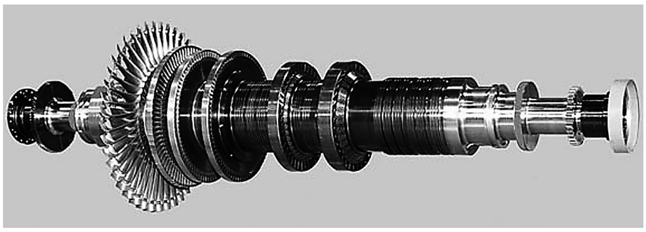

Application of Advanced Integral Shrouded Blades to High-Speed and High-Power Mechanical Drive Steam Turbines

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The steam turbine for driving a synthesis gas compressor in ammonia plants is one of the main products of Mitsubishi Heavy Industries, Ltd. (MHI). MHI has reestablished a leading position for the steam turbine in the global markets by the following procedures: (1) Improvement of the reliability by applying the advanced integral shrouded blade (ISB) including an innovative combined structure to the control stage, and (2) Upgrading of the turbine performance and efficiency by adopting high-performance profiles. The advanced ISB have been used not only for the control stage but also for the intermediate stage. They have been applied in a large number of MHI turbines for the past several years. All the turbines using these blades are already operating successfully. The results obtained from rotating blade shaker tests, cascade tests, and profile performance tests using air turbine during the development process are briefly introduced in this paper.

1. Introduction

The MHI steam turbines that drive the synthesis gas compressors used in ammonia plants (hereafter referred to as SYN. GAS turbines) hold a leading position in global markets represented by the advantage of high reliability, compact design (single casing), and both end driving.

In recent years, however, strict demands regarding turbine performance have been made by customers in order to reduce plant operating costs. Since MHI's SYN. GAS turbines have met market needs without having to be completely up-graded over several years in the past because MHI placed emphasis on reliability, development of next generation models that aim at reinforcing cost-competitiveness on performance have become necessary.

In particular, since the control stage rotating blades were used under high speeds and high powers, the height of the rotating blades was limited to 20 mm or less so as to secure reliability. As a result, the stationary nozzle gauging angle became larger, diagram efficiency was decreased, aspect ratio was reduced, and profile performance was lowered.

In order to recapture the advantage of SYN. GAS turbines, MHI has newly developed a high-performance profile by reducing the gauging angle to improve the profile aspect ratio. This was done assuming that the advanced ISB would be applied to the control stage of the turbine. However, the well-known twist back effect of long height blades with torsional profile cannot be applied to this control stage blade without torsional profile, since the blade is of a parallel short type. ("Parallel" means that the cross sectional profile of the blade is uniform in the height direction.) MHI has developed a innovative mechanism to combine together adjacent blades by themselves, and has succeeded in improving the blade to become practically identical with an endless grouped blading structure.

This paper presents a concept of the combined structure of the new mechanism, the results of FEM analysis of blade strength, the results of rotating blade shaker tests done using a test rotor, and the results of profile performance tests using air turbine.

Table 1 Major specifications of SYN. GAS turbine

Max. power	Max. rotational speed	Main steam pressure	Main steam temperature	Max. inlet flow
22 000 kW	12 000 rpm	110 kgf/cm^{2}g	510°C	250 T/H

2. Specifications of steam turbines for driving synthesis gas compressors

Table 1 shows the specifications of the SYN. GAS turbine applied at the control stage of the turbine. **Fig. 1** shows a cross sectional view of the turbine. **Table 2** shows the specifications of the newly developed and conventional control stage blades and nozzles.

