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Investigation on Active Axial Thrust Balancing for a Turbopump

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Abstract
A 10 ton class closed cycle engine of staged combustion using liquid oxygen and kerosene is being developed as the upper stage engine of the next KSLV(Korea Space Launch Vehicle) following model to increase thrust and specific impulse. Since turbopump pressure in a staged combustion closed cycle engine is largely increased, an active thrust balancing system has to be needed to control the turbopump axial force. Two types of active thrust balancing systems with one axial control gap and two axial control gaps were investigated through pressure and leakage flow analysis to improve the axial thrust control capability.

1. Introduction

A 7 ton class open cycle liquid rocket engine of gas generator type using liquid oxygen and kerosene has been developed and verified through serial ground hot firing tests to be applied as the third stage engine of the space launch vehicle in the KSLV(Korea Space Launch Vehicle)-II program. To increase thrust and specific impulse of the upper stage engine, a 10 ton class closed cycle engine of staged combustion type using the same propellants is going to be developed in the following KSLV-III program. Outlet pressure of turbopump in a staged combustion closed cycle engine is largely increased due to two stage combustion through preburner and main combustor sequentially compared to a gas generator open cycle engine with single stage combustion. The current tubopump for the KSLV-II presents 85 bar head rise in an oxidizer pump and 114 bar head rise in a fuel pump. Since the axial thrust developed by the turbopump is not quite high, a passive thrust balancing is applied so that ball bearings support the axial force. On the other hand, head rise of the next generation turbopump will be increased up to 260 bar in an oxidizer pump and up to 300 bar in a fuel pump. As the axial thrust increases with the higher pump pressure, bearing failure risk also increases due to excessive bearing axial load. Therefore, an active thrust balancing has to be needed to control the axial force.

There are two types of active thrust balancing systems. One is a balancing system with one axial control gap and the other is a balancing system with two axial control gaps [1,2]. A balancing system with one axial control gap is priorly considered for the newly developing turbopump due to easiness of design and manufacturing despite lower control capability. The other balancing system with two axial control gaps is also considered as an alternative model since it has superior control capability despite complexity of design and manufacturing in contrast to the balancing system with one axial control gap.

Up to now, several researches have been performed to investigate static characteristics of active balancing systems and design method has been well established through analytical approach. S. Maier et al. [2] carried out a research on the active balancing systems of the liquid rocket engine Lox turbopump using analytical and numerical methods. The fluid rotation ratio was calculated considering the throughflow effect of rotor-stator cavity internal flow in the analytical model and the pressure profile of the cavity was acquired reflecting the flow centripetal effect. Rotor and stator wall frictions and throughflow angular momentum in the cavity were taken into account to obtain the fluid rotation ratio as a function of radial position. T. Shimura et al. [3] calculated the axial thrust of rocket pump active balancing system with two axial control gaps reflecting the transported momentum into balance chamber and the wall boundary layer conditions. The analytical results are compared with those of the CFD calculation and the experimental data. Radial grooves installed on the stationary wall were studied to affect balancing performance by reducing fluid angular velocity as swirl brake.

In the present study, two types of active thrust balancing systems with one axial control gap and two axial control gaps respectively are applied for design of the following turbopump through pressure and leakage flow

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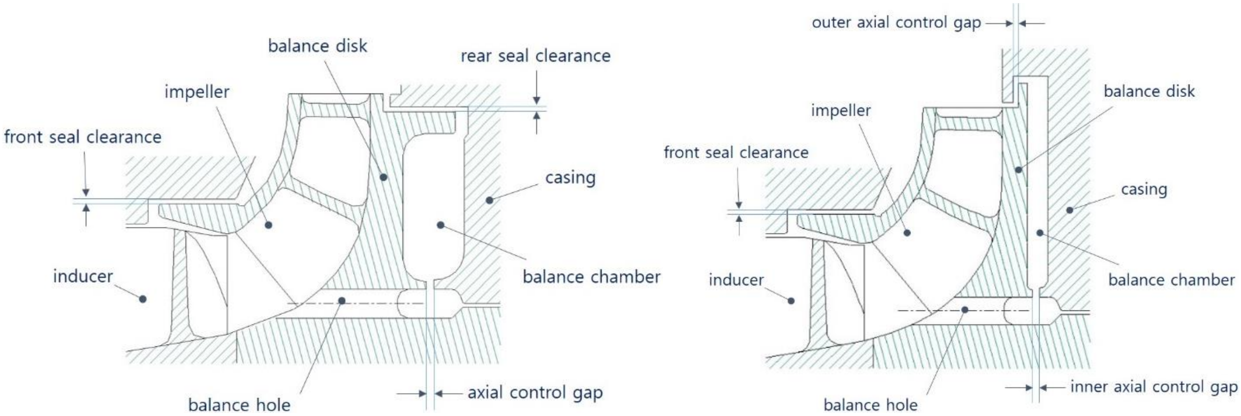
investigation using the analytical method based on references [2] and [4]. Axial force of the balancing system is obtained from the converged solution of pressure by matching the pump head rise to the pressure difference summation of total flow passages including balance chamber flow passage and additional secondary flow passage for bearing cooling flow. To decide force equilibrium position and movement range of a rotor system during the balancing, design parameters such as clearance and roughness of annular seal and axial control gap are investigated through the one dimensional flow analysis. A parametric research is performed to extend the axial thrust control span and improve the control sensitivity for various design parameters. Effect on the bearing cooling flow rate is also examined according to the axial thrust balancing design.

2. Axial Thrust Balancing Design and Analysis

The liquid rocket engines for the KSLV-II are gas generator type open cycle engines and the turbopumps use passive type control for thrust balancing. Although a passive type balancing system imposes an unbalanced force on a thrust bearing, the unbalanced axial force is not too big for the thrust bearing to endure because of relatively low outlet pump pressure for a gas generator type open cycle engine. The axial thrust is adjusted to a moderate level of force by radially positioning the shoulder seals of the impeller front and rear sides and installing the swirl brake on the casing. The thrust bearing is also verified through autonomous tests under the imposed axial force condition to ensure margin of safety and durability.

