An Improvement of Performance in Steam Turbine by Developing Three-Dimensionally Designed Blades*

Kiyoshi SEGAWA**, Yoshio SHIKANO**,
Norio YASUGAHIRA**, Kuniyoshi TSUBOUCHI**
and Naoaki SHIBASHITA***

From the viewpoint of reducing power generation costs and saving energy resources, improved efficiency of thermal power plants is a critical issue. Steam turbines, which are the main plant equipment, have been developed which provide a high level in blade performance. Further enhancement of blade performance, however, requires refinement of the blade shape and the control flow pattern in the radial direction. Recently, development of three-dimensionally designed blades has been active, with goals of raising the internal efficiency and being able to the control flow pattern in the radial direction not only for stationary blades, but also for moving blades⁽¹⁾⁻⁽³⁾. In order to improve the flow pattern through the blade rows, the authors have proposed three-dimensionally designed stationary blades⁽⁴⁾⁻⁽⁷⁾, which can be used in low pressure (LP) and high and intermediate pressure (HIP) turbines. This paper describes the development of three-dimensionally designed blades, in which performance of turbine stages including moving blades has been improved. The new blades reduce the end-wall losses in the HIP stages where there is a serious problem with secondary flow.

Key Words: Cascade, Secondary Flow, Numerical Analysis, Air Turbine Test, End-wall loss, Flow Pattern, Stage Efficiency

1. Development of Three-Dimensionally Designed Blades

It is important to reduce end-wall loss near inner and outer casings in HIP stages having a short blade height. The main factors to cause the end-wall loss are the inlet boundary layer, passage vortex and crossflow. The passage vortex results from development of a horseshoe-like vortex, which is produced by interactions between the inlet end-wall boundary layer and the blade leading edge, and crossflow is generated by imbalance in the boundary layer between centrifugal force and the pressure gradient in the circumferential direction. It is important to

reduce the blade load by biasing flow toward the end-wall, so as to lower the end-wall loss. But reducing the blade load near the end-wall lowers the flow rate in the vicinity of the midspan, and that decreases blade performance there.

It has been confirmed that three-dimensionally curved stacking in the circumferential direction gives blade force to fluids in the vicinity of the end-wall and controls the blade load distribution in the radial direction, although the flow yaw angle is deflected from the designed one. Therefore, the three-dimensional design of blades has to be optimized by considering both the blade load and the deflection of flow yaw angle distribution in the radial direction. This means the most important subject for improvement of flow pattern must be realized by considering not only the design of the blade itself, but also the effect of the turbine stage, consisting of both stationary and moving blades.

The three-dimensionally designed blades

* Received 13th July, 1997

** Power & Industrial Systems R & D Division, Hitachi, Ltd., 7-2-1 Omika-cho, Hitachi-shi, Ibaraki-ken 319-1221, Japan

*** Hitachi Works, Hitachi, Ltd., 3-1-1 Saiwai-cho, Hitachi-shi, Ibaraki-ken 317-8511, Japan

Fig. 1 Comparison of blade configurations

developed by the authors and conventional blades are shown in Fig. 1. The conventional stationary and moving blades have straight trailing edge counters, while the new stationary blade has three-dimensionally curved stacking in the circumferential direction, and the new moving blade has a higher reaction blade section with a larger outlet mechanical angle than that of the conventional one near the root and tip, and a smaller outlet angle near the midspan.

2. Computational Results of Three-Dimensional Turbulent Stage Flow Analysis

Three-dimensional Navier-Stokes analysis which assumes steady interaction between stationary and moving blade flows, is made to verify the stage performance of the three-dimensionally designed blades. A control volume method with a body-fitted curvilinear coordinate system⁽⁸⁾ is introduced for space discretization, while a damping surface technique⁽⁸⁾ is employed for time integration. A low Reynolds number k - ϵ turbulence model⁽⁹⁾ is used to consider the turbulence effect. Computational conditions are shown in Table 1.

Table 1 Computational conditions

Inlet total pressure P_0 (MPa)	0.142
Inlet total temperature T_0 (K)	353
Outlet static pressure P_s (MPa)	0.103
Velocity ratio W/V_0	0.5

The velocity ratio, based on the rotational velocity divided by the theoretical velocity obtained from stage heat drop, is about 0.5. Total pressure, total temperature and flow directions are specified in the radial direction on the stage inlet boundary and are assumed constant in the circumferential direction. Static pressure at the root is described on the stage outlet boundary. A periodic boundary condition is used on upstream and downstream periodic surfaces. Non-slip and adiabatic conditions are used on the blade surfaces and end-wall boundaries.

