533.695.5:621-226.2

The Wind Tunnel Tests of the Steam and Gas-Turbine Blades*

By Busuke HUDIMOTO** and Shigeo KUBOTA***

Experiments on the flow through cascade of steam turbine blades were recently carried out and the results obtained are summarized in this report. All of the experiments were carried out on the blade profiles of the impulse type turbine, and investigations were made on the following subjects.

1. Comparison of the performances of the several typical blade profiles,
2. Effect of the blade turning angle,
3. Effect of Reynolds Number and the blade surface roughness,
4. Effect of the boundaries of jet from the turbine nozzle.

1. Experimental Apparatus and Definitions

The present experiments were carried out with an ordinary low speed cascade wind tunnel, which is schematically shown in Fig. 1. Air

Fig. 1 Cascade test section

stream from a nozzle B contracts at the inlet of a channel formed by two fixed parallel walls and two parallel adjustable side walls D_1 and D_2 , and then enters into the cascade of blades C to be tested. After passing through the cascade, air stream flows into the atmosphere forming a free jet. Static pressure at several points on a blade, velocities and static pressures upstream and downstream of the cascade were measured. In particular, Pitot traverse and yaw-meter traverse were carried out along the line AA shown in Fig. 1.

The projected length and the span of the blade on a plane perpendicular to the cascade axis are 100 mm and 300 mm respectively, and the stream velocity of the leaving stream is approximately 30 m/s.

* Received 15 th March, 1954.

** Professor, Kyoto University, Yoshida-Honmachi, Sakyo-ku, Kyoto.

*** Research Engineer, Kawasaki Dockyard Co., Higashi-Kawasaki-cho, Ikuta-ku, Kobe.

From the measured pressure distribution on the blade surface, the force acting on a blade was calculated by graphical integration and from the survey of the leaving stream, deviation angle and loss of energy were estimated.

The following notations and definitions are used in the present report (See Fig. 2).

Fig. 2 Notations for the flow through cascade

- w_1 : Velocity, upstream of the cascade,
- w_2 : Velocity, downstream of the cascade,
- β_1 : Blade inlet angle or angle between cascade-axis and tangent line drawn to the skeleton line of the blade profile at the leading edge,
- β_2 : Blade outlet angle, or the same angle as β_1 , but at trailing edge,
- γ_1 : Angle between directions of w_1 and cascade-axis,
- γ_2 : Angle between directions of w_2 and cascade-axis,

α : Attack angle = $\beta_1 - \gamma_1$,
 $\Delta\beta$: Deviation angle = $\gamma_2 - \beta_2$,
 ρ : Density of fluid,
 p : Static pressure,
 p_1 : Static pressure upstream of cascade,
 p_2 : Static pressure downstream of cascade,
 P_z : Component of force acting on a blade in the direction of the cascade axis,
 P_x : Component of force acting on a blade perpendicular to the cascade axis,
 l : Projected length of blade,
 t : Pitch or blade spacing in the direction of the cascade axis,
 λ : t/l .

The coefficients C_z and C_x which represent the magnitude of forces P_z and P_x are defined as follows :

$$P_z = C_z \frac{\rho}{2} w_1^2 l,$$

$$P_x = C_x \frac{\rho}{2} w_1^2 l.$$

By the law of momentum, C_z and C_x can also be expressed as follows :

$$C_z = \lambda \sin 2\gamma_1 (1 + \tan \gamma_1 \cot \gamma_2), \quad \dots\dots\dots (1)$$

$$C_x = \lambda \left(\frac{\sin^2 \gamma_1}{\sin^2 \gamma_2} - 1 \right). \quad \dots\dots\dots (2)$$

The velocity coefficient ψ is defined as follows :

$$\psi = w_{2m}/w_{20},$$

where, w_{20} is the theoretical outlet velocity without energy loss, and w_{2m} is the actual mean outlet velocity. If the energy loss in the cascade is expressed in the form of $\xi \cdot \frac{\rho}{2} w_{20}^2$, then there exists the following relation,

$$\psi = 1 - \frac{\xi}{2},$$

so the energy loss can be obtained from velocity coefficient and *vice versa*.

2. Blade Profiles Used

The blade profiles used in the experiments are shown in Fig. 3. The inlet blade angle β_1 and the outlet blade angle β_2 of every blade tested are shown in Table 1. The blade profiles of A_1 - A_7 were prepared for the purpose of comparing the performances of the several typical blade profiles. Therefore, there exists no special relations whatsoever among their camber lines and thickness distributions. The blade profiles of A_1 - A_4 have almost symmetric camber lines with respect to the cascade axis, and their leading edges have rather sharp ends, while the blade profiles of A_5 - A_7 have asymmetrical camber lines, and they have rounded leading edges. The blade profiles of B_1 - B_4 have different turning angles, and their forms of the camber line and

Fig. 3 Blade profiles

the thickness distribution are changed by some systematic rules.

