NEW TECHNOLOGY ON HIGH PERFORMANCE AND RELIABILITY OF MECHANICAL DRIVE STEAM TURBINES

by

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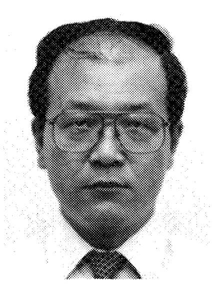
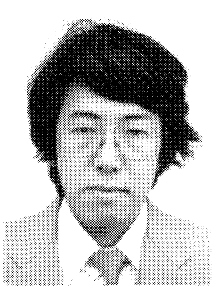


Tomoyoshi Sasaki joined Hiroshima Engine Works, Mitsubishi Heavy Industries, Limited, in 1974, after receiving his Bachelor and Masters degrees in Mechanical Engineering from the Kyushu University. He has been engaged in the design and development of steam turbines since 1983.



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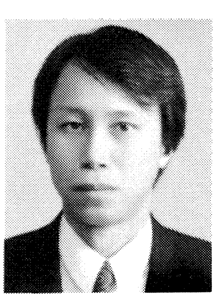

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ABSTRACT

The new design technology of the mechanical drive steam turbines used in the chemical plant industry is presented. In order to improve the performance of high speed turbines, the new high pressure ratio impulse stage has been developed. Low pressure blade design technology was also established by using the flow analysis technology. With respect to the high reliability of the turbines, the new analysis technology for steam whirl vibration characteristics, direct lubrication bearing, and erosion prevention technology have been developed.

INTRODUCTION

With respect to the mechanical drive steam turbines used in the chemical plant industry, research and development are currently underway to improve the efficiency, not only to cope with the increase in energy costs, but also to meet the requirements for saving resources and environmental protection.

Also, in order to improve the efficiency of compressors, it is necessary to increase the speed of compressors and turbines.

Under these circumstances, the authors' company has been making efforts to improve the efficiency and to increase the speed of mechanical drive turbines. As a result of the advancement of computers, new technologies have also been developed in recent years.

The technical achievements recently made for improvement of efficiency and reliability of turbines are presented. New turbines which have been developed by utilizing these new technologies are explained.

PERFORMANCE IMPROVEMENT TECHNOLOGY

Aiming to improve the efficiency of mechanical drive turbines, the researchers have been, for many years, developing performance improvement technologies, such as those shown below:

- High performance blade design technology
- Optimum flow pattern design technology
- Leakage control design technology [1]
- Moisture loss reduction technology [2]

As a part of these technologies, under high blade design and optimum flow pattern design technology, a new high pressure ratio impulse stage was developed to increase turbine speed, and to design more compact turbines. Low pressure (LP) blade design technology was also established by using the flow analysis technology which has advanced remarkably in recent years. These developments are introduced in the following articles.

High Performance HP, LP stages

In these compact designed turbines, and with a reduced number of stages, turbine stages are operated at a high pressure ratio, and, therefore, are required to have a high performance in such conditions. However, the nozzle exit flow becomes supersonic for the impulse turbine stage with high pressure ratio, therefore, the performance of such stages decreases due to an increase in shock loss, if such stages have the conventional convergent nozzle.

For this reason, a convergent-divergent nozzle (C-D nozzle) was used in order to improve the performance in the operating range with high pressure ratios. The high pressure ratio impulse turbine was successfully developed by performing the cascade design, cascade tests, and turbine tests.

Nozzle Cascade Performance

The C-D nozzle was designed as shown in Figure 1 and tested for the cascade performance in comparison with conventional convergent nozzle.

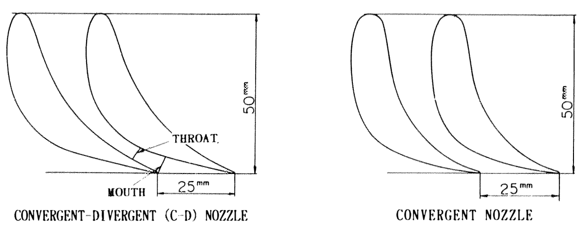

Figure 1. Nozzle Profile.

The conventional convergent nozzle is designed to develop the high performance in the range of the exit Mach number $M_2 = 0.6 \sim 0.8$. Whereas, the new C-D nozzle has been designed to develop the best performance for $M_2 = 1.2 \sim 1.4$. In designing the C-D nozzle, the area ratio (mouth/throat ratio) is ordinarily selected to match the exit Mach number. However, in the case of the new design, the area ratio was selected to create some underexpansion, in order to prevent the increase of losses due to the shock wave generated by overexpansion.

Shown in Figure 2 is the relationship between the profile loss and the Mach number on the cascade test of the C-D nozzle, designed according to such a principle.

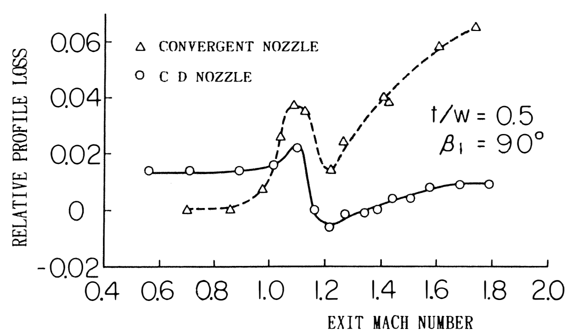

Figure 2. Comparison of Profile Losses between Convergent and Convergent-Divergent Nozzle.

The test results are summarized as follows:

- The C-D nozzles have a good profile performance in the range of the exit Mach number $M_{2th} = 1.0 \sim 1.8$, as compared with convergent nozzle. The difference in performance increases as the Mach number increases.
- The convergent nozzle has a better performance than the C-D nozzle in the range of $M_{2th} < 1.0$.
- In the range of $M_{2th} = 1.35 \sim 1.45$, that is, at the design point, the C-D nozzle has a better performance than convergent nozzle by about 3.5 ~ 4.0 percent in the profile loss coefficient.