The counter lines of the total pressure distributions at outlets of the stationary and moving blade rows are shown in Fig. 2. For both stationary and moving blade rows, the total pressures are normalized by the stage inlet total pressure and the inlet relative total pressure at the midspan of conventional moving blade rows, respectively. The total pressure level at the outlet of the newly designed moving blade is higher than that of the conventional blade, in particular, total pressure drop is restrained in the vicinity of the end-walls. Then, the end-wall loss is reduced in this region.

The total-to-static efficiency distributions in the radial direction are shown in Fig. 3. Efficiencies are normalized by the midspan efficiency of the conventional blade. Efficiency of the newly designed blade is remarkably improved near the end-walls and has nearly the same values as the conventional one near the midspan. In case of newly designed blade, secondary flow is restrained in these regions and efficiency distribution is improved in the radial direction due to changes in flow patterns using the three-dimensionally designed blade.

The degree of reaction distributions in the radial direction are shown in Fig. 4. For the newly designed blade, the degree of reaction increases in the vicinity of the root, and the increasing rate of the degree of reaction decreases toward the tip, compared with conventional one. It is found that the degree of reaction is uniformed in the radial direction due to the control of stationary blade load by newly designed blade.

Figure 5 shows the axial velocity distributions at the outlet of stationary blade rows. For the newly designed stationary blade, axial velocity increases in the vicinity of the root. This is because the flow,

Stationary blade

Moving blade

(1) Conventional blades

Stationary blade

Moving blade

(2) New blades

Fig. 2 Computational results of total pressure distributions

Fig. 3 Stage efficiency distributions

Fig. 4 Degree of reaction distributions

which is affected by blade force of this blade, goes toward the end-walls, and flow rate increases in that region. On the other hand, axial velocity decreases at the non-dimensional blade height (h/NH) of 0.3 to 0.8.

The axial velocity distributions at the outlet of moving blade rows are shown in Fig. 6. For the newly designed moving blade, axial velocity tends to increase in the vicinity of the end-walls, compared

with the conventional one, while it decreases in the vicinity of the midspan, because the flow is biased by the change in the outlet mechanical angle distribution.

The results of the three-dimensional turbulent stage flow analysis show that the two newly designed blades are effective for reducing end-wall losses, and possibly raising stage efficiency.

Fig. 5 Axial velocity distributions at the outlet of stationary blade rows

Fig. 6 Axial velocity distributions at the outlet of moving blade rows

3. Verification of Stage Performance by Air Turbine Tests

3.1 Air turbine test facility

Air turbine tests are performed in order to verify stage performance of the newly designed blades. The turbine (Fig. 7) has a single stage, in which working fluid is supplied from a 2700 kW air compressor. Rotational speed is controlled by an electric dynamometer and output power is detected by a torque meter. The stage flow rate is measured by an orifice flow meter installed in the inlet pipe. A photograph of the air turbine test facility is shown in Fig. 8 and a meridional section schematic is shown in Fig. 9.

Stage performance is obtained from measuring the total pressure and flow angle distributions at the stage inlet and the outlet of the blade rows by traversing five-hole pitot tubes in the radial direction. The pitot tubes are automatically traversed in the circum-

Fig. 7 Schematic diagram of test turbine

Fig. 8 Air turbine test facility

Fig. 9 The meridional section schematic of test turbine

ferential and radial directions at the outlet of stationary blade rows and only in the radial direction at the stage inlet and the outlet of moving blade rows. Data acquisitions are carried out by using an online computer system. Three types of blade families are

Fig. 10 Improved internal efficiency for the newly designed blades

Fig. 11 Results of air turbine performance test

adopted as test models. Aspect ratios (AR s) of moving blades are 1.5, 1.9 and 2.6, respectively. Total performance tests are used to get the effects of aspect ratio. Results are described for a flow measurement test when the aspect ratio is 1.5.

3.2 Total performance tests

In order to test the total performance of the turbine stage, the pressure ratio based on inlet and outlet pressures is kept constant during the experiments and velocity ratio changes from 0.4 to 0.55 by varying the rotational speed. The results are compared at the design point velocity ratio of 0.5. The improved internal efficiencies of the newly designed blades, based on those of conventional blades for each aspect ratio, are shown in Fig. 10. The newly designed blades give improvements in internal efficiency from about 0.4 to 1.9%, compared with conventional ones. As the effect of end-wall loss is greater for the HIP stage, the three-dimensional design is especially

Fig. 12 Axial velocity distributions at the outlet of stationary blade rows

important for a short blade to improve efficiency.

Figure 11 shows normalized internal efficiency distributions of the newly designed blades with aspect ratio of 1.5. Internal efficiencies are improved by increasing the aspect ratio, due to a relative reduction in the end-wall loss region in the root-to-tip flow path. Improvements in internal efficiency at aspect ratios of 1.9 and 2.6 are about 0.7 and 2.7 at the design point, respectively.