For the next generation of KSLV, a staged combustion type closed cycle engine is going to be applied to improve specific impulse of engine. Since the staged combustion cycle engine largely increases pump outlet pressure, an active type balancing system is required to deal with excessive axial force for various operating conditions. Active type balancing systems control axial thrust and automatically find balancing point by regulating leakage flow through axial gap of impeller back side. There are two types of active thrust balancing systems as shown in figure 1.



(a) Balancing with one control gap (b) Balancing with two control gaps

Figure 1: Active axial thrust balancing systems

A balancing system with one control gap has merit of simple design and easy manufacturing but lower control ability. As the clearance of the single control gap increases, the slope of axial force change decreases so that automatic axial thrust control is not working well in areas with large clearance. On the other hand, a balancing system with two control gaps has merit of superior control ability despite complex design and difficult manufacturing. If one control gap increases then the other control gap decreases in the balancing system with two control gaps therefore two-way control ability is acquired.

Axial force F imposed on the balance disk of figure 1 is obtained from equation 1 by integrating impeller back side pressure $\mathrm{p(r)}$ from impeller hub r_{hub} to impeller tip r_{tip}.

$$\mathrm{F} = 2\pi \int_{r_{hub}}^{r_{tip}} p(r) r dr \quad (1)$$

The impeller back side pressure $\mathrm{p(r)}$, balance chamber pressure of figure 1, is calculated from equation 2 by subtracting the fluid centrifugal term from the impeller outlet pressure.

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

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where $P_{r_{tip}}$ is the impeller outlet pressure, ρ the fluid density, ω the impeller rotational speed and k the fluid swirl ratio obtained from the fluid core angular velocity divided by the impeller rotational speed.

Using the analytical model of references [2] and [4], the fluid swirl ratio k is determined as a function of radial position considering the balance chamber configuration, the rotor and stator friction coefficients, the impeller rotational speed, the fluid viscosity and the leakage flow ratio in equation 3.

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

where $\bar{r}$ is the nondimensional radius $\bar{r} = r/r_{tip}$, Q the leakage flow rate, Re_u the rotational Reynolds number $Re_u = \omega r_{tip}^2/\nu$, ν the fluid kinematic viscosity and k_0 the constant swirl ratio when the leakage flow is zero. If the leakage flow exits, the angular momentum induced by the impeller outlet flow rotation is introduced into the balance chamber. Since the inflow angular momentum is preserved in the balance chamber, the swirl ratio k increases as the radial position goes down to the centre of rotor. Whereas if the leakage flow does not exist, the swirl ratio k remains constant k_0 because of no additional angular momentum. The constant swirl ratio k_0 is represented in equation 4.

$$k_0 = \frac{1}{1+\left(\frac{r_w}{r_{tip}}\right)^2\sqrt{\left(\frac{r_w}{r_{tip}}+5\frac{t_{ax}}{r_{tip}}\right)\left(\frac{c_{f,w}}{c_{f,r}}\right)}} \tag{4}$$

where r_w is the casing wall radius of balance chamber, t_{ax} the casing wall axial length of balance chamber, $c_{f,w}$ the shear friction coefficient of casing wall and $c_{f,r}$ the shear friction coefficient of rotor. The shear friction coefficient c_f is represented in equation 5 as given in Gülich [5].

$$c_f = \frac{0.136}{\left[-log_{10}\left(0.2\frac{\epsilon}{r_{tip}}+\frac{12.5}{Re_u}\right)\right]^{2.15}} \tag{5}$$

where ϵ is the equivalent sand roughness obtained from maximum roughness depth dividing by the equivalence factor.

The radially inwards volumetric leakage flow Q is represented at the axial control gap as given in equation 6.

$$Q_{axgap} = 2\pi r_i s \bar{c}_i \tag{6}$$

where r_i is the axial control gap inner radius and s is the axial control gap clearance. The leakage flow radial velocity $\bar{c}_i$ is calculated at the axial control gap inner radius as given in equation 7.

$$\bar{c}_i = \sqrt{\frac{\frac{2}{\rho}\Delta P_{axgap} - 2\omega^2\int_{r_i}^{r_a} k(r)^2 r dr}{\zeta_E\left(\frac{r_i}{r_a}\right)^2 + \frac{\lambda r_i}{2s}\left[1-\left(\frac{r_i}{r_a}\right)\right] + \zeta_A}} \tag{7}$$

where ΔP_{axgap} is the radially pressure drop at the axial control gap, r_a the axial control gap outer radius, ζ_E the axial control gap inlet loss coefficient, λ the axial control gap friction coefficient and, ζ_A the axial control gap outlet loss coefficient. The leakage flow rate is obtained from the axial control gap pressure drop after subtracting the centrifugal effect radially then dividing by total resistance of the axial control gap. The radial pressure distribution at the axial control gap is given in equation 8.

$$P_{axgap}(r) = P(r_a) - \frac{\rho}{2}\bar{c}_i^2\left[\zeta_E\left(\frac{r_i}{r_a}\right)^2 + \frac{\lambda r_i^2}{2s}\left(\frac{1}{r}-\frac{1}{r_a}\right) + \zeta_A\left(\frac{r_i}{r}\right)^2 + \frac{2\omega^2}{\bar{c}_i^2}\int_r^{r_a} k(r)^2 r dr\right] \tag{8}$$

One dimensional analysis program for active thrust balancing with one axial control gap is conducted on the flow chart shown in figure 2. Pressure drops of each element such as impeller rear annular seal, balance camber and axial control gap are calculated using leakage flow rate, clearance configuration, fluid property, friction coefficients and swirl ratio. After summation of the pressure drops, the total pressure drop of all the elements is compared with the impeller pressure rise. Until the difference between the total pressure drop of all the elements and the impeller pressure rise decreases to a desired value, the one dimensional analysis is iteratively conducted to obtain the converged leakage flow rate and radial pressure profile.