Table 2 Blade and nozzle specifications at control stage

		Basic diameter	Profile height	Profile width	Aspect ratio	Gauging
Stationary nozzle	Newly developed	ϕ 428 mm	20.0 mm	15.0 mm	1.3	0.26
	Conventional	ϕ 470 mm	12.5 mm	25.0 mm	0.5	0.39
Rotating blade	Newly developed	ϕ 425 mm	25.0 mm	50.0 mm	0.5	0.36
	Conventional	ϕ 466 mm	16.0 mm	50.0 mm	0.3	0.44

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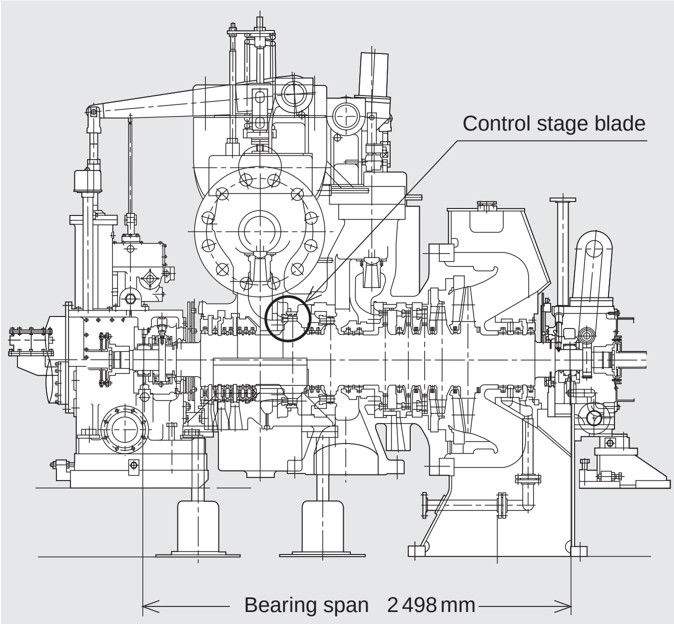

Fig. 1 Cross-sectional view of SYN. GAS turbine

The aspect ratios of the stationary nozzles and rotating blades are remarkably increased, that is, they are 2.7 times that of the conventional type for stationary nozzles and 1.6 times that of the conventional type for rotating blades.

3. Combined structure of innovative mechanism (endless grouped blading structure)

The endless grouped blading structure, which utilizes the twist back effect of the conventional ISB (long height blades), cannot be adopted in the control stage blade considered here because it is a parallel short profile. In order to improve the ISB to be equivalent to endless grouped blades, the combined structure of a new mechanism different from the conventional structure must be developed (hereafter this type of blade is referred to as a parallel short ISB to distinguish it from the conventional ISB).

In the combined structure of the new mechanism of the parallel short ISB, the rotating blades (in which the contact surface between adjacent blades of the shroud section is tilted) are inserted on a tilt in the profile convex side by using the clearances between the blade roots and rotor grooves. In this system, the blades move from a tilted state to a raised state due to the centrifugal force acting on the blades during rotation. On the other hand, by inserting blades with a tilted state, as shown in **Fig. 2**, the tangential pitch of the shroud section can be designed in a tilted state by a geometrical relation, to be greater than in the raised state. Accordingly, when the increased amount of pitch at the shroud section obtained geometrically is greater than a physically extended distance, the shroud contact surface between the adjacent blades can be held in a combined state even during rotation without being physically separated. The physically extended distance means the amount between the contact surfaces of adjacent blades at the shroud section when both surfaces are separated by centrifugal force and thermal expansion during actual operation.

Though the increased amount of pitch can be adjusted by changing the tilted angle of the blades and the tilted angle of the contact surface of the shroud section, basically, the tilted angle of the blades is determined and restricted by the types of blade roots and rotor grooves. Accordingly, the pitch is adjusted by only the tilted angle of the contact surface.

Since the blades are tilted and assembled in this system, it can be considered that the blades will not rise during rotation, due to an excess of the increased ammount of pitch. In this case, the blade root may enter into one side local contact state as abnormal condition and excessive stress may occur locally in the blade root. To solve this problem, the increased amount of pitch must be properly designed so that the blades can be brought into a raised state during rotation.

