Development of Destabilization force Reduction Structure for Industrial Centrifugal Compressors

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In the field of turbomachinery such as centrifugal compressors and steam turbines, the aim has been to reduce the size and number of stages by increasing the speed and load. As a result, however, the rotordynamics stability margin tends to decrease due to the increase in the fluid destabilization force (hereinafter referred to as destabilization force) generated in the impeller and blade area, as well as the sealing areas. In order to increase the speed and load of turbomachinery, it is indispensable to improve the rotordynamics stability by reducing the destabilization force. On the other hand, it is difficult to predict the destabilization force, which is related to the flow phenomena in a narrow area and small flow rate. Therefore, we have developed a structure for reducing the destabilization force for centrifugal compressors using numerical analysis technology.

1. Introduction

As plant capacity increases, the size of turbomachinery such as centrifugal compressors and steam turbines operated in plants also increases and accordingly the installation space and operating costs tend to increase. In order to reduce these factors, it is effective to reduce the size and the number of stages by increasing the speed and load of turbomachinery. However, in order to do so, a structure to reduce the destabilization force is necessary for the purpose of avoiding an increase in destabilization force and the decrease in rotordynamics stability margin due to the higher speed and load.

The structure of a centrifugal compressor is as shown in **Figure 1**. The suctioned gas is boosted by the rotation of the impeller fastened to the rotor and is discharged through the diffuser and scroll downstream of the impeller. Part of the gas discharged from the impeller flows into the shroud gap, passes through the seal fin installed in the impeller eye and then mixes with the mainstream gas.

In some cases of centrifugal compressors and steam turbines with seal fins, the destabilization force is generated from the seal fins and it is known that the destabilization force is roughly proportional to the swirl velocity of the gas flowing into the seal fins[(1)]. **Figure 2** gives an example of installing a swirl canceller as a structure that reduces the destabilization force in a centrifugal compressor. The swirl canceller has a structure in which part of the gas in the scroll flows into the seal fins in the direction opposite to the swirl direction to reduce the swirl brakes of the gas passing through the seal fins. The gas that has flowed into the seal fins from the scroll through the swirl canceller reflows into the impeller and therefore the workload of the impeller is increased. As a result, the power is increased. Since the increase in power leads to an increase in operating costs, it is necessary to minimize the increase in power by applying a destabilization

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force reduction structure.

This report presents the development of a destabilization force reduction structure that suppresses the increase in power due to the application of numerical analysis technology.