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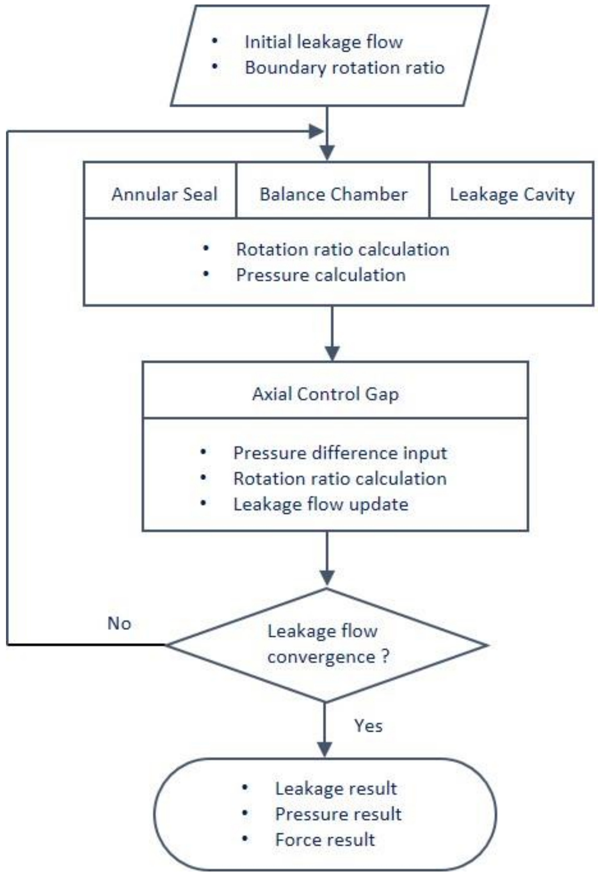

Figure 2: Analysis program flow chart

As for active thrust balancing with two axial control gaps, the converged solution is also obtained in the same way as the previous described analytical method. However, the impeller rear annular seal element is replaced by the outer axial control gap element to apply two axial control gaps. Leakage flow direction in the outer axial control gap is outwards opposite to inwards direction in the inner axial control gap therefore the centrifugal effect of leakage flow is applied in reverse to the clearance pressure of the outer axial control gap. Axial force direction imposed on the balance disk in the outer axial control gap is also opposite to that in the balance chamber so that careful consideration is taken into account to calculate the axial thrust.

3. Results and Discussion

3.1 Analysis Verification

One dimensional analysis program is developed and the calculated results are compared with those of reference [2]. Two types of balancing systems with one axial control gap and two axial control gaps are computed and compared for verification of the developed program. The liquid oxygen pump with 45 bar pressure rise is examined at 10,000 rpm. The numerical results compared with those of reference [2] are shown in figure 3 for the balancing system with one control gap and in figure 4 for the balancing system with two control gaps, respectively.

As for the balancing system with one control gap, axial force decreases and leakage flow rate increases as clearance of the one control gap increases as depicted in figure 3. On the contrary, if clearance of the one control gap decreases, axial force increases and leakage flow rate decreases. As for the balancing system with two control gaps, axial force also decreases as clearance of the inner control gap increases with the similar trend of the balancing system with one control gap but the sensitivity of axial force depending on clearance is higher. Leakage flow rate shows a different aspect that it has the smallest amount when any of axial control gaps is closed as the impeller extremely moves in either direction of front and rear sides. It reaches maximum value where the impeller is placed around the middle position of movement width at the state of similar clearances in the both of inner and outer control gaps. The present analysis results show good agreements with the analysis and CFD results of reference [2].

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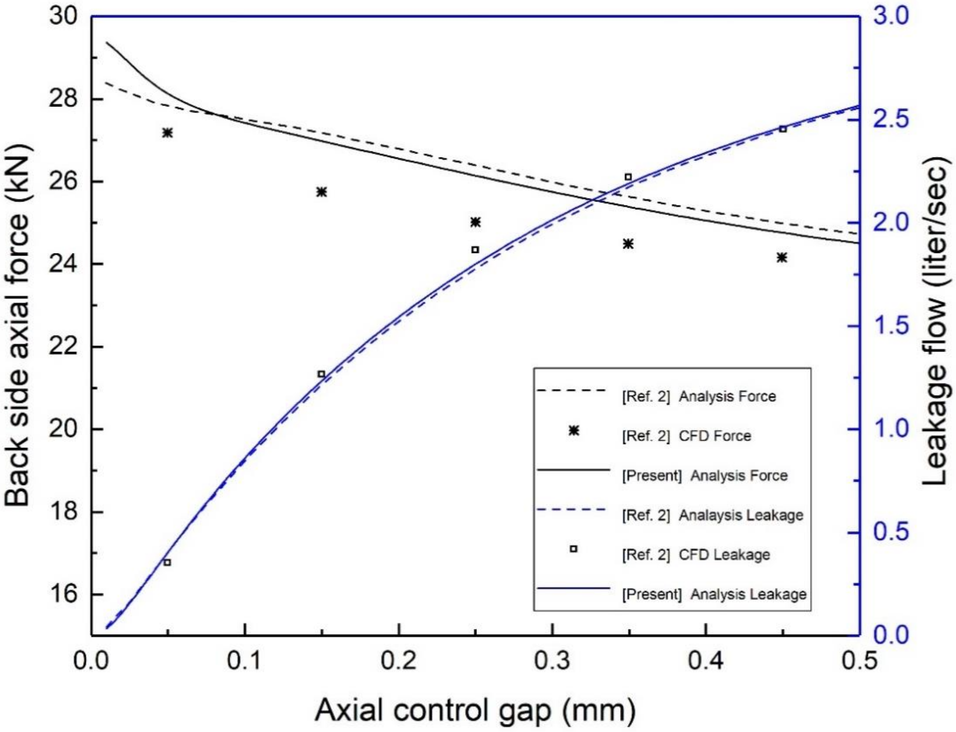