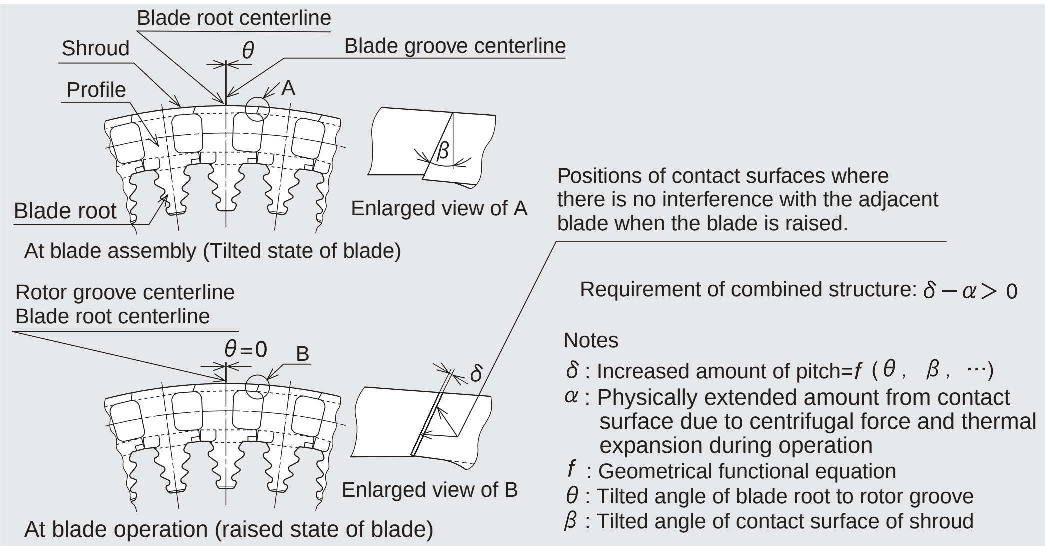

Fig. 2 Combined structure of parallel short ISB

4. Static strength of advanced ISB

The static strength of the parallel shorter ISB that should be examined in the combined structure described above is considered to be critical in the following state.

When the blades are considered to still remain in the raised state acquired during rotation even after rotation has stopped, the physically extended amount of the shroud section due to centrifugal force and thermal expansion becomes zero. As a result, the total increased amount of pitch acts as a compressive stress on the contact surface of the shroud section.

The distribution of stresses in the blade while in a stationary state was evaluated by FEM analysis. The results are described below.

The distribution of stress during rotation was also evaluated in the same manner, and it was confirmed that the distribution of stress does not have any particular problems with respect to strength. Since the distribution of stress is less than stress levels around the shroud section in a stationary state, it has been omitted in this paper.

Table 3 shows the analysis conditions. The maximum values taking into account manufacturing tolerances is used as the increased amount of the pitch at the shroud section in the analysis.

Fig. 3 shows the minimum principal stress distribution determined by FEM analysis. As shown in this figure, compressive stresses occur on the profile concave side on the upper surface of the shroud section and that tensile stresses occur on the profile convex side. Though the maximum compressive stress, which occurs in the shroud and around the center section of the profile convex side, exceeds the yield stress of the blade material by approximately 50 %. This excessive stress can be reduced due to deformation of the structure because it is a local type and a displacement constraint type. Accordingly, it could be determined that these stresses do not have any potential problems with respect to strength.

5. Rotating blade shaker test and vibratory strength

To verify the combined structure of the parallel short ISB, the tapping test for a single blade in free-staging state and rotating blade shaker test using a test rotor were carried out.

In addition, the vibratory strength of the blade was analyzed based on the results of above tests and the mode analysis conducted by FEM.

5.1 Tapping test

The tapping test was carried out to measure the natural frequencies and mode shapes of the blade in a single free-standing state.

This test was carried out in a state in which one blade was inserted in a groove of the test rotor. **Table 4** shows the natural frequencies of the minimum order modes.