The preceding points can be confirmed also from the schlieren photographs of cascade shown in Figure 3. The photographs show

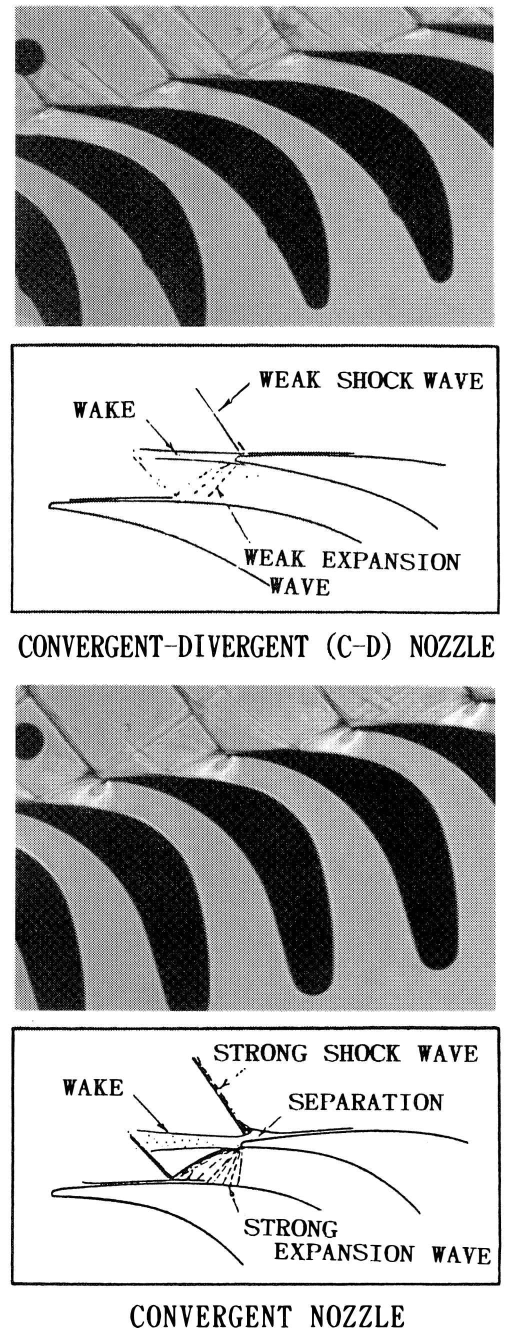

Figure 3. Comparison of Color Schlieren Photos between Convergent and Convergent- Divergent Nozzle.

that a sudden expansion is occurring at the throat of the cascade with the convergent nozzle, however, such a sudden expansion can not be found in the flow passage from the throat to the exit with the C-D nozzle. It is considered that by making steam to expand gradually in the flow passage from the throat to the exit, the shock wave generated at the blade passage exit can be suppressed and the losses in the high Mach number range can be reduced.

Moving Blade Cascade Performance

In the high pressure ratio impulse stage, the Mach number at the moving blade inlet and exit increases as the nozzle exit Mach number increases. Therefore, the nozzle and also the moving blade were designed to get high performance at the supersonic range and cascade tests were performed for the new moving blade in comparison with the conventional blade.

The profile of the new moving blade is shown in Figure 4. The new blade was designed to prevent overexpansion in the flow in the supersonic range, thus to improve performance, by optimizing the curvature radius on the suction surface at the inlet and exit.

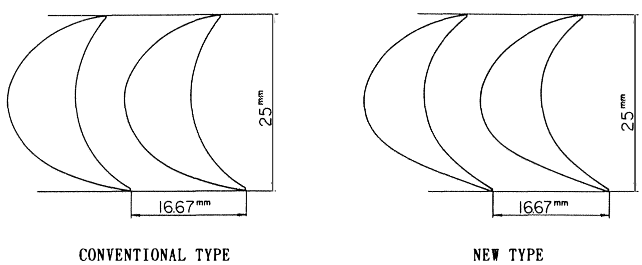

Figure 4. Blade Profile.

The relationships between profile loss coefficients and Mach numbers are shown in Figure 5 for the new blade and conventional blade. It can be seen from the figure that the new blade has a better performance than the conventional blade in the range of the exit Mach number, $M_{2th} > 0.8$, and the performance is better by two percent at the maximum, in the profile loss coefficient at the design point with $M_{2th} = 0.85 \sim 1.1$.

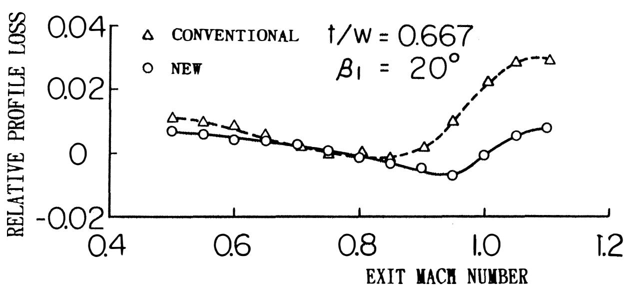

Figure 5. Comparison of Profile Losses between New and Conventional Blade.

Internal Efficiency of Turbine

In order to compare the performance of the new high pressure ratio turbine stage with that of a conventional turbine stage, steam turbine tests were performed. The new stage was made by combining the C-D nozzle with the new moving blade, both confirmed for the high performance at the high Mach number range through cascade tests.

A comparison of the internal efficiency for both turbines at the stage. Mach number $M_{st} = 1.4$ is shown in Figure 6. The internal efficiency characteristics of the new turbine in the range of $M_{st} = 0.6 \sim 1.74$ are shown in Figure 7. From these figures, the following points can be found:

• The efficiency of the new turbine is higher by about 3.5 ~ 4.0 percent than that of the conventional turbine at the design point ($M_{st} = 1.4$).

With the new turbine, the internal efficiency in the range of $M_{st} = 1.2 \sim 1.4$ is higher by about two percent than that at $M_{st} = 1.0$, showing that the performance is improved in the range of a high Mach number.

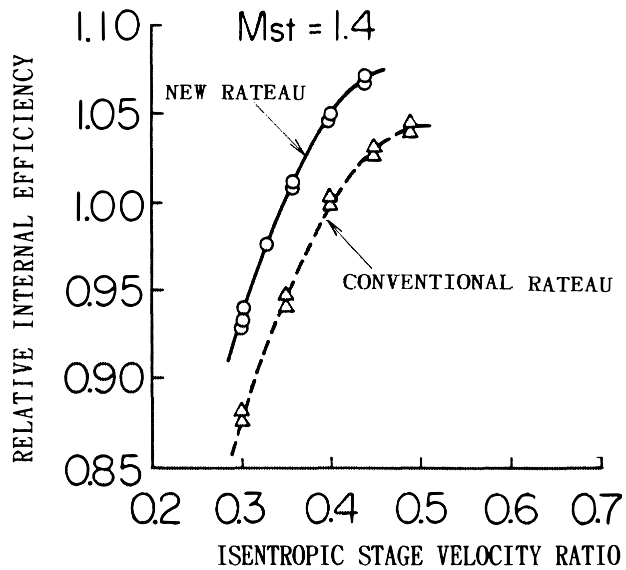

Figure 6. Comparison of Internal Efficiencies between New and Conventional Rateau Stages.