Figure 3: Axial force and leakage flow comparison for a balancing system with one control gap

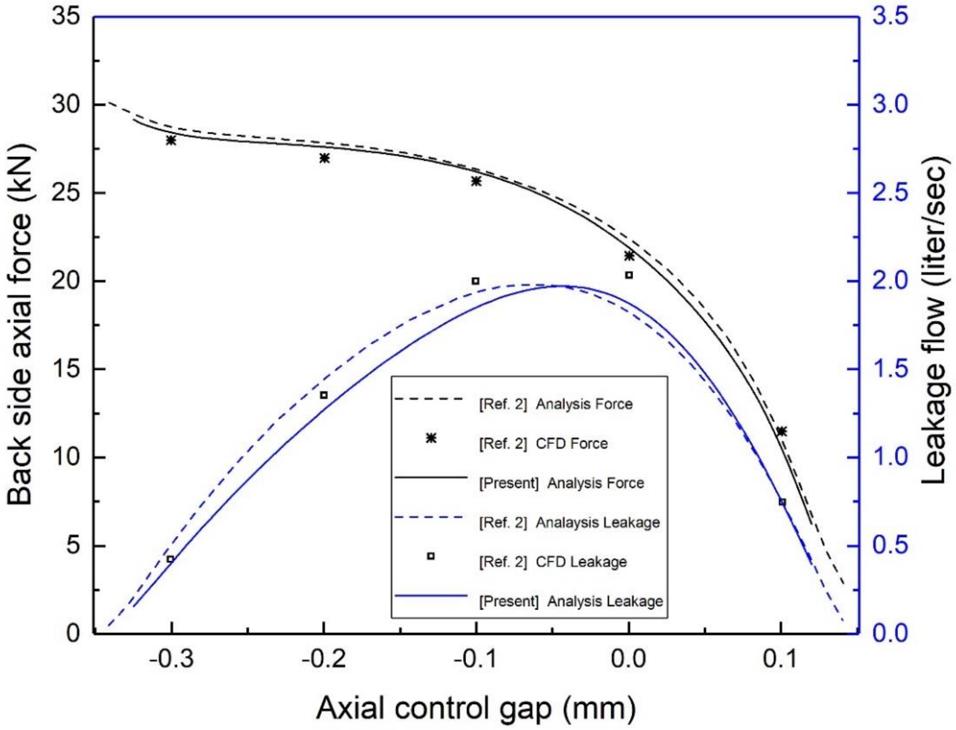

Figure 4: Axial force and leakage flow comparison for a balancing system with two control gaps

3.2 Active Axial Thrust Balancing System Design with One Control Gap

An active axial thrust balancing system with one control gap is preferentially designed from a base model with a passive axial thrust balancing system. Figure 5 shows two balancing systems with leakage flow path for a passive type

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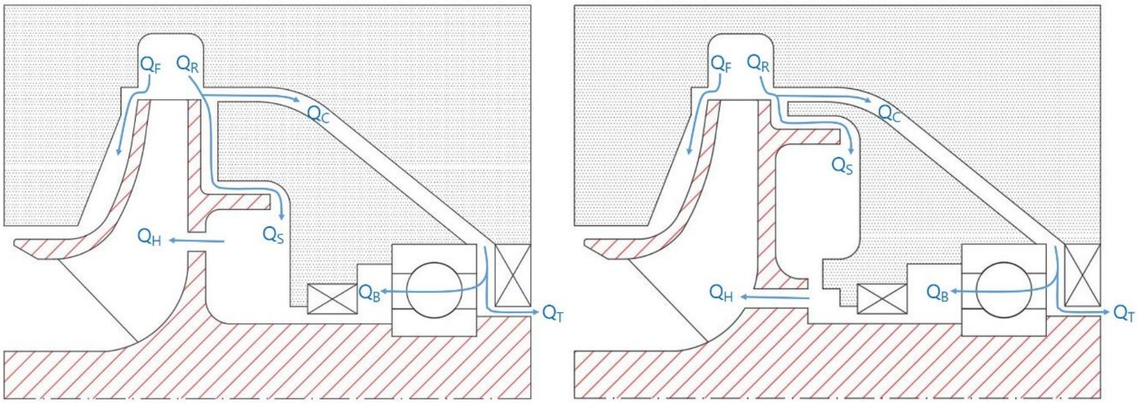
and an active type, respectively. Due to balance chamber placed between rear shoulder seal and axial gap, balance hole for return flow in the active type is positioned lower than that in the passive type. In addition to a flow path through rear shoulder seal, another flow path through bearing for cooling is also described in the figure and considered in analysis. One dimensional analysis is performed for an oxidizer pump using liquid oxygen and the pump builds 260 bar pressure rise and operates at 29,000 rpm.



(a) Passive balancing system (b) Active balancing system

Figure 5: Passive and active axial thrust balancing systems with leakage flow path

Figures 6 and 7 show nondimensional axial force and leakage flow ratio as a function of clearance of axial control gap for the active axial thrust balancing system with one control gap. The impeller front side force and the back side force for the passive axial thrust balancing system of the base model are also presented for comparison in figure 6. In the same manner, the impeller front seal leakage ratio and the rear seal leakage ratio for the passive type are compared with the results of the active type in figure 7.