3. Profiles of A-Series

Although the profiles of A_1 - A_7 have no mutual relations among them, all of their blade angles having similar value as shown in Table 1. Accordingly, their performance characteristics will be discussed altogether in this paragraph.

Table 1

	A_1	A_2	A_3	A_4	A_5	A_6	A_7	B_1	B_2	B_3	B_4
β_1	28°	28°	28°	30°	34°	36°	34°	45°	37.5°	30°	25°
β_2	24°	24°	24°	30°	30°	30°	30°	45°	37.5°	30°	25°

Fig. 4 $C_z, C_x \sim \gamma_1$, A-Series

The coefficients of the forces acting on the blades are obtained by the graphical integration of the pressure distribution around the blade, and an example is shown in Fig. 4 in the case of $\lambda=0.6$, the abscissa being the inlet angle γ_1 , the values of coefficients C_z and C_x can also be calculated by equations (1) and (2) with the measured values of the angles γ_1 and γ_2 . Those values obtained from the measured pressure distributions coincide fairly well with those calculated by equations (1) and (2).

In Fig. 5, the velocity coefficient ψ is shown

Fig. 5 $\psi \sim \lambda$, A-Series

as a function of λ for the blades A_1 - A_7 and B_3 , when $\alpha=0^\circ$. It can be seen from this figure that there exists a value of λ which gives a maximum of velocity coefficient for each profile, and the optimum value of λ falls in the range from 0.6 to 0.7.

Figure 6 illustrates an example of the relation

Fig. 6 Pressure distribution and distribution of radius of curvature

between the pressure distribution and the curvature of the blade surface in the case of blade profile A_6 , when $\lambda=0.6$ and $\alpha=0^\circ$, the distance

from the leading edge along the upper surface of the blade is taken on the axis of abscissae, and the pressure on the blade surface and the radius of curvature of the blade surface are taken on the axis of ordinates. As this blade profile is constructed with some segments of circular arcs, it has necessarily discontinuities of the radius curvature on some points of the surface. It can be seen from Fig. 6 that the pressure distribution makes remarkable jump at these discontinuities. Accordingly, it is recommendable to reduce the said discontinuities under a certain limit.

4. Profiles of B-Series

The results of the tests carried out on four blade profiles B_1 - B_4 to find the effect of the turning angle on the performance characteristics of the cascade are as follows:

Fig. 7 Pressure distribution around blade profiles, B-Series

Figure 7 shows the pressure distributions along the surface of these blades, when $\lambda=0.7$ and $\alpha=0^\circ$. For easier comparison,

$$c_{pn} = \frac{2(p-p_1)}{\rho w_n^2} \cdot \frac{1}{\cot \gamma_1 + \cot \gamma_2},$$

is taken on the axis of ordinates where w_n is the component of velocity normal to the cascade axis. It can be seen from Fig. 7 that the pressure distribution around the blade varies systematically, depending presumably on both the variation of the camber line and that of thickness.

In Fig. 8, the values of C_z and C_x are plotted as a function of λ , when $\alpha=0^\circ$ and -10° .

Figure 9 shows the relation between the

Fig. 8 $C_x, C_z \sim \lambda$, B-Series Fig. 9 $\Delta\beta \sim \lambda$, B-Series

deviation angle $\Delta\beta$ and λ , when $\alpha = -10^\circ$. It can be understood from this figure that $\Delta\beta$ increases proportionally to λ and Fig. 10 shows the relation between $\Delta\beta$ and β_1 at $\alpha = 0^\circ$ and -10° , and it will be noted that $\Delta\beta$ also increases with β_1 .

Figure 11 shows the relation between the velocity coefficient ψ and the turning angle θ , where $\theta = \pi - \beta_1 - \beta_2 + \alpha - \Delta\beta$. Figure 11 (a) shows the case of $\frac{2(p_1 - p_2)}{\rho w_1^2} = 0$ and (b) shows that of

0.4. Here, those values of velocity coefficients when blade is stalled or flow is bubbled are omitted from these figures, and those values when the blade is nearly stalled are shown by broken

Fig. 10 $\Delta\beta \sim \beta_1$, B-Series Fig. 11 $\psi \sim \theta$, B-Series

lines in (a). Accordingly, the velocity coefficient seems to lie in the region bounded by two lines in figure (c), coefficient for larger λ lying near the upper limiting line and that for smaller λ near the lower one.