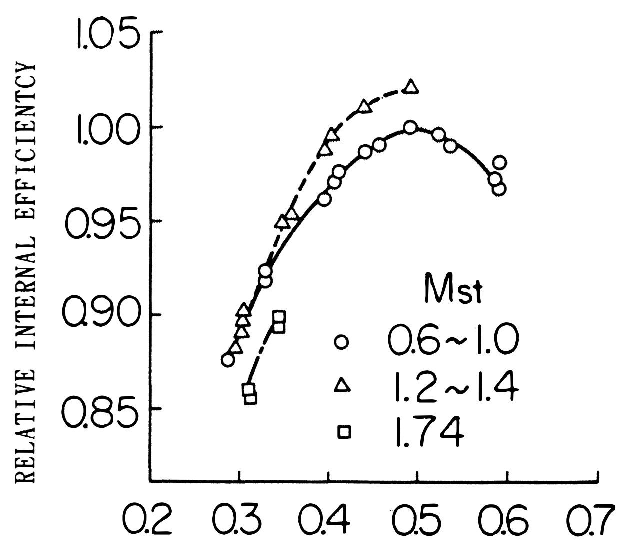

Figure 7. Internal Efficiency of New Rateau Stage.

Example of Application

As an example, the performance is shown in Figure 8 of the high pressure ratio impulse (Rateau) stage used for the mechanical drive turbine in comparison with that of the conventional turbine. The following points can be noted from the figure:

• With the new turbine stage (four stages), the peak point in performance appears in the range of turbine speed higher than that with the conventional stage (six stages).

• In the operating range of conventional turbine (8000 ~ 10000 rpm), the new four stage turbine has almost the same efficiency as the conventional five stage turbine.

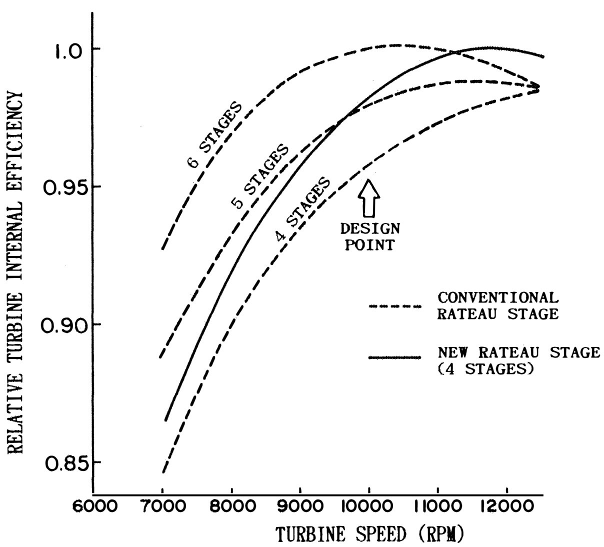

Figure 8. Comparison of Internal Efficiency between Conventional and New Rateau Stage for Actual Turbine.

High Performance LP Stage
Optimum Flow Pattern Design

In the case of LP turbine stages, the blade height is larger and the tip hub ratio is smaller as compared with the case of HP turbine stages. Therefore, the flow pattern varies to a large extent in the direction of blade height. For this reason, the Q3D flow analysis is used for the flow pattern design, since the curvature of streamlines on the meridional plane can be taken into consideration. By using this method, it is intended to improve performance through an accurate flow pattern evaluation.

An example is shown in Figure 9 of the calculation for the meridional plane streamlines for the mechanical drive turbine by using the Q3D flow analysis. A detailed evaluation of cascade performance becomes possible by using such methods of analysis.

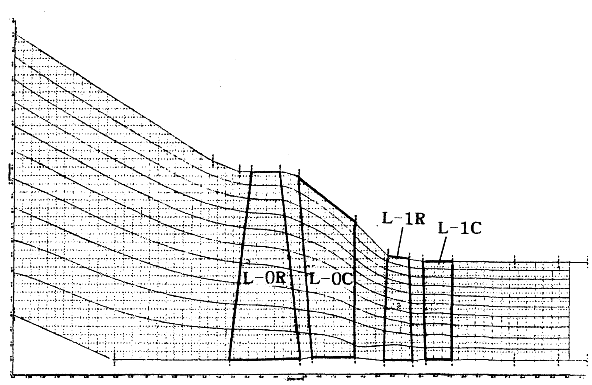

Figure 9. Q3D Flow Analysis Result of LP Turbine.

The main measures for the performance improvement which can be achieved through this analysis are as follows:

• Decrease of mismatching of rotor blades

• Selection of an optimum blade profile appropriate for the exit Mach number

• Reduction of leaving loss through an optimum gauging selection

Full 3D Aerodynamic Design

The nonviscous, full 3D flow analysis [3], has been advanced for practical use as a design tool in recent years, and is being used for further improvement of blade performance, together with the Q3D flow analysis previously mentioned.

In this full 3D aerodynamic design, the flow pattern at the blade tip is changed by performing 3D stacking of the stationary blade being bent in a bow shape in the direction of the blade height (Figure 10).

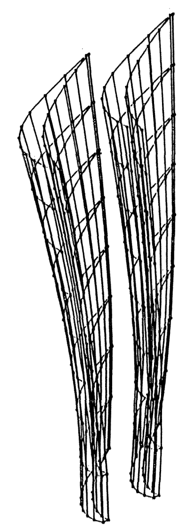

Figure 10. Bow Shape Nozzle.

Particularly for the flow pattern on the meridional plane, static pressure increases on the base side at the moving blade inlet, (Figure 11), and this effect of pressure rising cause the increase of blade inlet angle, as shown in Figure 12.

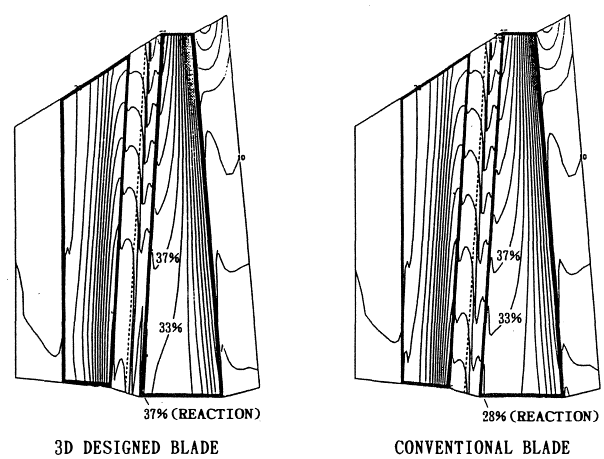

Figure 11. Comparison of Merdonal Plane Pressure Distribution between 3D Designed Blade and Conventional Blade.