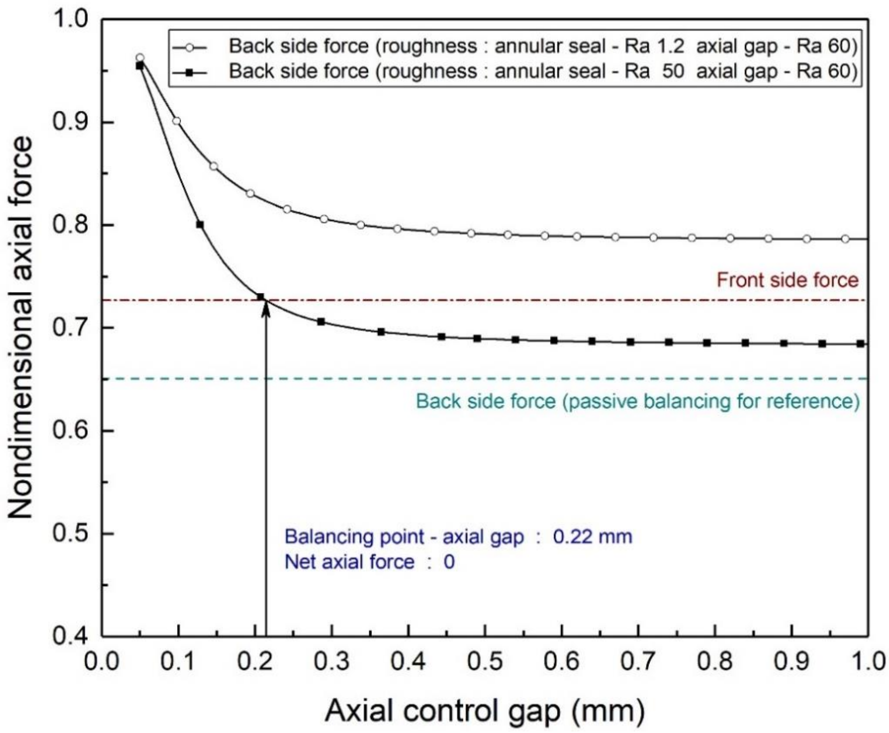


Figure 6: Axial force of active axial thrust balancing system with one control gap

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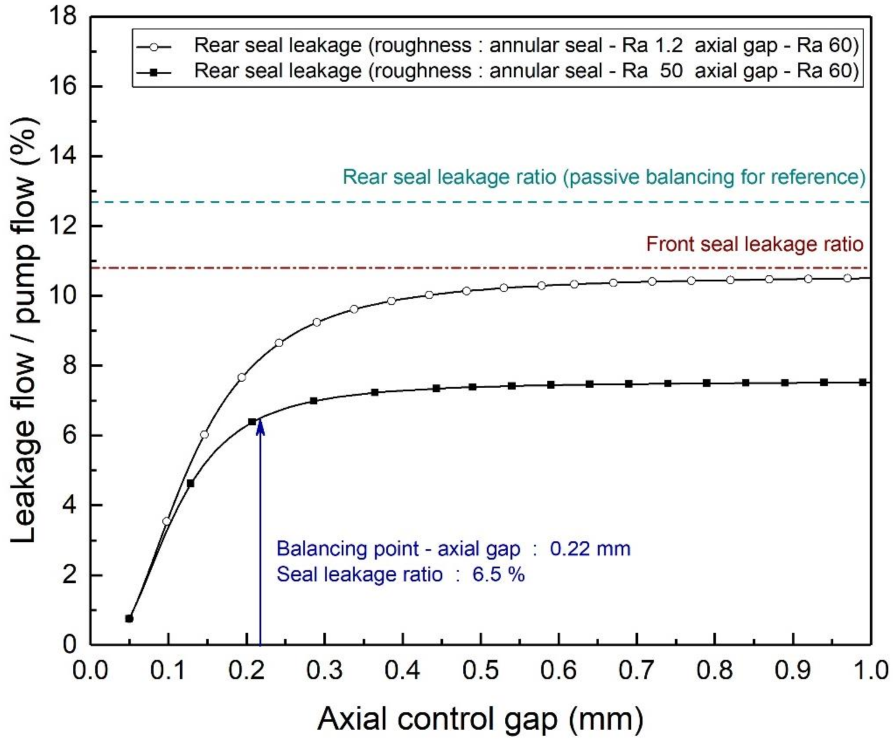

Figure 7: Seal leakage flow ratio of active axial thrust balancing system with one control gap

As for the active type, two cases are investigated according to surface roughness variation of the rear shoulder seal and axial control gap. The first case is for surface roughness Ra 1.2 of the rear shoulder seal and Ra 60 of the axial control gap. The second case is for surface roughness Ra 50 of the rear shoulder seal and Ra 60 of the axial control gap. Surface roughness Ra 1.2 is chosen for machined surface. Surface roughness Ra 50 and Ra 60 are chosen to impose proper resistance on leakage flow.

It shows very large axial force due to high pressure preservation in the balance chamber where the axial control gap is very small. As axial control gap gradually increases, leakage flow increases and pressure of the balance chamber decreases therefore axial force decreases. The narrow clearance area where axial force and leakage flow are changing with respect to axial control gap variation is the possible range for active axial thrust balancing. If axial control gap exceeds certain level, axial force and leakage flow are not changing anymore. Those are converging to some level instead because the axial control gap loses control ability due to very small gap resistance in the wide clearance area. Thus axial thrust and leakage flow are determined by resistance of the rear shoulder seal instead of the axial control gap.

The active thrust balancing point is decided as the back side force equals to the front side force by clearance adjustment of the axial control gap. As shown in figure 6, there is no balancing point in the case of rear shoulder seal with smooth roughness Ra 1.2 since the back side force is larger than the front side force in entire range of axial control gap. At unbalanced state of axial force, bearing has to support all net axial thrust. However, if the rear shoulder seal has coarse roughness Ra 50, a balancing point exists because the back side force decreases further at the fully open state of axial control gap compared to the smooth roughness case of the rear shoulder seal. The axial thrust balancing is obtained around 0.22 mm clearance of the axial control gap in the coarse roughness Ra 50 case of the rear shoulder seal and the net axial force is zero at the balancing state. Instead of adjusting the roughness increase of the rear shoulder seal, the similar result can be obtained by reducing the radial clearance of the rear shoulder seal. From an initial state with 0.5 mm clearance of the axial control gap, the impeller firstly moves backwards in the direction of decreasing clearance because the front side force is larger than the back side force at the initial gap position. After the first movement of the impeller, the back side force increases due to narrow clearance of the axial control gap so that the impeller starts to move forwards reversely to reach the balancing point with clearance 0.22 mm of the axial control gap. The leakage flow rate is about 6.5 % of the pump total flow rate at the balancing point that is a reduced value with approximately half of the leakage flow rate for the base model with the passive balancing system.