5. Effects of Reynolds Number and Roughness of Blade Surface

It is a well-known fact that the performance characteristics of the cascade of the turbine blades are deteriorated seriously whenever Reynolds number Re is less than about 10^5 , and roughness of the blade surface also affects on the blade characteristics, and some examples of the expe-

rimental results are explained in this paragraph.

In Fig. 12 (b), the line 1' shows the relation between the velocity coefficient and Reynolds

Fig. 12 $\Delta\beta, \psi \sim Re$, A-Series

number for the blade profile A_2 when $\lambda=0.6$ and $\alpha=0^\circ$. It may be seen from this figure that there is no effect of Reynolds number on the velocity coefficient in this profile and also only minor change in the pressure distribution on the blade surface is observed in varying Reynolds number. The lines 1 in Fig. 12 (a) and (b) show the same relation for the blade A_6 , when $\lambda=0.6$ and $\alpha=0^\circ$, and they show also the deterioration of the performance characteristics at low Reynolds number in regard to both the outlet angle and the velocity coefficient. The pressure distributions on the blade surface of the profile A_6 at several Reynolds number are shown in Fig. 13, in the case of $\lambda=0.6$ which shows that the distribution along the upper surface of the blade varies gradually with Reynolds number.

The blade A_6 , surface of which was entirely covered with fine sand with a mean diameter of 0.89 mm i. e. 0.0089 l, was tested at $\lambda=0.6$ and $\alpha=0^\circ$. In this case, the velocity coefficient and the deviation angle $\Delta\beta$ become as shown by lines 2 in Fig. 12 (a) and (b). The velocity coefficient diminishes by about 5 percent and deviation angle increases by about 1° compared with the blade profile with smooth surface. In other experiments the blade A_7 was tested at $\lambda=0.6$ and $\alpha=0^\circ$, the surface of which was partially made rough by covering with fine sand with the diameter

Fig. 13 Pressure distribution around blade for various Re numbers

of 0.49 mm i. e. 0.0049 l. The experimental results so obtained show that the effect is serious when the region of the leading edge of the blade or the upper surface near the leading edge is rough, while it is not so serious when the lower surface of the blade is rough. This can be easily understood from the relation between the boundary layer thickness and the energy losses.

6. The Influence of the Jet Stream Boundary

The flow passing through the row of turbine blades is actually three-dimensional. Accordingly there are some differences in comparison with the two-dimensional flow. The prominent difference is caused by the existence of so-called "secondary flow" that appears in the flow path between the blades. It has been presumed that the difference between the nozzle height and the blade height or span will affect the flow more or less. In the steam turbine of impulse type, the blade height b is larger than the nozzle height h , and the ratio $(b-h)/l$ of the high-pressure stage is larger than that of the low-pressure stage, hence, the effect of the difference $b-h$ seems to be stronger at high-pressure stage. To find this effect a series of experiment was carried out on the profile A_5 .

Our experimental apparatus for this purpose is illustrated schematically in Fig. 14. A clearance $a/l=0.15$ is set up between the nozzle exit and the inlet of the cascade.

Fig. 14 Test section for three dimensional research

The experiment is carried out for the several combinations of the nozzle height h , the blade height b , and the pitch-chord ratio λ , in two cases of attack angle $\alpha=0^\circ$ and 6° .

The notations used for this experiment are same as shown in paragraph 1, except the additional definition of the coefficients of the force. In this case, the components of the force are defined as follows:

$$P_z = C_z \frac{\rho}{2} w_1^2 l b,$$

$$P_x = C_x \frac{\rho}{2} w_1^2 l b,$$

where b is the blade height or span, when distance y is measured in the span-wise direction, coefficients of C_z and C_x can be expressed by the following integrals:

$$C_z = \frac{1}{b} \int_0^b c_z(y) dy,$$

$$C_x = \frac{1}{b} \int_0^b c_x(y) dy,$$

where coefficients of $c_z(y)$ and $c_x(y)$ are determined by the graphical integration of the pressure

Fig. 15 C_z, C_x distribution along blade span

distributions on the blade surface obtained at each y .

Figure 15 shows the typical example of the distribution of C_z and C_x along the blade span. In this figure, white dots represent the values at $\alpha=0^\circ$, the black dots at $\alpha=6^\circ$, and the vertical broken lines show the positions of the inside walls of the nozzle.

As can be seen from this figure, c_z has a flat distribution at the middle part of the blade span, but shows a sharp drop near the end of the boundary of the jet from the nozzle, the larger the $(b-h)$, the more serious such tendency becomes, contrarily, C_x shows a tendency of increasing in the region of the jet boundary.

In accordance with such tendency, the circulation around the blade changes along the blade span, as shown in Fig. 16, in which Γ is the circulation.

