{tip}}$, as shown in equation (1).

$$p(r) = P_{r_{tip}} - \rho\omega^2 \int_r^{r_{tip}} k^2(r) r dr \quad (1)$$

where $P_{r_{tip}}$ is the impeller outlet pressure, r is the radial position in the balance chamber, r_{tip} is the impeller tip radius, ρ is the fluid density, ω is the angular velocity of the shaft, and k is the rotational speed ratio, representing the ratio of the fluid rotational speed to the impeller rotational speed.

By considering the through-flow entering the balance chamber, the angular momentum of the fluid is determined. Then, the rotational speed ratio k can be obtained from the equilibrium solution with the friction torque of the rotor and stator surfaces surrounding the balance chamber. The rotational speed ratio k with respect to the radial position is expressed as shown in equation (2).

$$dk/d\bar{r} = 0.07875\left(\pi\omega\bar{r}^{1.6}/QRe_u^{0.2}\right) r_{tip}^3 \{[(1-k_0)k/k_0]^{1.75} - |1-k|^{1.75}\} - 2k/\bar{r} \qquad (2)$$

where $\bar{r}$ is the nondimensional radius $\bar{r} = r/r_{tip}$, Q is the leakage flow rate, Re_u is the rotational Reynolds number $Re_u = \omega r_{tip}^2/\nu$, ν is the fluid kinematic viscosity, and k_0 is the constant swirl ratio when the leakage flow is zero. The detailed equations are described in Reference [5]. Subsequently, the axial load *F* on the impeller's rear shroud is calculated through the integration of the pressure $p(r)$ across the area, specifically from the impeller hub radius r_{hub} to the impeller tip radius r_{tip}.

$$F = 2\pi \int_{r_{hub}}^{r_{tip}} p(r) r dr \qquad (3)$$

3. Numerical results

Figure 2 shows the nondimensional net axial force and leakage flow rate calculated by varying the inner control gap clearance in the active thrust balancing system. It can be observed that as the inner control gap clearance narrows, the net force increases and the leakage flow rate decreases. Conversely, as the inner control gap clearance increases, the net force decreases. The leakage flow rate, however, increases until the midpoint of the inner control gap and declines thereafter. The reason for this is that when the inner control gap clearance narrows, the resistance to flow increases, which simultaneously reduces the leakage flow rate and raises the pressure in the balance chamber. In the case where the inner control gap clearance widens, the outer control gap clearance narrows in opposition, which also leads to a decrease in leakage flow rate. However, because the balance chamber is exposed to the downstream low-pressure region while the upstream high-pressure region is obstructed, the balance chamber pressure and the net force decreases. The leakage flow rate reaches its maximum at a clearance position intermediate to the inner and outer control gaps where their clearances are similar. The swirl breaker(SB) affects the net force and leakage flow rate compared to the case without it because it serves to suppress the fluid swirl inside the balance chamber. When the swirl breaker is present, the slope of the net force change with respect to the inner control gap clearance increases at the load equilibrium position. This is advantageous as it