5.2 Rotating blade shaker test

Fig. 4 shows the state of the parallel short ISB after they are completely assembled in the test rotor. The parallel short ISB were also designed and manufactured for

Table 3 Analysis Conditions

Rotational speed, Room temperature	Increased amount of pitch	Coefficient friction of contact surface	Analysis method
0 rpm	0.103 mm	0.2	Nonlinear elastic

Table 4 Natural frequency of tapping test of single free-standing blade (stationary, room temperature)

Primary Tangential	Primary axial	Primary torsional
3 700 Hz	7 392 Hz	8 192 Hz

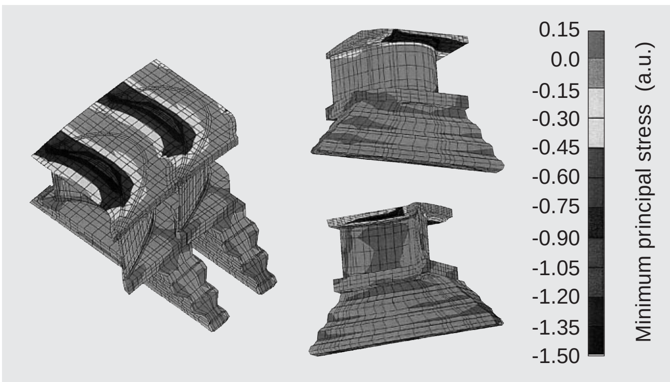

Fig. 3 Results of FEM analysis (minimum principal stress distribution)

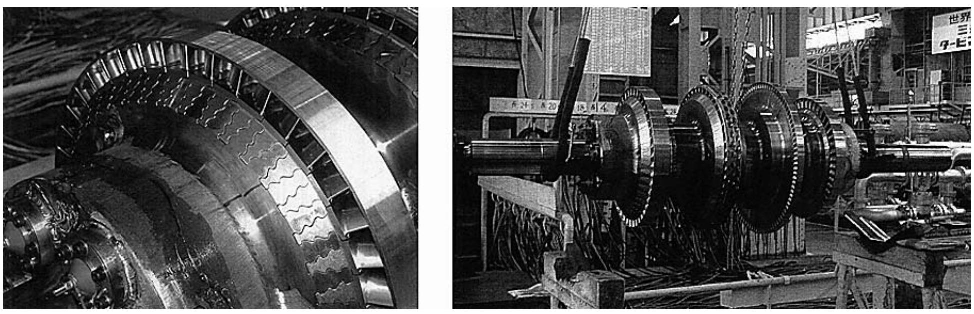
Fig. 4 Assembled test rotor of parallel short ISB