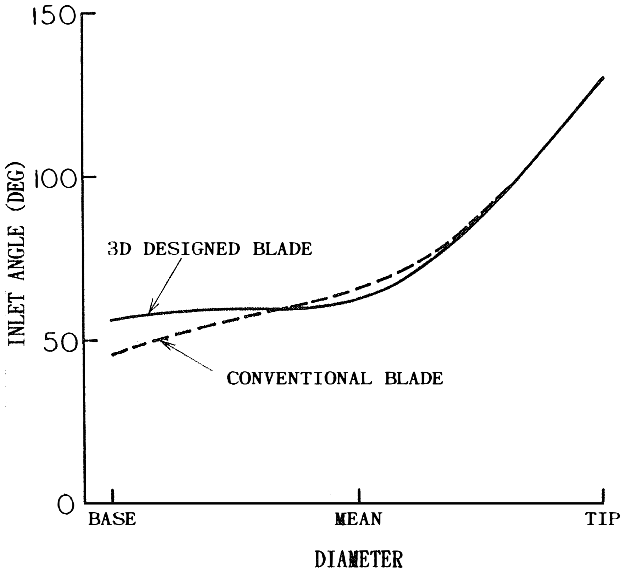

Figure 12. Comparison of Blade Inlet Angle between 3D Designed Blade and Conventional Blade.

By such increase in static pressure and inlet angle on the base side at the moving blade inlet, that is, by 3D effect, the secondary flow losses and the profile losses of the stationary and moving blades can be reduced.

Introduction of Viscous Flow Analysis

Flow analysis has been advanced remarkably in recent years, and it is becoming possible to analyze the blade performance without performing any cascade tests. Particularly, the accuracy of loss estimation by means of the 2D viscous flow analysis is fairly high, therefore, blade profiles are often optimized using this method.

As an example, a comparison is shown in Figure 13 of the density distributions in the stationaly blade obtained by the flow analysis and by testing. This figure shows clearly that the result of calculation coincides with that of testing. Furthermore, in Figure 14, an example is shown of obtaining the relationship between the profile performance and pitch/chord of stationary blade by using the 2D viscous flow analysis. This figure is to evaluate the change in performance with the pitch/chord within the 0.4 and 1.0 range. It can be found that the best performance is obtained with the pitch/chord in the range between 0.6 and 0.8.

The results of such analysis are applied for the LP turbine stages, and also for the HP and IP turbine stages as necessary for a further improvement of efficiency.

HIGH RELIABILITY TECHNOLOGY

High rotating speed is required for the mechanical drive turbine, depending upon the characteristics of driven machines such as the compressor. This tendency is being promoted as a result of the requirement for higher efficiency mentioned previously.

Therefore, it has become necessary more and more to improve the vibration characteristics of the rotor, the bearing performance, and the reliability of the blades, etc. Some examples in this respect are explained in the following article.

Rotordynamics

Turbines and compressors are being designed according to the Q-factor design method [4], which was proposed previously, and the sufficient reliability of these machines is secured against

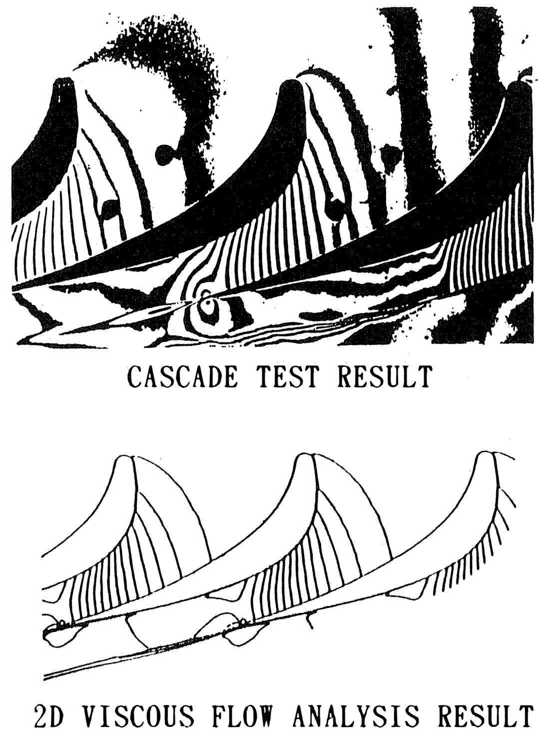

Figure 13. Comparison of Density Contour between Cascade Test and 2D Viscous Flow Analysis Results.

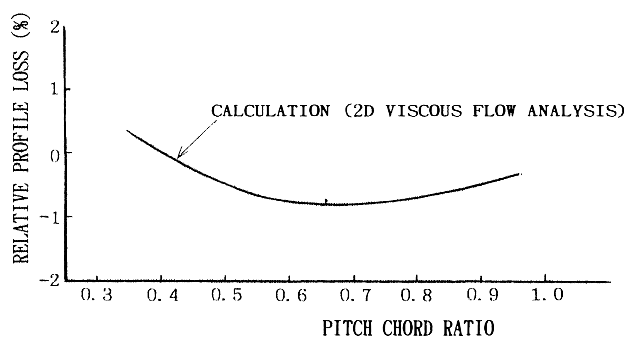

Figure 14. Optimum Pitch Chord Ratio of Stationary Blade.

critical speeds and oil whipping. With respect to compressors, it has become possible to reflect the subsynchronous whirl phenomena in the design and centrifugal compressors with a satisfactory stability have been developed [5, 6]. As a similar problem, turbines have the phenomena of steam whirl. Design methods have been developed for turbines separately against self excited vibration and forced vibration, due to steam whirl as follows.

Self Excited Vibration

The exciting force due to steam is generated at the blade itself, at the blade tip seal, at the gland seal, etc. It is necessary to provide damping against such exciting forces at the bearings .

The company stability criterion shown in Figure 15 to determines whether a self excited vibration will occur or not. The Thomas equation is used for calculation of the torque exciting force which a blade itself possesses and the modified Kostute's equation is used for calculation of the exciting forces for the tip seals and for the labyrinth gland seals. The latter equation is the one also used for the centrifugal compressors.

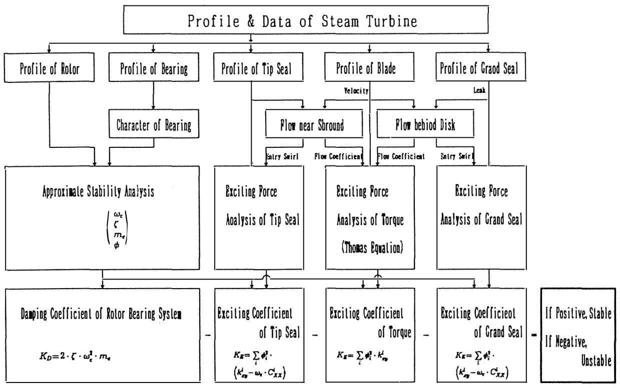

Figure 15. Block Diagram of Steam Whip Stability Analysis Method.