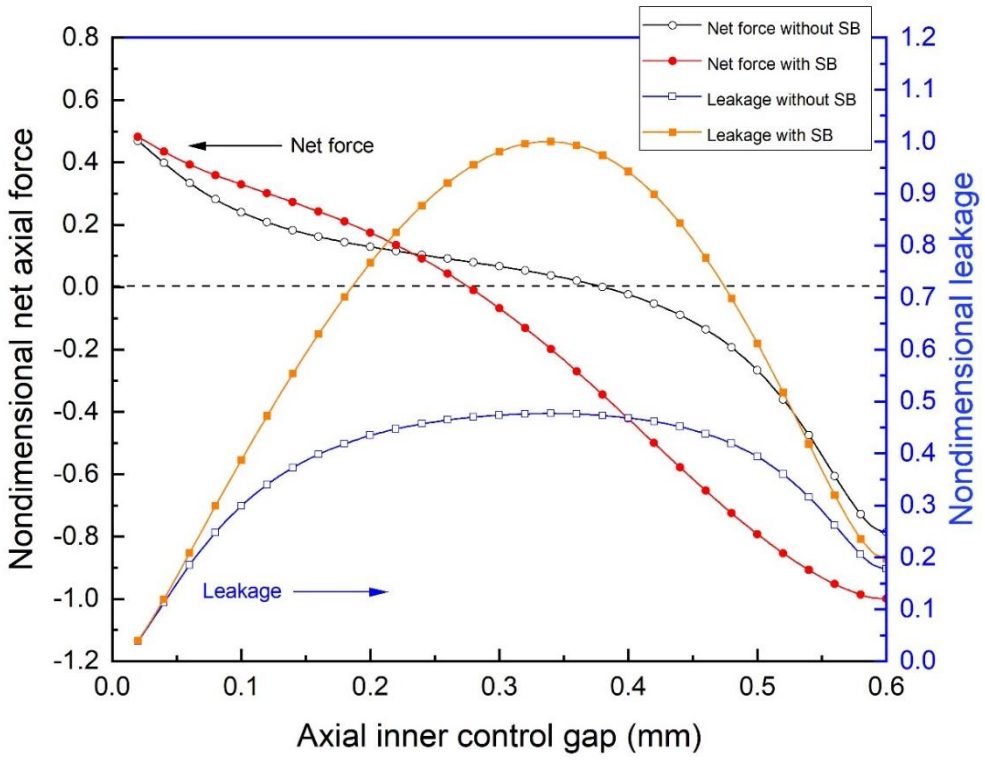


Figure 2 - Two-gap control balancing characteristics with swirl breaker.

increases the restoring force toward the equilibrium point. However, it has the disadvantage of increasing the leakage flow rate, necessitating the optimal design of the swirl breaker.

Figure 3 shows the net force difference calculated by varying the inner control gap clearance in an active balancing system, for different cases where the leakage flow rate passing through a swirl breaker, placed on the stator, is altered. The baseline represents the case without a swirl breaker. Cases with a swirl breaker are divided into three scenarios in which the through-flow passing through the swirl breaker accounts for 30%, 60%, and 90% of the total leakage, respectively. The net force difference represents the result obtained by subtracting the net force of the baseline case from the net force obtained in each analysis case. The swirl breaker was configured to be positioned in the upper one-third region when the balance chamber is divided radially into three zones. Near the inner control gap of 0 mm, all three cases with the swirl breaker show the same net force as the baseline. However, as the inner control gap increases, the change in net force difference becomes larger as the leakage flow rate through the swirl breaker increases. Within the range where the inner control gap is approximately smaller than 0.3 mm, the net force is observed to be higher in cases incorporating a swirl breaker. Conversely, when the inner control gap exceeds 0.3 mm, the net force is comparatively lower for cases with a swirl breaker. When the inner control gap is 0.6 mm (making the outer control gap 0 mm), the net forces are not identical across all cases, showing slight discrepancies, which contrasts to the 0 mm case. When the outer control gap is close to 0 mm, similar to when the inner control gap approaches 0 mm, the leakage flow rate decreases significantly. Therefore, the swirl breaker's role is not prominent, and the net force should be the same regardless of the swirl breaker's presence. However, in the model used for this analysis, the outer control gap has a tiny radial clearance, so even if the axial gap disappears, a small amount of leakage flow persists, leading to a difference in net force. Due to the two control gaps (inner and outer), the variation of the net force difference exhibits a sine wave pattern with two maximum peaks, returning to the baseline at both ends of the total clearance and near the center. When the outer control gap is larger than the inner control gap, the balance chamber pressure is influenced by the upstream high-pressure region, and the swirl breaker amplifies this influence, causing an increase in pressure. Conversely, when the inner control gap is larger than the outer control gap, the balance chamber pressure is influenced by the downstream low-pressure region, and the swirl breaker amplifies this