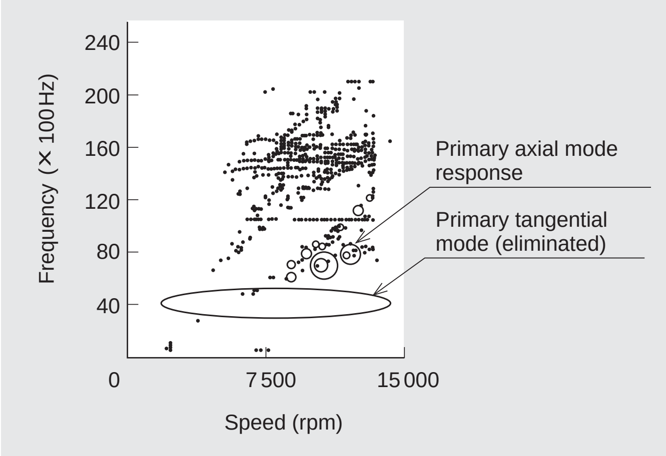

Fig. 5 Campbell diagram

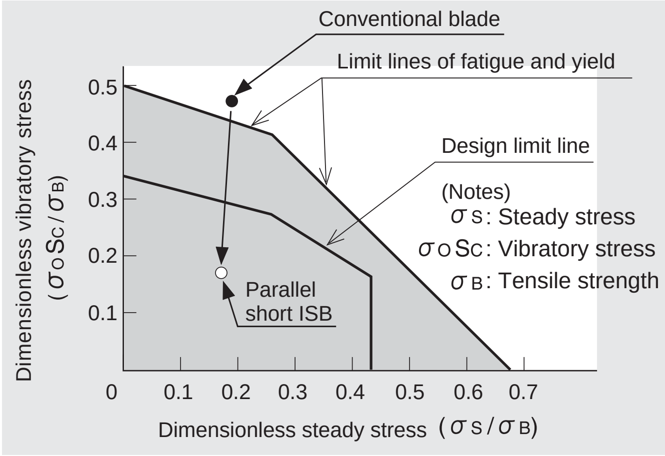

Fig. 7 Goodman diagram

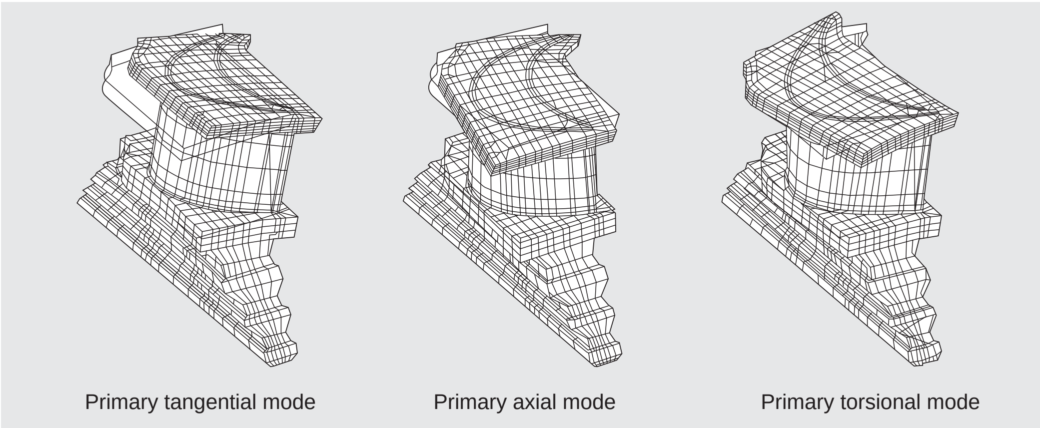

Fig. 6 Results of calculated mode shapes of minimum order in different directions

different blade profiles other than those used in the control stage. Tests of these blades were also conducted at the same time.

The blade shaker test under rotating was carried out in a vacuum chamber in an MHI facility. The blade was excited during rotation by air nozzles until it rotated at a maximum speed of 13 200 rpm, at which point the vibration was measured. In the measurement method, signals were transmitted from semiconductor strain gauges installed on the blade to an FM telemeter and received by equipment in the measurement chamber.

Fig. 5 shows the Campbell diagram of the parallel short ISB for the control stage obtained by the rotating blade shaker test. As shown in this diagram, responses occur near the frequency of 7 000 Hz, and the primary axial responses in the tapping test of the single free-standing blade can be considered as appearing in the figure. On the other hand, since the primary tangential responses (near 3 700 Hz; matches the direction of steam thrust force), which were critical with respect to blade strength, were disappeared, it could be confirmed that the combined structure of this parallel short ISB performs effective functions on our expectation.

5.3 Evaluation of vibratory stress

Vibratory stress during actual operation was evaluated based on the responses of the parallel short ISB in the rotating blade shaker test and the mode analysis by FEM. Since this blade is used at the control stage of the turbine, vibratory stress was evaluated under the partial admission state, namely the shock load response.

Fig. 6 shows each mode of the minimum order in the tangential, axial, and torsional directions, which was acquired by the mode analysis.

On the other hand, the response of the primary tangential mode was disappeared in the rotating blade shaker test, and the responses of the mode in the axial and torsional directions were measured. However, since the effective excitation force (due to steam thrust force) is small for the primary torsional mode, resonance in the response of a shock load with the primary axial mode was evaluated for the vibratory stress of the parallel short ISB.

In the Goodman diagram shown in **Fig. 7**, the strengh level of this advanced ISB is evaluated, and the strength level of the conventional type blade (shroud packed blade) is also plotted in the same manner.