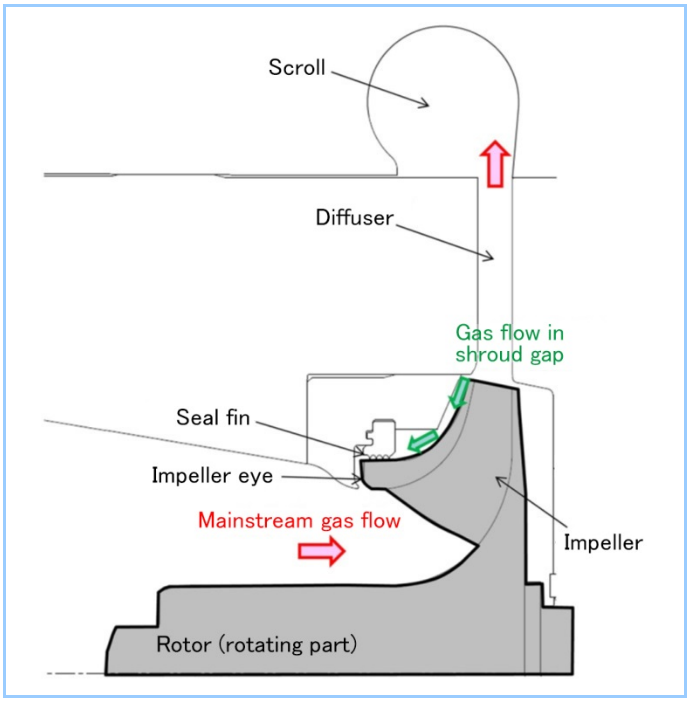

Figure 1 Centrifugal compressor

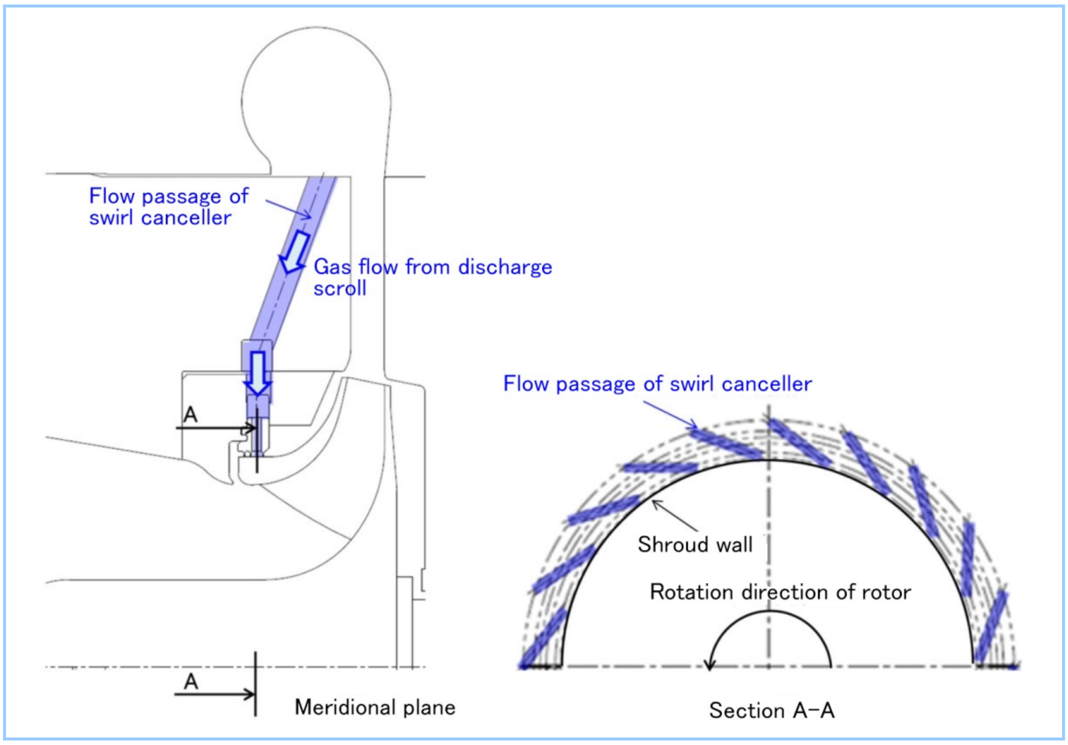

Figure 2 Swirl canceller

2. Examination details

2.1 Examination target

Figure 3 depicts the centrifugal compressor examined and **Table 1** lists its main specifications. This centrifugal compressor is a motor-driven multi-shaft, multi-stage (5-stage configuration) type with a built-in speed-increasing gear. Each stage is driven by the power of the motor transmitted to each pinion shaft via the speed-increasing gear. The examination described in this report was conducted for the 5th stage, where the discharge pressure is the highest, at about 5.8 MPa.

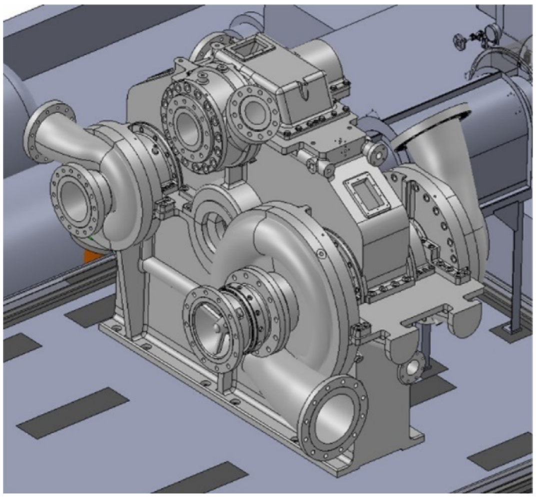

Figure 3　Multi-shaft, multi-stage centrifugal compressor with built-in speed-increasing gear