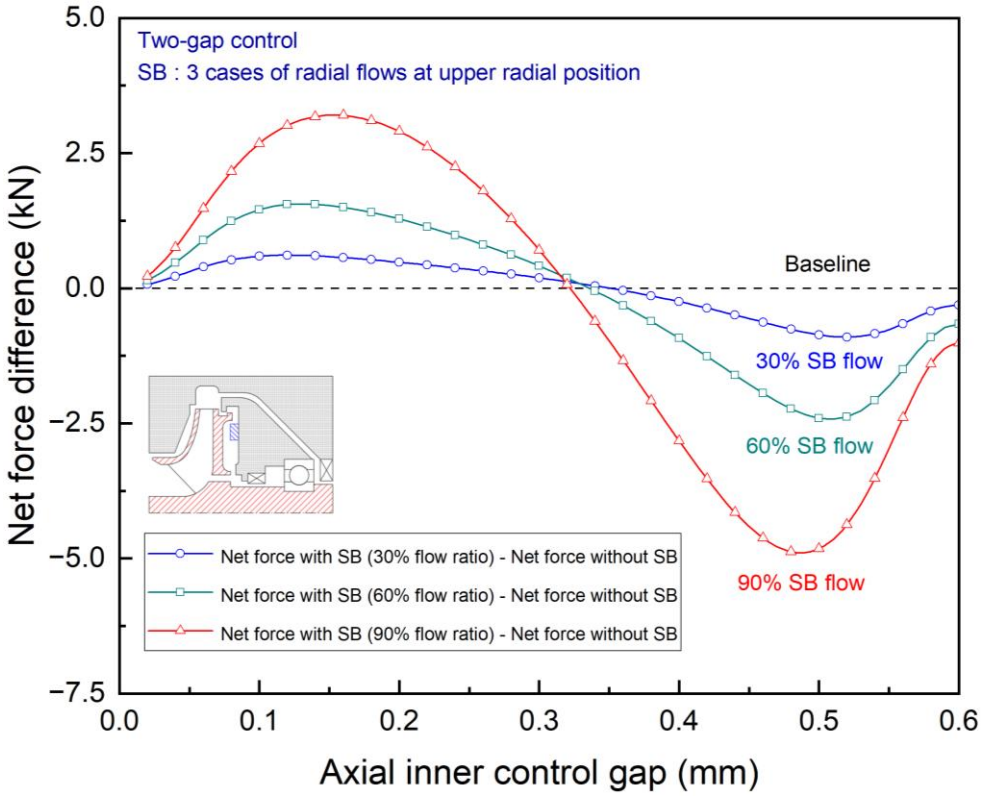


Figure 3 – Effect of the swirl breaker through-flow rate on the axial force.

influence, causing a decrease in pressure. It can be seen that as the leakage flow rate through the swirl breaker increases, leading to a greater swirl breaker effect, the difference between the maximum and minimum values of the net force increases, thereby increasing the force control capability.

Figure 4 shows the net force difference calculated by varying the inner control gap clearance for cases where the radial position of the swirl breaker is changed in an active balancing system. The effect of the swirl breaker's radial position on the net force change was examined by radially dividing the balance chamber region into three sections (upper, middle, and lower) and placing the swirl breaker in one of these sections. In each analysis, the through-flow rate passing through the swirl breaker was uniformly set to 90% of the total leakage. The analysis results indicate that the placement of the swirl breaker at the lower position has the largest impact on the net force change according to the variation in the inner control gap, followed by the middle placement, and lastly, the upper placement shows the smallest impact. The reason why the net force change is greater when the swirl breaker is positioned towards the lower side (i.e., closer to the axis of rotation) within the balance chamber is that the swirl of the flow entering the balance chamber increases as it approaches the axis of rotation due to the conservation of angular momentum, thus resulting in a more significant suppression effect by the swirl breaker. Consequently, the most effective results can be obtained by placing the swirl breaker in the region where the swirl of the flow entering the balance chamber is greatest.