It has been revealed that the vibration, as shown in Figure 16, occurs with a 15 MW independent generator turbine, manufactured about 20 years ago. When the output is increased over 12 MW, the vibration with a component of 36 Hz suddenly increases. When analyzed according to the above mentioned criterion, the turbine becomes unstable at 11 MW, as shown in Figure 17. Therefore, the vibration was judged as a steam self excited vibration (steam whip), and a swirl controller was installed in the gland seal part where the exciting force was the greatest. As a result, self excited vibration disappeared up to 15 MW, securing a stable operation.

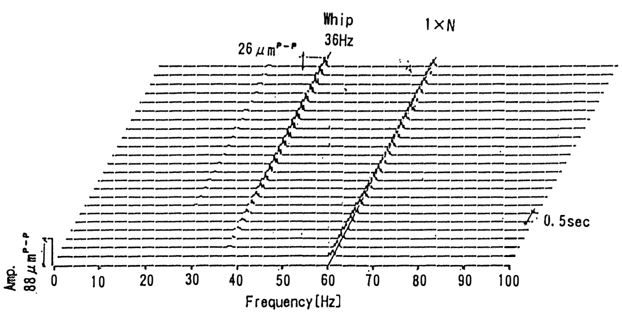

Figure 16. Water Fall Diagram.

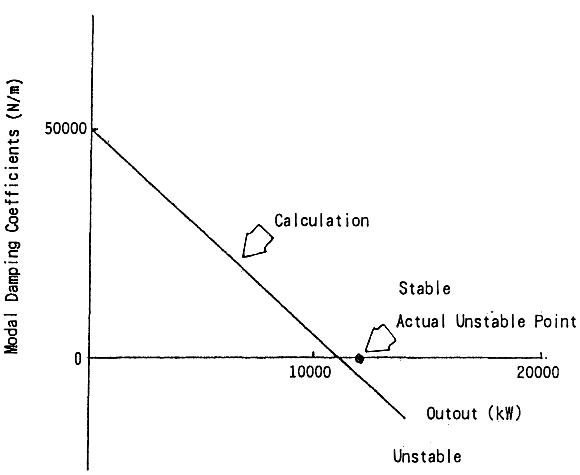

Figure 17. Steam Whip Stable Analysis Result.

By utilizing the method explained above, the company designs steam turbines which have sufficient stability also against steam whipping.

Forced Vibration

As a type of forced vibration due to steam, the vibration due to partial admission is well known [7]. In order to evaluate the exciting force due to partial admission, tests were performed using a test turbine shown in Figure 18. As a result, the following points have been clarified.

- It has been confirmed by actuating the turbine under operation that the vibration characteristics (natural frequency and damping ratio) of rotors do not change due to the valve opening.

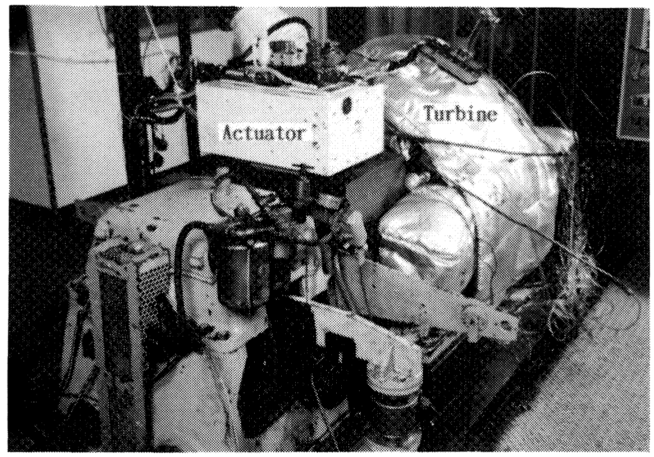

Figure 18. Turbine Test Facility with Actuator.

- By measuring the magnitude of swirl at the tip seal inlet, the information was fed back for analysis of exciting force at the tip seal.
- The vibration component with the natural frequency in the horizontal direction is amplified with the valve opening in the range of approximately zero through seven percent, as shown in Figure 19.
- The magnitude of vibration, according to the third step above, does not change, depending on the size of the space after the control stage.
- The vibration component is amplified where the reaction at partial admission is 0 ~ 0.1.

The vibration due to partial admission is considered as selective resonance and it has been made possible to evaluate the steam forced vibration by formulating the steam exciting force as follows:

$$Fr = Kr \cdot f(Rp) \cdot \Delta P \cdot D \cdot H$$

where

Fr = steam exciting force
Kr = nondimensional radial load factor at full admission
ΔP = pressure drop through a stage
D = mean diameter of rotor blade
H = height of rotor blade
f = factor at partial admission
Rp = reaction

Reliability of Bearings

Rotational speed required of turbines is 10000 ~ 15000 rpm in the case of the compressor drive in ammonium synthesis plants, for

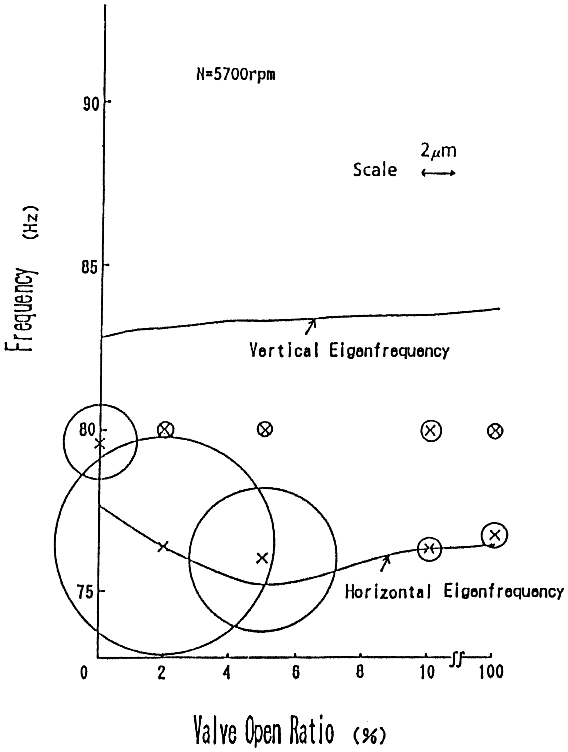

Figure 19. Partial Admission Test Result.

example, and the peripheral speed becomes 90 m/s at journal bearings and 180 m/s at thrust bearings [8]. The pressure at bearing surface is ordinarily taken at 10 ~ 20 kgf/cm^2(1 ~ 2 MPa) under operating conditions.