3.3 Flow measurement test

Flow conditions of turbine stage can be obtained by pitot tube measurements. The performance test results at aspect ratio of 1.5 are described, because improvement of internal efficiency is the highest at the design point.

Axial velocity distributions in the radial direction at the outlet of stationary blade rows are shown in Fig. 12. Velocities are normalized by the midspan velocity of each case. The axial velocity of the conventional blade decreases near the end-walls due to the thick boundary layer flow, while for the newly designed blade, the axial velocity tends to become greater near the root and tip. This is because the new stationary blade is designed using the concept of bowed stacking and mass flow increase near the end-walls.

Energy loss coefficient distributions of stationary blade rows in the radial direction are shown in Fig. 13, which are normalized by the midspan energy loss of the conventional blade. The energy loss of the newly designed blade decreases at all radial locations, compared with the conventional blade. In particular, the load distribution of the stationary blade decreases near the end-walls due to the three-dimensional effect, so the loss decreases in that region.

Figure 14 shows incidence angle distributions of

Fig. 13 Energy loss distributions at the outlet of stationary blade rows

Fig. 15 Relative yaw angle distributions at the outlet of moving blade rows

Fig. 14 Incidence angle distributions of moving blade rows

Fig. 16 Axial velocity distributions at the outlet of moving blade rows

moving blade rows in the radial direction. The incidence angles of conventional and newly designed blades are within about ± 5 deg and the incidence characteristics are good. Two dimensional cascade experiments show that increase in energy loss within this range is small.

Relative yaw angle distributions at the outlet of moving blade rows are shown in Fig. 15. Values are normalized by the midspan yaw angle of each case. The relative yaw angle distribution of the conventional blade is deflected near the root due to the effect of secondary flow, while that of the newly designed blade is restrained near the root, due to the new mechanical angle distribution in the radial direction.

Axial velocity distributions at the outlet of moving blade rows are shown in Fig. 16. Values are normalized by the midspan velocity of each case. For the conventional blade, velocity defect is great near the root due to the end-wall boundary layer and

secondary flow, while the axial velocity of the newly designed blade is large near the root, and the effects of the end-wall boundary layer and secondary flow decrease.

Figure 17 shows total-to-static efficiency distributions in the radial direction, normalized by the mid-span efficiency of the conventional case. Efficiency of the newly designed blade is remarkably improved in the vicinity of the end-walls compared with the conventional one due to improvement of the flow pattern of both stationary and moving blades in the radial direction as shown in the previous figures.

4. Conclusions

Three-dimensionally designed blades, which are able to properly control flow pattern, have been developed for use in the HIP stages of steam turbines. From the results of both three-dimensional turbulent stage flow analysis and air turbine tests, it was found

Fig. 17 Total-to-static efficiency distributions

that the three-dimensionally designed blades offered improvements in efficiency of about 0.4 to 1.9%, compared with conventional blades. These high performance three-dimensionally designed blades are being retrofitted to HIP stages of actual steam turbines.

References

- (1) Cofer, J.I., IV, Advances in Steam Path Technology, ASME Journal of Engineering for Gas Turbines and Power, Vol. 118(1996), p. 337.
- (2) Hesketh, J.A., et al., Modernization of Steam Turbines for Improved Performance, GEC Review, Vol. 11, No. 2(1996).
- (3) Takenaga, H., et al., Performance Improvement of Geothermal Turbine by Application of F3D Impulse Blades, ASME PWR-Vol. 26(1994), p. 81.
- (4) Shikano, Y., et al., An Investigation of Leaned Nozzle Effects on Low Pressure Steam Turbine Efficiencies, ASME PWR-Vol. 10(1990), p. 151.
- (5) Sato, T., et al., Up-to-Date Technology of Efficiency Improvement for Steam Turbine, Sponsored by IEA and EPRI, Washington, D.C., March (1990).
- (6) Otawara, Y., et al., Technical Developments for Improved Performance of Steam Turbine, ICOPE-93, (1993), p. 367.
- (7) Yamazaki, Y., et al., The Effects of Leaned Nozzle on Steam Turbine Long Blade Stage, Proceedings of 31th Turbomachinery Conference in Sendai, (1993), (in Japanese), p. 91.
- (8) Shikano, Y., et al., Numerical Analysis of Three-Dimensional Compressible Turbulent Flows in a Turbine Stage, Trans. Jpn. Soc. Mech. Eng., (in Japanese) Vol. 53, No. 496, B(1987), p. 3622.
- (9) Chien, K.Y., Predictions of Channel and Boundary-Layer Flows with a Low-Reynolds-Number Turbulence Model, AIAA J., Vol. 20, No. 1(1982), p. 33.