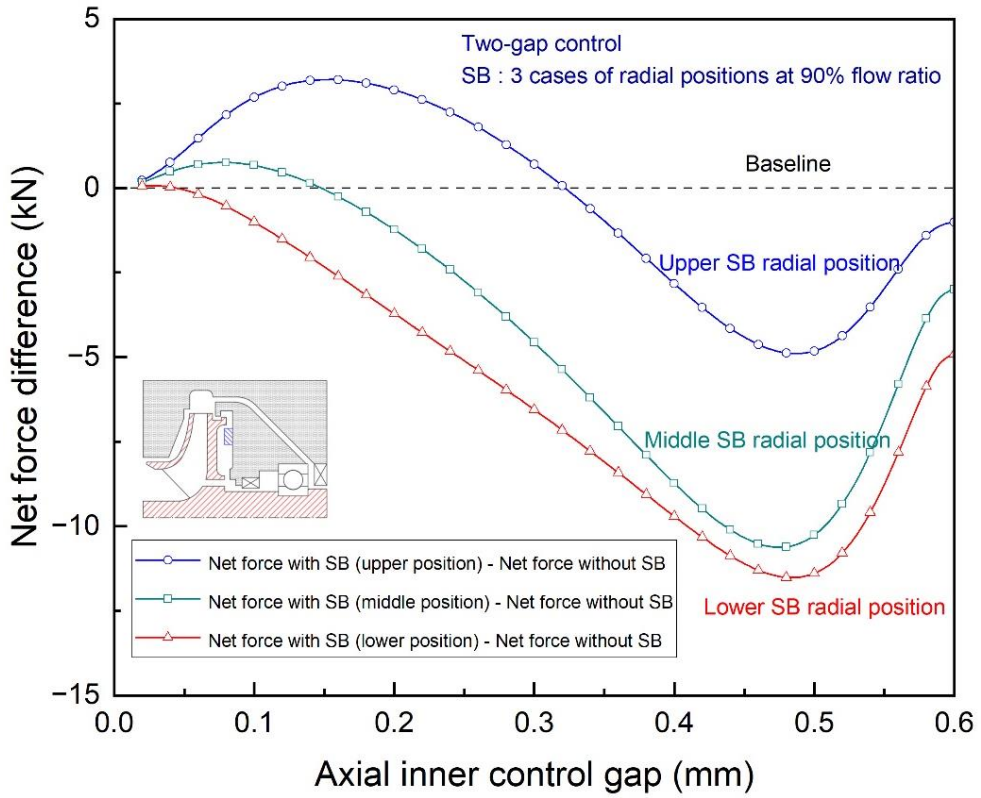


Figure 4 – Effect of the swirl breaker radial position on the axial force.

Figure 5 shows the radial pressure profile on the impeller rear shroud with the inner control gap clearance of 0.15 mm. In this case, the outer control gap clearance is 0.45 mm, which indicates that the balance chamber is closed at the bottom and open at the top, making it susceptible to the high pressure at the impeller outlet. The flow rate through the swirl breaker was set to 90% of the total leakage. Pressure profiles for two cases - with the swirl breaker at the upper and lower positions - were analyzed and compared with the pressure profile without a swirl breaker. The radial position and back-side pressure were nondimensionalized by the impeller tip radius and impeller outlet pressure, respectively. With the inner control gap being narrow and the outer control gap wide, the balance chamber pressure is influenced by the high-pressure region at the impeller's upper section. A swirl breaker placed at the upper position reduces the swirl of incoming leakage and consequently

increases balance chamber pressure. On the other hand, a swirl breaker placed at the lower position reduces both the balance chamber pressure and the downstream pressure of the inner control gap when compared to the case without a swirl breaker. This is inferred to be because the lower-positioned swirl breaker amplifies the influence of the downstream low-pressure region, exerting a significant impact on the pressure profile despite the small inner control gap clearance. It is observed

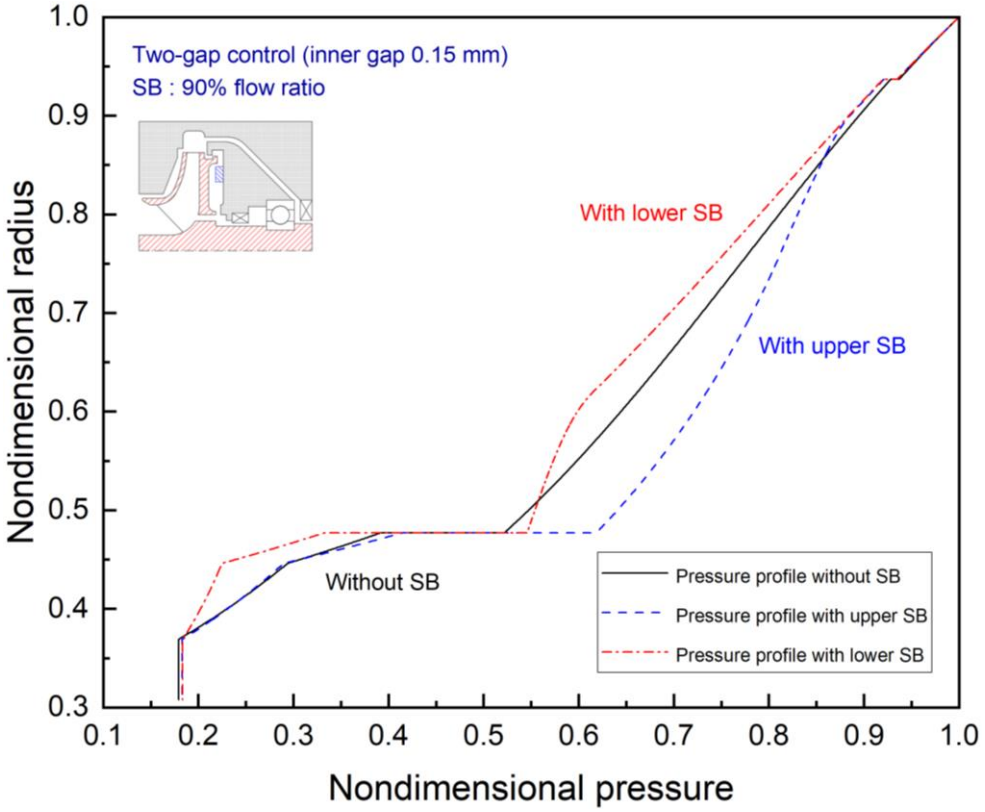


Figure 5 – Pressure profile of the impeller back side at inner gap 0.15 mm.