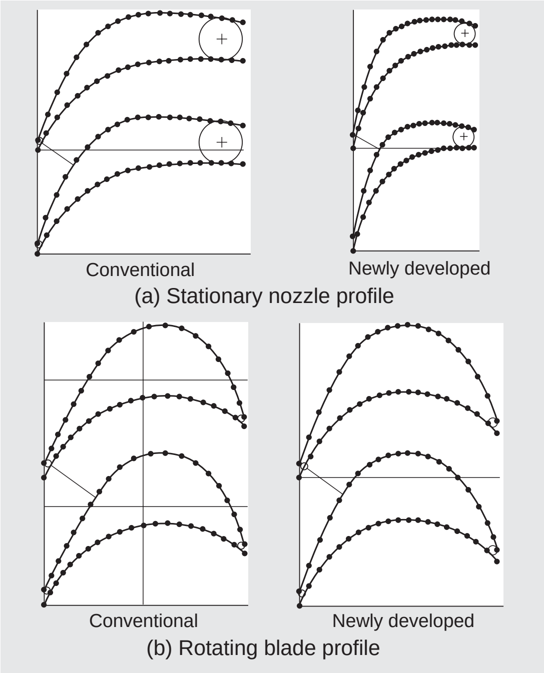

Fig. 8 Comparison of profiles between newly developed type and conventional type

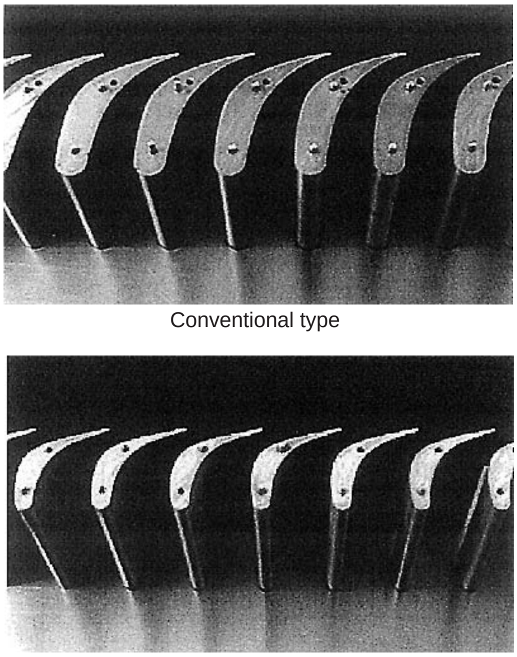

Fig. 9 Stationary nozzle cascade test model

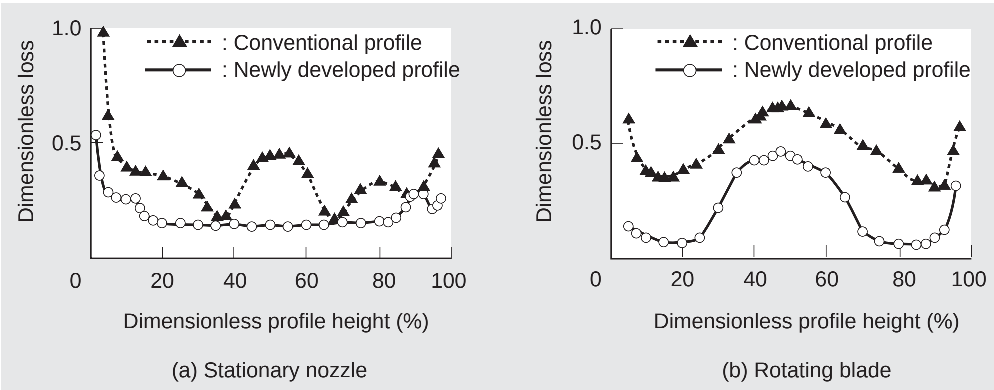

Fig. 10 Results of cascade test

This diagram indicates that the evaluated point of the conventional blade exceeded the allowable limit, and the blade cannot be used. However, the vibratory stress level of the parallel short ISB was greatly reduced to approximately 1/3 that for the conventional blade and was determined to have sufficient strength. As a result, the vibratory strength of the parallel short ISB can be evaluated as having a sufficient reliability that is significantly increased compared with that of conventional blades.