Table 1　Main specifications of centrifugal compressor

Number of stages	5 stages
Process Gas	Air
Discharge pressure	About 5.8 MPa

2.2　Mechanism of destabilization force generation

As shown in **Figure 4**, when the eccentricity of the rotor occurs in the positive direction (toward θ=0 degree) of the X-axis for some reason, the clearance of the sealing fins of the impeller eye becomes small in a certain region (around θ=0 degree) and large in another region (around θ=180 degree). Since the clearance distribution becomes non-uniform circumferentially, the flow rate distribution of the fluid passing through the seal fins becomes non-uniform in the same manner and so does the static pressure distribution. The phase of the static pressure distribution deviates from the phase of the clearance distribution and the degree of the phase deviation changes depending on the swirl velocity of the fluid passing through the seal fins. A force acting from a region with high static pressure to a region with low static pressure (force pushing the rotor in the positive direction of the Y-axis) is generated and this becomes a destabilizing force (destabilization force F) that vibrates the rotor, which reduces the rotordynamics stability margin. Furthermore, the circumferential static pressure distribution generated by the seal fins propagates through the upstream shroud gap, so that the circumferential static pressure distribution also occurs in the shroud gap, which causes the generation of destabilization force as in the case of the seal fins, resulting in a further reduction of the rotordynamics stability margin.

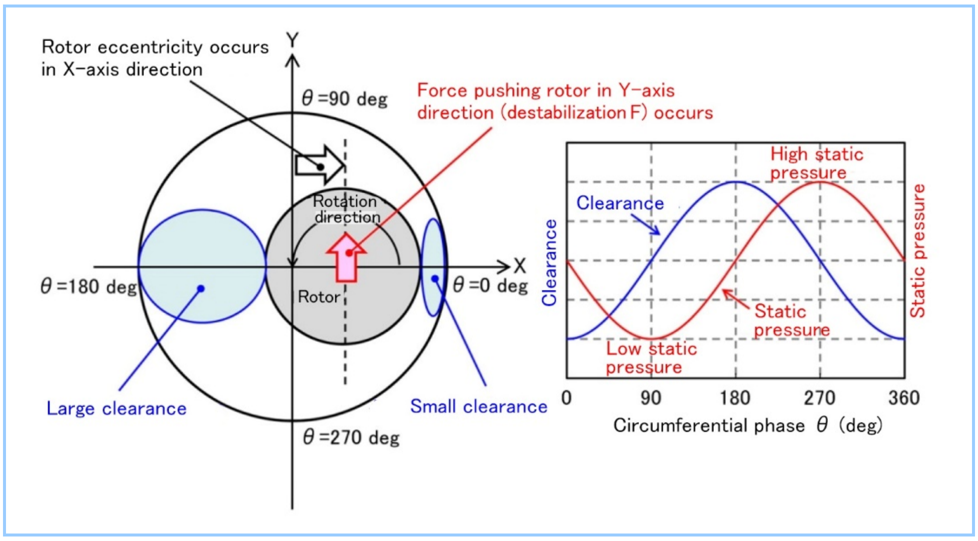

Figure 4　Schematic of circumferential clearance distribution and static pressure distribution caused by rotor eccentricity

2.3　Numerical analysis method

In order to evaluate the destabilization force of the examined centrifugal compressor, the internal flow was evaluated using computational fluid dynamics (CFD). As shown in **Figure 5**, the analysis was conducted for the shroud gap and the seal fin area, using a computational grid of about

45 million nodes. In order to simulate the circumferentially non-uniform seal clearance caused by the rotor eccentricity, the analysis was performed with the shroud wall surface made eccentric by 10% of the seal clearance. This numerical analysis was performed using the general-purpose three-dimensional viscous flow analysis code ANSYS CFX and the turbulence model SSTk-ω.