During the automatic balancing, variation of bearing cooling flow rate is also a matter of interest and the result is represented according to clearance change of the axial control gap in figure 8. As shown in the result, the bearing cooling flow rate is not affected by the seal surface roughness and clearance change of the axial gap and represents

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almost constant value with about 8 % of the pump total flow rate. The early passive balancing model also has a similar bearing cooling flow rate with 7.9 %.

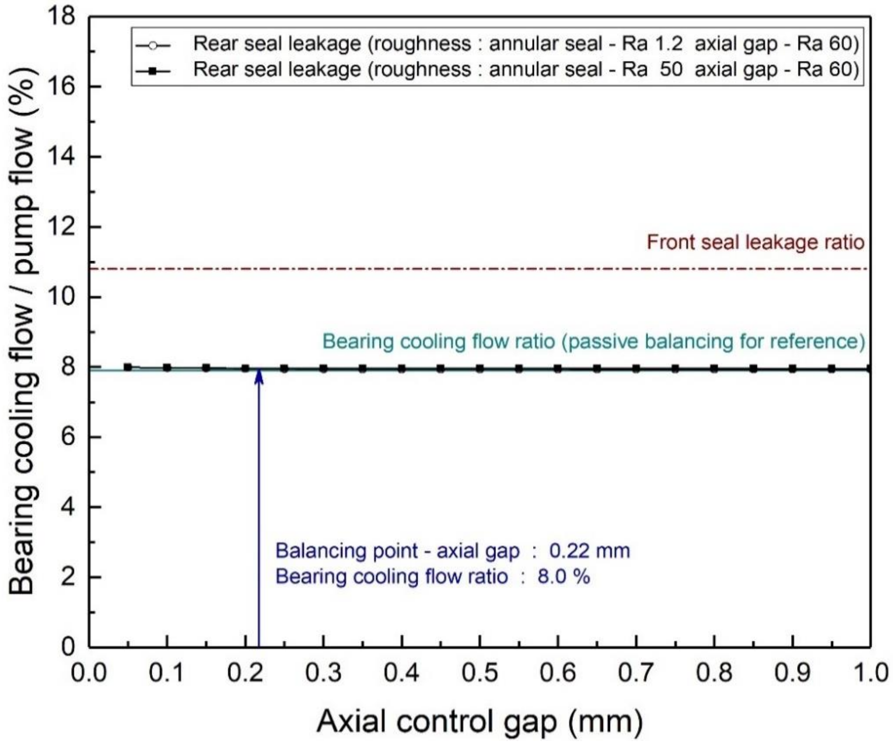
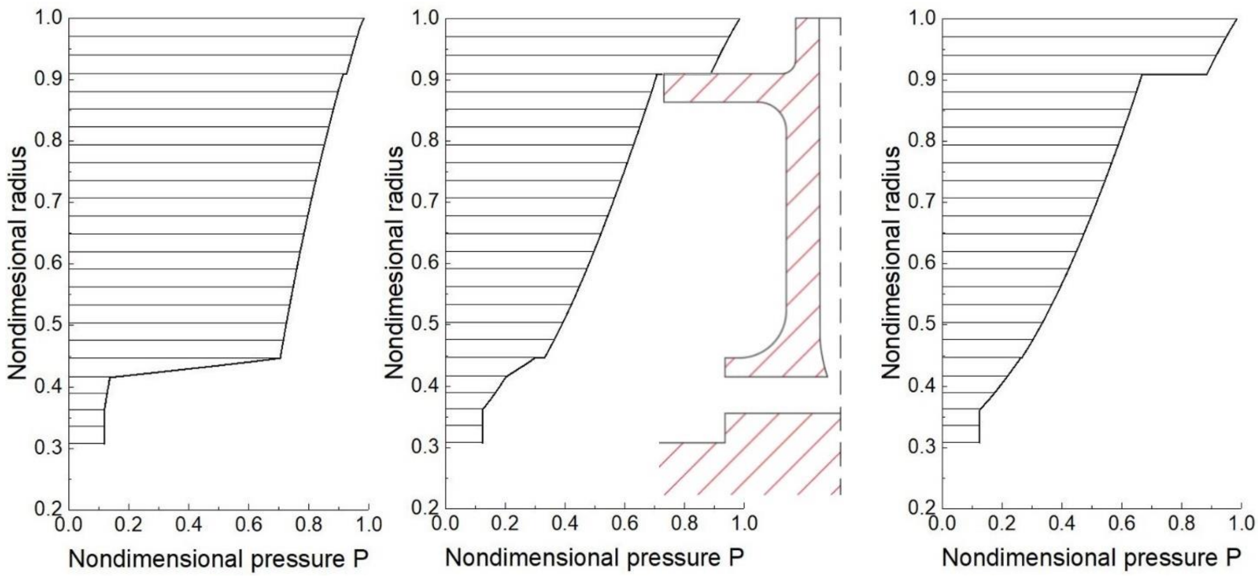


Figure 8: Bearing cooling flow ratio of active axial thrust balancing system with one control gap



(a) small gap (b) balancing gap (c) large gap

Figure 9: Pressure profile of impeller back side at various clearances of axial control gap for active axial thrust balancing system with one control gap