The major factors required of bearings are as follows:

- Bearings should be capable of sustaining high peripheral speeds and high loads.
- Bearings should be stable in operation.
- Bearings should have adequate dynamic characteristics.
- Bearings should be such that can be handled easily.

In view of the increase in rotational speed of mechanical drive turbines, the most important subject of the company concerning journal bearings and thrust bearings is reducing the bearing temperature which affects on the load bearing capability. The following methods can be considered for reduction of bearing temperatures:

- To increase lubricating oil quantity.
- To improve the overall heat transfer coefficient of bearing pads.
- To lubricate bearings effectively.

With the first method, reduction of bearing temperature can not be expected substantially, even with a surplus supply of lubricating oil if sufficient oil is being supplied for lubrication of sliding surface. For the second method, forced cooling of bearing pads and use of materials with high thermal conductivity can be considered. However, forced cooling of bearing pads requires complicated facilities and is not suitable to make machines compact. Use of materials with high thermal conductivity, copper for example, can be expected to reduce the bearing temperature by 5.0 ~ 10°C. However, the third method is expected to have the greatest effect on reduction of bearing temperatures,therefore, the company has been utilizing a direct lubrication method to supply cold lubricating oil to bearing pads, by providing oil supply nozzles between bearing pads.

The developed direct lubrication journal bearing and thrust bearing are shown in Figures 20 and 21, respectively. With respect to the effect of direct lubrication on reduction of bearing metal temperature, is shown in Figure 22, with the journal bearing. An example with thrust bearing is shown in Figure 23.

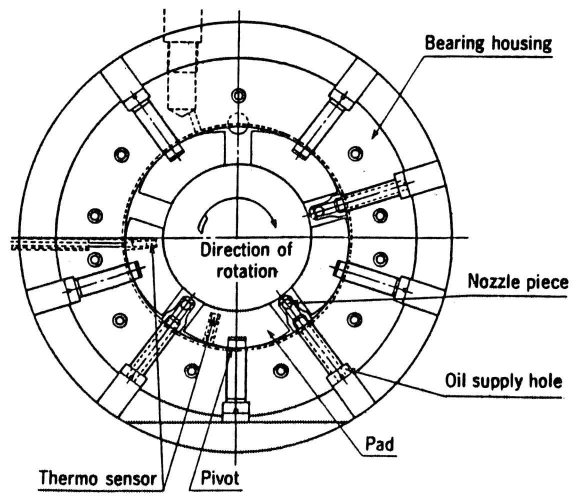

Figure 20. Direct Lubrication Type Tilting Pad Journal Bearing by MHI.

	Conventional bearing	Direct lubrication bearing
Lubricating system	Oil flood type, without nozzles	With nozzles, non-flood at periphery of thrust collar
No. of pads	12	12
Outer dia. of bearing	250mm	250mm
Inner dia. of bearing	125mm	125mm

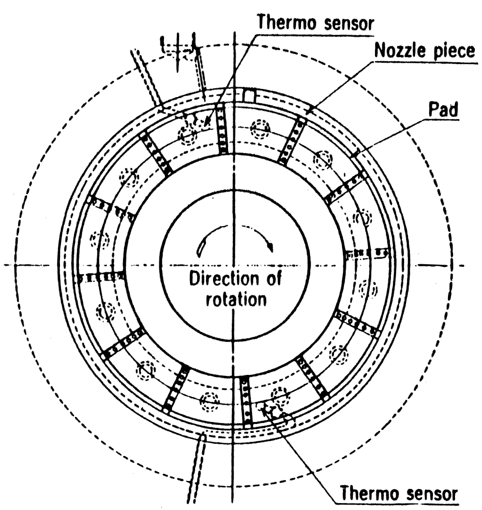

Figure 21. Direct Lubrication Type Bearing (9 7/8 in) Tested.

These bearings have been used for many mechanical drive turbines manufactured by the authors' company.

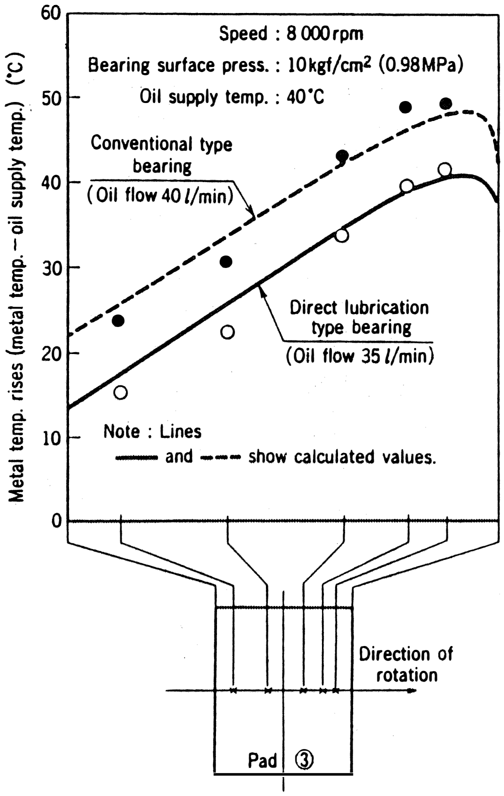

Figure 22. Metal Temperature Profile on Bottom Pad.

The specifications are shown in Table 1 for the compressor drive turbine used in a synthetic ammonia plant. The specification of the bearings are shown in Table 2. The operation data of these bearings are shown in Figure 24. The result was satisfactory with the journal bearing and the thrust bearing both at about 85 ~ 95°C, under normal operating conditions.

High Reliability LP Blade

Super computers are now used for strength analysis of stationary blades and moving blades. The vibration mode shown in Figure 25

Table 1. Principal Characteristics of the Turbine.

Speed	10360 rpm
Output	26000 kW
Inlet steam temperature	482°C
Inlet steam pressure	102 kgf/cm^2 (10.0 MPa)
Extraction pressure	38.5 kgf/cm^2 (3.8 MPa)
Exhaust pressure	0.15 ata (13kPa)
Drive system	Double shaft drive

Table 2. Principal Characteristics of Bearings.