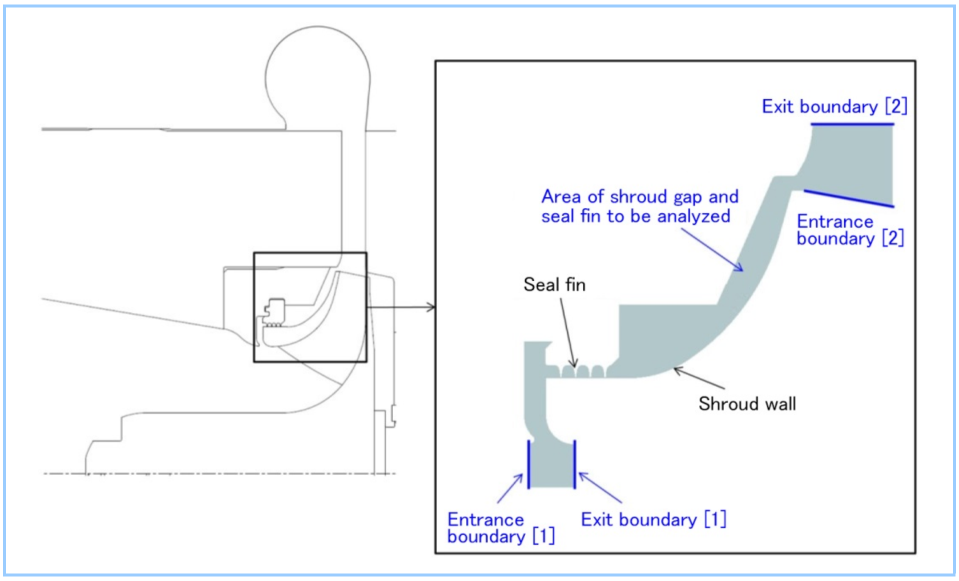

Figure 5 Analysis model

3. Results of examination

3.1 Making destabilization force reduction structure specific

As described in chapter 1, it is necessary to reduce the swirl velocity of the gas passing through the seal fins in order to reduce the destabilization force occurred at the seal fins and the shroud gap. Therefore, in this research, we have made a structure specific that reduces the swirl velocity of gas passing through the seal fins by installing a swirl brakes (baffle plate) in the shroud gap.

Figure 6 illustrates the swirl brakes shape examined this time. To establish a structure where the swirl brakes is installed on the inner diameter side of the shroud gap, the location of the swirl brakes was determined based on the following two points: (1) it is estimated that toward the inner diameter side of the shroud gap, the flow path area becomes narrower and the ratio of the flow path area occupied by the swirl brakes increases, so the pressure loss during gas passage increases and the swirl velocity reduction effect increases and (2) due to the swirl velocity reducing effect of the swirl brakes, the area where the difference in rotation speed between the gas and the shroud wall surface occurs is made as narrow as possible, which reduces the friction loss. Numerical analysis was conducted for this shape and an evaluation of destabilization force and power was performed.

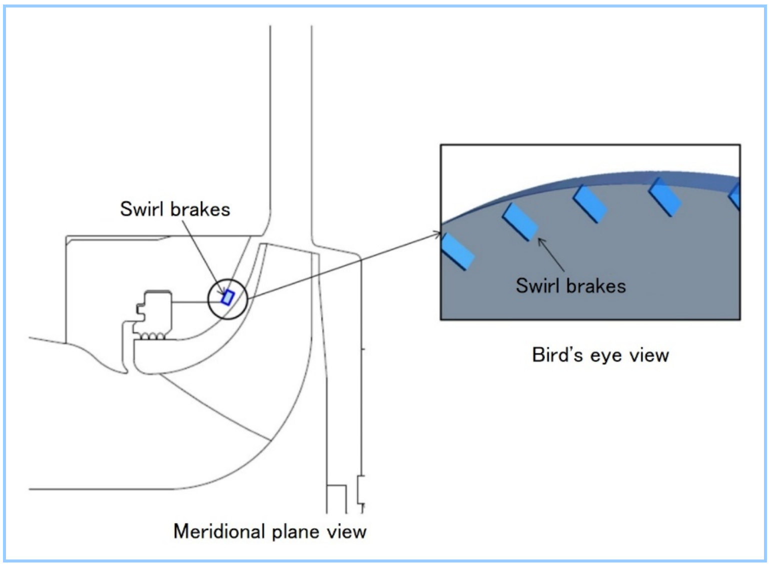

Figure 6 Swirl brakes

3.2 Evaluation results of applying destabilization force reduction structure

Figure 7 compares the destabilization force with and without the swirl braked. It was confirmed that the swirl brakes significantly reduces the destabilization force when present.

Figure 8 compares the swirl velocity distribution in the shroud gap with and without the swirl brakes. It can be seen that in the case where the swirl brakes are installed, by installing it on the inner diameter side, the swirl velocity was reduced from the position of the swirl brakes until the gas flows into the swirl brakes. As such, a high swirl velocity (a state where the difference from the rotation speed of the shroud wall surface is small) is maintained and friction loss is reduced, as expected.