Figure 9 shows pressure profiles of the impeller back side with respect to the nondimensional radius normalized by the impeller tip radius for three clearance cases of small gap 0.05 mm, balancing gap 0.22 mm and large gap 0.7mm. The back side pressure decreases as the radial position decreases from the impeller tip to the rear shoulder seal due to centrifugal effect. The result shows the pressure drop through the rear shoulder seal and then the pressure decreases again from the rear shoulder to the axial control gap. At the entrance of the axial control gap, the pressure drop appears

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due to inlet loss effect and the pressure gradually decreases as the leakage flow passes through the axial control gap and balance hole. The pressure decreases as the radial position decreases because the centrifugal force of the fluid decreases as the radius decreases. The constant pressure is formed from the hub to the bottom of the balance hole.
The pressure profile of the impeller back side changes depending on clearance of the axial control gap under the same pressure rise of the impeller. If the clearance of the axial control gap is small, most pressure drops appear in the axial control gap and pressure drops in the balance chamber above the axial control gap, including the rear shoulder seal, are relatively small. On the other hand, if the clearance of the axial control gap is large, pressure drop decreases in the axial control gap and increases in the balance chamber and the rear shoulder seal. This is because the larger the clearance of the axial control gap, the lower the resistance to the flow rate, indicating that the axial control gap can not control the flow rate anymore. Since the effect of the axial control gap becomes insignificant as the clearance increases, the pressure profile of the impeller back side converges to a uniform shape. As a result, the axial thrust and the leakage flow rate also show convergent values.

3.3 Active Axial Thrust Balancing System Design with Two Control Gaps

An active axial balancing system with two control gaps can be formed by adding an outer axial control gap instead of a shoulder seal above the rear of the impeller. In this system, when one control gap becomes smaller or larger, the other control gap becomes larger or smaller on the contrary, so gap control is performed in both directions. Therefore, the loss of control ability, which is a disadvantage revealed in the active balancing system with one control gap, does not appear in the active balancing system with two control gaps.

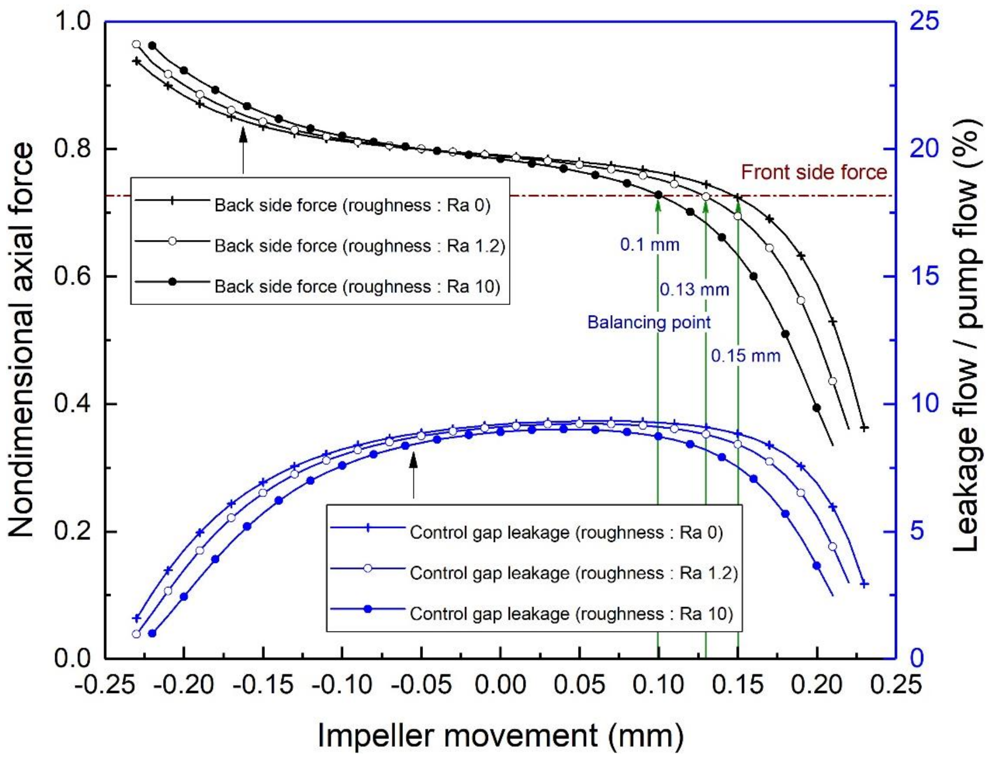

Figure 10: Axial force and leakage flow ratio of active axial thrust balancing system with two control gaps for three roughness cases

Figure 10 shows the nondimensional axial force and leakage flow ratio calculated by changing the inner control gap for an active balancing system with inner and outer axial control gaps that can move by $\pm$0.25 mm, respectively. In the graph, the x-axis represents the movement of the impeller. If it moves from the neutral position in the (-) direction, the inner control gap decreases, and if it moves in the (+) direction, the inner gap increases. That is, the inner gap is 0 mm at the x-axis coordinate -0.25 mm and +0.5 mm at the x-axis coordinate +0.25 mm. At this time, the outer gap changes to the opposite of that of the inner gap. As in the case of single control gap, if the inner gap gradually increases from the closed state, the back side force gradually decreases and the leakage flow ratio gradually increases. When the inner gap and outer gap approach the neutral position, the amount of change in back side force and leakage flow ratio decreases, indicating a flat value. When the inner gap increases again from the neutral position, the back side force