No. of pads	Journal bearing 5	Thrust bearing 12
Size	Dia. 120 mm	Outer dia. of collar 9 7/8 inch
Peripheral speed	65 m/s	116 m/s
Bearing surface press.	20 kfg/cm^2 (2.0 MPa)	24 kgf/cm^2 (2.4 MPa)

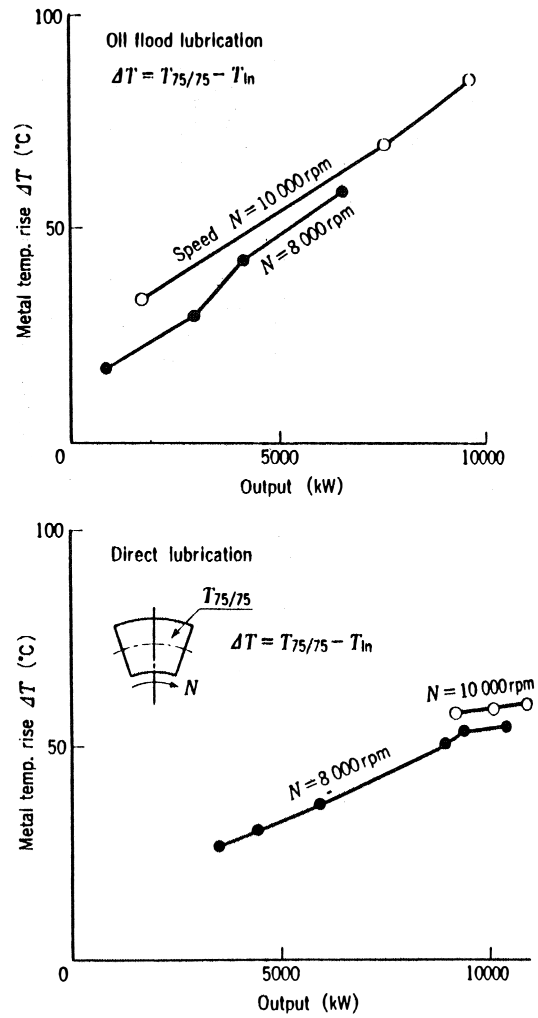

Figure 23. Bearing Metal Temperature in Flood Lubrication and Direct Lubrication.

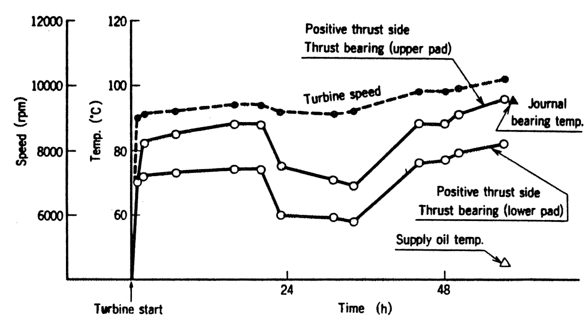

Figure 24. Operational Data.

is that of a blade group consisting of six moving blades found together with shrouds. This is an example of calculation results using super computers. This vibration is an asymmetrical mode (i.e., so-called fishtail mode). Vibrations of such a mode of high order could be a problem with LP stages and with long blades. The authors' company has been using the endless grouped blading shown in Figure 26 for long blades for many years to avoid the problem.

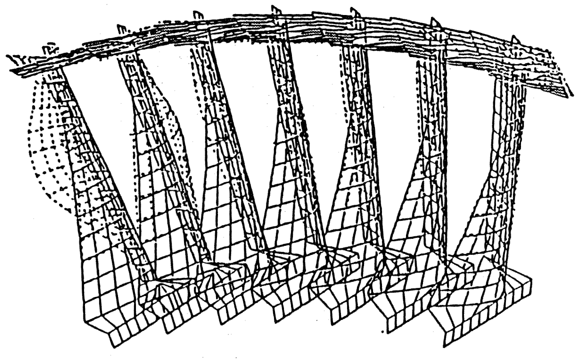

Figure 25. Vibration Mode of a Blade Group.

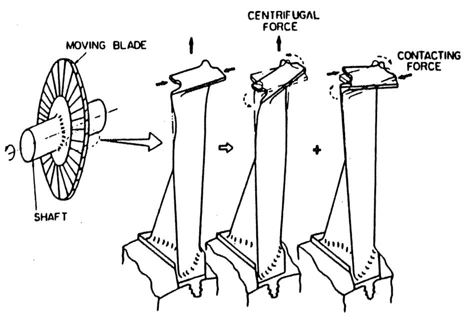

Figure 26. Principle of Endless Grouped Blading.

Among various factors, the drain attack erosion becomes a serious problem when the turbine speed is increased. An evaluation formula for drain attack erosion is:

$$dM/dW = k \cdot \exp(C \cdot V)$$

where

M = depth of erosion (μm)
W = total weight of water particles which impact a unit area (kg/cm2)
V = impact velocity of water particles (m/s)
K,C = constants depending on materials

Constant / Material	C	K
12% Cr	12.1×10^{-3}	0.056
Ti-6Aℓ-4V	8.2×10^{-3}	0.14
Stellite 6B	3.25×10^{-3}	0.56
Ti-15Mo-5Zr	8.53×10^{-3}	0.019

In order to decrease the erosion, the following methods may be considered;

• Reduce the weight of water particles, W, by optimizing the design of the grooved stationary blades and drain catchers.

• Make the material constants, K and C, smaller.

For the second method, stellite was applied on the moving blade surface by brazing in the past, however, because of problems, the fatigue limit of the moving blade was lowered as a result of brazing, and stellite could hardly stay adhered in the case of twisted blades. For this reason, the company has developed a process to apply Cr-TiN on the moving blade surface by ion plating, and actually has been using this process turbines [9].

DEVELOPMENT OF HIGH SPEED HIGH PERFORMANCE TURBINES

The company has completed the design of a high efficiency, high speed turbine for a synthetic gas compressor drive, using the new technologies mentioned previously. The cross section of the new turbine is shown in Figure 27 and the specifications are shown in Table 3. This turbine is of a single casing construction and with the direct lubrication journal and thrust bearings.

Table 3. Specification of New Turbine

Item	Specification
Turbine Speed × Power	(NOR) 14,800 rpm × 20,000 kW (MCR) 15,500 rpm
Steam Condition	Inlet 130 ata × 513 °C Extraction 47~40 ata Exit 0.1~0.2 ata

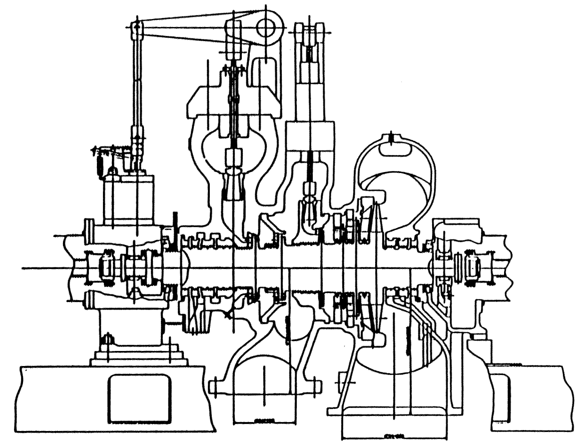

Figure 27. Cross Section of New Turbine.