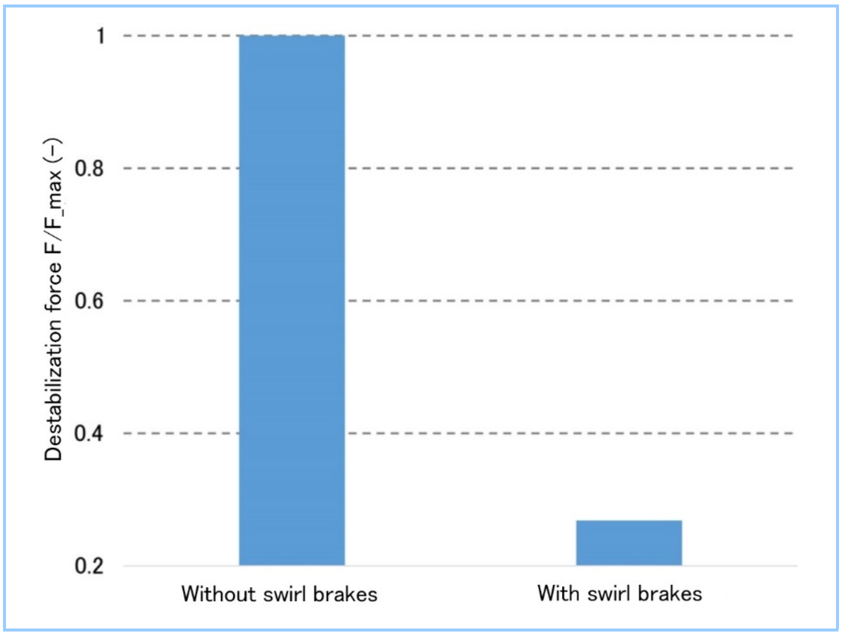

Figure 7 Comparison of destabilization force with and without swirl brakes

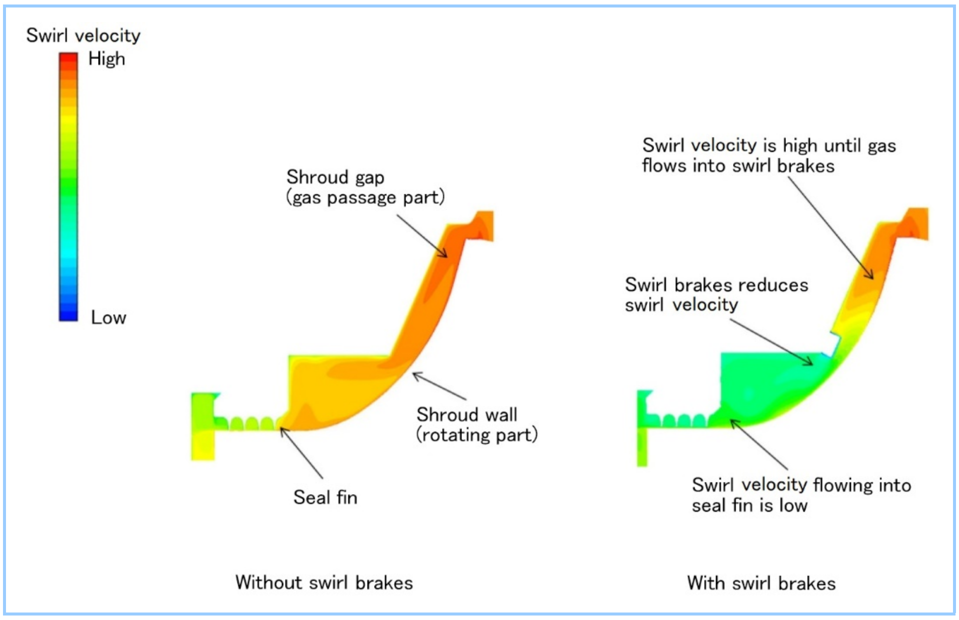

Figure 8 Swirl velocity distribution in shroud gap

Figure 9 compares power with and without the swirl brakes. It was confirmed that in the case where the swirl brakes are installed, the power increase was about 0.6% compared to the case were thr swirl brakes are not installed. In the case where the swirl brakes are installed, there is almost no increase in the flow rate of gas passing through the seal fins that occurs when a swirl canceller is installed and it is considered that the increase in power due to the increase in the workload of the impeller is small.

Figure 10 summarizes the relationship between the power and the destabilization force obtained in this examination. Although the description is omitted in this report, this figure also gives the analysis results in the case where a swirl canceller—a conventional destabilization force reduction structure—was shown. It was confirmed that in the case where a swirl canceller installed, the destabilization force was reduced but it was accompanied by a large increase in power as a contradictory event and that in the case where the swirl brakes are installed in contrast, the destabilization force can be greatly reduced while suppressing the increase in power.