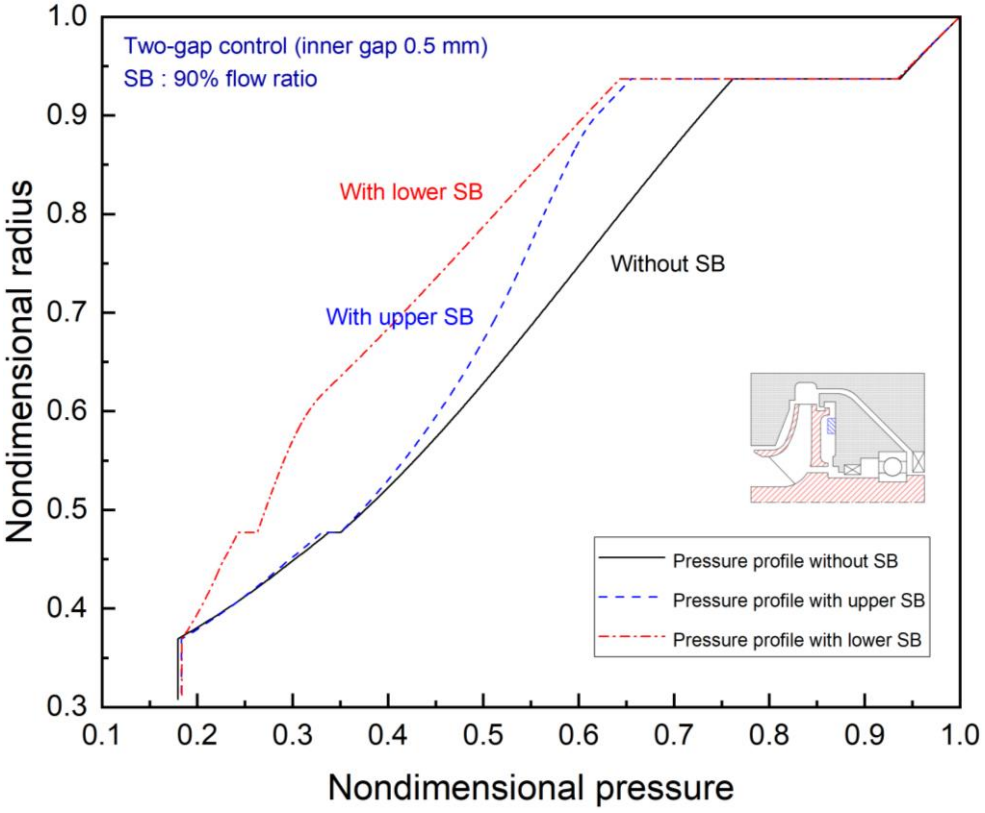


Figure 6 – Pressure profile of the impeller back side at inner gap 0.5 mm.

that the swirl breaker installed at the upper position, unlike the one at the lower position, does not affect the pressure in the region below the inner control gap.

Figure 6 shows the radial pressure profile on the impeller rear shroud when the inner control gap clearance is widened to 0.5 mm under the same analysis conditions as the previous case. In this case, the outer control gap clearance becomes 0.1 mm. Consequently, the balance chamber is closed at the top and open at the bottom, making it susceptible to the low pressure downstream of the inner control gap. As the inner control gap opens, the balance chamber is influenced by the low-pressure region. The placement of a swirl breaker introduces an effect that suppresses the flow swirl, leading to an observable additional pressure reduction compared to the case without a swirl breaker. It can be confirmed that when the swirl breaker is at the top, it only affects the pressure drop in the upper part of the balance chamber. However, when the swirl breaker is at the bottom, it influences the pressure drop across the entire region, from the upper part of the balance chamber down to the region below the inner control gap.

4. Conclusions and future work

A numerical analysis was performed to configure an active thrust balancing system for controlling the excessive axial thrust in turbopumps and to enhance its control performance. A one-dimensional analysis program was developed to obtain converged solutions for the pressure distribution and leakage flow rate corresponding to changes in the control gap clearance through iterative calculations. A swirl breaker was additionally incorporated into the balancing system to improve its load control performance, and the analysis results confirmed that the load control capability is increased due to the swirl breaker. Further parameter studies under various conditions will be carried out to determine the optimal design.

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