Fig. 16 Distribution of circulation along blade span

As is shown in Fig. 15 (a), when s_1 and s_2 are distances measured from the nozzle wall to the positions where C_z begins to drop and C_x begins to rise respectively, they seem to depend chiefly on the value of $(b-h)$ according to the present experiments. Figure 17 shows the mean

Fig. 17 s_1, s_2 and $s_3 \sim \frac{b-h}{l}$

values of s_1/l and s_2/l , the abscissa being $(b-h)/l$.

Figure 18 shows the values of C_z and C_x , and it will be noted that there are two values of C_z

Fig. 18 $C_z, C_x \sim \frac{b-h}{l}$

at $(b-h)/l=0$. The lower value is obtained in this experiment and the higher value is obtained by an experiment without slit between the exit of the nozzle and the inlet of the cascade i. e. $\alpha=0$. The difference between these two values is the reduction of force P_z caused by the leakage through the slit.

The velocity coefficient ψ , which also represents the energy loss, is calculated by averaging the values of the outlet velocity distribution, which is measured all over the section at the cascade outlet, and an example so obtained is shown in Fig. 19. It can be seen that in the case of $h/l=2.5$, the loss is greater by 6 percent

Fig. 19 $\psi \sim \lambda$

than in the case of two-dimensional flow, and at $(b-h)/l=0.5$ by 14 percent.

The direction of the outlet flow from the three-dimensional cascade is largely influenced by the presence of the secondary flow. Figure 20

Fig. 20 $\Delta\beta$ distribution along blade span

shows the distribution of the deviation angle along the blade span, and in this figure, the white dots are the values at $\alpha=0^\circ$ and the black dots at $\alpha=6^\circ$ and the chain line shows the values at $\alpha=0^\circ$ determined by a photographic method using wool tuft.

If s_3 is measured, as shown in Fig. 20 (a),

Fig. 21 δ distributions along blade span

from the surface of the inside wall of the nozzle to a position where $\Delta\beta$ is a mean value of its maximum and minimum, it can reasonably be considered that s_3 represents the approximate position of the center of the secondary flow vortex. The value of s_3 can also be roughly expressed as a function of $(b-h)/l$ just as s_1 and s_2 , and it is shown in Fig. 17.

The span-wise deflection of the leaving stream was also measured by the same photographic method and Fig. 21 shows the span-wise deflection angle δ plotted along the blade span, where the white dots show the values on the lower side of the blade surface and the black dots indicate those on the upper side.

7. Conclusions

The conclusions of our report are briefly summarized as follows:

1. The performance of the blade profile that has nearly symmetrical camber with a sharp leading edge is not satisfactory.
2. The optimum pitch-chord ratio is nearly $\lambda=0.65$ so long as our experiment is concerned.
3. The deterioration of the performance by the roughness of the blade surface is fairly remarkable.
4. The effects of the jet boundary depend on the difference between the blade span and nozzle height.

621.615.01:532.595

Experiments on the Surging of a Blower*

By Yoshinori SHIMOYAMA** and Tadayasu ITO***

This paper deals with experiments on surging phenomena.

The results may be summarized as follows:

1. The air column in the pipe-line vibrates in a certain form resembling that of free vibration.
2. The more rapid revolutions of a blower become, the more violent the surging grows, and the more important role the characteristic of a blower plays in determining the amplitude of the surging.
3. When the length and area of a cross section of the pipe-line are fixed, the surging is apt to be built up according to the volume of the tank.
4. The feature of the surging is considerably influenced by the conditions under which the energy is dissipated from vibrating air column.

1. Introduction

When a blower is in a small discharge or cut-off condition, a violent oscillation of air-pressure is apt to occur in the pipe-line. The term "surging of a blower" has been given to this phenomenon which, up to the present, has been only partially studied. Theoretically, it is clear that surging is a self-sustained oscillation of flow in the pipe-line caused by the negative characteristic of the blower⁽¹⁾ (characteristic of increasing pressure with increased discharge).

Experimentally few researches⁽²⁾ have been conducted up to the present. This leaves the obvious duty for some one to carry out systematic experiments, and clarify the actual state in which the surging occurs, by considering the effects of the dimensions of the pipe-line, tank and accessories of pipe-line on the surging and the relation between the form of characteristic curve and the surging.

The present authors deal with the surging phenomena rather more qualitatively but with some quantitative emphasis.

2. Apparatus and procedure

The authors use single-stage centrifugal blowers with no guide vane. The dimensions of

* Received 18 th June 1956.

** Professor of the Faculty of Engineering of Nagoya University.

*** Lecturer of the Faculty of Engineering of Nagoya University, Furôchô, Chikusa-ku, Nagoya.