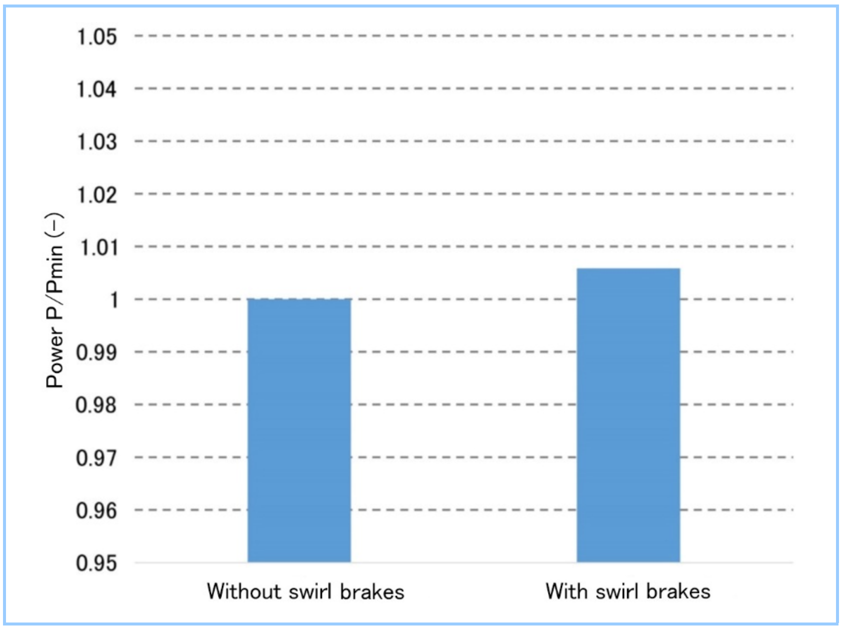

Figure 9 Comparison of power with and without swirl brakes

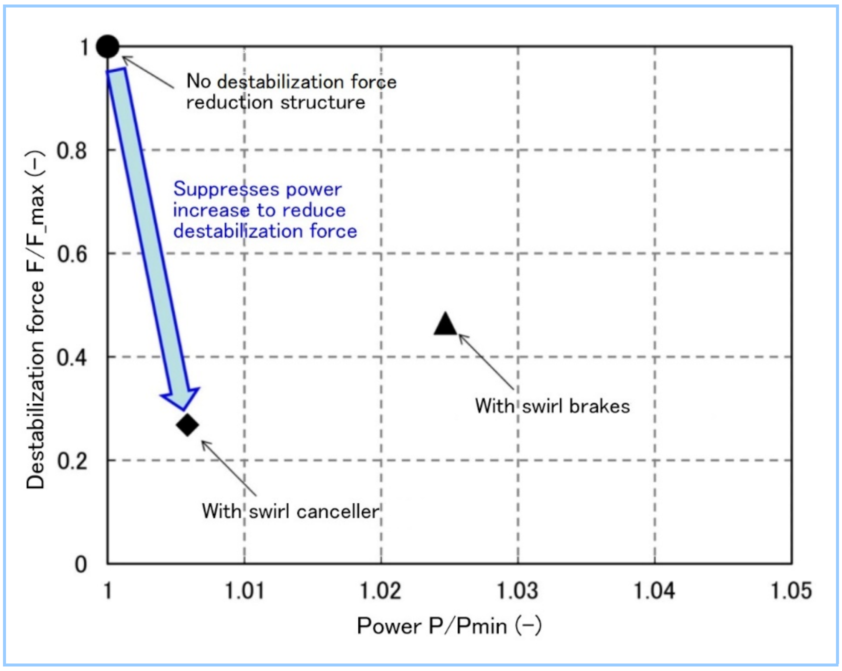

Figure 10 Relationship between power and destabilization force with and without destabilization force reduction structure

4. Conclusion

We have developed a swirl brakes structure for centrifugal compressors that can greatly reduce the destabilization force while suppressing the increase in power by using numerical analysis technology. In the future, we would like to promote understanding of the internal flow of the impeller shroud gap and swirl brakes by using elemental tests and the confirmation of the effect of installing the swirl brakes, in order to work on the verification of the validity of numerical analysis technology and the development of further destabilization force reduction technology.

References

(1) Iwasaki M. et al., EFFECT OF PARTIAL ADMISSION AND SWIRL BRAKES ON DESTABILIZATION FORCE OF LABYRINTH GAS SEAL, ASME Turbo Expo, GT2020-14155