The performance has been improved to have three percent higher internal efficiency than that of conventionally designed turbines by incorporating the following items:

• Optimization of blade profiles by means of flow analysis.

• Elimination of the inlet guide loss by adopting welded nozzles for diaphragms.

• Prevention of steam leakage by filling radial fins on rotor blades.

The performance and the rotor vibration characteristics of this turbine were confirmed with an air turbine testing unit shown in Figure 18.

The performance of the new test turbine was improved by about four percent than that of the conventional test turbine. With respect to the vibration characteristics, it has been confirmed also from the result of theoretical analysis mentioned previously that the new turbine is on the safety side against possibilities of steam whirl as compared with conventional turbines as shown in Figures 28 and 29.

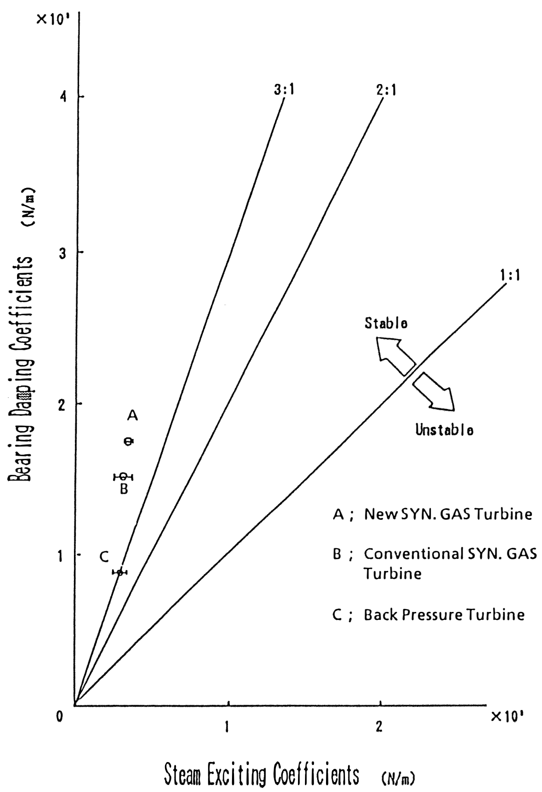

Figure 28. Steam Stability Analysis Results.

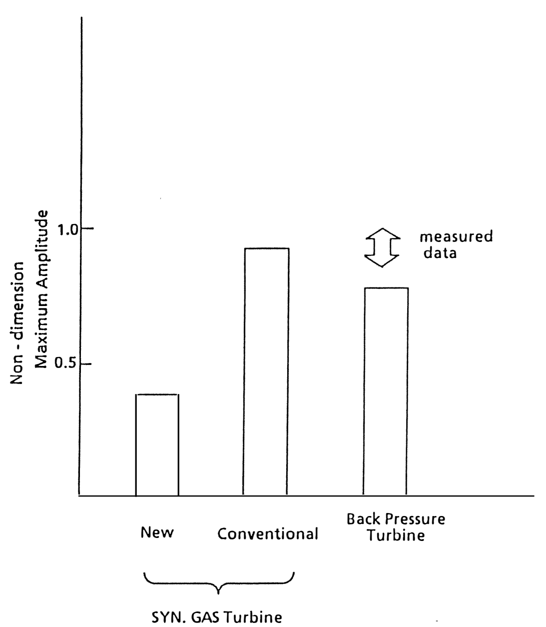

Figure 29. Predicted Maximum Amplitude of Steam Whirl at Partial Admission.

CONCLUSION

Explanations were presented of the results of studies carried out for the following items in order to meet the requirements of mechanical drive turbines for higher efficiency and higher speed.

- Improvement of performance of high speed turbines by adopting high pressure ratio impulse turbine stages (C-D nozzle).
- Improvement of performance of the LP blades by means of full 3D aerodynamic design technology.
- Improvement of reliability by means of the analysis technology for steam whirl vibration characteristics.
- Improvement of reliability by using direct lubrication bearings.
- Improvement of reliability by means of the erosion prevention technology.

The company has been applying these technologies for various turbines.

REFERENCES

1. Takenaga, H., Katayama, K., et al., "Performance Improvement of Mechanical Drive Steam Turbines," *Proceedings of 12th Turbomachinery Symposium,* Turbomachinery Laboratory, Department of Mechanical Engineering, Texas A&M University, College Station, Texas (1983).
2. Yoshida, K., et al., "Development of Measuring Techniques and Countermeasure on Wet Steam for Steam Turbine," The 1990 International Joint Power Generation Conference PWR-10, pp. 237-244 (1990).
3. Denton, J.D., "A Method of Calculating Fully Three Dimensional Inviscid Flow through Any Type of Turbomachine Blade Row," VKI LS 1983-06 (1983).
4. Shiraki, K., Kanki, H., "A New Vibration Criteria for High Speed Large Capacity Turbomachinery," *Proceedings of 8th Turbomachinery Symposium,* Turbomachinery Laboratory, Department of Mechanical Engineering, Texas A&M University, College Station, Texas (1979).
5. Kanki, H., Morii, S., "Theoretical and Experimental Study on the Destabilizing Force by Labyrinth Seal," 4th Workshop on Rotordynamic Instability Problems in High-Performance Turbomachinery, Turbomachinery Laboratory, Department of Mechanical Engineering, Texas A&M University, College Station, Texas (1986).
6. Kanki, H., Katayama, K., et al., "High Stability Design for New Centrifugal Compressor," 5th Work Shop on Rotordynamic Instability Problems in High-Performance Turbomachinery, Turbomachinery Laboratory, Department of Mechanical Engineering, Texas A&M University, College Station, Texas (1988).
7. Nosyleko, D., et al., "Reduction of Low Cycle Vibration of 800MW Turbo-Generator," Energetics and Electrification 3 [177], in Russian, pp. 1-2, (1983).
8. Sasaki, T., Mouri, Y., et al., "Performance Improvement of Bearings in Mechanical Drive Steam Turbines by Direct Lubrication," MHI Technical Review, 24, 3, (1987).
9. Ebara, R., Nakajima, H., et al., "Application of Ion-Plated Cr-TiN multilayer Coating To Steam Turbine Blades," *Proceedings of the 16th Turbomachinery Symposium,* Turbomachinery Laboratory, Department of Mechanical Engineering, Texas A&M University, College Station, Texas (1987).