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decreases and the leakage flow ratio also decreases. This trend does not appear in the case of single control gap. In the case of single control gap, when the control gap becomes larger than a certain level, there is no further change in the back side force and leakage flow ratio. However in the case of two control gaps, the outer control gap decreases while the inner control gap increases so that the back side force and leakage flow ratio can be consistently controlled by the outer control gap. Figure 10 also shows the calculation results for three cases of Ra 0, Ra 1.2, and Ra 10 in order to examine the effect of the roughness of the control gap on the balancing behavior. It can be seen that as the roughness of the control gap increases, the slope of the back side force change becomes steeper and the leakage flow ratio decreases. The point at which the force is balanced in the case of Ra 0 is when the impeller moves +0.15 mm and the inner control gap becomes 0.4 mm. The balancing position of the Ra 1.2 case is at +0.13 mm impeller movement, i.e. 0.38 mm inner control gap, and that of the Ra 10 case is at +0.1 mm impeller movement, i.e. 0.35 mm inner control gap. As for the control gap where the force is balanced, the inner gap decreases and the outer gap increases as the roughness increases. At the balancing position, the leakage flow through the balance chamber shows a similar value of 8.7 to 8.8 % in all three roughness cases. If the surface roughness of the control gap is processed to an appropriate level, it is thought that the desired automatic balancing performance can be obtained.

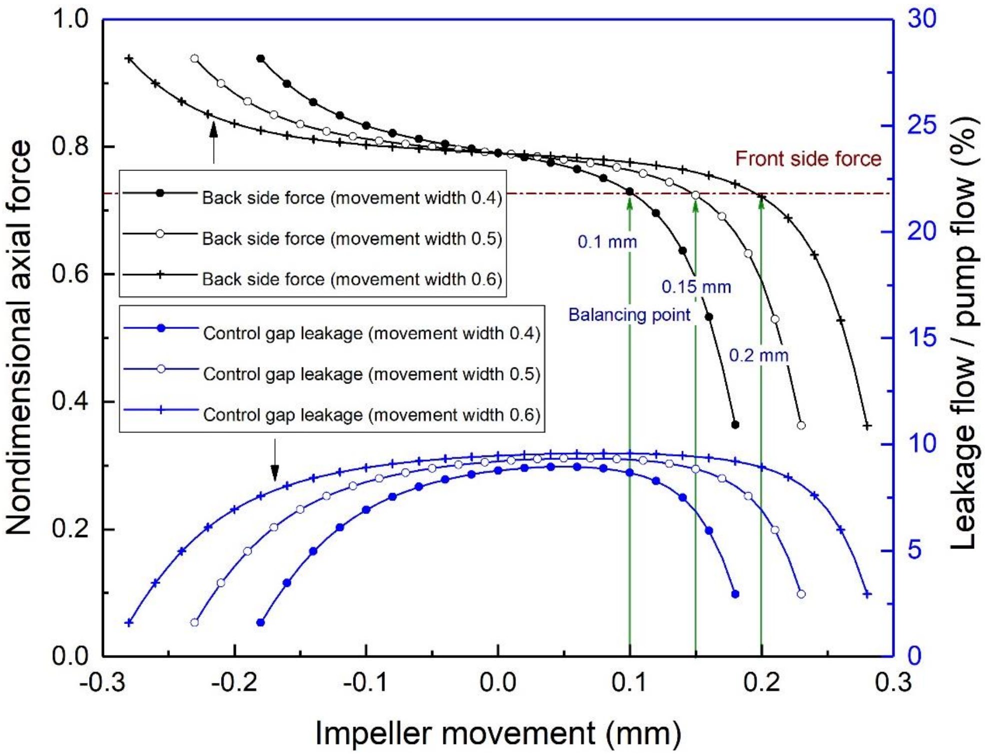

Figure 11: Axial force and leakage flow ratio of active axial thrust balancing system with two control gaps for three axial movement cases

Figure 11 shows the calculated nondimensional axial force and leakage flow ratio by increasing the control gap width where the impeller can move to $\pm$0.2 mm (total 0.4 mm), $\pm$0.25 mm (total 0.5 mm) and $\pm$0.3 mm (total 0.6 mm) in the case of control gap roughness Ra 0. The back side force change shows the steepest slope when the control gap has a travel width of $\pm$0.2 mm (total 0.4 mm). The point at which the force is balanced is when the impeller moves +0.1 mm and the inner control gap becomes 0.3 mm (outer control gap 0.1 mm) in the case of the control gap width $\pm$0.2 mm (total 0.4 mm). The balancing position is obtained when the impeller moves +0.15 mm and the inner control gap is 0.4 mm (outer control gap 0.1 mm) in the case of gap width $\pm$0.25 mm (total 0.5 mm), and when the impeller moves +0.2 mm and the inner control gap is 0.5 mm (outer control gap 0.1 mm) in the case of gap width $\pm$0.3 mm (total 0.6 mm), respectively. In all the cases, it can be seen that balancing is achieved when the outer control gap is 0.1 mm. The leakage flow rates flowing to the rear of the impeller at the balancing positions show similar values of 8.7 to 8.9 %. As the movement width of the impeller increases to $\pm$0.25 mm (total 0.5 mm) and $\pm$0.3 mm (total 0.6 mm), the back side force becomes flat near the neutral position of the control gap and the balancing point cannot be determined as a single point. Therefore, the impeller axial position for the force equilibrium may become unstable.

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4. Conclusion

In order to investigate the active thrust balancing performance of the turbopump, parametric research was performed on the case with one axial control gap and the case with two axial control gaps. One dimensional analysis program was developed to analyse pressure and leakage flow for the active balancing systems. The balancing point at which the force equilibrium occurs was obtained from the numerical analysis. In the case of an active balancing system with one axial control gap, as the single gap increases beyond a certain level, the balancing control ability decreases. Moreover, if the leakage flow resistance of the impeller rear shoulder seal is too small, it may not be possible to find a balancing point where the force equilibrium is established. In the case of an active balancing system with two axial control gaps, it can be seen that as the leakage flow resistance of the control gaps increases and the movement width decreases, the slope of change for the axial force and leakage flow becomes steeper. If the axial movement width of the impeller increases too much during the balancing, the axial force becomes flat near the neutral position, which may make it difficult to find the balancing point as a single point.

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