6. Evaluation of advanced ISB performance

The following measures were taken to improve the performance of the parallel short ISB.

(1) The gauging of the profile was reduced to improve diagram efficiency.
(2) Profile loss was reduced by improving the acceleration pattern in the flow path.
(3) Secondary flow loss was reduced by increasing the aspect ratio of profile.
(4) Exhaust loss was reduced by increasing the annular area.
(5) Tip leak loss was reduced by installing radial fins on the outer surface of the rotating blade at shroud section.

Fig. 8 shows a comparison between the conventional profiles and the newly developed profiles. The newly developed stationary nozzle is formed so that the thickness of the profile is reduced less than that of the conventional form so as to shorten the profile width. The rotating blade has been so formed that flow velocity distribution is smoothed to reduce profile loss.

Next, cascade tests were performed in order to verify the performance of the newly developed profiles. **Fig. 9** shows the stationary nozzle cascade test model, and **Fig.10** shows the test results. In the newly developed profiles, loss is significantly reduced to less than that of conventional blades by improving the profile shape and the effect of an increase in the height of the blade and nozzle.

In addition, overall performance was measured by using an air turbine to verify performance at the control stage. Based on the results obtained by this test and the cascade test results, the performances of the conventional control stage and the newly developed one were compared with each other using one-dimensional performance calculations.

As a result, the amount loss of the newly developed control stage was highly improved and estimated as being 0.57 that of the conventional ones, as shown in **Fig. 11**. It could be also confirmed that newly developed type obtained higher performance at control stage.

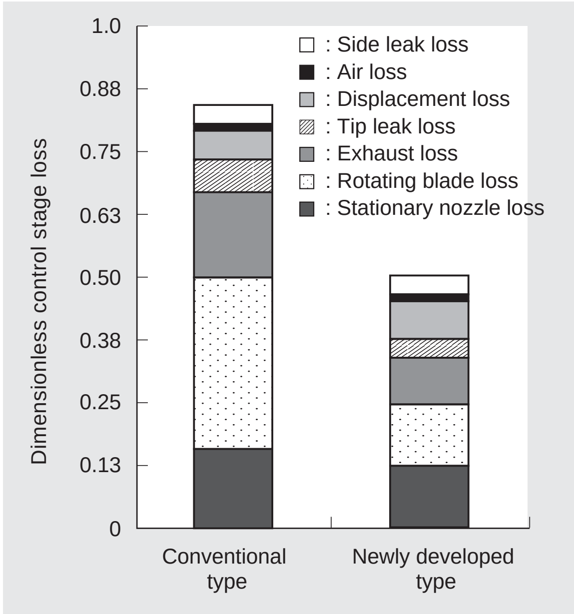

Fig. 11 Results of performance analysis

7. Conclusion

To improve the performance and reliability of MHI's SYN. GAS turbines, an advanced ISB was applied to the control stage of the turbine, and high performance profiles were developed with an improved profile aspect ratio. In addition, MHI succeeded in improving the parallel short blade to be practically equivalent to an endless grouped blading by developing a combined structure of the innovative mechanism.

The following results could be obtained for the rotating blade shaker tests, cascade tests, and profile performance tests. As a result of these efforts, it was completely possible to reestablish a leading position for the SYN. GAS turbines and to recover the competitiveness of these turbines in the global markets.

(1) Vibratory stress was greatly reduced in the parallel short ISB at the control stage to approximately 1/3 that of conventional blade, and the reliability of the blade with respect to strength could be remarkably increased.

(2) Ccontrol stage loss was considerably reduced to 0.57 compared with the losses of conventional stage thanks to effects to improve the profile and other characteristics.

The parallel short ISB was applied not only to the control stage but also to the intermediate stage, and was applied to MHI's SYN. GAS turbines and many other high-speed turbines since several years ago. All of these turbines are already operating successfully. MHI will promote the application of the parallel short ISB even further in the future.


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